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ROBERT STIRLING AND HIS FAMILY

Michael Stirling (grandfather)

A farmer of Dunblane, Perthshire, Scotland. Reputed to have invented the first rotary threshing machine in 1756 or 1758. Built a water mill.

Robert Stirling (uncle)

Assistant minister at Methven in 1763, later minister at Crieff. Died in 1787. His son, Michael, born in 1780, also entered the ministry.

Peter Stirling (father)

Peter Stirling was a farmer of Cloag, near Methven, Perthshire. He had two sons, Robert and James.

Robert Stirling

Robert Stirling was born in Cloag on October 25, 1790. He studied at the University of Glasgow, as is mentioned in the Fastie, but also at Edinburgh University (1805–6 and 1808). In 1805 he took classes in Latin and Greek, and in 1806 he studied advanced Latin and Greek, logic and mathematics, metaphysics and rhetoric. There is no record of his classes in 1808 and it is possible that it is not the same Robert Stirling who attended in that year, as owing to a change in the system of keeping records at that time his place of origin is not given. Robert Stirling was 15 years old when he went to Edinburgh. This age, or even younger, was quite normal for entering the university in those days.

Robert Stirling was licensed to preach by the Presbytery of Dunbarton on July 4, 1815. He was presented to the second charge at Kilmarnock, Ayrshire, by the Commissioner of the Duke and Duchess of Portland, and was ordained to the Ministry on September 19, 1816. He immediately became second or junior minister of Laigh Kirk, Kilmarnock, and two months later he applied for his first patent for the Stirling air engine and the heat regenerator—or economiser, as he himself called it. The patent assigned to him was No 4081 of 1816.

On July 10, 1819, he married Jane, the eldest daughter of William Rankine, a wine merchant at Galston, and five years later he had been 'translated' to Galston as minister of the church there. In 1827, and again in 1840, he and his brother James re-patented the air engine.

Early in 1840 the University of St Andrews, the oldest of Scottish

for his additional scientific achievements, as is stated in some places. Without any doubt this was a glorious day for the minister of Galston, who nonetheless remained in the village of his ministry, gaining widespread respect by the quality of his life and work—a quality shown by his care for the victims of the 1848–1849 cholera epidemic.

Ill-health at last forced him from the pulpit in 1876, and he died in 1878.

Patrick, William, Robert and James (sons)

Four of Robert Stirling's sons were engineers, and all made their mark in the engineering world. Patrick, born in 1820, gained fame as a locomotive engineer with the Great Northern Railway Company. His famous eight feet single locomotives have been photographed and described by engineers from all parts of the world. In August, 1895, these engines took an Aberdeen express over the 188 miles between London and York in 188 minutes. William, born in 1822, and Robert, born in 1824; were engineers in Peru. James, the youngest son, born in 1835, was a locomotive engineer with the Glasgow and South-Western Railway Company. He introduced many improvements in locomotive power.

James Stirling (brother)

James Stirling, the younger brother of the inventor, originally also studied for the Church at Glasgow and at Edinburgh. Earlier, however, he had directed his attentions towards mechanical engineering, and he eventually achieved local renown as an engineer. In 1827 improvements to the air engine were patented (No 5465) jointly by him and his brother. An engine based on Patent No 8652, of about 25 kW (35 horsepower), was constructed by him at the Dundee Foundry Company.

(From *The Stirling Engine*, J. Zarinchang, 1972; further fascinating details of the life and works of Robert Stirling are given by Edelman (1969).)

This book is dedicated to all who have contributed to the renaissance of the Stirling engine over the past forty years, especially workers at the Philips Research Laboratories, Eindhoven, and in particular, Dr. R. J. Meijer and Dr. J. W. L. Köhler.

REV. DR. ROBERT STIRLING (1790–1878)

Minister, Church of Scotland, and Inventor Extraordinaire.

Wrote in 1876 '. . . *These imperfections have been in a great measure removed by time and especially by the genius of the distinguished Bessemer. If Bessemer iron or steel had been known thirty five or forty years ago there is scarce a doubt that the air engine would have been a great success. . . It remains for some skilled and ambitious mechanist in a future age to repeat it under more favourable circumstances and with complete success. . . .*'

Photograph of a painting of Robert Stirling in

PREFACE

A few years have gone by since the completion of my earlier work *Stirling cycle machines*. Much has happened in that interval. It is timely to attempt, not simply a revised and updated version, but rather the production of a more substantial work. The earlier book was prepared from a collection of notes put together hurriedly for a one-day seminar during sabbatical leave at the University of Bath, in England, where I was remote from my reference collection. The slender volume was surprisingly well received, reprinted several times, published as a paperback, and translated into Russian.

This time I have no escape clause for this book was written with more leisure at the University of Calgary with full access to all my references. Not only that, but by the award of an Isaac Killam Memorial Fellowship, the University relieved me of all my teaching and administrative responsibilities during the period of its composition. I am most appreciative of this. It is my hope that the ensuing text will justify the respite from my ordinary labours, the burden of which undoubtedly fell on the shoulders of my colleagues in the Department of Mechanical Engineering.

Chapter 11, 'Free-piston Stirling engines,' was contributed by William Beale, the father of the free-piston Stirling engine. Chapter 20, 'Model Stirling engines', was contributed by Andrew Ross, an attorney fast becoming distinguished in the model Stirling engine fraternity. I was most happy to have these specialists participate, and wish to make clear that credit for all that is contained therein rests entirely with them.

The very extensive bibliography on Stirling engines contained herein was prepared by my daughter Josephine, my wife Ann, and my secretaries Karen Odegard and Marlene Stewart. This most tedious and exacting burden was lifted entirely from my shoulders in a most gratifying fashion.

Similarly the index was prepared with great dedication by Marlene Stewart. Her work greatly facilitated this important but exacting last lap.

I have many others to thank for much help over the past twenty years in the course of my work on Stirling engines. First, I owe to Aubrey Burstall, Professor Emeritus of the University of Newcastle-on-Tyne, my thanks for introducing me to Stirling engines and providing early opportunities. I owe much to an earlier mentor, W. E. Durney, of Wolverton Technical College, for rousing in me an abiding interest in engineers and their works. Specialists in the field who contributed to my understanding include my old and close friends, Ted Finkelstein and William Beale. Others are Bill Martini, Worth Percival, Rolf Meijer, Jan Köhler, and the

work. I have gained much from the writings and the many kindnesses of Zacharias, Kuhlman, Carlqvist, Lia, and their staff and colleagues. By their lucid writings Joe Smith at M.I.T., Bill Gifford of the University of Syracuse, Costa Rallis at Johannesburg, Allen Organ at Kings College, London, and their students have helped me understand many things.

At Calgary I am grateful for the many opportunities I have to share my thoughts with John Kentfield and to gain much from his optimistic, inquisitive and resourceful perception of the nature of things. Peter Glockner, Head of the Department of Mechanical Engineering and all my colleagues there, by their interest and encouragement, helped substantially in bringing the book to completion.

In keeping with current trends I have attempted to produce a 'bilingual' volume in SI units with the customary British units following in parentheses. Much of the labour of unit translation was accomplished by my able graduate student, Amoozegar-Faisie. On my instructions he adopted a cavalier approach to 'rounding off' that I expect will offend the purists. The responsibility for this is all mine.

The ladies of the office, particularly Karen Odegard, Jean Buckton, Marlene Stewart, Betty-Ann Maylor, and Ida Pfisterer all deserve my appreciation and thanks for their efforts in producing fair copy. Bert Unterberger and his girls have done great things with the illustrations herein.

Despite meticulous attention to detail by my assistants and myself there will be, no doubt, numerous errors and omissions for which I accept full responsibility. Many wrote to advise me of deficiencies in my earlier work. Where it was appropriate I have made the correction or rectified an omission. I appreciate the help of all who wrote to me before and thank, in advance, all who will discover and advise me of mistakes in the present work.

My children, Josephine and Christopher, and my students, have all contributed by sacrificing much of my time, interest, and effort they could have reasonably expected to be their due.

Finally I have to offer my greatest appreciation and thanks to my wife, Ann. She is undoubtedly the best-informed, or, at any rate, the most talked-at Canadienne in the field of Stirling engines. She has managed to survive and, with her unfailing good humour, somehow makes it all seem in the end to be worthwhile.

University of Calgary
September, 1978

G. WALKER

CONTENTS

1 INTRODUCTION

DEFINITION

A Stirling engine is a mechanical device which operates on a *closed* regenerative thermodynamic cycle, with cyclic compression and expansion of the working fluid at different temperature levels. The flow is controlled by *volume changes*, and there is a net conversion of heat to work or vice versa.

This generalized definition embraces a large family of machines with different functions, characteristics, and configurations. It includes both rotary and reciprocating machines, utilizing mechanisms of varying complexity. It covers machines capable of operating as prime movers, heat pumps, refrigerating engines, or pressure generators.

Other machines exist which operate on an *open* regenerative cycle, where the flow of working fluid is controlled by *valves*. For convenience, these may be called Ericsson engines. Unfortunately the distinction is not widely established in practice and the name 'Stirling engine' is frequently indiscriminately applied to all types of regenerative machines.

NOMENCLATURE

Stirling engines are frequently called by other names, including hot-air or hot-gas engines, or one of a number of designations reserved for particular arrangements of engine, i.e. Heinrici, Robinson, or Rankine-Napier. The result is a general lack of clarity in the nomenclature. It may be argued, convincingly, that the designation 'Stirling cycle' should be reserved for a particular idealized thermodynamic cycle, and the name 'Stirling engine' for a particular form of machine (which, incidentally, *does not* work on the Stirling cycle, a situation that does nothing to improve clarity). A preferred generic title would be 'regenerative thermal machine'. It is almost certainly too late for logic to prevail, and the name 'Stirling engine' will continue to be widely and indiscriminately used. However, a clear distinction should always be made between machines where the flow is controlled by (a) volume changes (Stirling engines) and (b) valves (Ericsson engines), because they have radically different characteristics. In this work we are concerned principally with Stirling engines and take only passing note of Ericsson machines.

Use of the term 'Stirling engine' as the generic title for closed-form regenerative thermal engines is comparatively recent. It is believed to have originated at the Research Laboratories of Philips at Eindhoven about 1954 (Meijer 1978)†. Up to that time it was customary simply to refer

† Private communication: for other references, see the bibliography at the end of the book for complete source details.

to hot-air engines and perhaps to preface this with the name of the inventor, i.e. Buckett, Wenham, Lehmann, Stirling, etc. The change of working fluid from air to helium or hydrogen at Philips in the mid-fifties made the term 'hot-air engine' inappropriate. The alternative title, 'Philips hot-gas engine' proved less attractive than 'Stirling engine' and so this latter has passed into general usage.

EARLY HISTORY

Stirling and Ericsson engines have a long history, which has been well surveyed by Finkelstein (1959), by Zarinchang (1972), and by Ross (1977).

The earliest air engine recognized by Zarinchang (1972) was the 'atmospheric fire-wheel' of Amontons in France, 1699. Another early air-engine by H. Wood in 1759 used a modified Newcomen atmospheric steam engine '*operating on the principle of hot or rarefied air...*'. In 1797 Glazebrook described an open-cycle machine operating on what may be distinguished as the first hot-gas cycle with a compound working fluid. Shortly thereafter, in 1801, Glazebrook again, introduced the original use of a closed cycle with repeated use of the same working fluid.

In 1807 Sir George Cayley, a Yorkshire squire, constructed an open-cycle, internal-combustion, hot-air engine that was probably the first engine of this type to work properly. Cayley was the pioneer aeronautical engineer and wisely recognized that he was unlikely to get any of his gliders to fly under power with a contemporary steam engine. He thus invented the new form of air engine and must have been exceedingly disappointed to find it equally unsuited as an aircraft engine. A century was to elapse before a suitable power plant enabled the Wright brothers to make the historic flight denied Cayley.

Robert Stirling, a minister of the Church of Scotland and originator of the regenerative heat exchanger, invented the closed-cycle regenerative engine in 1816 and remained actively involved with his brother, James, in its development for many years. As with all engineering developments they were beset by limitations of materials and at the end of his life, in 1876, Robert was led to write:

> '*These imperfections have been in a great measure removed by time and especially by the genius of the distinguished Bessemer. If Bessemer iron or steel had been known thirty-five or forty years ago there is scarce a doubt that the air engine would have been a great success... It remains for some skilled and ambitious mechanist in a future age to repeat it under more favourable circumstances and with complete success...*'

From the viewpoint of a century later, with public interest in Stirling engines rising to unprecedented levels, this was a wonderfully prophetic

Contemporaneously with Stirling, the Swedish inventor John Ericsson, working in England, introduced the open-cycle regenerative air engine in a variety of forms. The family of engines where the flow is controlled by valves are designated as Ericsson engines in recognition of his work. Ericsson was a great engineer and prolific inventor. He is accorded the honour of inventing the screw propellor and participated in the early development of railways in England. He was one of the contestants in the celebrated Rainhill trials of steam locomotives won by George Stephenson's *Rocket*. Ericsson later went to the United States and established the manufacture of both open- and closed-cycle regenerative engines. He continued a long and active career on a broad engineering front and was well known for his spectacular marine engineering feats.

Throughout the nineteenth century thousands of hot-air engines were made and used in a wide variety of sizes and shapes in Britain, Europe, the U.S.A., and other parts of the world. They were reliable and reasonably efficient. More importantly, they were safe compared with contemporary reciprocating steam engine installations and their associated boilers, which exploded with depressing regularity, due to poor materials and imperfect jointing techniques.

Many of the hot-air engines made were small, low-power machines of 100 W ($\frac{1}{4}$hp) to 4 kW (5 hp). Some large machines were also made however. The most notable was undoubtedly the enormous marine engine built by Ericsson in 1853 having four cylinders 4.2 m (14 ft) in diameter with a stroke of 1.5 m (5 ft), running at 9 revolutions per minute and producing about 220 kW brake power (300 hp). A contemporary report in the New York *Daily Times* of 12 January 1853, includes an account by the correspondent of his riding up and down on the pistons of this monster. The engine was designed for 330 kW (450 hp) and its performance was disappointing when installed in a ship called *The Ericsson*. The engine was replaced subsequently by steam engines but the ship was ill-fated and capsized in a squall in New York harbour. The story of the big engine has been well told by Ferguson (1961).

The internal combustion engine, in the form of the hot bulb gas engine was invented about the middle of the nineteenth century. Subsequently it was developed in the form of the gasoline, spark-ignition engine and the oil, compression-ignition engine. Later, at the turn of the century, the electric motor was invented and developed. Together, the internal combustion engine and electric motor gradually superseded both steam and Stirling engines in small sizes. However, a glance at the Sears-Roebuck catalogue for the early 1900s shows that in the United States, hot-air engines could be bought 'off-the-shelf' much as small gasoline engines or electric motors are today. Further, the heat-engine textbooks of that era

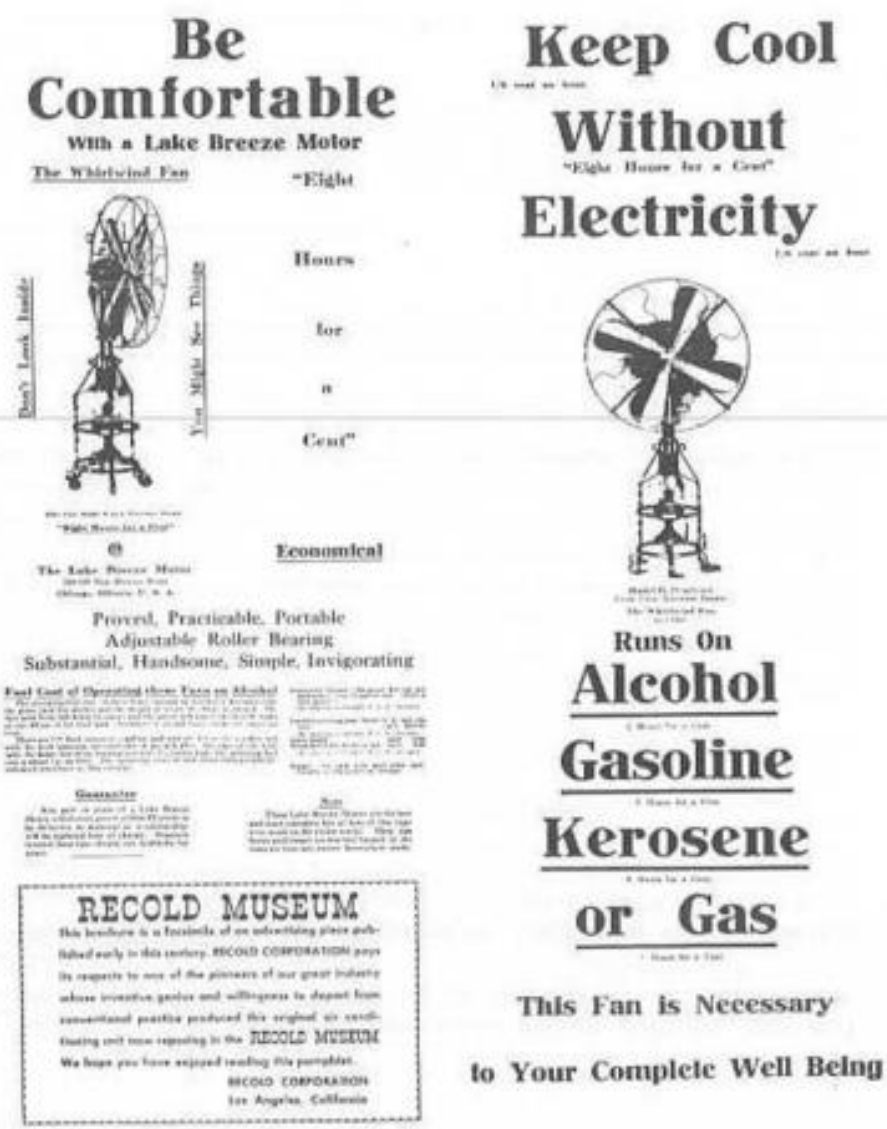


FIG. 1.1. Facsimile reproduction of an advertising piece for a Stirling-engine ventilation fan, *circa* 1900.

internal combustion engines. Nevertheless, by the time of the First World War, hot-air engines were no longer available commercially in large quantity, although production of machines for special purposes continued for many years.

One of the widest uses of hot-air engines in small sizes was to drive ventilating fans and water pumps. A facsimile reproduction of an advertising piece of 1900 is shown in Fig. 1.1. The production of similar machines in substantial quantities was carried on in England to the late 1940s largely for export to tropical countries. The author has a fond vision of a missionary after a long day in the field returning to his bungalow, pouring a mint julep, and relaxing on the stoep under the cooling breeze of his kerosene-driven Stirling engine.

Even today it is by no means unusual to come across hot-air engines. In 1971 for example a complete, new, and unused Bailey engine (*circa* 1902) of 3.6 kW (5 hp) rating, in its original factory packing was sold at a Welsh country-house auction. An Ericsson pumping engine was recently removed from a lakeside estate near Ottawa in Canada and refurbished for exhibition in the foyer of an Ottawa machine works. In the United States one occasionally finds hot-air engines still fulfilling useful purposes.

There are many collections of historical engines. Perhaps the most extensive is that at the Ford Museum in Greenfield Village, Dearborn, Michigan. The Smithsonian Institute in Washington has a notable collection, as does the Science Museum in London, and the Museum of Technology in Paris. Three or four hot-air engines may be found operating daily on a routine basis at the Museum of Science and Technology in Birmingham, England. The Ericsson Museum in Philadelphia has many items and memorabilia of intense interest concerned with John Ericsson. There are also many private collections and in the U.S. Mid-West it is commonplace to see hot-air engines displayed alongside steam engines at county and state fairs.

High quality facsimile reproductions of old-time hot-air engine catalogues and other advertising pieces have recently been published by Alan G. Philips of Orlando, Florida, and by Robert Huxtable of Lansing, Michigan. Models of hot-air engines are still made in large numbers by amateur craftsmen and are also obtainable commercially from the sources listed in the back of this volume.

REBIRTH OF THE STIRLING ENGINE

The present renaissance of interest in Stirling engines is due almost entirely to workers at the Philips Research Laboratory in Eindhoven. Work on small Stirling engines started there in the mid-1930s. The objective was to provide a small, quiet, thermally activated, electric-power generator for radios in areas of the world without regular power

supplies. It is said the choice between steam and hot-air engines was made following a chance visit to the Museum of Technology in Paris by one of the technical directors of the laboratory where he saw some of the old hot-air engines displayed. He believed, rightly, that modern materials and technology could elevate the hot-air engine to a performance undreamt of in earlier engines.

Work on Stirling engines continued during the German occupation in the Second World War and the first information about it was released soon after the war. Brilliant research and engineering resulted in the development of small generator sets far superior to any of the old hot-air engines. However, by the early 1950s the invention of the transistor and improvements in dry batteries had eliminated the original need. Recognizing the significance of their work, though, the Philips engineers switched their efforts to engines of higher powers, and the invention of the 'rhombic drive' and roll-sock seal led to the development of a family of single and multiple cylinder engines ranging in power up to hundreds of horsepower. This work has been reported periodically by Meijer (1969d) in a number of papers and more recently by van Beukering and Fokker (1973).

In 1958 the General Motors Corporation of Detroit made a licensing agreement with Philips which continued until 1970. Percival (1974) has given much interesting detail about the work done at General Motors in this period. They were concerned with Stirling engines for space and underwater power, vehicle and surface vessel propulsion, and stationary power.

Following relinquishment by General Motors of the licence agreement with Philips, the Ford Motor Co. of Detroit became licensees (in 1971) and after some preliminary work and evaluation have embarked (in 1977) on a seven-year substantial ($180M) programme of development for automotive Stirling engines funded by the U.S. Department of Energy.

Other licences were granted by Philips to United Stirling AB of Malmo, Sweden (in 1968) and to a West German consortium of the two diesel engine companies MAN and MWM (1967). United Stirling is a consortium of Swedish interests, principally Kockums, the great Swedish shipbuilders of Malmo, and FFW, a Swedish government company oriented to defence interests. In the early 1970s there was substantial cooperative effort between Philips and the European licensees in the development of transport-vehicle engines, and marine and underwater power systems. Later, the European licensees chose to follow a more independent path.

Zacharias (1974) has provided an interesting account of developments at MAN/MWM. Since about 1974, they have ceased much public discussion of their activities. However, it is understood that their programme of development is continuing unabated with concentration of effort on engines of 370 kW (500 hp) to 740 kW (1000 hp) for underwater power systems.

The Swedish group, United Stirling, concerned primarily in the early days with heavy-vehicle engines, have expanded their interests to the passenger car field. They did some work in cooperation with Ford but in 1978 a second substantial ($100M) programme for automotive Stirling engines was initiated in the United States by the Department of Energy and involving the two United States companies Mechanical Technology, Inc. of Latham, New York, and American Motors Corp., Detroit, Michigan, in partnership with United Stirling of Sweden. Hallare and Rosenqvist (1977) have provided an interesting historical review of the United Stirling work.

To manage these two automotive programmes, the Stirling Engine Project Office was established at the National Aeronautics and Space Administration (NASA) Lewis Research Centre, Cleveland, Ohio and to develop competence in the field an independent programme of study and development was started at NASA Lewis about 1975.

The External Combustion Engine Project was initiated by the U.S. Department of Energy in 1977 to demonstrate the use of 370 kW (500 hp) to 1480 kW (2000 hp) Stirling engines burning coal and other 'alternative' fuels with high efficiency, including municipal, industrial, and agricultural wastes. Management for this programme is the responsibility of the Argonne National Laboratory, Illinois.

In the early 1960s William Beale, a professor at the Ohio University, Athens, Ohio, invented the free-piston Stirling engine and with remarkably single-minded purpose brought the device into commercial production in the form of a small demonstration engine in the early 1970s. Further development followed with substantial funding to perfect a gas-fired air-conditioning pump unit now in the final pre-production stages. Beale's company, Sunpower Inc., in association with M.T.I., Latham, N.Y. is engaged in producing free-piston Stirling engines for other applications, notably as nuclear-fuelled space power plants and as solar-powered electric generators.

The General Electric Company in the United States is also developing Beale-type free-piston Stirling engines and are also said to be in the process of acquiring a Philips licence.

Substantive work on miniature engines for artificial hearts has been funded for over a decade by the United States National Institutes of Health, Cardiovascular Devices Branch, Department of Health, Education, and Welfare, Bethesda, Maryland. This broad investigative programme of power sources for artificial hearts has encompassed three

separate Stirling engine developments at the Donald Douglas Laboratories, Richland, Washington, at the Aerojet Liquid Rocket Co., Sacramento, California, and at the Thermo-electron Corp., Waltham, Massachusetts. A separate programme of Stirling engine artificial heart development is being carried out jointly by Westinghouse and Philips, funded by the U.S. Department of Energy.

Research on Stirling engines for solar, space, and underwater power, and the development of a basic laboratory research engine is in progress at the California Institute of Technology, Jet Propulsion Laboratory, Pasadena, California.

At the Atomic Energy Research Establishment, Harwell, England, a free-piston Stirling engine was developed in the early 1970s and is believed to be in commercial production as both a radioisotope or fossil-fuelled electric-power generator for navigation aids or other remote unattended locations requiring low-power electric supplies.

The Swedish company FFW, a parent of United Stirling, has developed a 10 kW (13.6 hp) power generator for recreational vehicles and other uses and is said to be planning to introduce the unit commercially in the United States in 1978 through the Stirling Power Systems Corp. of Ann Arbor, Michigan.

As a result of public concern about noise, air pollution, and energy conservation there has in recent years been increasing interest in Stirling engines. This is manifest in the soaring number of papers published annually, a total of 70 in 1977, and in 1978, 25 at the 13th Intersociety Energy Conversion Engineering Conference alone. Many original and innovative developments to the theory and practice of Stirling engines are being made in an increasing number of industrial companies, universities, and government research establishments in North America, Europe, South Africa, and elsewhere. A directory of the known Stirling engine activities was prepared in 1978 by William Martini (Martini 1978a) and is reproduced in abbreviated form herein.

REFRIGERATION MACHINES

Stirling engines operate well as cooling engines. The possibilities of this were recognized as early as 1834, by John Herschel, and in 1876 Alexander Kirk described a refrigerating machine that had been in use for ten years. However, it was not until the late 1940s that serious effort was directed to the commercial development of Stirling-cycle cooling engines. Again, this was undertaken by the Philips Company at Eindhoven under the direction of Dr. J. Köhler. The first cooling engine (an air liquefier) was introduced in 1953. Since that time, further research has resulted in the development of a variety of cryogenic cooling engines, covering a wide range of cooling capacities, and has led to the manufacture of associated equipment for cryogenic research and industrial applications. So far, Stirling-cycle cooling engines have proved more suitable for the cryogenic (extremely low temperature) range, rather than the higher temperature range (of domestic and industrial interest) which is dominated, at present, by 'Freon' vapour-compression refrigerating machines.

Other manufacturers have entered the small (and miniature) cryogenic cooling engine market, including Malakar Labs. Inc., Hughes Aircraft Co., Texas Instruments, and the Martin Marietta Corp. in the United States. These companies, together with North American Philips Inc. (who specialize in miniature cryogenic coolers), have as their principal interest the provision of small cooling engines for electronic applications, mainly in infra-red detection equipment for a variety of military and civil purposes.

Other reciprocating regenerative cryogenic cooling engines have been developed, principally the Collins helium liquefier, by A. D. Little Inc., and a variety of Gifford–McMahon machines. All these machines have valves and, in accordance with the definition adopted earlier, must be classified as Ericsson-cycle machines. This is not to suggest that such machines are unimportant. The development, by Samuel Collins at M.I.T., of a relatively inexpensive and reliable expansion-engine, capable of liquefying helium, was among the most significant advances in cryogenic engineering, opening up the possibilities of helium research on a broad front. The future benefits of this research in terms of superconducting electric-power transmission and miniaturized electronics are incalculable.

Early in the composition of this present work the decision was made to abandon the concept of a single volume embracing the Stirling engines for both power generation and cooling applications. The mass of information available is simply too great for adequate compression in a single volume. Furthermore, the readers of one are rarely interested in details of the other. Thus it seemed sensible, in fact became inevitable, that separate volumes were required, one for Stirling engines used for power purposes and another for the cooling application. This book is the one dealing with power applications. A companion volume is in preparation for the cooling engines.

PROSPECTS FOR THE FUTURE

At this time (1978) the future prospects of the Stirling engine appear brighter than ever before. In independent surveys of *alternative* automotive propulsion systems, the Stirling engine always emerges as a favoured unit with high potential for development. The characteristics of low noise, low levels of emission products, multifuel capacity, long life, comparable

size and weight, good partload performance, and favourable torque characteristics combine to form an impressive challenge to internal combustion engines. For this reason the United States Government has seen fit to underwrite the cost of a comprehensive seven-year development programme for automotive Stirling engines carried out by two independent, competent, and capable engineering teams. However the spark-ignition and compression-ignition engines in their variety of forms are formidable opponents so long as the distillate fuels, gasoline or diesel oil, remain available in relative plenty. As these supplies decline, a move to the all-electric economy will become inevitable. Perhaps it is in these circumstances that the Stirling engine, with thermal battery charged overnight, will emerge to become the dominant automotive propulsion system of the future.

Its use for stationary power generation in a broad power spectrum appears virtually assured. It is likely to find increasing use in heat pumps and refrigeration systems. An excellent start has been made on Stirling-engine artificial hearts. It will likely remain the dominant system for small and intermediate size cryogenic and low temperature refrigeration systems.

2 IDEAL THERMODYNAMIC CYCLES

SOME ELEMENTARY CONSIDERATIONS

THE first and second Laws of Thermodynamics appear to apply to all thermal power machines, including Stirling engines. Unfortunately no way to demonstrate the first and second laws in some simple but irrefutable fashion has been devised. Equally, of course, it is completely outside human experience for a machine to behave in contravention of these fundamental laws, despite the aspirations of many inventors. Proposals for perpetual motion machines *always* contravene the first or second law.

Belief in the Laws of Thermodynamics is closely akin to religious faith. If one has it, all is explainable. If one doubts, nothing is explainable. An understanding and belief in the laws of thermodynamics is necessary to appreciate the delights of regenerative thermal engines.

The first Law of Thermodynamics

The first Law, a restatement of the Law of Conservation of Energy, denies the possibility of an engine (or some thermodynamic 'black box') to exist, from which power, or work, can be drawn continuously, without replenishment. The first Law requires that at least as much energy (in any form) shall be *supplied* to the machine as is *taken* from it. Let us consider air and petrol, supplied to a spark-ignition engine. Firstly, they combine in a combustion process, and the hot gases drive the engine. Of the energy supplied in the fuel, about one-third goes to useful work output from the engine, another third goes to the cooling system, and the remaining third leaves the exhaust as low-grade thermal energy. If the petrol supply is terminated, the engine stops. This is a direct application of the first Law of Thermodynamics, and a matter of common experience.

The second Law of Thermodynamics

The second Law of Thermodynamics is, perhaps, less well understood. One statement of this Law is that it is not possible to construct a system which will operate in a cycle, extract heat from a reservoir and do an equivalent amount of work on the surroundings. The first Law says that the work produced can *never be greater* than the supplied heat, while the second Law goes further, and says that it *must always be less*. In the spark-ignition engine, it is the second Law which denies the possibility of converting all the energy in the supplied petrol to useful work. Some of the energy must be 'wasted' in the form of heat which is rejected to the cooling system or the exhaust.

These bold statements will suffice for our purpose here. For fuller discussion of the first and second Laws of Thermodynamics, and for

follows, the reader is referred to any standard text on engineering thermodynamics, e.g. Wallace and Linning† (1968).

Thermal efficiency

The ratio of the work produced W to the energy supplied Q is called the thermal efficiency η, so that $\eta = W/Q$. In many applications, it is important to maximize the thermal efficiency, since this represents the fraction of 'useful' energy obtained from that energy which is purchased in the form of gallons of petrol or oil. It is of interest, therefore, to establish the maximum possible value of thermal efficiency, bearing in mind the limitation of the second Law of Thermodynamics that it must always be less than unity.

Carnot efficiency

For any given situation, the theoretical maximum thermal efficiency depends only on the maximum and minimum temperature of the cycle, and is given by

$$\eta_{max} = (T_{max} - T_{min})/T_{max}.$$

This relationship is so important that it is given the special name 'Carnot efficiency'. It is the highest possible value, and is attained when all heat transfers to, or from, the system occur at the constant temperatures of T_{max} or T_{min} respectively.

P–V AND *T–S* DIAGRAMS

The processes which occur in the simplest thermal machine are still, however, so complicated that it is not possible to calculate precisely what is happening. Instead, a theoretical model is assumed, in which the various events are idealized to the extent necessary to make analysis of their operation possible. In this way, the operation of most types of machines may be simulated by the assumption of a repeated sequence of thermodynamic processes, called a *cycle*. Usually, each process is assumed to be one in which changes in the thermodynamic functions are occurring as the fluid moves from one state to another, but one of the functions is maintained constant. The important thermodynamic functions here are pressure (P), volume (V), temperature (T), internal energy (U), enthalpy (W) and entropy (S).

A cycle, consisting of a sequence of processes in which one of the thermodynamic functions is maintained constant while the others change, can be graphically represented in a variety of ways. Two of these are of importance in aiding the analysis of the operation of thermal machines.

† Wallace, F. J. *and* Linning, W. A. (1968). *Basic engineering thermodynamics.* Sir Isaac Pitman and Son Ltd., London.

These are the pressure–volume (P–V) and the temperature–entropy (T–S) diagrams.

These two diagrams are important because *areas on the P–V diagram represent work done* and *areas on the T–S diagram represent heat transferred.* As an example, consider Fig. 2.1, which shows a piston in a closed-ended cylinder. Some gas is trapped in the volume contained between the end of the cylinder and the piston, and can be said to be at a state represented by the point A, shown on the pressure–volume and temperature–entropy planes. If this gas were now heated through the cylinder walls, from some external source, a number of different things might happen. If the piston were fixed, the volume would remain constant, and heating the gas would result in increases in the pressure and temperature, as shown in Fig. 2.1(a). The supplied heat would be the

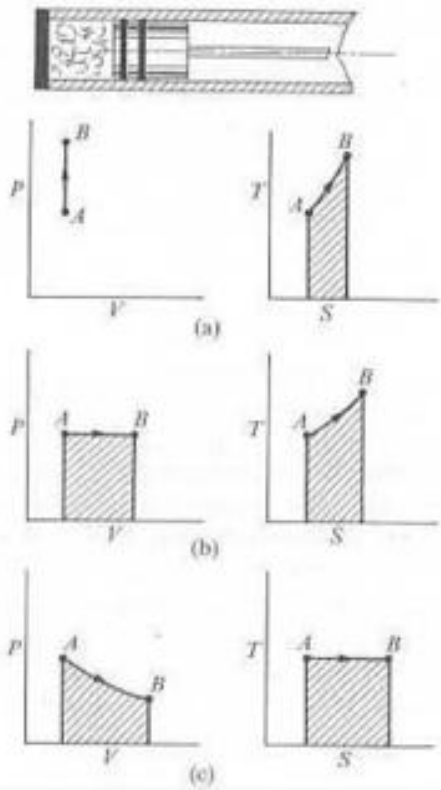


FIG. 2.1. Thermodynamic processes on P–V and T–S diagrams.
(a) Constant-volume heating.
(b) Constant-pressure heating.
(c) Constant-temperature heating.

shaded area shown on the T–S diagram. Any work done would be the area of the P–V diagram: in this case there is no change in volume so no work is done. If, instead, the piston was free to move, and the process of heat addition was regulated, so as to maintain the pressure or temperature constant, the P–V and T–S diagrams shown in Fig. 2.1(b) and 2.1(c), would result. In both these cases, work is *done* by the gas in expanding to a larger volume as the heat is added to the system.

THE CARNOT CYCLE

The Carnot cycle is a thermodynamic cycle comprised of four processes, occurring sequentially, as shown in Fig. 2.2.

To consider the operation of an ideal Carnot-cycle engine, let us assume that we have a cylinder and piston, as shown in Fig. 2.2. We assume that the cylinder is perfectly insulated, and that the piston can move, with no friction and no leakage of the working fluid from the cylinder. The cylinder head is a component that can be perfectly conducting or perfectly insulating, as we choose.

For the start of the cycle we will assume the piston to be at the *outer dead point* (O.D.P.), so that the volume contained within the piston and cylinder combination is a maximum. The pressure and temperature (T_{min}) of the working fluid are at their minimum values, and are represented on Fig. 2.2(a) by the point 1. We let the piston move towards the cylinder head, so that compression occurs, shown by the process 1–2 on Fig. 2.2(a). For this process, we assume that the cylinder head is perfectly conducting, and that the heat-transfer rate is infinite, so that the process occurs isothermally (constant temperature). Work is done *on* the gas, represented by the shaded area on the P–V diagram; heat is *abstracted* from the working fluid, represented by the shaded area on the T–S diagram. In this case, since the process is isothermal, the amount of the heat transferred is exactly equal to the work done (in comparable units).

For the second process, isentropic compression, shown in Fig. 2.2(b), the cylinder head is made perfectly insulating. As the piston continues to move towards the cylinder head, heat can no longer be abstracted from the working fluid, and so, ideally, the entropy remains constant. This process results in a decrease in the volume, and in increases in both the pressure and temperature. The work done on the gas is the shaded area on the P–V diagram, but there is no heat transferred. The remaining two processes, isothermal expansion from 3 to 4 and isentropic expansion from 4 to 1 then follow, and are shown on Fig. 2.2(c) and Fig. 2.2(d), respectively.

If these four diagrams are combined, the resultant P–V and T–S diagrams are as shown in Fig. 2.2(e). The shaded area, enclosed by the

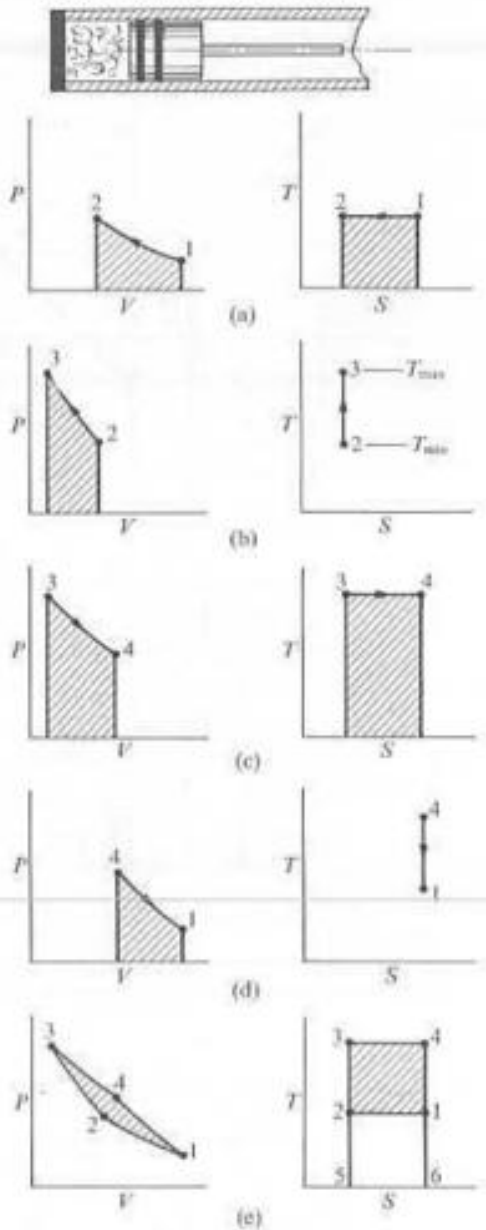


FIG. 2.2. The Carnot cycle.
(a) Isothermal compression.
(b) Isentropic compression.
(c) Isothermal expansion.

the cycle. Similarly, on the T–S diagram, the area 3–4–5–6 is the heat supplied to the cycle. The area 1–2–3–4 is the amount converted to work, and the area 1–2–5–6 is the 'waste heat' of the cycle. It is clear, from this diagram, why the Carnot cycle has the highest possible thermal efficiency. Given temperature limits T_{max} and T_{min}, no possible sequence of thermodynamic processes could result in a larger ratio of the areas 1–2–3–4 and 3–4–5–6, so that the efficiency, $\eta = W/Q$ = area 1–2–3–4/area 3–4–5–6 must be a maximum.

Absolute temperatures must be used in thermodynamic analysis. The zero temperature on the T–S diagram is −273 °C (= 0 K) or −460 °F (= 0 °R), so that the 'waste-heat' area 1–2–5–6 may be very appreciable.

It is clear that the efficiency of the Carnot cycle (and this generally applies to all engines) can be improved by (a) increasing T_{max} and (b) decreasing T_{min}. The ultimate maximum value of T_{max} is governed by the materials used to construct the engine, this is called the 'metallurgical limit'. The lowest possible value of T_{min} is that temperature at which cooling water or air is available, generally, the ambient atmospheric temperature.

In practice, it is not possible to construct Carnot-cycle engines. There are no materials which are perfectly insulating or conducting, and all pistons sliding in cylinders do have friction and leakage losses. However, the most serious difficulty arises because isothermal and isentropic processes for a gas (say, air), have slopes that are so little different, when compared on a P–V diagram, that the area of the P–V diagram shown in Fig. 2.2(e) becomes negligibly small, unless very high pressures and very long piston strokes are used This would result in a tremendously heavy engine, which would be quite unable to produce sufficient work to overcome its own friction losses. Despite this lack of practicality, the Carnot cycle is useful in a preliminary study of the operation of an engine. Furthermore, with some modifications (which change it to the Rankine cycle), the Carnot cycle is representative of the mode of operation of liquid-vapour machines, such as reciprocating steam-engines, steam turbines, or 'Freon' refrigerating plants.

THE STIRLING CYCLE

The Stirling cycle is similar, in some respects, to the Carnot cycle. It is illustrated in Fig. 2.3.

Consider a cylinder containing two opposed pistons, with a regenerator between the pistons. The regenerator may be thought of as a thermodynamic sponge, alternately releasing and absorbing heat. It is a matrix of finely-divided metal in the form of wires or strips. One of the

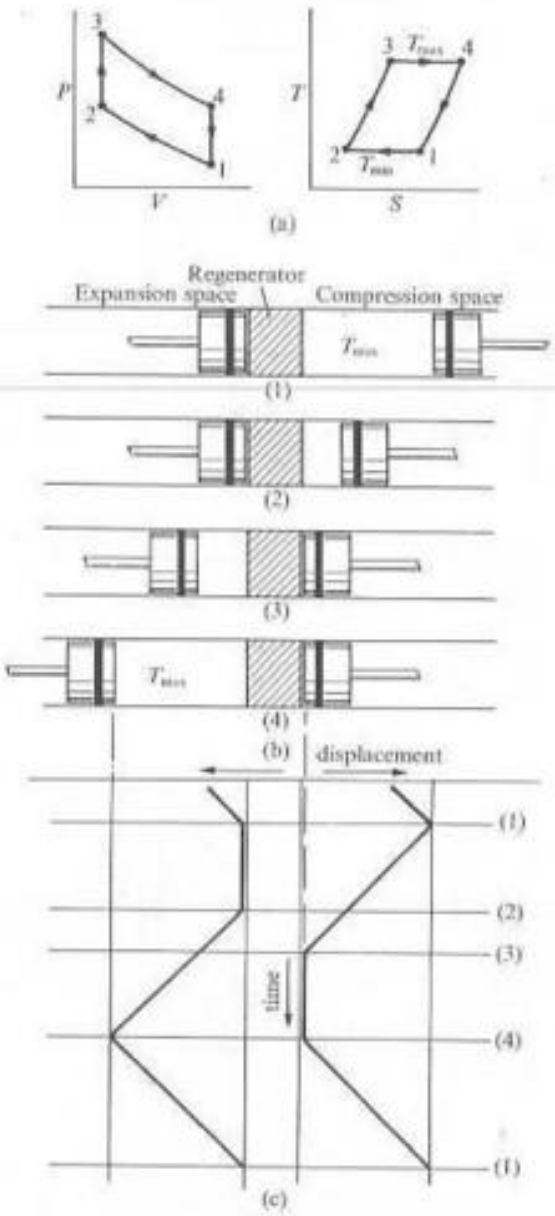


FIG. 2.3. The Stirling cycle.
(a) P–V and T–S diagrams.
(b) Piston arrangement at the terminal points of the cycle.

expansion space, and is maintained at a high temperature T_{max}. The other volume is called the *compression space*, and is maintained at a low temperature T_{min}. There is, therefore, a temperature gradient ($T_{max} - T_{min}$) between the ends of the regenerator, and it is assumed that there is no thermal conduction in the longitudinal direction. As in the Carnot cycle, it is assumed that the pistons move without friction or leakage loss of the working fluid enclosed between them.

To start the cycle, we assume that the compression-space piston is at the outer dead point, and the expansion-space piston is at the inner dead point, close to the face of the regenerator. All the working fluid is then in the cold compression space. The volume is a maximum, so that the pressure and temperature are at their minimum values, represented by 1 on the P–V and T–S diagrams, shown in Fig. 2.3. During compression (process 1–2), the compression piston moves towards the inner dead point, and the expansion-space piston remains stationary. The working fluid is compressed in the compression space, and the pressure increases. The temperature is maintained constant because heat Q_c is abstracted from the compression-space cylinder to the surrounds.

In the transfer process 2–3, both pistons move simultaneously, the compression piston towards (and the expansion piston away from) the regenerator, so that the volume between them remains constant. Therefore, the working fluid is transferred, through the porous metallic matrix of the regenerator, from the compression space to the expansion space. In passage through the regenerator, the working fluid is heated from T_{min} to T_{max}, by heat transfer from the matrix, and emerges from the regenerator into the expansion space at temperature T_{max}. The gradual increase in temperature in passage through the matrix, at constant volume, causes an increase in pressure.

In the expansion process 3–4, the expansion piston continues to move away from the regenerator towards the outer dead point; the compression piston remains stationary at the inner dead point, adjacent to the regenerator. As the expansion proceeds, the pressure decreases as the volume increases. The temperature remains constant because heat Q_E is added to the system from an external source.

The final process in the cycle is the transfer process 4–1, during which both pistons move simultaneously to transfer the working fluid (at constant volume) back, through the regenerative matrix form and the expansion space, to the compression space. In passage through the matrix, heat is transferred from the working fluid to the matrix, so that the working fluid decreases in temperature, and emerges at T_{min} into the compression space. Heat transferred in the process is contained in the matrix, for transfer to the gas in process 2–3 of the subsequent cycle.

The cycle is composed, therefore, of four heat-transfer process.

Process 1–2: isothermal compression; heat transfer *from* the working fluid at T_{min} to the external dump.

Process 2–3: constant volume; heat transfer *to* the working fluid from the regenerative matrix.

Process 3–4: isothermal expansion; heat transfer *to* the working fluid at T_{max} from an external source.

Process 4–1: constant volume; heat transfer *from* the working fluid to the regenerative matrix.

If the heat transferred in process 2–3 has the same magnitude as in process 4–1, then the only heat transfers between the engine and its surroundings are (a) heat supply at T_{max} and (b) heat rejection at T_{min}. This heat supply and heat rejection at constant temperature satisfies the requirement of the second Law of Thermodynamics for maximum thermal efficiency, so that the efficiency of the Stirling cycle is the same as the Carnot cycle, i.e. $\eta = (T_{max} - T_{min})/T_{max}$. 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 increases 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.

A comparison of the P–V diagrams of a Carnot and Stirling cycle, between given limits of pressure, volume, and temperature, is shown on Fig. 2.4. The shaded areas 5–2–3 and 1–6–4 represent the additional work made available by substituting constant-volume processes for isentropic processes. The isothermal processes (1–5 and 3–6) of the Carnot

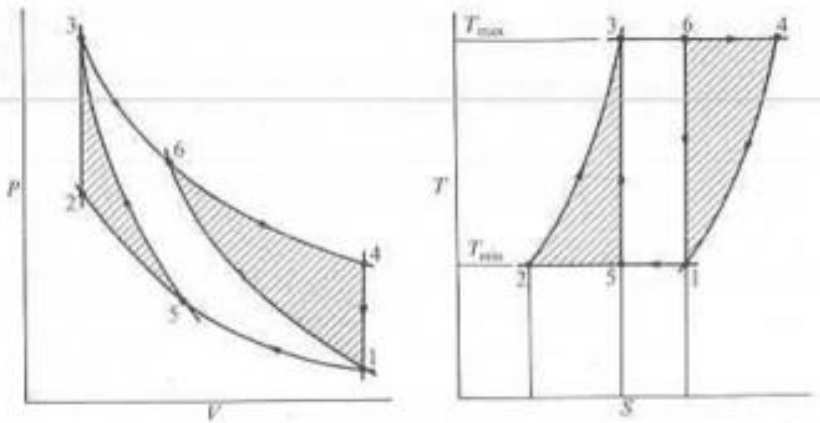


FIG. 2.4. Stirling and Carnot cycles. The Stirling and Carnot cycles are superimposed, with common values for the maximum and minimum temperatures, pressures, and volumes. Hatched areas on the P–V plane represent the increased work output of the Stirling cycle. Hatched areas on the T–S plane represent increased heat-transfer of the Stirling cycle.

cycle are extended to process 1–2 and 3–4, respectively, so that the quantities of heat supplied to—and rejected from—the Stirling cycle are increased in the same proportion as the available work. The *fraction* of supplied heat which is converted to work (the efficiency), is the same in both cycles.

The Ericsson cycle

In the Ericsson cycle, the processes of constant-volume regenerative heat transfer, described above, are replaced by constant-pressure regenerative processes. This leads to the P–V and T–S diagrams shown in Fig. 2.5. The efficiency of the cycle is the same as that of the Carnot cycle but, as in the Stirling cycle, the net available work and the quantities of heat transferred are much greater, for given limits of pressure, volume, and temperature.

The Stirling cycle as a prime mover

In the previous discussion, heat, at some high temperature T_{max}, was supplied to the cycle. Part of the heat was converted to work, and part was rejected, as heat, at a low temperature T_{min}. This describes a cycle operating as a prime mover, a machine producing work from a high-temperature energy source, and rejecting heat at a low temperature.

The Stirling cycle as a refrigerating machine

The same ideal machine which was used to describe the operation of the Stirling cycle as a prime mover can be used to describe the operation of the cycle as a refrigerating machine. The only difference is that the

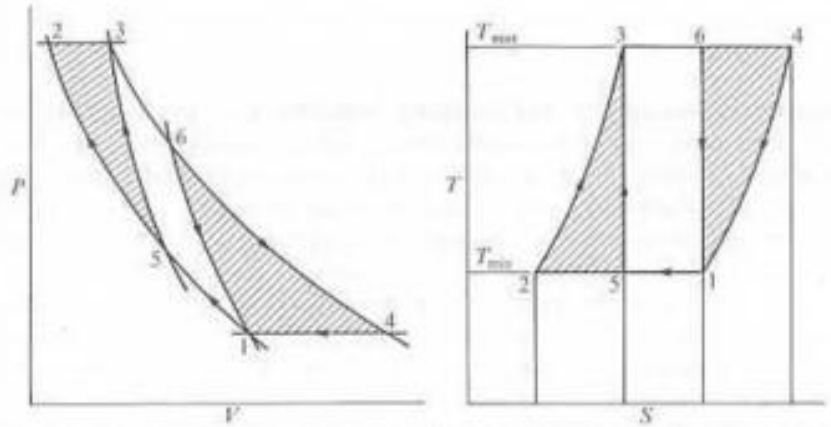


FIG. 2.5. Ericsson and Carnot cycle. The Ericsson and Carnot cycles are superimposed, with common values for the maximum and minimum temperatures, pressures, and volumes. As before, the hatched areas represent the increased work output and heat-transfer of the Ericsson cycle.

temperature of the heat supplied from the external source during expansion is *lower* then the temperature at which heat is rejected from the working fluid during compression. This is illustrated in Fig. 2.6, where P–V and T–S diagrams for the prime mover and refrigerating machine are superimposed.

When the Stirling-cycle machine is operating as a refrigerator, heat is lifted from the cold zone during the expansion process 3'–4'. The work of compression (area 1–2–5–6) is the same for both the prime mover and refrigerator. The work of expansion (area 4'–3'–5–6), in the case of a refrigerator is less than the compression work and work equivalent to area 1–2–3'–4', from an external source, is necessary to drive the cycle. During transfer from the compression space to the expansion space, in process 2–3', the working fluid experiences a *decrease* in temperature, and a corresponding increase in temperature during the alternate transfer process 4'–1.

The performance of a refrigerator is assessed in terms of its coefficient of performance (COP), where

$$\text{COP} = \text{heat lifted/work done} = T_{ref}/(T_{min} - T_{ref}).$$

The COP of the Stirling, Ericsson, and Carnot cycles are the same, for given temperature limits, but the *refrigerating* capacity of the Stirling and Ericsson cycles are much greater than the Carnot cycle, for given pressure and volume limits.

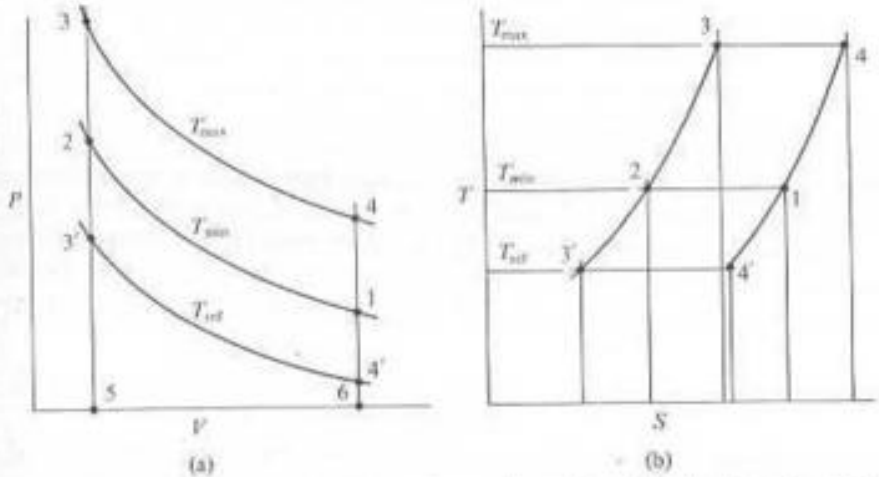


FIG. 2.6. Stirling cycle as prime mover and as cooling engine. In both prime movers and cooling engines the *compression* process occurs at temperature T_{min}. The *expansion* process occurs at a temperature of (a) T_{max}, in the prime mover, and (b) T_{ref} in the cooling engine. Heat is supplied at a high temperature to produce useful work in the prime mover. Heat is abstracted to refrigerate, in the cooling engine, and a net input of work is required.

The Stirling cycle as a heat pump

As a heat pump, the Stirling cycle operates exactly as it did in the refrigerating machine described above, with the temperature of the expansion space T_{ref} less than the temperature of the compression space T_{min}. The difference between operation as a heat pump and refrigerating machine is that both T_{ref} and T_{min} are increased. In both the prime mover and refrigerator application, T_{min} is the ambient atmospheric temperature at which cooling water is available, whereas, in the case of the heat pump, T_{min} is the temperature at which heat is rejected from the system, and is the useful product, for heating a concert hall or office building. Therefore, for a heat pump, T_{min} is *above* the ambient atmospheric temperature, and heat is supplied to the cycle (at T_{ref}), from atmospheric air or river water, at approximately the ambient atmospheric temperature.

A comparison of the Stirling cycle's performance as a heat pump and a refrigerator is drawn in Fig. 2.7. In both cases, work from an external source is required to drive the cycle, and is equivalent to area 1–2–3′–4′. In

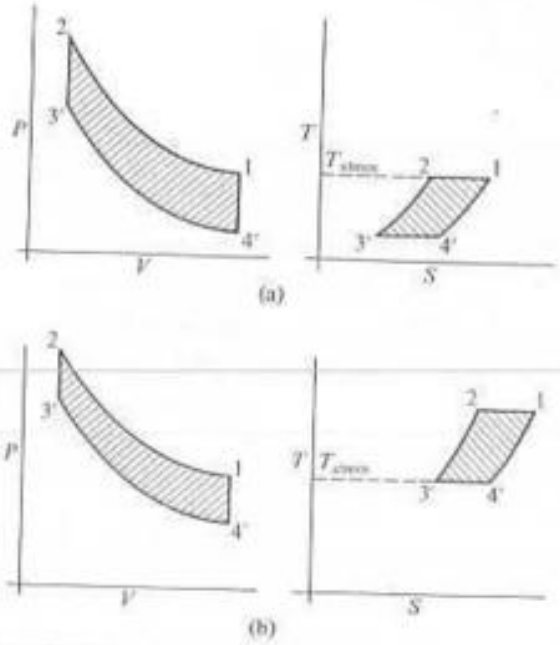


FIG. 2.7. Stirling cycle as a heat pump and a refrigerating machine. (a) heat pump. (b) refrigerating machine. When operating as a heat pump or refrigerating machine, the expansion process occurs at a temperature less than that for the compression process, and a net input of work is required. In the heat pump, expansion is at atmospheric temperature, and the heat rejected during compression, at a high temperature, is the useful output of the cycle. In the refrigerating machine, the heat supplied, during expansion at a low temperature, is the useful refrigerating capacity of the cycle.

the case of the heat pump, the useful product is the heat rejected at temperature T_{min}, and the performance of a heat pump is therefore assessed as

$$\text{COP}_{hp} = (\text{heat rejected})/(\text{work done}) = (T_{min})/(T_{min} - T_{ref}).$$

This is the inverse of the thermal efficiency, whereas the coefficient of performance of a refrigerator, namely,

$$\text{COP}_{ref} = (T_{ref})/(T_{min} - T_{ref}),$$

is *not* the inverse of thermal efficiency.

The Stirling cycle as a pressure generator

Systems closely related to the Stirling cycle have been proposed, and are under investigation, where the objective is to pump a fluid, and to increase the pressure. When the fluid to be pumped is a liquid (or gas), separated by a diaphragm (or piston) from the working fluid in the Stirling cycle device, the system can be classified as a Stirling cycle which is working as a prime mover. In other instances, where the working fluid is itself the fluid to be compressed and pumped, there are, invariably, valves or other flow-controlling devices; these systems cannot be classified as Stirling-cycle machines, using the limited definition proposed in the Introduction. Nevertheless, they are discussed in the literature as Stirling-cycle machines. In most cases, fluid is added to the cycle when the pressure is low, and withdrawn at a higher pressure. Heat is supplied at a high temperature, and rejected at a low temperature. The work during expansion is greater than the work during compression, by an amount equivalent to the 'pump' work of the compressed fluid.

COMPOSITE CYCLES

The Reitlinger cycle

The most general form of idealized thermodynamic cycle for Stirling and other regenerative engines consists of (a) two isothermal processes where the heat is supplied to and drawn from the system, (b) two polytropic processes. In the ideal reversible heat engine it is essential that external heat exchange occurs only at the upper and lower temperature limits of the cycle. Heat reception and heat rejection during the polytropic phase must therefore be allowed for by heat storage in a regenerator.

This generalized cycle was first analysed by Reitlinger (1876). The Reitlinger cycle can exist in an infinite variety of forms distinguished by the character of the polytropic regenerative phases.

Most introductions to applied thermodynamics emphasize the singularity of the Carnot cycle as an ideal cycle with

Now it can be seen that the Carnot cycle is simply a Reitlinger cycle in which the polytropic phases are adiabatic isentropic in character. This rather forbidding classification simply means that the processes connecting the isothermal heat supply and rejection occur without any heat transfer (adiabatic) and at constant entropy (isentropic).

Two other special cases which have been given names are the Stirling and Ericsson cycle where the regenerative processes occur at constant volume and constant pressure respectively.

An infinite number of other thermodynamic cycles may be postulated in which the regenerative processes are neither at constant volume or pressure, nor are they isentropic. All these cycles are characterized by the same thermodynamic efficiency, simply:

$$\eta = \frac{\text{Heat supplied} - \text{Heat rejected}}{\text{Heat supplied}} = \frac{T_{max} - T_{min}}{T_{max}}.$$

The Rallis isothermal cycle

Another special case of the generalized Reitlinger cycle was developed by Rallis (1976)†. The Rallis isothermal cycle is a composite cycle defined by two isothermal processes bounded by regenerative processes that occur partly at constant volume and partly at constant pressure as shown in Fig. 2.8.

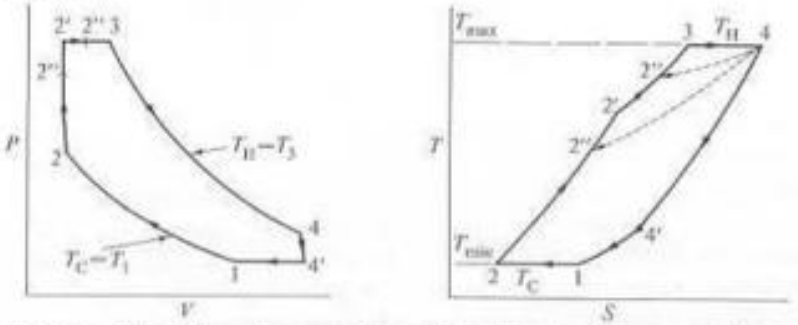


FIG. 2.8. The Rallis isothermal regenerative cycle. In this cycle the processes of compression and expansion are isothermal. The regenerative processes take place partly at constant-volume and constant-pressure conditions. If the regenerative process is imperfect the working fluid is heated regeneratively to temperature $T_{2'}$ and supplementary heating is required to temperature T_3.

It is evident that the Stirling and Ericsson cycles are special cases of the Rallis composite cycle and some other special cases can also be defined. It is convenient therefore to develop analytical expressions for the general form of the Rallis cycle and then to reduce these to the special cases as required.

† Private communication

The following treatment was contributed by Rallis (1976):

Nomenclature:

T_i = absolute temperature of the working fluid at state i
v_i = specific volume of the working fluid at state i
p_i = pressure of the working fluid at state i
W = specific external work done per cycle
Q_{ext} = specific external heat supplied per cycle
R = characteristic gas constant
c_p = specific heat at constant pressure of working fluid
c_v = specific heat at constant volume
$\bar{p}_m$ = mean effective pressure
γ = ratio of specific heats c_p/c_v

For convenience in analysis we define the following non-dimensional parameters:

$\tau = T_3/T_1$ the temperature ratio
$r_c = v_1/v_2$ the compression volume ratio
$r_e = v_4/v_3$ the expansion volume ratio
$\alpha = T_2/T_{2'} = p_3/p_2$ the constant volume heating ratio
$\beta = T_3/T_{2'} = v_3/v_2$ the constant pressure heating ratio
$\rho = T_4/T_{4'} = p_4/p_1$ the constant volume cooling ratio
$\sigma = T_4/T_1 = v_4/v_1$ the constant pressure cooling ratio
$\varepsilon = (T_{2'} - T_1)/(T_3 - T_1)$ the regenerator effectiveness
$\eta = W/Q_{ext}$ the thermal efficiency
$\xi = \bar{p}_m/p_1$ specific power of the cycle

Because $T_2 = T_1$ and $T_4 = T_3$ in this isothermal cycle it follows that:

$$\alpha\beta = \rho\sigma = \tau \tag{2.1}$$

$$r_e = \sigma r_c/\beta = \alpha r_c/\rho \tag{2.2}$$

and

$$T_{2'} = [\varepsilon(\tau - 1) + 1]T_1 \tag{2.3}$$

The external work done per cycle is:

$$\begin{aligned} W &= p_3(v_3 - v_2) + RT_3 \ln r_e - p_1(v_4 - v_1) - RT_1 \ln r_c \\ &= p_3v_3 - p_3v_2 + RT_3 \ln r_e - p_1v_4 + p_1v_1 - RT_1 \ln r_c \\ &= RT_3 - \alpha p_2v_2 + RT_3 \ln r_e - \sigma p_1v_1 + RT_1 - RT_1 \ln r_c \\ &= R[T_3 - \alpha T_1 + T_3 \ln r_e - \sigma T_1 + T_1 - T_1 \ln r_c] \\ &= RT_1[\tau - \alpha + \tau \ln(\alpha r_c/\rho) - \sigma + 1 - \ln r_c] \\ &= RT_1[\tau\{1 + \ln(\alpha\sigma r_c/\tau)\} - (\alpha + \sigma + \ln r_c) + 1] \end{aligned} \tag{2.4}$$

The heat supplied externally per cycle for $T_1 \leq T_{2'} \leq T_{2'}$ is given by:

$$Q_{ext} = c_v(T_{2'} - T_{2'}) + c_p(T_3 - T_{2'}) + RT_3 \ln r_e$$

and for $T_{2'} \leq T_{2'} \leq T_3$

$$Q_{ext} = c_p(T_3 - T_{2'}) + RT_3 \ln r_e$$

Now

$$T_1 \leq T_{2'} \leq T_{2'}$$

implies $T_1 \leq [\varepsilon(\tau-1)+1]T_1 \leq \alpha T_1$

or $1 \leq [\varepsilon(\tau-1)+1] \leq \alpha$

and

$$T_{2'} \leq T_{2'} \leq T_3$$

implies $\alpha T_1 \leq [\varepsilon(\tau-1)+1]T_1 \leq T_3$

or $\alpha \leq [\varepsilon(\tau-1)+1] \leq \tau$

Thus for $T_1 \leq T_{2'} \leq T_{2'}$

$$\begin{aligned} Q_{ext} &= c_v[\alpha - \{\varepsilon(\tau-1)+1\}]T_1 \\ &\quad + c_p[T_3 - \alpha T_1] + RT_3 \ln(\alpha r_c/\rho) \\ &= RT_1\left[\frac{\alpha - \varepsilon(\tau-1) - 1}{(\gamma-1)} + \frac{\gamma(\tau-\alpha)}{(\gamma-1)} + \tau \ln(\alpha r_c/\rho)\right] \\ &= \frac{RT_1}{(\gamma-1)}[(\alpha-1) - \varepsilon(\tau-1) + \gamma(\tau-\alpha) + \tau(\gamma-1)\ln(\alpha r_c/\rho)] \end{aligned} \tag{2.5}$$

whilst for $T_{2'} \leq T_{2'} \leq T_3$

$$\begin{aligned} Q_{ext} &= c_p[T_3 - \{\varepsilon(\tau-1)+1\}T_1] + RT_3 \ln(\alpha r_c/\rho) \\ &= RT_1\left[\frac{\gamma\{\tau - \varepsilon(\tau-1) - 1\}}{(\gamma-1)} + \tau \ln(\alpha r_c/\rho)\right] \\ &= \frac{RT_1}{(\gamma-1)}[\gamma(1-\varepsilon)(\tau-1) + \tau(\gamma-1)\ln(\alpha r_c/\rho)] \end{aligned} \tag{2.6}$$

The thermal efficiency for $T_1 \leq T_{2'} \leq T_{2'}$ is given by:

$$\eta = \frac{(\gamma-1)[\tau\{1+\ln(\alpha r_c/\rho)\} - (\alpha+\sigma+\ln r_c)+1]}{\gamma(1-\varepsilon)(\tau-1) + \tau(\gamma-1)\ln(\alpha r_c/\rho)} \tag{2.7}$$

whilst for $T_{2'} \leq T_{2'} \leq T_3$

$$\eta = \frac{(\gamma-1)[\tau\{1+\ln(\alpha r_c/\rho)\} - (\alpha+\sigma+\ln r_c)+1]}{\gamma(1-\varepsilon)(\tau-1) + \tau(\gamma-1)\ln(\alpha r_c/\rho)} \tag{2.8}$$

The effective work output per cycle may be represented by the indicated mean effective pressure, defined by : $\bar{p}_m$ = W/stroke volume

$$\begin{aligned} \bar{p}_m &= \frac{RT_1[\tau\{1+\ln(\alpha r_c/\rho)\} - (\alpha+\sigma+\ln r_c)+1]}{(v_4 - v_2)} \\ &= \frac{RT_1 r_c[\tau\{1+\ln(\alpha r_c/\rho)\} - (\alpha+\sigma+\ln r_c)+1]}{v_1(\sigma r_c - 1)} \end{aligned}$$

or in dimensionless form with respect to the pressure at state 1 as

$$\begin{aligned} \xi &= \bar{p}_m/p_1 \\ &= \frac{r_c[\tau\{1+\ln(\alpha r_c/\rho)\} - (\alpha+\sigma+\ln r_c)+1]}{(\sigma r_c - 1)} \end{aligned} \tag{2.9}$$

Now consider some special cases:

(a) *Ideal Stirling cycle*

Here

$$\beta = \sigma = 1$$

hence

$$\alpha = \rho = \tau$$

and

$$r_e = r_c = r$$

and since

$$T_1 \leq T_{2'} \leq T_{2'}(=T_3)$$

$$\begin{aligned} \eta &= \frac{(\gamma-1)[\tau\{1+\ln r\} - (\tau+1+\ln r)+1]}{(\tau-1) - \varepsilon(\tau-1) + \tau(\gamma-1)\ln r} \\ &= \frac{(\gamma-1)(\tau-1)\ln r}{(1-\varepsilon)(\tau-1) + \tau(\gamma-1)\ln r} \end{aligned} \tag{2.10}$$

and for $\varepsilon = 1$, $\eta = 1 - (1/\tau)$ the Carnot efficiency. Also:

$$\begin{aligned} \xi &= \frac{r[\tau(1+\ln r) - (\tau+1+\ln r)+1]}{(r-1)} \\ &= \frac{r(\tau-1)\ln r}{} \end{aligned} \tag{2.11}$$

(b) *Ideal Ericsson Cycle*

Here

$$\alpha = \rho = 1,$$

hence

$$\beta = \sigma = \tau$$

and

$$r_e = r_c = r$$

and since

$$T_2 (= T_1) \leqslant T_{2'} \leqslant T_3$$

$$\eta = \frac{(\gamma-1)[\tau\{1+\ln r\}-(1+\tau+\ln r)+1]}{\gamma(1-\varepsilon)(\tau-1)+\tau(\gamma-1)\ln r}$$

$$= \frac{(\gamma-1)(\tau-1)\ln r}{\gamma(1-\varepsilon)(\tau-1)+\tau(\gamma-1)\ln r} \qquad (2.12)$$

and for $\varepsilon = 1$, $\eta = 1-(1/\tau)$ the Carnot efficiency. Also

$$\xi = \frac{r[\tau\{1+\ln r\}-(1+\tau+\ln r)+1]}{(\tau r-1)}$$

$$= \frac{r(\tau-1)\ln r}{(\tau r-1)} \qquad (2.13)$$

(c) *Ideal constant-volume heating, constant-pressure cooling, cycle*

Here

$$\beta = \rho = 1,$$

hence

$$\alpha = \sigma = \tau$$

and

$$r_e = \alpha r_c = \tau r_c,$$

hence since

$$T_1 \leqslant T_{2'} \leqslant T_2 (= T_3)$$

$$\eta = \frac{(\gamma-1)[\tau\{1+\ln(\tau r_c)\}-(\tau+\tau+\ln r_c)+1]}{(\tau-1)-\varepsilon(\tau-1)+\tau(\gamma-1)\ln(\tau r_c)}$$

$$= \frac{(\gamma-1)[\tau\{\ln(\tau r_c)-1\}-\{\ln r_c-1\}]}{(1-\varepsilon)(\tau-1)+\tau(\gamma-1)\ln(\tau r_c)}$$

$$= \frac{(\gamma-1)[\tau(\ln\tau+\ln r_c)-\tau-\ln r_c+1]}{(1-\varepsilon)(\tau-1)+\tau(\gamma-1)(\ln\tau+\ln r_c)}$$

$$= \frac{(\gamma-1)[\tau(\ln\tau-1)+(\tau-1)\ln r_c+1]}{(1-\varepsilon)(\tau-1)+\tau(\gamma-1)(\ln\tau+\ln r_c)} \qquad (2.14)$$

and for $\varepsilon = 1$

$$\eta = 1-\frac{(\tau+\ln r_c-1)}{\tau\ln(\tau r_c)}$$

Also

$$\xi = \frac{r_c[\tau\{1+\ln\tau r_c)-(\tau+\tau+\ln r_c)+1]}{(\tau r_c-1)}$$

$$= \frac{r_c[\tau\{\ln(\tau r_c)-1\}-(\ln r_c-1)]}{(\tau r_c-1)} \qquad (2.15)$$

(d) *Ideal constant-pressure heating, constant-volume cooling, cycle*

Here

$$\alpha = \sigma = 1,$$

hence

$$\beta = \rho = \tau$$

and

$$r_e = r_c/\tau$$

hence since

$$T_2 (= T_1) \leqslant T_{2'} \leqslant T_3$$

$$\eta = \frac{(\gamma-1)[\tau\{1+\ln(r_c/\tau)\}-(1+1+\ln r_c)+1]}{\gamma(1-\varepsilon)(\tau-1)+\tau(\gamma-1)\ln(r_c/\tau)}$$

$$= \frac{(\gamma-1)[\tau\{1+\ln(r_c/\tau)\}-(1+\ln r_c)]}{\gamma(1-\varepsilon)(\tau-1)+\tau(\gamma-1)\ln(r_c/\tau)} \qquad (2.16)$$

and for $\varepsilon = 1$

$$\eta = 1-\frac{(1-\tau)+\ln r_c}{\tau\ln(r_c/\tau)}$$

Also

$$\xi = \frac{r_c[\tau\{1+\ln(r_c/\tau)\}-(1+\ln r_c)]}{(r_c-1)} \qquad (2.17)$$

The Rallis adiabatic cycle

Rallis (1976)† has also investigated another composite cycle in which the processes of compression and expansion are assumed to be adiabatic (without heat transfer) rather than the isothermal processes assumed

† Private communication.

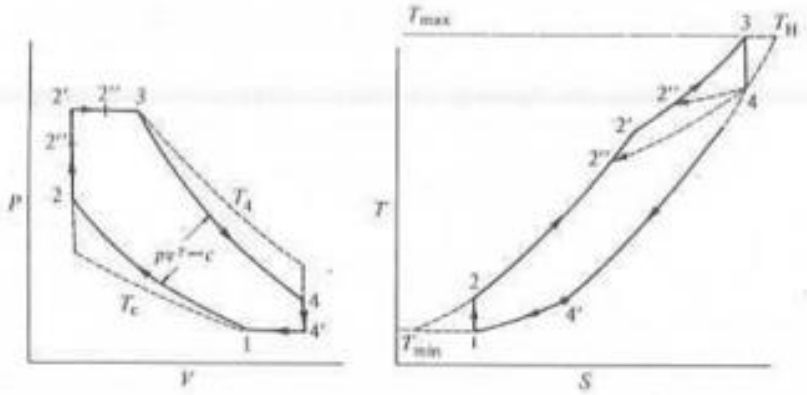


FIG. 2.9. The Rallis adiabatic regenerative cycle. In this cycle the processes of compression and expansion are adiabatic. The regenerative processes take place partly at constant-volume and partly constant-pressure conditions. The working fluid is heated regeneratively to temperature $T_{2'}$ and the cycle heat is supplied to increase the temperature to T_3.

above. The cycle is shown in Fig. 2.9. As before another whole family of special cases may be derived from the generalized case.

In practical engines it is difficult to achieve isothermal compression and expansion. Such processes usually turn out to be polytropic but in most cases can be adequately represented as adiabatic for the idealized type of analysis considered here.

The Rallis adiabatic cycle is very similar indeed to a composite cycle identified by Hutchinson (1955) as the Walker composite cycle. Professor Rallis has pointed out that Professor Walker, who originated this cycle, was in fact his predecessor at the School of Engineering, University of Witwatersrand, South Africa. Rallis attributes the development of the cycle to Professor Walker. However Walker's interest appears to have been limited to internal combustion engines and as such he did not consider regenerative processes. As these are fundamental, it appears necessary to distinguish between the adiabatic composite cycle with regenerative processes (Rallis) and without regenerative processes (Walker).

The following treatment was contributed by Rallis (1976).

For convenience in analysis we define the same non-dimensional parameters as used previously except that

$$\varepsilon = (T_{2'} - T_2)/(T_4 - T_2)$$

Note that this definition of regenerator effect is not the same as used previously.

Whilst

$$T_2 = r_c^{\gamma-1} T_1$$
$$T_{2'} = \alpha T_2 = \alpha r_c^{\gamma-1} T_1$$
$$T_3 = \beta T_{2'} = \alpha\beta r_c^{\gamma-1} T_1 = r_c^{\gamma-1} T_4 = \tau T_1$$
$$T_{4'} = \sigma T_1$$
$$T_4 = \rho T_{4'} = \sigma\rho T_1$$

hence

$$\alpha\beta r_c^{\gamma-1} = \sigma\rho r_c^{\gamma-1} = \tau \tag{2.18}$$

also

$$r_e = v_4/v_3 = (v_4/v_1)(v_1/v_2)(v_2/v_3) = \sigma r_c/\beta \tag{2.19}$$

and

$$T_{2'} = [\varepsilon\sigma\rho + (1-\varepsilon) r_c^{\gamma-1}] T_1 \tag{2.20}$$

The external work done per cycle here is

$$\begin{aligned} W &= p_3(v_3 - v_2) + c_v(T_3 - T_4) - p_1(v_4 - v_1) - c_v(T_2 - T_1) \\ &= RT_3 - \alpha RT_2 + c_v(T_3 - T_4) - \sigma RT_1 + RT_1 - c_v(T_2 - T_1) \\ &= R\left[T_3 - \alpha r_c^{\gamma-1} T_1 + \left(\frac{T_3 - \sigma\rho T_1}{\gamma - 1}\right) - \sigma T_1 + T_1 - \left(\frac{r_c^{\gamma-1} T_1 - T_1}{\gamma - 1}\right)\right] \\ &= RT_1\left[\tau - \alpha r_c^{\gamma-1} + \left(\frac{\tau - \sigma\rho}{\gamma - 1}\right) - \sigma + 1 - \left(\frac{r_c^{\gamma-1} - 1}{\gamma - 1}\right)\right] \\ &= c_v T_1[\gamma(\tau + 1) - \sigma(\rho + \gamma - 1) - \{\alpha(\gamma - 1) + 1\} r_c^{\gamma-1}] \end{aligned} \tag{2.21}$$

The heat supplied externally per cycle for $T_2 \leq T_{2'} \leq T_{2'}$ is given by:

$$\begin{aligned} Q_{ext} &= c_v(T_{2'} - T_{2'}) + c_p(T_3 - T_{2'}) \\ &= RT_1\left[\frac{\alpha r_c^{\gamma-1} - \varepsilon\sigma\rho - (1-\varepsilon) r_c^{\gamma-1}}{\gamma - 1} + \frac{\gamma(\tau - \alpha r_c^{\gamma-1})}{\gamma - 1}\right] \\ &= \frac{RT_1}{(\gamma - 1)}[(\alpha - 1 + \varepsilon - \alpha\gamma) r_c^{\gamma-1} - \varepsilon\sigma\rho + \gamma\tau] \\ &= c_v T_1[\gamma\tau - \varepsilon\sigma\rho - \{\alpha(\gamma - 1) + (1-\varepsilon)\} r_c^{\gamma-1}] \end{aligned} \tag{2.22}$$

Whilst for $T_{2'} \leq T_{2'} < T_3$

$$\begin{aligned} Q_{ext} &= c_p(T_3 - T_{2'}) \\ &= c_p T_1[\tau - \varepsilon\sigma\rho - (1-\varepsilon) r_c^{\gamma-1}] \end{aligned} \tag{2.23}$$

Thus the thermal efficiency for the case $T_2 \leqslant T_{2'} \leqslant T_{2'}$ is given by

$$\eta = \frac{\gamma(\tau+1) - \sigma(\rho+\gamma-1) - \{\alpha(\gamma-1)+1\}r_c^{\gamma-1}}{\gamma\tau - \varepsilon\sigma\rho - \{\alpha(\gamma-1)+(1-\varepsilon)\}r_c^{\gamma-1}} \tag{2.24}$$

whilst for $T_{2'} \leqslant T_{2'} < T_3$

$$\eta = \frac{\gamma(\tau+1) - \sigma(\rho+\gamma-1) - \{\alpha(\gamma-1)+1\}r_c^{\gamma-1}}{\gamma[\tau - \varepsilon\sigma\rho - (1-\varepsilon)r_c^{\gamma-1}]} \tag{2.25}$$

and the indicated mean effective pressure

$$\bar{p}_m = \frac{c_v T_1[\gamma(\tau+1) - \sigma(\rho+\gamma-1) - \{\alpha(\gamma-1)+1\}r_c^{\gamma-1}]}{(v_4 - v_2)}$$

$$= \frac{RT_1 r_c[\gamma(\tau+1) - \sigma(\rho+\gamma-1) - \{\alpha(\gamma-1)+1\}r_c^{\gamma-1}]}{v_1(\gamma-1)(\sigma r_c - 1)}$$

Hence

$$\xi = \frac{r_c[\gamma(\tau+1) - \sigma(\rho+\gamma-1) - \{\alpha(\gamma-1)+1\}r_c^{\gamma-1}]}{(\gamma-1)(\sigma r_c - 1)} \tag{2.26}$$

Evidently for regeneration to occur $T_{2'} \geqslant T_2$

$$\varepsilon\sigma\rho + (1-\varepsilon)r_c^{\gamma-1} \geqslant r_c^{\gamma-1}$$

or

$$r_{c\max} \leqslant (\sigma\rho)^{1/(\gamma-1)} \tag{2.27}$$

Now consider some special cases

(a) *Constant-volume cycle*

Here

$$\beta = \sigma = 1$$

and

$$r_e = r_c = r$$

hence

$$\alpha = \rho = \tau/r^{\gamma-1}$$

and since

$$T_2 \leqslant T_{2'} \leqslant T_2 (= T_3)$$

$$\eta = \frac{\gamma(\tau+1) - \{(\tau/r^{\gamma-1}) + \gamma - 1\} - [(\tau/r^{\gamma-1})(\gamma-1)+1]r^{\gamma-1}}{\gamma\tau - (\varepsilon\tau/r^{\gamma-1}) - \{(\tau/r^{\gamma-1})(\gamma-1)+(1-\varepsilon)\}r^{\gamma-1}}$$

$$= \frac{\tau[1-(1/r^{\gamma-1})] - (r^{\gamma-1}-1)}{\tau[1-(\varepsilon/r^{\gamma-1})] - (1-\varepsilon)r^{\gamma-1}} \tag{2.28}$$

At $\varepsilon = 0$ this reduces to the constant volume air standard cycle

$$\eta = 1 - (1/r^{\gamma-1})$$

whilst for $\varepsilon = 1$ it reduces to

$$\eta = 1 - (r^{\gamma-1}/\tau)$$

Now

$$\xi = \frac{r\{\tau[1-(1/r^{\gamma-1})] - (r^{\gamma-1}-1)\}}{(\gamma-1)(r-1)} \tag{2.29}$$

and

$$r_{\max} = [\tau/r_{\max}^{\gamma-1}]^{1/(\gamma-1)} = \tau^{1/(\gamma-1)}/r_{\max}$$

or

$$r_{\max} = \tau^{1/2(\gamma-1)} \tag{2.30}$$

(b) *Constant-pressure cycle*

Here

$$\alpha = \rho = 1$$

and

$$r_e = r_c = r$$

hence

$$\beta = \sigma = \tau/r^{\gamma-1}$$

and since

$$T_{2'} (= T_2) \leqslant T_{2'} < T_3$$

$$\eta = \frac{\gamma(\tau+1) - (\tau/r^{\gamma-1})\gamma - \gamma r^{\gamma-1}}{\gamma[\tau - (\varepsilon\tau/r^{\gamma-1}) - (1-\varepsilon)r^{\gamma-1}]}$$

$$= \frac{\tau[1-(1/r^{\gamma-1})] - (r^{\gamma-1}-1)}{\tau[1-(\varepsilon/r^{\gamma-1})] - (1-\varepsilon)r^{\gamma-1}} \tag{2.31}$$

which is the same as for the constant volume cycle, as are the values at $\varepsilon = 0$ and 1 and the value of $r_{\max}$.

However the specific indicated mean effective pressure here is given by

$$\xi = \frac{r[\gamma(\tau+1) - (\tau/r^{\gamma-1})\gamma - \gamma r^{\gamma-1}]}{(\gamma-1)[(\tau\eta/r^{\gamma-1}) - 1]}$$

$$= \frac{\gamma(1 - r^{\gamma-1}\tau)}{(\gamma-1)(\tau - r^{\gamma-2})} \tag{2.32}$$

(c) *Constant volume heating, constant pressure cooling cycle*

Here

$$\beta = \rho = 1,$$

hence

$$\alpha r_c^{\gamma-1} = \sigma r_e^{\gamma-1} = \tau$$

but

$$r_e = \sigma r_c,$$

then

$$\alpha = \tau / r_c^{\gamma-1}$$

and

$$\sigma = \tau / r_e^{\gamma-1} = (\tau / r_c^{\gamma-1})^{1/\gamma}$$

and since

$$T_3 \leqslant T_{2'} < T_2 (= T_3)$$

$$\eta = \frac{\gamma(\tau+1) - (\tau/r_c^{\gamma-1})^{1/\gamma}\gamma - \{(\tau/r_c^{\gamma-1})(\gamma-1)+1\}r_c^{\gamma-1}}{\gamma\tau - \varepsilon(\tau/r_c^{\gamma-1})^{1/\gamma} - \{(\tau/r_c^{\gamma-1})(\gamma-1)+(1-\varepsilon)\}r_c^{\gamma-1}}$$

$$= \frac{(\tau - r_c^{\gamma-1}) - \gamma[(\tau/r_c^{\gamma-1})^{1/\gamma} - 1]}{(\tau - r_c^{\gamma-1}) - \varepsilon[(\tau/r_c^{\gamma-1})^{1/\gamma} - r_c^{\gamma-1}]} \qquad (2.33)$$

At $\varepsilon = 0$ this reduces to:

$$\eta = 1 - \frac{\gamma[(\tau/r_c^{\gamma-1})^{1/\gamma} - 1]}{(\tau - r_c^{\gamma-1})}.$$

Whilst at $\varepsilon = 1$ it reduces to

$$\eta = \frac{(\tau - r_c^{\gamma-1}) - \gamma[(\tau/r_c^{\gamma-1})^{1/\gamma} - 1]}{(\tau - r_c^{\gamma-1}) - [(\tau/r_c^{\gamma-1})^{1/\gamma} - r_c^{\gamma-1}]}$$

Now

$$\xi = \frac{r_c[(\tau - r_c^{\gamma-1}) - \gamma\{(\tau/r_c^{\gamma-1})^{1/\gamma} - 1\}]}{(\gamma-1)[(\tau r_c)^{1/\gamma} - 1]} \qquad (2.34)$$

and

$$r_{c\max} = (\tau^{1/\gamma}/r_{c\max}^{(\gamma-1)/\gamma})^{1/(\gamma-1)} = \tau^{1/\gamma(\gamma-1)}/r_{c\max}^{1/\gamma}$$
$$= \tau^{1/(\gamma^2-1)} \qquad (2.35)$$

(d) *Constant-pressure heating, constant-volume cooling cycle*

Here

$$\alpha = \sigma = 1,$$

hence

$$\beta r_c^{\gamma-1} = \rho r_e^{\gamma-1} = \tau$$

but

$$r_e = r_c/\beta,$$

thus

$$\beta = \tau / r_c^{\gamma-1}$$

and

$$\rho = \tau / r_e^{\gamma-1} = \tau^\gamma / r_c^{\gamma(\gamma-1)}$$

and since here

$$T_2 (= T_2) \leqslant T_{2'} < T_3$$

$$\eta = \frac{\gamma(\tau+1) - \{(\tau^\gamma/r_c^{\gamma(\gamma-1)}) + \gamma - 1\} - \gamma r_c^{\gamma-1}}{\gamma[\tau - (\varepsilon\tau^\gamma/r_c^{\gamma(\gamma-1)}) - (1-\varepsilon)r_c^{\gamma-1}]}$$

$$= \frac{\gamma(\tau - r_c^{\gamma-1}) - \{(\tau^\gamma/r_c^{\gamma(\gamma-1)}) - 1\}}{\gamma[(\tau - r_c^{\gamma-1}) - \varepsilon\{(\tau^\gamma/r_c^{\gamma(\gamma-1)}) - r_c^{\gamma-1}\}]} \qquad (2.36)$$

At $\varepsilon = 0$ this reduces to:

$$\eta = 1 - \frac{(\tau/r_c^{\gamma-1})^\gamma - 1}{\gamma(\tau - r_c^{\gamma-1})}$$

whilst at $\varepsilon = 1$ it reduces to:

$$\eta = \frac{\gamma(\tau - r_c^{\gamma-1}) - \{(\tau/r_c^{\gamma-1})^\gamma - 1\}}{\gamma[(\tau - r_c^{\gamma-1}) - \{(\tau/r_c^{\gamma-1})^\gamma - r_c^{\gamma-1}\}]}$$

Now

$$\xi = \frac{r_c[\gamma(\tau - r_c^{\gamma-1}) - \{(\tau/r_c^{\gamma-1})^\gamma - 1\}]}{(\gamma-1)(r_c - 1)} \qquad (2.37)$$

and

$$r_{c\max} = \rho^{1/(\gamma-1)} = (\tau^\gamma/r_{c\max}^{\gamma(\gamma-1)})^{1/(\gamma-1)}$$
$$= \tau^{\gamma/(\gamma^2-1)} \qquad (2.38)$$

Note that in the foregoing, where we have used regenerator effectiveness rather than efficiency, in case (d) the maximum quantity of heat which can be rejected to the regenerator is $c_v(T_4 - T_2)$ whilst the quantity picked up by the gas is $c_v(T_{2'} - T_2)$. Thus the maximum value which the regenerator effectiveness can have is obtained by equating these two terms.

EFFECTS OF ADIABATIC PROCESSES

Using the above theory Rallis, Urieli, and Berchowitz (1977) compared ideal regenerative cycles having isothermal and adiabatic processes of compression and expansion. They limited their comparison to cycles with constant volume regenerative processes. The cycle with isothermal compression and expansion was the ideal Stirling cycle. The cycle with adiabatic compression and expansion they called the pseudo-Stirling cycle. For both cycles they assumed that, in some cases, the regenerative process was less than ideal and assigned arbitrary values to the regenerator effectiveness.

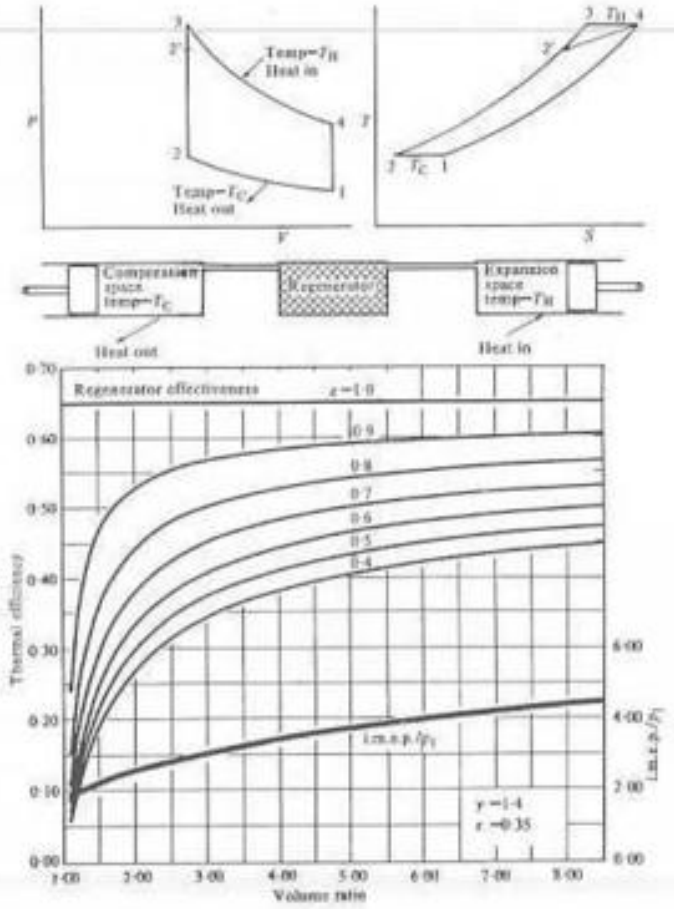


FIG. 2.10. Ideal Stirling engine (isothermal compression and expansion processes). Thermal efficiency and specific power as a function of volume ratio at different values of regenerator effectiveness (after Rallis *et al.* 1977).

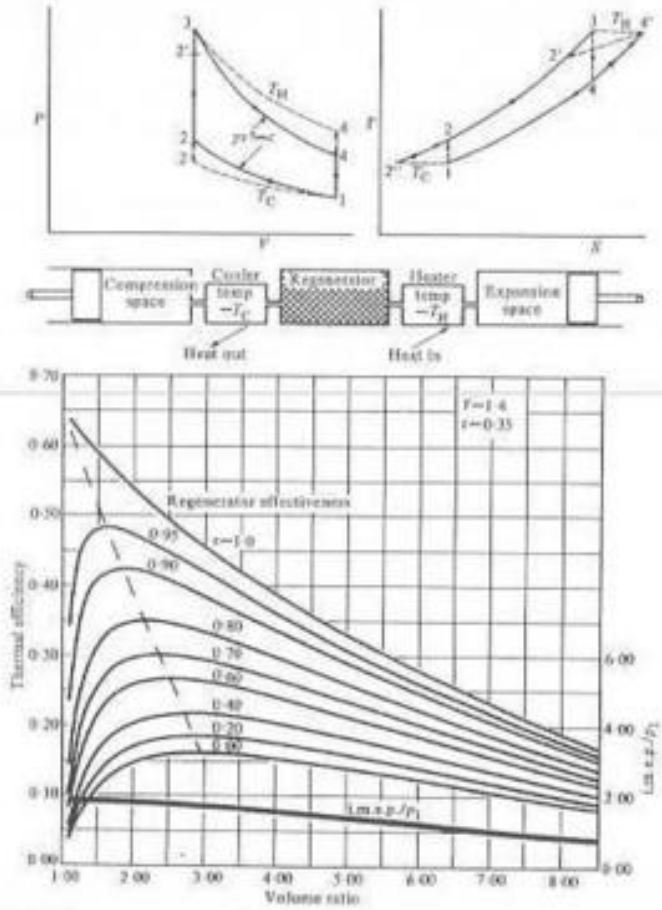


FIG. 2.11. Pseudo-Stirling cycle (adiabatic compression and expansion). Thermal efficiency and specific power as a function of volume ratio at different values of regenerator effectiveness (after Rallis *et al.* 1977).

For the purpose of comparison a temperature ratio $\tau = T_{min}/T_{max} = 0.35$ was assumed. For all cases the ratio of specific heats $\gamma = 1.4$ was used.

The results obtained for the two ideal cycles are reproduced in Fig. 2.10 (for the isothermal Stirling cycle) and Fig. 2.11 (for the adiabatic pseudo-Stirling cycle).

In each figure the thermal efficiency η and specific power,† (i.m.e.p.)/p_1 are given as a function of the volume ratio v_{max}/v_{min}. Comparison of these

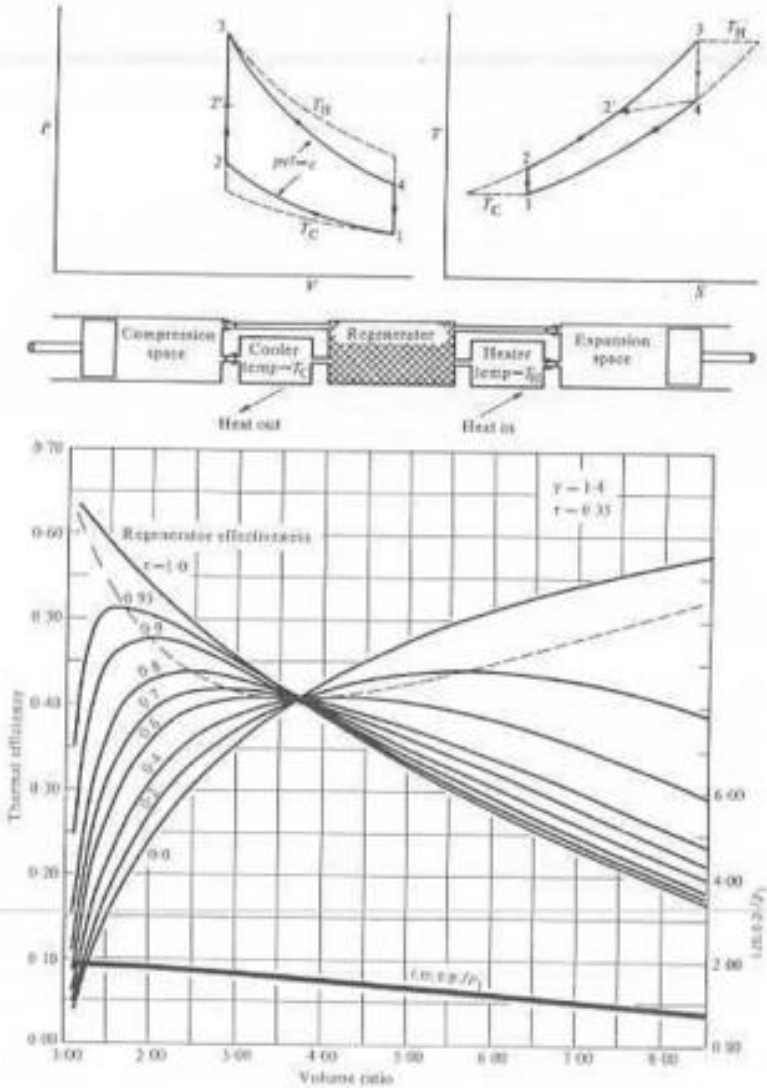


FIG. 2.12. Ported regenerative constant-volume cycle (adiabatic compression and expansion with bypass ports for cooler and heater). Thermal efficiency and specific power as a function of volume ratio at different values of regenerator effectiveness (after Rallis *et al.* 1977).

two figures illustrates in the clearest possible way the profoundly deleterious effect on efficiency and output that follows a departure from isothermality.

The ported constant-volume regenerative cycle

Rallis *et al.* (1977) extended their study to include a cycle in which alternative bypass ports were provided between the regenerator and the compression and expansion spaces.

The mechanical arrangement of such a system is shown in Fig. 2.12. Fluid *from* the compression space flowing *to* the regenerator bypasses the cooler. Similarly fluid flowing *from* the expansion space *to* the regenerator bypasses the heater. The system is applicable to the pseudo-Stirling cycle with adiabatic processes. As Rallis *et al.* remark

'... it seems intuitively wasteful to cool the working medium after heating by compression and to reheat it after cooling by expansion ...'

The ported engine was conceived by Finkelstein (1952) for engines with a gaseous working fluid. A similar scheme for flow regulation was used by Malone (1930) in his liquid engines. So far as is known however the study by Rallis *et al.* was the original theoretical investigation of the phenomena.

Using the same temperature ratio $\tau = 0.35$ and $\gamma = 1.4$ as before the results they obtained are presented in Fig. 2.12. The specific power is of course the same as for the pseudo-Stirling cycle but significant improvement in the thermal efficiency was gained over the whole spectrum of regenerator effectiveness values.

In their paper Rallis *et al.* (1977) provide further details of their study including conceptual diagrams for reduction to practice of the ported engine. It does appear to be a concept worth further investigation and hopefully experimental work in the area will be undertaken in the future.

3 PRACTICAL REGENERATIVE CYCLE

IDEAL CYCLE

THE Stirling cycle is a highly idealized thermodynamic cycle. It consists of four thermodynamic processes including two isothermal expansion and compression processes and two constant-volume processes.

This ideal cycle is discussed in detail in Chapter 2 along with a number of other alternative cycles all of which are as idealized as the Stirling cycle. Engines cannot be constructed to operate on these idealized cycles. Nevertheless they provide a model for comparison with the way in which practical engines may operate.

It was assumed in Chapter 2 that all the processes of the ideal cycles were thermodynamically reversible, i.e. that the fluid was everywhere at the same instantaneous equilibrium condition. It was assumed also that the processes of compression and expansion were isothermal or isentropic. The first assumption requires infinite rates of heat transfer between the cylinder walls and the working fluid. The latter assumption, of an isentropic process, requires zero heat transfer between the cylinder walls and the working fluid.

It was further assumed that the whole mass of working fluid in the cycle was, at any particular time, all in the compression space or the expansion space. The effects were neglected of any voids in the regenerative matrix, the cylinder clearance space, and any pockets in the cylinder or connecting ducts.

The two pistons were caused to move in some idealized discontinuous way to achieve the prescribed working fluid distribution. All aerodynamic and mechanical friction effects were ignored. Finally in most of the ideal cycles discussed in Chapter 2 regeneration was assumed to be perfect. This implies infinite rates of heat transfer between the working fluid and the regenerative matrix and, further, that the heat capacity of the regenerative matrix is infinite. Under such conditions the temperatures of the gas and the matrix are, at any point, the same and remain constant regardless of the direction of fluid flow.

PRACTICAL CYCLE

In any practical engine, all these factors and others combine to reduce the thermal efficiency to well below the Carnot value of the ideal cycle. The actual thermal efficiency may be quoted as a fraction of the theoretical Carnot efficiency; this ratio is called the relative efficiency,

$$\eta_r = \text{actual thermal efficiency}/\text{Carnot thermal efficiency}$$

A value in excess of 0.4 for the relative efficiency is evidence of a well-designed machine. The maximum achievable value is about 0.7.

To illustrate the discussion of the ideal cycle, a mechanical arrangement was assumed of two opposed pistons, with an interposed regenerator. The two-piston machine is one of several different mechanical arrangements which are to be considered in detail later. One practical version of a two-piston machine is shown in Fig. 3.1. It consists of a Vee engine, with both pistons coupled to a common crankshaft. The spaces above the pistons constitute the compression and expansion volumes; they are coupled by a duct, containing the regenerator and additional heat exchangers.

In the operation of this engine, a significant departure from ideality arises as a consequence of the continuous, rather than discontinuous, motion of the pistons. This results (as shown in Fig. 3.2) in a P–V

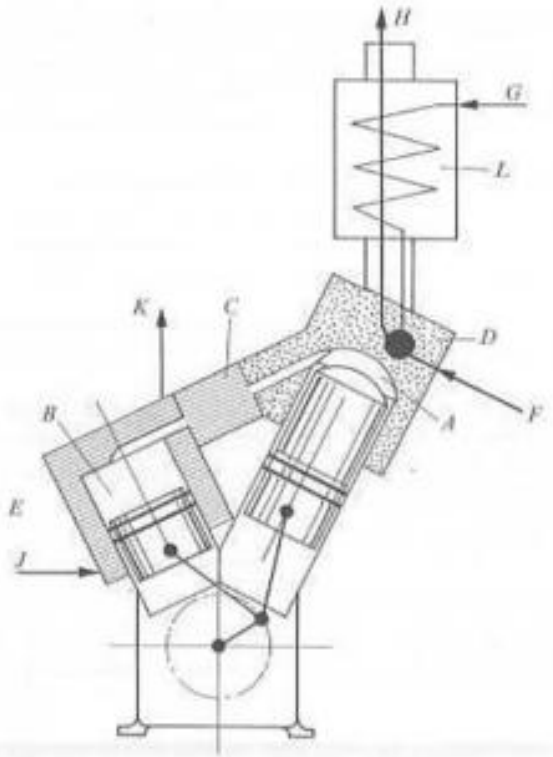


FIG. 3.1. Diagram of practical opposed-piston Stirling engine. *A*—expansion space, *B*—compression space, *C*—regenerator, *D*—heater, *E*—cooler, *F*—fuel inlet, *G*—air inlet, *H*—exhaust products of combustion, *J*—water inlet, *K*—water outlet, *L*—exhaust-gas inlet.

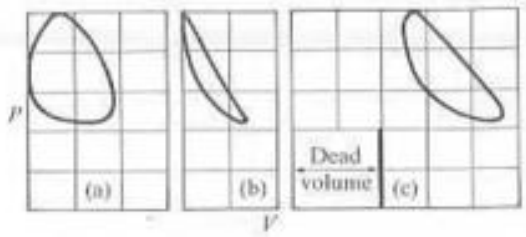


FIG. 3.2. Pressure—volume diagrams for practical engine. (a) Expansion-space diagram. (b) Compression-space diagram. (c) Total working-space.

diagram which is a smooth continuous envelope. The four processes of the ideal cycle are not sharply defined.

The processes of compression and expansion do not take place wholly in one or other of the two spaces, so that three P–V diagrams may be drawn, one for the compression space, one for the expansion space, and one for the total enclosed volume, which includes the 'dead' space. The 'dead' space is defined as that part of the working space not swept by one of the pistons, and includes cylinder clearance spaces, void volumes of the regenerator and other heat exchangers, and the internal volume of associated ducts and ports. The P–V diagram for the expansion space represents the total positive work of the cycle, whereas the diagram for the compression space represents the compression (or negative) work of the cycle. The difference in the areas of these diagrams is the net cycle output, the 'indicated' work available for overcoming mechanical-friction losses and for providing useful power to the engine crankshaft.

In a cycle where the processes of compression and expansion are isothermal and there are no friction losses, the difference in the area of the expansion- and compression-space diagrams will be found to be exactly equal to the area of the P–V diagram for the total working space. In a practical engine, of course, this equality does not obtain, because aerodynamic-flow losses in the regenerator and other heat-exchangers cause differences in the pressure of the working fluid in the compression and expansion spaces. Flow losses are important, because (as shown in Fig. 3.3), they cause a decrease in the area of the expansion-space P–V diagram, resulting in (a) a decrease in the net cycle output (and, hence, in efficiency) of a prime mover and (b) a decrease in the cooling capacity and the COP of a refrigerating machine.

The sinusoidal piston motion results in the working fluid being distributed in a cyclically time-variant manner throughout various temperature ranges, and it is not possible to draw a meaningful T–S diagram for the total mass of the working fluid. It is possible to draw T–S diagrams for

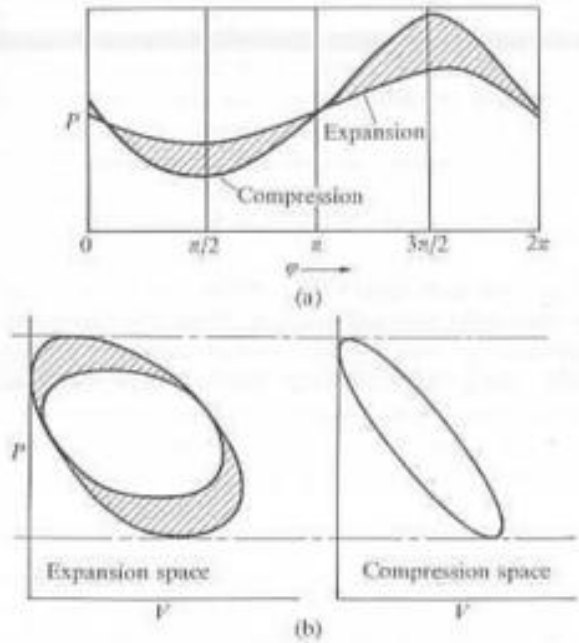


FIG. 3.3. Effect of aerodynamic-flow loss on engine work. (a) Pressure—time diagram for pressure variation in the expansion and compression spaces. The difference in pressure is the flow loss in the regenerator and other heat exchangers. (b) Pressure—volume diagrams for the compression and expansion spaces. The hatched area on the diagram for the expansion space represents the work effectively lost by flow losses in the regenerator and other heat exchangers.

ture range to another, but no convenient way has been found to combine these multiple diagrams.

The processes of compression and expansion are not isothermal, another major departure from ideality. In an engine, running at a reasonable speed (say, 1000 rev/min), it is likely that the processes are nearer adiabatic (no heat-transfer) than isothermal (infinite heat-transfer). In order to improve the situation special heat exchangers are often incorporated (as shown in Fig. 3.1), including (a) a heater, adjacent to the expansion space, imparting heat *to* the working fluid, and (b) a cooler, adjacent to the compression space, abstracting heat *from* the working fluid. Despite the advantages of improved heat transfer, the provision of such heat exchangers imposes some penalties. Additional aerodynamic-flow losses are likely, with the consequent deleterious effect on performance, as discussed above. The dead volume will be increased by the void volume of the heater and cooler, and this has a critical effect on the

heated, not only when flowing from the regenerator *to* the expansion space, but also when flowing *from* the expansion space to the regenerator. Similarly, the working fluid is cooled when flowing from, as well as to, the compression space. The provision of one-way flow systems is possible, but adds much complication to the machine.

Considerations of increased flow loss and void volume (along with considerations of cost, size, and weight) combine to produce a compromise heat exchanger design. Consequently, substantial differences may exist between the temperatures at which heating (combustion products) and cooling (water or air) is available and the temperatures experienced by the working fluid. This is illustrated diagrammatically in Fig. 3.4, and

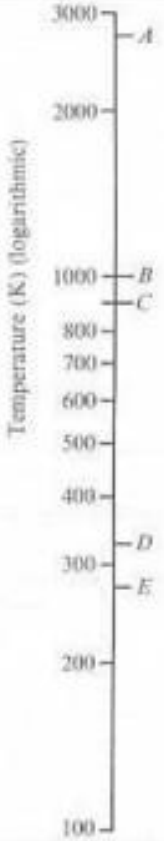


FIG. 3.4. Characteristic temperature regime in a fossil-fuelled, water-cooled Stirling engine. *A*—Temperature of combustion products, *B*—Temperature of heater walls, *C*—Mean temperature in expansion space, *D*—Mean temperature in compression space, *E*—Temperature of cooling water and cooler walls.

might be considered representative of the temperature range established in a fossil-fuelled water-cooled regenerative engine. The temperatures of the combustion products and cooling water are 2800 K (5040 °R) and 280 K (504 °R), respectively. The metallurgical limit of the materials used for the expansion cylinder and heater is (say), 1000 K (1800 °R). This provides for a steep temperature gradient of 2800 to 1000 K (5040 to 1800 °R) between the combustion products and cylinder wall, with the potential for high rates of heat transfer. Further temperature gradients of (say), 100 K (180 °R) between the working fluid and the expansion space, and 50 K, between the working fluid and the compression space, might exist, so that the cyclic temperature excursion of the working fluid varies from $(280+50)=330$ K (594 °R) to $(1000-100)=900$ K (1620 °R). Whereas the Carnot- (or Stirling-) cycle efficiency for the system *might* be calculated as

$$\eta_c = (2800-280)/2800 = 2520/2800 = 90 \text{ per cent,}$$

to give a more realistic picture it *should* be calculated as

$$\eta_c = (900-330)/900 = 570/900 = 63 \text{ per cent.}$$

This example demonstrates one of the major difficulties in the commercial application of Stirling engines—one shared by the gas turbine—the question of materials. Some parts of the machine (the heater and expansion space), are exposed, continuously, to a high temperature, and are subject, therefore, to the metallurgical limit of the heater and expansion cylinder materials.

The allowable temperature excursion of the working fluid in a Stirling engine is limited to a fraction of that permissible in an internal-combustion engine using an Otto or Diesel cycle, where the maximum cycle temperatures are attained only momentarily. Thus, although regenerative cycles between given temperature limits are thermodynamically more efficient than Otto or Diesel cycles, in practice regenerative engines are compared with gas (or oil) engines operating with radically different temperature limits.

Not all the heat available from combustion of the fuel and air can be transferred to the working fluid, since this would require a very large heater. The heat passing to exhaust in the combustion products of a Stirling engine represents a direct loss, because it must be paid for in terms of gallons of oil (or cubic feet of gas burned), but has served no useful purpose in the engine. An important engine accessory, therefore, is another heat exchanger (the exhaust/air preheater), used to warm the incoming air by heat transferred from the exhaust gas. This heat exchanger can be of the recuperative type or the regenerative type. In the recuperative type, the two fluids, exhaust gas and incoming air, are

separated by walls into separate ducts. In the regenerative type, the fluids flow alternately, and usually in contraflow, through the same porous matrix. It is important to distinguish carefully between the regenerative heat/exchanger, incorporated as an integral part of the engine, and the recuperative (or regenerative) heat exchanger, used as an accessory of the engine for exhaust/air preheating.

The continuous motion of the reciprocating elements, the non-isothermal compression and expansion processes, the limited heat transfer in cooling and heating devices, the exhaust-stack loss, the increased dead space, and aerodynamic-flow loss together constitute the principal reasons for the failure of most practical Stirling engines to fulfil their designer's hopes and ambitions. Other causes of disappointment include deficiencies in regenerator operation, high mechanical-friction losses, temperature equalization as a result of relatively massive conduction paths, and fluid leakage owing to imperfectly designed (or imperfectly operating) seals.

4 THEORETICAL ANALYSIS OF STIRLING ENGINES

INTRODUCTION

THEORETICAL analyses of Stirling engines have been developed with varying degrees of sophistication.

The most simple analysis is that for the ideal Stirling cycle, where the thermodynamic cycle comprises two isothermal and two constant-volume regenerative processes. However this involves such gross idealization of the process occurring in an actual Stirling engine as to be suitable only for the most elementary, preliminary design calculations.

A more realistic analysis was devised by Gustav Schmidt (1871). This has become the classical analysis of the cycle and is believed to give a more reasonable approximation of engine performance. Nevertheless, the analysis remains very highly idealized, so that in practice the indicated performance of an engine will likely be no better than 60 per cent of the predicted Schmidt cycle performance and, often, a good deal less.

The reason for this is, primarily, that the Schmidt cycle presumes the processes of compression and expansion to be isothermal. In practical engines running at 1000 or more revolutions per minute, this is not the case. The processes of compression and expansion in the engine cylinders are more nearly adiabatic than isothermal. This innocent sounding difference wreaks a profound redistribution of the cyclic mass variation of the working fluid and, consequently, on the performance of the engine.

Following the introduction of the Schmidt analysis in 1871, nearly a century was to elapse before Finkelstein (1960a) devised a generalized analysis permitting the theoretical investigation of engines with processes of compression and expansion other than isothermal. The Schmidt cycle with isothermal compression and expansion processes then became a special case of the generalized Finkelstein analysis. Another special case is the cycle with adiabatic processes in the engine cylinder, called here the Finkelstein adiabatic cycle. With the generalized theory other cases of *limited* heat transfer in the engine cylinders may also be considered.

Further developments of the adiabatic model to account for secondary effects, primarily aerodynamic-flow losses, and thermal effects are possible. These make the adiabatic model the preferred level of analysis for routine performance predictions. The analysis is sufficiently complex to require the use of digital computing machines, but only to a moderate extent, involving little computer time and reasonable expenditures.

program now installed on a commercial computer network and available for general use on payment of a royalty fee. Later in the closing years of the 1970s substantial efforts were devoted to Stirling engine simulation by other workers and a variety of advanced engine simulation programs have become available.

A comprehensive discussion of the generalized Finkelstein analysis and the subsequent nodal analysis programs is beyond the scope of this present work. Therefore in this chapter we shall attempt:

(a) to summarize equations for the ideal Stirling cycle,
(b) to present in reasonably comprehensive detail the Schmidt isothermal analysis,
(c) to outline the Finkelstein adiabatic analysis,
(d) to compare and comment briefly on the advanced level analyses for Stirling engines making reference to the source documents for those who wish to investigate further.

IDEAL STIRLING CYCLE

Equations for analysis of the ideal Stirling cycle are summarized below. Again the reader is advised that the gross idealization of the Stirling cycle precludes the use of these equations for anything other than the most elementary investigations.

The principal deficiency of the ideal cycle is that it assumes *all* the working fluid is, instantaneously, at the same condition in either the compression or expansion spaces. This implies that the internal void volume of the regenerator is zero and requires the pistons, or other reciprocating elements, to move with a discontinuous motion. The processes of expansion and compression are assumed to be isothermal and the effects of imperfect regeneration, and aerodynamic pressure drops are not considered.

A more complete treatment of the ideal cycle, contributed by Rallis, is reproduced in Chapter 2. With reference to Fig. 2.3 and the discussion of the ideal cycle given in Chapter 2 the equations for the ideal cycle may be summarized as follows:

Required data

(i) Some reference temperature and pressure, or volume, say, conditions, at state 1.
(ii) Temperature ratio $\tau = T_{min}/T_{max}$.
(iii) Volume ratio $r = V_{max}/V_{min}$.

For unit mass of working fluid, assumed to be a perfect gas, then $V_1 = RT_1/p_1$ from the characteristic gas equation.

Process parameters

(a) *Isothermal compression process* (1—2).

In this process, heat is abstracted from the working fluid and rejected from the cycle at the minimum cycle temperature. Work is done on the working fluid equal in magnitude to the heat rejected from the cycle. There is no change in internal energy, and there is a decrease in entropy.

$$p_2 = p_1 V_1/V_2 = p_1 r,\ T_2 = T_1 = T_{min}$$

Heat transfer (Q) = work done $(W) = P_1 V_1 \ln(1/r) = RT_1 \ln(1/r)$.
Change in entropy $(S_2 - S_1) = R \ln(1/r)$.

(b) *Constant-volume regenerative transfer process* (2—3).

In this process heat is transferred from the regenerative matrix to the working fluid, increasing the temperature from T_{min} to T_{max}. No work is done, and there is an increase in the entropy, and internal energy, of the working fluid.

$$p_3 = p_2 T_3/T_2 = p_2/\tau,\ V_3 = V_2.$$

Heat transfer $(Q) = C_v(T_3 - T_2)$.
Work done $(W) = 0$.
Change in entropy $(S_3 - S_2) = C_v \ln(1/\tau)$

(c) *Isothermal expansion process* (3–4).

In this process, heat is supplied to the cycle at a high temperature T_{max}, during expansion of the working fluid. Work is done, by the working fluid, equal in magnitude to the heat supplied. There is no change in the internal energy, but an increase in the entropy of the working fluid.

$$p_4 = p_3 V_3/V_4 = p_3(1/r);\ T_4 = T_3 = T_{max}.$$

Heat transfer (Q) = work done $(W) = p_3 V_3 \ln r = RT_3 \ln r$.
Change in entropy $(S_4 - S_3) = R \ln r$.

(d) *Constant-volume regenerative transfer process* (4—1).

In this process, heat is transferred from the working fluid to the regenerative matrix, decreasing the temperature of the working fluid from T_{max} to T_{min}. No work is done, and there is a decrease in the internal energy and entropy of the working fluid.

$$p_1 = p_4 T_4/T_1 = p_1 \tau,\ V_1 = V_4$$

Heat transfer $(Q) = C_v(T_1 - T_4)$.
Change in entropy $(S_1 - S_4) = C_v \ln \tau$

In the regenerative processes, the heat transferred from the matrix to the working fluid in process 2–3 is restored from the working fluid to the matrix in process 4–1. There is no net gain or loss of heat by the working fluid or the matrix. Therefore

the total heat supplied (at T_{max}) $= RT_3 \ln r$,
the total heat rejected (at T_{min}) $= RT_1 \ln (1/r)$.

$$\text{and thermal efficiency} = \frac{\text{heat supplied} - \text{heat rejected}}{\text{heat supplied}}$$

$$= \frac{\text{work done}}{\text{heat supplied}}$$

$$= (RT_3 \ln r - RT_1 \ln r)/RT_3 \ln r$$

$$= 1 - \tau.$$

This value corresponds to the Carnot efficiency between the same temperature limits.

THE SCHMIDT CYCLE

The classical analysis of the operation of Stirling engines is due to Schmidt (1871). The theory provides for harmonic motion of the reciprocating elements, but retains the major assumptions of isothermal compression and expansion and of perfect regeneration. It, thus, remains highly idealized, but is certainly more realistic than the ideal Stirling cycle. Provided a reasonable level of caution is exercised in interpretation, the predictions of the Schmidt theory can be a useful tool for engine design.

Principal assumptions of the Schmidt cycle

1. The regenerative process is perfect.
2. The instantaneous pressure is the same throughout the system.
3. The working fluid obeys the characteristic gas equation, $PV = RT$.
4. There is no leakage, and the mass of working fluid remains constant.
5. The volume variations in the working space occur sinusoidally.
6. There are no temperature gradients in the heat exchangers.
7. The cylinder-wall and piston temperatures are constant.
8. There is perfect mixing of the cylinder contents.
9. The temperature of the working fluid in the ancillary spaces is constant.
10. The speed of the machine is constant.

Nomenclature used in the following analysis†

A	= a factor $(\tau^2 + 2\tau\kappa \cos\alpha + \kappa^2)^{\frac{1}{2}}$.
B	= a factor $(\tau + \kappa + 2S)$.
K	= constant.
M	= total mass of working fluid.
N	= machine speed.
p	= instantaneous cycle-pressure.
p_{max}	= maximum cycle-pressure.
p_{mean}	= mean cycle-pressure.
p_{min}	= minimum cycle-pressure.
P	= engine output per cycle.
P_{mass}	= P/MRT_c dimensionless power parameter based on the unit mass of working fluid.
P_{max}	= $P/(p_{max} V_T)$, dimensionless power parameter, based on the maximum cycle-pressure and combined swept volume per cycle.
Q	= heat transferred to the working fluid in the expansion space, the heat lifted.
Q_{mass}	= Q/MRT_c the dimensionless cooling parameter, based on the unit mass of working fluid.
Q_{max}	= $Q/(p_{max} V_T)$, the dimensionless heat lifted, based on the maximum cycle pressure.
R	= characteristic gas constant of the working fluid.
S	= $(2X\tau)/(\tau + 1)$, reduced dead volume.
T_C	= temperature of the working fluid in the compression space, generally assumed to be 300 K.
T_D	= temperature of the working fluid in the dead space.
T_E	= temperature of the working fluid in the expansion space.
V_C	= swept volume in the compression space.
V_E	= swept volume in the expansion space.
V_D	= total internal volume of heat exchangers, volume of regenerator, and associated ducts and ports.
V_T	= $(V_C + V_E) = (1 + \kappa) V_E$, combined swept volume.
V_W	= $\frac{1}{2} V_E (1 + \cos\phi) + \frac{1}{2} V_C [1 + \cos(\phi - \alpha)] + V_D$, volume of total working space.
V_{Wmax}	= maximum volume of total working space.
X	= V_D/V_E, dead-volume ratio.

† *Note*: Lower case suffixes indicate instantaneous values of temperature, pressure volume, and mass. Upper case suffices indicate maximum (or constant) values. Thus: E or e refers to expansion space.

α = angle by which volume variations in the expansion space lead those in the compression space (in fractions of π radians, or degrees).
$\delta = (\tau^2+\kappa^2+2\tau\kappa\cos\alpha)^{\frac{1}{2}}/(\tau+\kappa+2S)$.
$\theta = \tan^{-1}((\kappa\sin\alpha)/(\tau+\kappa\cos\alpha))$.
$\kappa = V_C/V_E$, swept-volume ratio.
$\tau = T_C/T_E$, temperature ratio.
ϕ = crank angle.

Basic equations

Volume of expansion space $V_e = \frac{1}{2}V_E(1+\cos\phi)$. (4.1)

Volume of compression space $V_c = \frac{1}{2}V_C[1+\cos(\phi-\alpha)]$ (4.2)

$= \frac{1}{2}\kappa V_E[1+\cos(\phi-\alpha)]$. (4.3)

Volume of dead space, being that constant volume of the total working space not included in the volumes of the expansion or compression space,

$$V_D = XV_E. \tag{4.4}$$

Mass of working fluid in expansion space, $M_e = (p_e V_e)/(RT_e)$.
Mass of working fluid in compression space $M_c = (p_c V_c)/(RT_c)$.
Mass of working fluid in dead space $M_d = (p_d V_d)/(RT_d)$.
Since the total mass of the working fluid remains constant,

$$M_T = (p_e V_e)/(RT_e)+(p_c V_c)/(RT_c)+(p_d V_d)/(RT_d) = (KV_E)/(2RT_c). \tag{4.5}$$

If the instantaneous pressure is the same throughout the system, and equal to p, say, and if T_e and T_c are constant at T_E and T_C then, substituting for the volumes, eliminating R, and re-arranging,

$$K/p = (T_C/T_E)(1+\cos\phi)+\kappa[1+\cos(\phi-\alpha)]+(2V_D T_C)/(V_E T_D). \tag{4.6}$$

If the temperature variation in the dead space is linear in the axial direction, then the mean temperature

$$T_D = T_C+\tfrac{1}{2}(T_E-T_C) = (1+T_E/T_C)(T_C/2)$$

and, since $T_C/T_E = \tau$ then, from eqn (4.6)

$$K/p = \tau(1+\cos\phi)+\kappa[1+\cos(\phi-\alpha)]+2S \tag{4.7}$$

where S (the reduced dead volume) $= 2X\tau/(\tau+1)$.
To simplify eqn (4.7) consider first

$$x = x\cos\phi + z\sin\phi$$

then

$$y = \sqrt{r^2}\cos(\phi-\beta),$$

where

$$\tan\beta = z/x,\ z = r\sin\beta,$$

and

$$x = r\cos\beta,$$

since

$$\sqrt{r^2}\cos(\phi-\beta) = \sqrt{r^2}(\cos\phi\cos\beta+\sin\phi\sin\beta)$$
$$= r\cos\phi\cos\beta + r\sin\phi\sin\beta$$
$$= x\cos\phi + z\sin\phi.$$

Eqn (4.8) is similar in form to eqn (4.7) therefore, by analogy,

$$K/p = [(\tau+\kappa\cos\alpha)^2+(\kappa\sin\alpha)^2]^{\frac{1}{2}}\cos(\phi-\theta)+\tau+\kappa+2S$$
$$= (\tau^2+2\tau\kappa\cos\alpha+\kappa^2)^{\frac{1}{2}}\cos(\phi-\theta)+\tau+\kappa+2S, \tag{4.9}$$

where $\tan\theta = (\kappa\sin\alpha)/(\tau+\kappa\cos\alpha)$.
Let $A = (\tau^2+2\tau\kappa\cos\alpha+\kappa^2)^{\frac{1}{2}}$, $B = \tau+\kappa+2S$, and $\delta = A/B$, then

$$K/p = A\cos(\phi-\theta)+B$$

and

$$p = K/\{B[1+\delta\cos(\phi-\theta)]\}.$$

The instantaneous pressure p is
(a) a minimum, when $\phi = \theta$, i.e. $(\phi-\theta) = 0$,
(b) a maximum, when $\phi = (\theta+\pi)$, i.e. $(\phi-\theta) = \pi$,
therefore, $p_{min} = K/[B(1+\delta)]$, and $p_{max} = K/[B(1-\delta)]$.
Thus,

$$p = p_{max}(1-\delta)/[1+\delta\cos(\phi-\theta)] \tag{4.10a}$$
$$= p_{min}(1+\delta)/[1+\delta\cos(\phi-\theta)], \tag{4.10b}$$

and the pressure ratio

$$p_r = p_{max}/p_{min} = (1+\delta)/(1-\delta). \tag{4.11}$$

Mean cycle pressure

The mean cycle pressure is given by

$$p_{mean} = (1/2\pi)\int_0^{2\pi} p\,\mathrm{d}(\phi-\theta)$$

$$= (1/2\pi)\int_0^{2\pi}$$

which can be resolved to

$$p_{mean} = p_{max}[(1-\delta)/(1+\delta)]^{\frac{1}{2}}. \tag{4.13}$$

Heat transferred and work done

Since the processes of expansion and compression take place isothermally the heat transferred Q is equal to the work done P, therefore

$$Q = P = \int p \, dV.$$

If $V = \frac{1}{2}V_E(1+\cos\phi)$,

$$dV = -\tfrac{1}{2}V_E \sin\phi \, d\phi \tag{4.14}$$

and, if

$$p = p_{mean}[1 - \Delta\cos(\phi-\theta)], \text{ approximately,} \tag{4.15}$$

where

then $$\Delta = 2\delta/[1+(1-\delta^2)^{\frac{1}{2}}],$$

$$\begin{aligned} Q &= -\tfrac{1}{2}\int_0^{2\pi} \{p_{mean}V_E[1-\Delta\cos(\phi-\theta)]\sin\phi\}\,d\phi \\ &= -\tfrac{1}{2}p_{mean}V_E\int_0^{2\pi}[\sin\phi - \Delta(\cos\phi\cos\theta\sin\phi + \sin\theta\sin^2\phi)]\,d\phi \\ &= -\tfrac{1}{2}p_{mean}V_E[-\cos\phi - \Delta[-\cos\theta\cdot\tfrac{1}{2}\cos 2\phi + \sin\theta(\tfrac{1}{2}\phi - \tfrac{1}{4}\sin^2\phi)]]_0^{2\pi} \\ &= -\tfrac{1}{2}p_{mean}V_E\left[-\Delta\sin\theta\frac{\phi}{2}\right]_0^{2\pi} \\ &= +\tfrac{1}{2}\pi p_{mean}V_E\Delta\sin\theta. \end{aligned} \tag{4.16}$$

Expansion space

The variation in volume in the expansion space follows the equation

$$V_e = \tfrac{1}{2}V_E(1+\cos\phi),$$

which conforms to the required equation (eqn (4.14)), therefore the heat transferred in the expansion space, from eqn (4.16), is given by

$$Q = \pi p_{mean}V_E\,\delta\sin\theta/[1+(1-\delta^2)^{\frac{1}{2}}]. \tag{4.17}$$

There is no need to invoke the approximate expression for p in eqn (4.15). Instead, eqn (4.10b) is a suitable starting point, and can be combined with eqn (4.14) to calculate the heat transfer per cycle, Q, as follows:†

$$Q = \oint p\,dV = \oint \frac{-p_{min}(1+\delta)}{1+\delta\cos(\phi-\theta)}\cdot\tfrac{1}{2}V_E\sin\phi\,d\phi = \frac{-V_E p_{min}(1+\delta)}{2}I,$$

where

$$I = \oint \frac{\sin\phi\,d\phi}{1+\delta\cos(\phi-\theta)}$$

Let $\phi-\theta=\psi$, $\tan\frac{\psi}{2}=t$; then $d\phi = d\psi = 2\,dt/(1+t^2)$ and

$$\delta I = \oint\left\{\frac{\delta\sin\psi\cos\theta}{1+\delta\cos\psi} + \frac{\delta\cos\psi\sin\theta+\sin\theta}{1+\delta\cos\psi} - \frac{\sin\theta}{1+\delta\cos\psi}\right\}d\psi \tag{4.18}$$

$$\begin{aligned} &= [-\cos\theta\log(1+\delta\cos\psi)]_{\psi=0}^{2\pi} + \sin\theta[\psi]_{\psi=0}^{2\pi} - \sin\theta\oint\left\{\frac{2\,dt}{1+t^2+\delta(1-t^2)}\right\} \\ &= 2\pi\sin\theta - 2\sin\theta\oint\frac{dt}{(1+\delta)+(1-\delta)t^2} \\ &= 2\pi\sin\theta - \frac{2\sin\theta}{(1-\delta)}\left(\frac{1-\delta}{1+\delta}\right)^{\frac{1}{2}}\oint\frac{[(1+\delta)^{\frac{1}{2}}/(1-\delta)^{\frac{1}{2}}]\,dt}{(1+\delta)/(1-\delta)+t^2} \\ &= 2\pi\sin\theta - \frac{2\sin\theta}{(1-\delta^2)^{\frac{1}{2}}}[\tan^{-1}(t(1-\delta)^{\frac{1}{2}}/(1+\delta)^{\frac{1}{2}})]_{\psi=0}^{2\pi} \end{aligned}$$

In evaluating the definite integral note that $t=0$ at both limits; $\tan^{-1}(0)=0$, π etc. It is clear from eqn (4.18), if $\delta\to 0$, that the cyclic integral must take the value $-2\pi\sin\theta$ for its third term. Thus

$$\begin{aligned} \delta I &= 2\pi\sin\theta - 2\pi\sin\theta/(1-\delta^2)^{\frac{1}{2}} \\ &= 2\pi\sin\theta[1-1/(1-\delta^2)^{\frac{1}{2}}] \end{aligned}$$

and

$$Q = \frac{-V_E p_{min}(1+\delta)}{2}\cdot\frac{2\pi\sin\theta}{\delta}[1-1/(1-\delta^2)^{\frac{1}{2}}]$$

Substituting for p_{min} gives

$$Q = -V_E p_{mean}\pi\sin\theta[(1-\delta^2)^{\frac{1}{2}}-1]/\delta$$

When the numerator and denominator are both multiplied by $[(1-\delta^2)^{\frac{1}{2}}+1]$, this proves to be identical to eqn (4.17).

Compression space

The variation in volume of the compression space follows the equation

$$V_c = \tfrac{1}{2}\kappa V_E[1+\cos(\phi-\alpha)]$$

and, by a process similar to that above, we can obtain expressions for the pressure and volume in the required form, so that heat transferred in the compression space is given by

Dividing eqn (4.19) by eqn (4.17),

$$Q_c/Q = [\kappa \sin(\theta-\alpha)]/\sin\theta$$
$$= \kappa(\sin\theta\cos\alpha - \cos\theta\sin\alpha)/\sin\theta$$
$$= \kappa(\cos\alpha - \sin\alpha/\tan\theta),$$

but $\tan\theta = \kappa\sin\alpha/(\tau+\kappa\cos\alpha)$ and, therefore, $Q_c/Q = -\tau$.

The heat transferred in the expansion space is of opposite sign to the heat transferred in the compression space, and is numerically different by the temperature ratio τ. By analogy, the work done in the two spaces has the same relationship, $P_C = -\tau P_E$, and the net power is $P = P_E + P_C = (1-\tau)Q$.

In the case of the machine acting as a prime mover $T_E > T_C$, i.e. $\tau < 1$, and the thermal efficiency

$$\eta = (\text{heat supplied—heat rejected})/(\text{heat supplied})$$
$$= (Q-\tau Q)/Q = 1-\tau = (T_E - T_C)/T_E.$$

This corresponds to the Carnot efficiency.

When the machine acts as a refrigerator, $T_C > T_E$, i.e. $\tau > 1$, and

$$\text{the coefficient of performance} = \text{heat lifted/work done}$$
$$= Q/(Q-Q_c) = 1/(1-\tau)$$
$$= T_E/(T_E - T_C).$$

For a heat pump, $T_c > T_E$, i.e. $\tau > 1$, and the

$$\text{coefficient of performance} = \text{heat rejected/work done}$$
$$= Q_c/(Q-Q_c) = \tau/(1-\tau) = T_C/(T_E - T_C).$$

This corresponds to the inverse thermal efficiency.

Mass distribution in the machine

From the characteristic gas equation,

$$M = pV/RT,$$

where $p = p_{mean}(1-\delta^2)^{\frac{1}{2}}/[1+\delta\cos(\phi-\theta)]$.

(a) *Expansion space* $V_e = \frac{1}{2}V_E(1+\cos\phi)$.

The instantaneous mass of working fluid in the expansion space is given by

$$M_e = \tfrac{1}{2}V_E p_{mean}(1-\delta^2)^{\frac{1}{2}}(1+\cos\phi)/[RT_E(1+\delta\cos(\phi-\theta)]. \quad (4.20)$$

The rate of change of mass of working fluid in the expansion space is

$$dM_e/d\phi = V_E p_{mean}(1-\delta^2)^{\frac{1}{2}}\{\delta[\sin(\phi-\theta)-\sin\theta]-\sin\phi\}/2RT_E$$

(b) *Compression space* $V_c = \frac{1}{2}\kappa V_E[1+\cos(\phi-\alpha)]$.

The instantaneous mass of working fluid in the compression space is given by

$$M_c = \tfrac{1}{2}[\kappa V_E p_{mean}(1-\delta^2)^{\frac{1}{2}}(1+\cos(\phi-\alpha)]/RT_C[1+\delta\cos(\phi-\theta)]. \quad (4.21)$$

The rate of change of mass of working fluid in the compression space is

$$dM_c/d\phi = \kappa V_E p_{mean}(1-\delta^2)^{\frac{1}{2}}\{\delta[\sin(\phi-\theta)+\sin(\alpha-\theta)]$$
$$-\sin(\phi-\alpha)\}/2RT_C[1+\delta\cos(\phi-\theta)]^2$$

(c) *Dead space* ($V_D = XV_E$, constant).

The instantaneous mass of the working fluid in the dead space is given by

$$M_d = [XV_E p_{mean}(1-\delta^2)^{\frac{1}{2}}]/RT_D[1+\delta\cos(\phi-\theta)]. \quad (4.22)$$

The rate of change of mass of working fluid in the dead space is

$$dM_d/d\phi = [XV_E p_{mean}(1-\delta^2)^{\frac{1}{2}}\,\delta\sin(\phi-\theta)]/RT_D[1+\delta\cos(\phi-\theta)]^2.$$

Now $dM_e + dM_c + dM_d = 0$, so that the total mass of working fluid M_T is constant. Now,

$$M_T = V_E p_{mean}(1-\delta^2)^{\frac{1}{2}}\{\tau(1+\cos\phi)+\kappa[1+\cos(\phi-\alpha)]$$
$$+2S\}/2RT_C[1+\delta\cos(\phi-\theta)]$$

and, when $\phi = 0$,

$$M_T = V_E P_{mean}(1-\delta^2)^{\frac{1}{2}}[\tau+S+(\kappa/2)(1+\cos\alpha)]/RT_C(1+\delta\cos\theta). \quad (4.23)$$

Heat lifted and engine output in dimensionless units

(a) The heat lifted per unit mass of working fluid, combining eqns (4.17) and (4.23) is given by

$$Q_{mass} = Q/M_T RT_C = \pi\delta\sin\theta(1+\delta\cos\theta)/[(1-\delta^2)^{\frac{1}{2}}[1+(1-\delta^2)^{\frac{1}{2}}]$$
$$\times[\tau+(\kappa/2)(1+\cos\alpha)+S]\}. \quad (4.24)$$

Similarly, the net engine-output per unit mass of working fluid is given by

$$P_{mass} = P/M_T RT_C = (1-\tau)Q_{mass}. \quad (4.25)$$

(b) Non-dimensional expressions, in terms of characteristic pressures and volumes, may be devised as follows. The combined swept volume is given by

Combining this with eqns (4.13) and (4.17), then

$$Q_{max} = Q/(p_{max} V_T) = [\pi(1-\delta)^{\frac{1}{2}} \delta \sin\theta]/[(1+\kappa)(1+\delta)^{\frac{1}{2}}(1+(1-\delta^2)^{\frac{1}{2}})] \quad (4.26)$$

and

$$P_{max} = (1-\tau)Q_{max}. \quad (4.27)$$

THE FINKELSTEIN ADIABATIC CYCLE

Finkelstein (1960a) devised a generalized thermodynamic analysis of Stirling engines in which the processes of compression and expansion were not confined to isothermal conditions. In this generalized analysis the processes of compression and expansion *in the engine cylinders* could be specified to occur anywhere between the two limiting cases:

(a) isothermal (infinite rates of heat transfer between the working gas and the cylinder walls).
(b) adiabatic (zero rates of heat transfer between the working gas and the cylinder walls).

The model assumed that, in the heater and cooler, infinite heat transfer and isothermal conditions prevailed so that fluid in the heat exchangers was *always* at the upper temperature T_E or the lower temperature T_C. The temperature of the working fluid in the cylinders varied during the cycle and could be greater or less than T_E (in the expansion space) or T_C (in the compression space).

Finkelstein's generalized analysis retained all the other significant assumptions enumerated above for the Schmidt cycle so that it remained highly idealized. Nevertheless the possibility for non-isothermal processes represented the most significant theoretical development in nearly a century.

The limiting isothermal case of Finkelstein's theory corresponded exactly to the Schmidt cycle and equations were given for the adiabatic limiting case and for other cases intermediate between these limits. The theory was readily amenable to numerical analysis by standard methods. Significant simplification of the generalized analysis was attained by the assumption of adiabatic conditions (zero heat transfer) in the compression and expansion cases.

In his presentation, Finkelstein included only one set of numerical results. This referred to a heat pump with the temperature ratio $\tau = T_C/T_E = 2$. The coefficient of performance of 1.0 with isothermal processes was reduced to 0.543 with adiabatic processes. Similarly Stoddart (1960) found that a Stirling engine prime mover having a Schmidt cycle (Carnot) efficiency of 50 per cent with isothermal compression and expansion

Later Khan (1962) investigated the effect of variation in the principal design parameters and obtained numerical results for a large number of different cases with adiabatic compression and expansion processes. Some of these results were summarized by Walker and Khan (1965).

In the adiabatic cycle the thermal efficiency became a function, not only of temperature (as in the isothermal cycle), but also the swept-volume ratio, κ, the phase angle, α, and the dead volume ratio X. The power output was, of course, a function of all these parameters in both the isothermal and adiabatic cases. The results presented by Walker and Khan provided for the first time some indications of the effects on efficiency of design parameters other than temperature. One unexpected result was that with increase in the dead space of the engine the thermal efficiency of the engine *increased* even though the power output declined (as predicted by the Schmidt theory).

More recently Lee (1976) has reconsidered the Finkelstein adiabatic cycle including the preparation of a fast, well-behaved computer program in Fortran IV language. The program, containing extensive documentation to facilitate understanding is reproduced in entirety as an appendix to Lee's thesis†.

Fig. 4.1, reproduced from Lee (1976), shows the cyclic temperature variation of the working fluid in an adiabatic-cycle Stirling engine having temperatures of 1000 K in the heater (T_E) and 300 K in the cooler (T_C) so that $\tau = 0.3$. The phase angle $\alpha = 105°$, swept volume ratio $\kappa = V_C/V_E = 1.0$, and the dead volume ratio $X = V_D/V_E = 1.0$. It is interesting to note that the temperature of fluid in the expansion space was less than the heater temperature for much of the time. The mean temperature in the expansion space was 900 K, a difference of 100 K below the heat temperature T_C. In the compression space the mean temperature of the fluid coincided with the cooler temperature of 300 K.

Significant work using the adiabatic cycle approach was reported by Qvale and Smith (1969) and by Rios and Smith (1969). They consider a basic cycle with adiabatic compression and expansion and then separately assess the effects of irreversibilities. This approach allows for independent study of individual heat exchange components in a series of successive approximations that can be extended to the required degree of complexity.

Very little has been published so far about theoretical work on Stirling engines at the Philips Company or their licensees. Mention is made in the various papers about the extensive use of computers and the ability to

† Copies of the thesis by Lee may be obtained through the usual channels or by direct application to G. Walker, Department of Mechanical Engineering, University of Calgary.

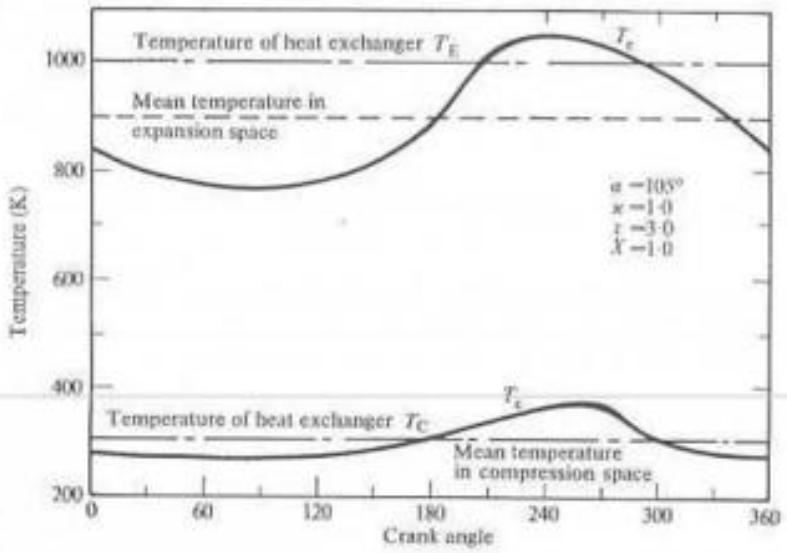


FIG. 4.1. Cyclic temperature variation of the working fluid in the expansion and compression spaces of a Stirling engine operating with adiabatic conditions in the two spaces (after Lee 1976).

predict engine performance to one or two per cent of test-bed performance. In this regard it is interesting to observe that Percival (1974) wrote:

'In 1969 it was concluded that there was still too much deviation between actual and predicted engine performance particularly for the 59 kW (80 hp) cylinder size. The real engine always rejected more heat and produced less power than the analytical engine.'

No doubt substantial improvements have taken place in the subsequent decade and there is no reason to doubt the claims made in several publications by workers at Philips of the ability to predict very closely the performance of actual engines.

Various straws in the wind gathered by the author over the years have led to the understanding that a family of computer programs has been developed by Philips. The various programs are thought to include some overall cycle-simulation programs for the complete engine at various levels of sophistication and complexity whereas others are available for the detailed simulation and design of single components or sub-assemblies in the engine, i.e. heater tubes, coolers, crank mechanisms etc.

So far as can be interpreted, the basis of the Philips thermodynamic-analysis simulation program is closely similar to that developed by Smith (reference above). This is basically the [illegible]

One of the main strengths of the Philips company is that close adherence over many years to similar design configurations has permitted the accumulation of extensive practical experience. On the basis of this, experience, 'fudge' factors of the proper magnitude can be judiciously applied to the analytical results to provide close and realistic simulation.

Indirect confirmation of Philips use of the adiabatic cycle with supplementary corrections was provided by Feurer (1973) of MAN/MWM, a Philips licensee, in his splendidly illuminating discussion of the effect of phase angle (α) on the power output and efficiency of a Stirling engine.

Feurer selected his engine parameters as representative of a single-cylinder Stirling engine built by MAN/MWM where '*measurements and calculations were largely in agreement.*' The principal parameters for the example were:

Working gas	Helium
Speed	1500 rpm
Mean pressure	12.2 M N/m^2 (120 atm.)
Cooler temperature (inside)	75 °C (167 °F)
Heater temperature (inside)	750 °C (1382 °F)
Carnot efficiency	66 per cent
Heater volume	100 cm^3 (6.1 in^3)
Cooler volume	56.5 cm^3 (3.4 in^3)
Regenerator volume	145.3 cm^3 (8.87 in^3)
Piston diameter	100 mm (3.9 in)
Stroke	50 mm (1.97 in)
Connecting rod length	100 mm (3.9 in)
Variable parameters:	
Phase angle	0–π rad (0–180 degrees)
Dead volume	0, 40, 100, 200 cm^3 (0, 2.4, 6.1, 12.2 in^3)
Diameter of connecting duct	100, 50, 20 mm (3.9, 1.97, 0.79 in).

He first calculated the power output and efficiency as a function of phase angle for the Schmidt cycle having the above design conditions and obtained the results reproduced in Figs. 4.2 and 4.3. These results refer to the case with zero dead volume, but in his paper Feurer gave results for nine other cases with different dead volumes. Also included in these figures were the results calculated for the adiabatic cycle.

Feurer then corrected the results for the adiabatic cycle to include the 'adiabatic residual losses', described as follows:

'When the semi-adiabatic machine was calculated, it was assumed that the temperature curve differed from the ideal temperature curve, but it was also assumed that this difference was in phase with the pressure curve, as in the case of the ideal processes. This difference—the influence on the pressure amplitude has already been allowed for—also has an influence on the phase displacement

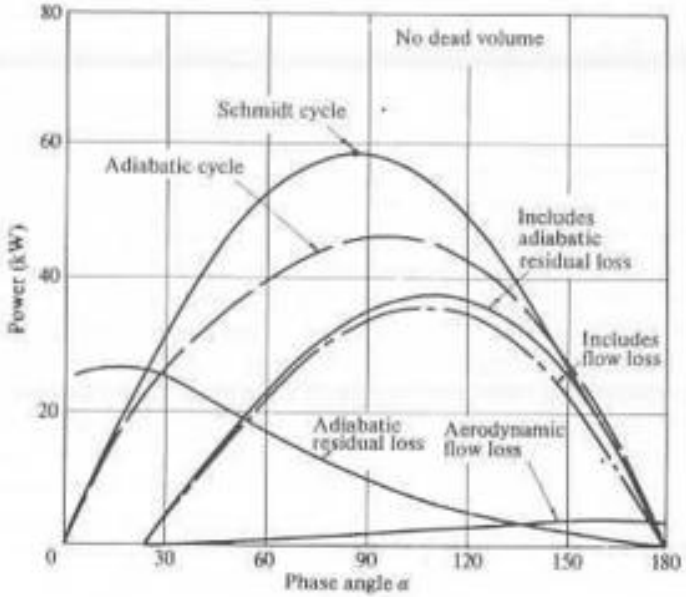


FIG. 4.2. Power output of a Stirling engine as a function of variation in the phase angle α (after Feurer 1973).

temperature in this volume element would result in a different mass, i.e. the deviating temperature curve would produce a different mass distribution in the cycle and this would result in a different pressure distribution. As compared with the old, ideal pressure distribution, this new pressure distribution has shifted by a certain amount. Thus, the following applies:

A difference between the ideal temperature in the cylinders, heat exchangers and connection spaces on the one hand, and the actual temperature in these components on the other inevitably results in a phase displacement between pressure and volume.

These losses, which are called 'adiabatic residual losses', are probably at their maximum when the largest pressure amplitudes occur on account of the phase displacement.

At a large phase angle of the volumes, i.e. when the volume between the pistons is only moved to and fro by the pistons and the pressure amplitude is only produced by the difference in temperature between the hot side and the cold side, this influence becomes negligible.'

The magnitude of the adiabatic residual loss is shown on Fig. 4.2. Its effect is shown on both Figs. 4.2 and 4.3 as the difference between the curves for the adiabatic cycle and that including the adiabatic residual loss.

A further correction was introduced to allow for the aerodynamic-flow loss resulting from the transfer of fluid about the engine. The magnitude of the aerodynamic-flow loss is also shown in Fig. 4.2. With zero dead

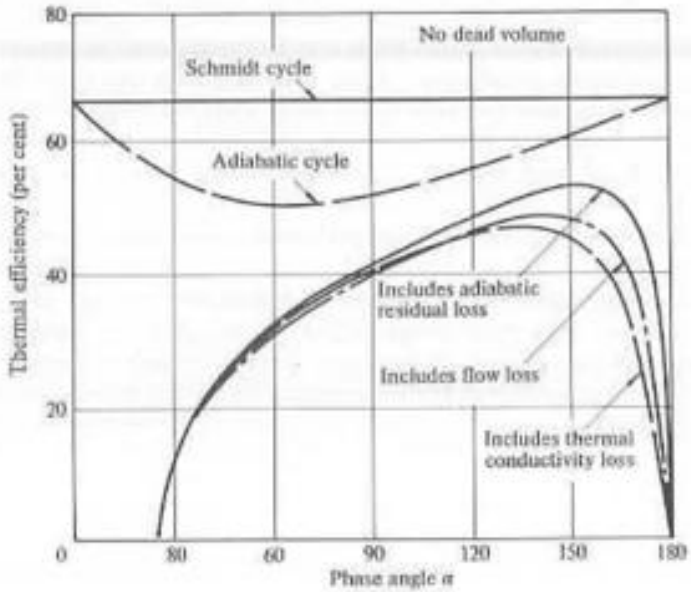


FIG. 4.3. Thermal efficiency of a Stirling engine as a function of variation in phase angle α (after Feurer 1973).

volume the flow loss was rather small as shown on Fig. 4.2 but was greater for the other cases considered by Feurer. Maximum volumes of the flow loss occurred at high values of the phase angle where the greatest mass flow rates of gas were involved.

Finally thermal losses by conduction were included. These had no effect on the engine power output and only a relatively minor effect on the efficiency as shown by Fig. 4.3.

The above is an abbreviated, inadequate, summary of the very important paper by Feurer, published simultaneously with another of equivalent significance (Zacharias 1973). Those responsible for the publication of these papers are warmly commended for their refreshing candour in presenting such fascinating material.

Hearsay has it, and the conference papers published in 1978 indicate, that a change in management policy at Philips will result in the future publication of material having greater significance than in the past. If true the trend is to be encouraged for they have been remarkably coy, revealing no more than the momentary '*flash of an ankle*'. The current heavy research investment of public funds in the U.S. will demand '*full frontal exposure*' and will, no doubt, flush out an interesting item or two

NODAL ANALYSIS

Nodal analysis of Stirling engines was pioneered by Finkelstein (1975a) but later, several other nodal analysis programs were prepared independently.

In nodal analysis programs the attempt was made to model the simultaneous energy and fluid flows occurring in the engine and thus simulate exactly the engine cycle and performance. This was achieved by writing and resolving equations for the conservation of mass, momentum and energy for particular nodes, cells or elements of the engine. The equations were too complex for general analytical solution and, thus, were solved numerically in terms of small incremental time steps. The equations were invariably reduced to one-dimensional form with additional simplifications introduced depending on the author.

A complete and comparative discussion of the various programs developed and available is beyond the scope of interest of this book (as well as beyond the competence of this author). Readers are therefore urged to consult the source documents referenced in the following notes.

All the nodal analysis programs are basically similar in their general approach. The design of engine to be simulated must be known in exact detail to the extent that the mechanical arrangement, cylinder-wall thickness and material, heat exchanger tube diameter, fin dimensions or matrix pore sizes are all specified. This design is then broken down, as experience dictates, into a number of nodes elements or control volumes. Some operating conditions must be specified such as charge pressure, and temperature of the energy source and sink.

Differential equations for the conservation of mass, momentum, and energy must be developed and generally converted to difference equations. Empirical formulae for the aerodynamic-friction and heat-transfer effects must be included, and an equation of state for the working fluid. A mathematically stable method must then be found for numerical solution of the difference equations to resolve the pressure, temperature, and mass distribution in the engine at the end of a particular time step, given the conditions at the beginning of the step.

The usual procedure for solution is to start with some initial arbitrary assumed conditions and then proceed through several engine cycles until quasi-steady state is achieved, that is when the instantaneous cyclic values of pressure, temperature, and mass distribution are not significantly different from the preceding cycle.

Finally the cyclic pressures and volumes are integrated to calculate work transfers. Heat flows are estimated and the overall thermal efficiency computed.

In most cases an isothermal Schmidt cycle-type calculation is included

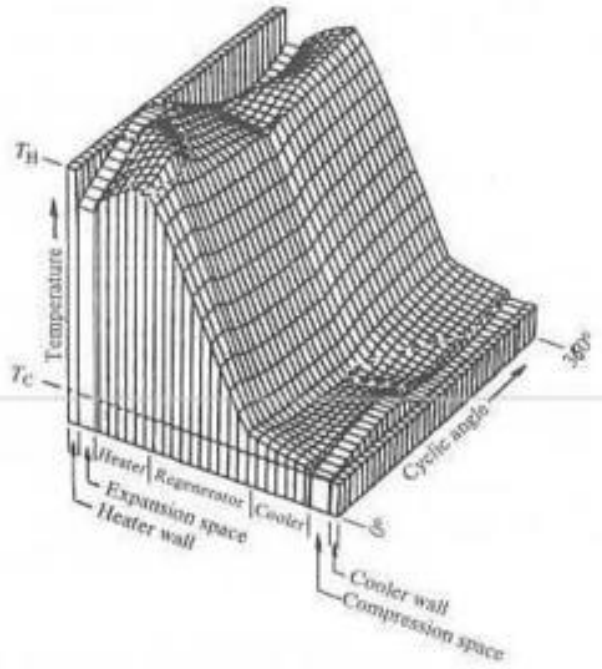


FIG. 4.4. Three-dimensional representation of the gas temperature versus node position and cyclic angle in a Stirling engine (after Schock 1978a).

The results are presented in a variety of formats. Typically the input data in terms of engine geometry and operating parameters is restated for comparison purposes. This is followed by tabulations at selected cyclic crank intervals of pressures, temperatures, velocities, mass content, and mass velocities at representative stations in the engine. Finally considerations of the overall cycle are presented, usually in terms of energy flows, work terms, heat transferred, or ratios of these such as the thermal efficiency.

Most programs now incorporate automatic plot routines so that, if required, the data can be presented pictorially as well as, or even instead of being tabulated. Some remarkable three-dimensional pictures can be produced in this way. Figs. 4.4 and 4.5 were given by Schock (1978a) and are typical of the interesting and informative representations achieved by computer automatic plot routines. These particular diagrams show the variation in temperature with node position or volume as a function of the cyclic angle.

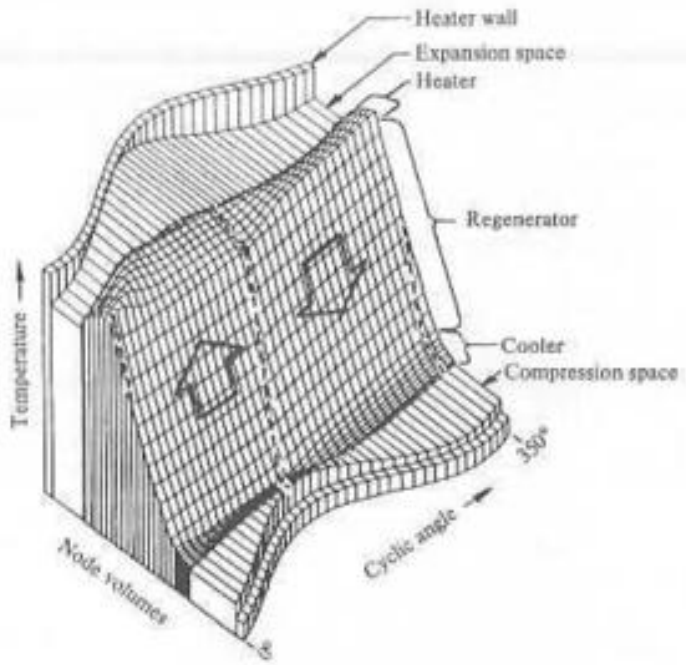


FIG. 4.5. Three-dimensional representation of the gas temperatures versus node volumes and cyclic angle in a Stirling engine (after Schock 1978a).

Finkelstein nodal analysis

Following the important development of the adiabatic analysis discussed above, Theodor Finkelstein continued working on increasingly sophisticated theoretical aspects of Stirling engines. This effort culminated during the late 1960s in the adaptation of the NASA Thermal Analysis Program (T.A.P.) to Stirling-cycle simulation and the creation of what is believed to be the first nodal analysis program.

The first practical application of the simulation program was at the University of Calgary in 1968/69 in support of the development of miniature cryogenic cooling engines for the British Ministry of Technology. Fig. 4.6 shows a simplified cross-section through the cooling engine. Fig. 4.7 shows the two-dimensional representation of the engine used for the nodal analysis simulation. This work was briefly discussed by Finkelstein, Walker, and Joshi (1971). Following some further refinement and development, Finkelstein (1975a) presented the theoretical basis for the analysis and described the numeric differencing technique for solution of

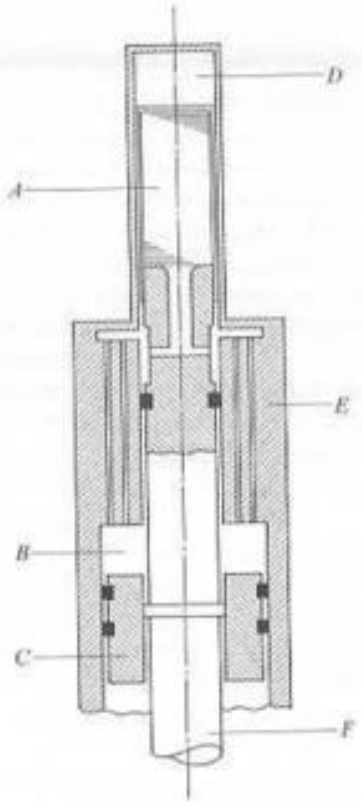


FIG. 4.6. Simplified cross-section of small Stirling-cycle cryogenic cooling engine. *A*—regenerator, *B*—compression space, *C*—piston, *D*—expansion space, *E*—cooler, *F*—displacer rod.

The Finkelstein nodal analysis program is now installed on a commercial computer network and available for general use on payment of a royalty fee. Potential users may obtain further details from Dr. T. Finkelstein, TCA Inc., P.O. Box 643, Beverley Hills, Ca.

Urieli nodal analysis

Israel Urieli, a mature graduate student working under the supervision of Professor C. Rallis at the University of Witwatersrand, has provided the most complete and comprehensive discussion (Urieli 1977) of Stirling-engine simulation by nodal analysis. His thesis is required reading for all those professionally interested in the field of Stirling-engine computer simulation. A good outline of the approach, and one more readily available than copies of the thesis, was provided by Urieli, Rallis,

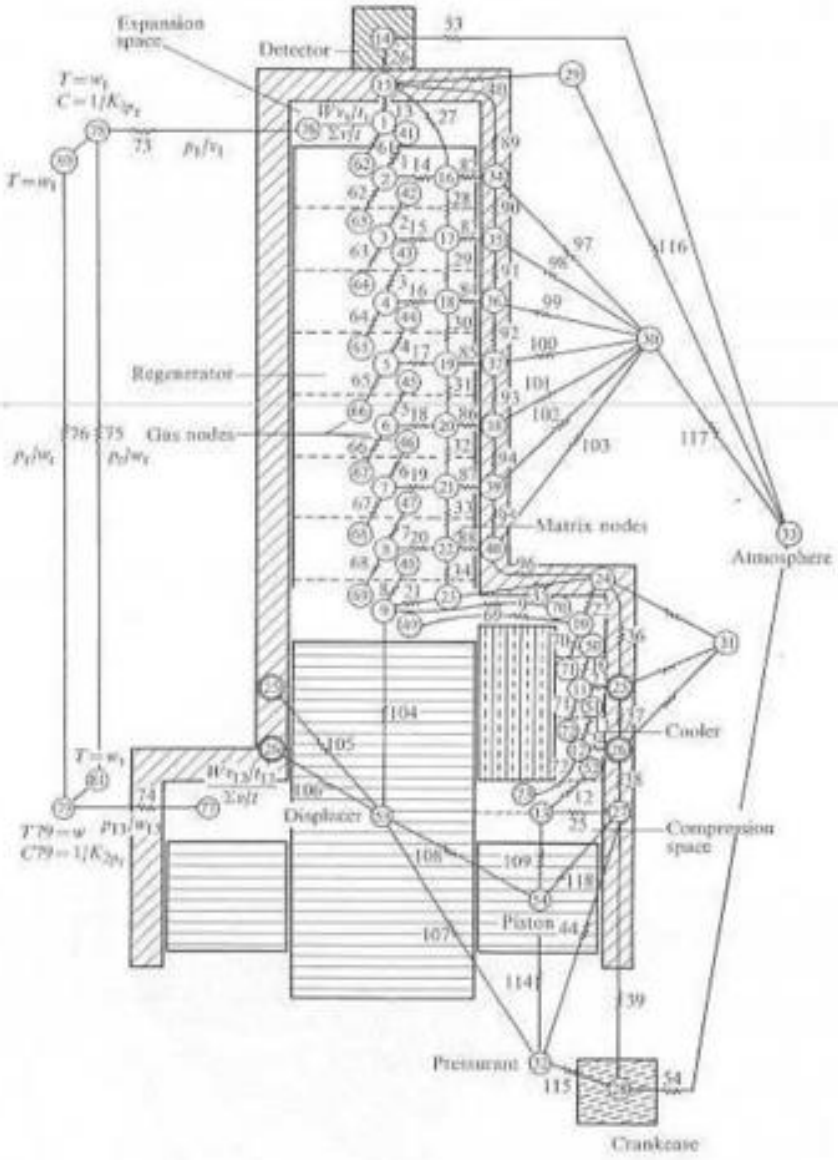


FIG. 4.7. Two-dimensional representation of the nodal analysis network for the small cryogenic cooling engine shown in Fig. 4.6.

The thesis contains a complete program listing in Fortran language of the complete Urieli program. The program is likely to be applicable to engines of a configuration other than the model assumed for investigation by Urieli. However experience with other computer programs suggests that very substantial effort would likely be required to establish a fully operational version of the Urieli program.

It is understood that work in the nodal analysis field continues at the University of Witwatersrand under the direction of Professor Rallis. Activity includes experimental work to validate the computer model and the investigation of alternative theoretical models (Berchowitz, Rallis, and Urieli 1977).

Martini (1978a) states that the Urieli thesis contains three errors in the main program and that corrections may be obtained from I. Urieli, Ormat Turbines, P.O. Box 68, Yavne, Israel.

Up-to-date information about his work on Stirling engines may be obtained on application to Professor C. Rallis, Department of Mechanical Engineering, University of Witwatersrand, South Africa.

Sunpower nodal analysis program

Gedeon (1978) has outlined the techniques of Stirling-engine simulation in use at Sunpower of Athens, Ohio, manufacturers of the Beale free-piston Stirling engines. Numerical simulation is an integral part of development at Sunpower. Engine refinement and program validation proceed hand-in-hand with routine daily exchanges between the staff engaged in laboratory and theoretical work.

It is understood that the Sunpower program is fast and fully validated. It is, in reality, a complex of programs capable of cycle simulation at different levels of sophistication as selected by the program controller. The program is self optimizing to the degree and according to criteria specified in the program input.

There is no doubt that in the field of Stirling engines, Sunpower has accumulated the greatest body of practical engine development and operating experience, supported by computer simulation, outside of Philips and their licensees. It is therefore gratifying to note that the Sunpower simulation programs are available for general use on a contract basis. Furthermore, active moves are afoot (Beale 1978†) to install the program on a commercial computer network and to organize 'schools' for education in the use of the programs. Further details of this and other aspects of Sunpower Stirling engine development may be gained from William Beale, President, Sunpower Inc., Athens, Ohio.

† Private communication

Schock nodal analysis program

Schock (1978a) has described the Stirling Nodal Analysis Program (SNAP) prepared by Fairchild Industries under contract to the U.S. Department of Energy in support of a Beale free-piston Stirling engine developed by Mechanical Technology Inc. and Sunpower Inc.

The Schock program appears to be closely similar in many respects to the Urieli program described above and developed simultaneously. It is well described in the draft report (Schock 1978a) referenced above and presumably intended for general distribution as a U.S. Department of Energy Contractor Report. An abbreviated description may be found in the paper (Schock 1978b) listed in the literature for the 1978 Intersociety Energy Conversion Engineering Conference.

It is expected that the program may be validated with reference to the performance of the free-piston engine described by Goldwater and Morrow (1977). Furthermore it is possible the program could be installed and made available for general use on a commercial computer network.

Details of the status of the program may be obtained from Alfred Schock, Energy Systems Department, Fairchild Space and Electronics Co., Germantown, Maryland.

Lewis Research Center nodal analysis program

Roy Tew and others (Tew 1978) at the NASA Lewis Research Center, Cleveland, Ohio, have developed a Stirling-engine nodal analysis program as part of the total DOE/NASA Stirling Engine Automotive Program. The description of the Lewis program given by Martini (1978) suggests that it may be less rigorous than the Schock or Urieli programs referenced above.

Simultaneously with the theoretical work, experimental studies are being carried out at the Lewis Research Center (Cairelli *et al.* 1977). It is thought to be likely that in the long run, the final answer to computer simulation and experimental validation will be placed in the public domain by the team at NASA Lewis Research Center. Readers are therefore urged to consult the current literature relating to the DOE/NASA Stirling Automotive Engine.

Further details may be obtained on application to Robert Ragsdale, Stirling Engine Project Office, NASA Lewis Research Center, 21000 Brookpart Road, Cleveland, Ohio.

Finegold/Vanderbrug nodal analysis program

Finegold and Vanderbrug (1977) have outlined a nodal analysis program for Stirling-engine simulation developed at the Jet Propulsion

for underwater applications. Present status and potential applications of the computer program are not known.

Organ nodal analysis program

Dr Allen Organ formerly of King's College, University of London, now with the School of Engineering, University of Cambridge, has published a series of recent papers indicative of a powerful nodal analysis type program in formation, if not yet fully developed. Readers are advised to consult recent contributions or contact Dr. Organ directly for an up-to-date appreciation of the status of development.

Philips/United Stirling/MAN/MWM nodal analysis programs

Several papers have hinted at the existence and use of nodal analysis type programs at Philips and their licensees. No details have been given. It is understood that for general design work, adiabatic-cycle programs are preferred over the more sophisticated nodal analysis programs.

SUMMARY

Stirling-engine simulation by nodal analysis is an expensive, time-consuming activity to be reserved for those professionally engaged in Stirling engine development and application, or for those academics engaged in the training and education of engineers at the graduate level.

At the present time no comprehensive comparison or evaluation is available of the various Stirling-engine simulation programs described above. Martini (1978a) has attempted the preparation of a design manual including a comparison of the various theoretical procedures. However he has unfortunately concentrated his efforts principally on isothermal type analyses with only passing reference to the more sophisticated nodal analyses.

Therefore at this time Stirling engine analysis is largely a horse-race where 'you pays yer money and yer takes yer choice'.

In terms of nodal analysis, the programs by Urieli and Finegold/Vanderbrug are completely listed in the references cited above. However neither of these programs has been extensively validated by reference to experimental data. Similarly the programs by Schock and by Tew are neither listed in the references cited and have yet to be extensively validated experimentally. Furthermore, at the time of writing neither program is installed and available commercially.

Finkelstein's program is available commercially on payment of moderate royalties but no adequate documentation of validity with reference to practical engines is available.

Finally, the Sunpower program is available commercially and has been validated extensively during the development of Beale-type free-piston engines. The applicability of the program to engines with crank mechan-

thought to present no particular problem, for it is generally reckoned in the trade that, from the aspect of simulation, free pistons are more difficult to handle than disciplined pistons.

Therefore, at this time, for those wishing to obtain a computer simulation or optimization of a Stirling engine concept (1978) the program of choice has to be the Sunpower version followed by Finkelstein model. The situation is extremely fluid and could change profoundly in a short time. One possibility is that Philips or their licensees could offer consultancy services to evaluate new concepts using their undoubted expertise and experience in the field. Various straws in the wind suggest this may be more likely a prospect than previously so that enquiries of this nature to North American Philips, United Stirling, or General Electric may well be appropriate.

Whatever the model selected it is certain that the few companies and individuals wishing to operate at the highest levels of Stirling engine simulation and optimization must be prepared to invest several weeks and considerable funds to establish the program procedure and to accomplish the work.

For those unable to afford the luxuries of nodal analysis a close study of the excellent manual by Martini (1978a) is recommended. The validity of his enthusiasm for so-called 'second order design' methods is not yet well established. Users of the manual are therefore cautioned to be discriminating. Of particular concern is Martini's view that an isothermal analysis corrected for various thermal and fluid effects can be an adequate basis for design.

The visceral feeling of this author is that the adiabatic cycle with the proper thermo/fluid corrections can be appropriate for most serious work in the design and optimization of Stirling engines. The adiabatic cycle does require the use of digital computers but not to an unconscionable degree and these are now so readily available that their use may be considered routine by most engineers. The adiabatic cycle, while a long way from actual conditions, is a good deal better than the isothermal case and requires only minor computing cost and time compared with nodal analysis. Readers sharing this view are therefore referred to the thesis by Lee (1976) and to the works of Rios, Qvale and Smith referenced below.

At yet an earlier stage in design isothermal analyses may be adequate. Martini (1978a) has developed techniques suitable for analysis using the small pocket calculators.

Finally at the start of a project it is usually sufficient to carry out a simple Schmidt cycle calculation using the equations given above. To attain the likely performance of a practical engine one simply divides the efficiency and power output by two (if one is an optimist) or by three (if one is a realist).

5 PRELIMINARY ENGINE DESIGN

INTRODUCTION

Power output: the Beale number concept

WILLIAM BEALE observed several years ago that the power output of many Stirling engines conformed approximately to the simple equation:

$$P = 0.015 p \times f \times V_0 \tag{5.1}$$

where P = engine power (watts),
p = mean cycle pressure (bar),
f = cycle frequency or engine speed (hertz),
V_0 = displacement of power piston (cm^3).

This can be rearranged as $P/(pfV_0) = \text{constant}$. The equation was found by Beale to be approximately true for all types and sizes of Stirling engines for which data were available including free-piston machines and those with crank mechanisms. In most instances the engines operated with heater temperatures of 650 °C and cooler temperatures of 65 °C.

The combination $P/(pfV_0)$ is a dimensionless group that may be called the Beale number. Now it is self evident that the Beale number will be a function of both heater and cooler temperatures. Recent work by the author leads to the conclusion that the relationship of Beale number to heater temperature may be of the form shown in Fig. 5.1 by the full line. Although the relationship is shown for the sake of clarity as a single line it must of course be understood that the relationship shown is a gross approximation and particular examples of engines that depart widely may be cited. Nevertheless, a surprisingly large number of engines will be found to lie within the bounds of the confidence limits drawn on either side of the proposed relationship. Well-designed, high-efficiency units with low cooler temperatures will be concentrated near the *upper* bound. Less well designed units of moderate efficiency with high cooler temperatures will be located at the *lower* extremity.

The applicability of a simple relationship such as the Beale number is clearly limited. Nevertheless, it provides a handy guideline for both 'back-of-envelope' style calculations for new projects and for newcomers to the field.

Many have an overly sanguine view of the capability of Stirling engines. One frequently hears proposals to convert small internal combustion engines to Stirling engines operating on low pressure air and using furnace heat or solar energy to produce several kilowatts of power. The Beale number is useful for dispelling such high hopes. For example, consider a small internal combustion engine having two cylinders with

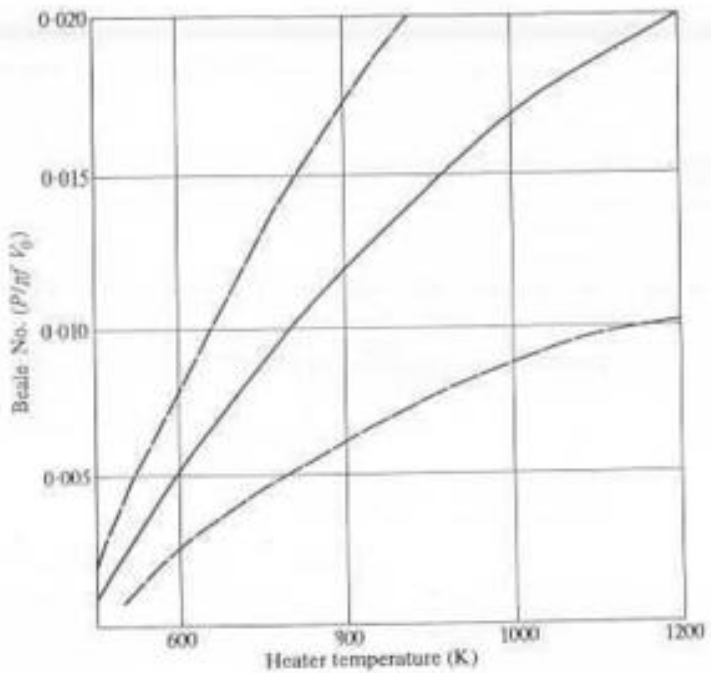


FIG. 5.1. The Beale number as a function of heater temperature. A design guideline for estimating the power output (P) of Stirling engines in watts.

diameter and piston strokes of, say, 5 cm. The engine may be modified to equip the two cylinders with heaters, regenerators, coolers and displacers to operate as a Stirling engine. Assume the working fluid to be at a low pressure, say 2 bar (0.2 MN/m^2). Then, the power output at a speed of 1200 revolutions per minute may be estimated using the Beale number concept as:

$$P = 0.015 p \times f \times V_0$$

$$= 0.015 \times 2 \times \frac{120}{6} \times \frac{\pi}{4} \times 5^2 \times 5$$

$$= 75 \text{ watts approximately.}$$

Clearly such a low power will come as a disappointment to our enthusiastic newcomer. Furthermore, it is likely that the mechanical friction will be greater than the power generated and so the engine will never run or, if it does, will barely be self-sustaining.

At the other end of the scale consider the concept of a large Stirling

agricultural or industrial waste for stationary power generation. Assume, arbitrarily, an output of 746 kW (1000 horsepower) per cylinder with a mean cycle pressure of 20 MN/m^2 (200 bar), with helium working fluid and a frequency of 30 cycles per second, corresponding to a speed of 1800 revolutions per minute.

The size of engine cylinder required may then be computed from the Beale number. There is some evidence that the Beale number is conservative for large engines. Furthermore, in such a high capital cost application, a sophisticated design with adequate cooling might be expected. Therefore perhaps it would be reasonable to double the value of the constant in eqn (5.1) from 0.015 to 0.03. Finally, it is advantageous at this stage to recall, from the aspect of seal, bearing, and piston-ring wear, the attractions of an 'oversquare' engine, i.e. a large bore and short stroke. Here we will arbitrarily assume a stroke that is one half the piston diameter.

Therefore, the piston swept volume:

$$V_0 = \frac{\pi}{4} D^2 \times \frac{D}{2}. \tag{5.2}$$

So the Beale number $P/(pfV_0) = 0.03$ may be rearranged as:

$$V_0 = \frac{P}{0.03pf} \tag{5.3}$$

or

$$(\pi D^3)/8 = \frac{P}{0.03pf} \tag{5.4}$$

so that

$$D = \sqrt[3]{\left(\frac{8P}{0.03\pi pf}\right)} \tag{5.5}$$

In our example

$$P = 1000 \times 746 \text{ watts}$$

$$p = 20 \text{ MN/m}^2 \text{ (200 bar)}$$

$$f = 30$$

so that

$$D = \sqrt[3]{\left(\frac{8 \times 7.46 \times 10^5}{0.03 \times \pi \times 200 \times 30}\right)}$$

$$= 21.9 \text{ cm}$$

Therefore, the diameter of the cylinder required is 22 cm and the piston

Thermal efficiency

Similarly crude approximations for thermal efficiency of Stirling engines may be used. The most efficient engines approach values a little in excess of 60 per cent of the thermodynamically ideal Carnot value. Most are nearer 50 per cent of the Carnot value and very little has to be overlooked or eliminated before the efficiency is down to 40 or even 30 per cent of the Carnot value. Design modifications to developed engines so as to effect economies or simplification in production should, therefore, always be carefully assessed for the effect on efficiency.

Depending on one's philosophy as an optimist or as a realist, a probable value of 40 to 50 per cent of the Carnot efficiency may be assumed. The thermal efficiency of an engine may then be estimated simply as the ratio:

$$\eta = \frac{\text{Power output}}{\text{Heat supplied}} = 0.5\left(\frac{T_{max} - T_{min}}{T_{max}}\right) \qquad (5.6)$$

Of course now the problem arises as to what values to use for the temperatures. As a general rule, for engines equipped with non-special high-temperature steels for the hot parts, an average temperature of 600 °C may be the maximum tolerable. For the cooler, in temperate zones with adequate water cooling available, a temperature in the range 20–80 °C may be achieved. Let us assume a value of 20 °C. Then the engine efficiency may be estimated:

$$\eta = 0.5\left[\frac{(600+273)-(20+273)}{(600+273)}\right]^{\dagger}$$

or even more approximately:

$$\eta = 0.5\left(\frac{900-300}{900}\right) = 33 \text{ per cent}$$

This must be recognized as a rather high efficiency to be achieved only by

† Note the use of absolute temperatures in computing the thermal efficiencies. Temperatures measured (by thermocouple or mercury in glass thermometers) in degrees Centigrade may be converted to the thermodynamic scale temperatures, in Kelvin, by the addition of 273, thus:

$$100\,^{\circ}\text{C} = (100+273)\text{K}.$$

Similarly, temperatures measured in degrees Fahrenheit must be converted for thermodynamic calculations to the absolute temperature degrees Rankine by the addition of 460. Thus

$$212\,^{\circ}\text{F} = (212+460)\,^{\circ}\text{R}.$$

Single degrees have exactly the same magnitude of temperature change in the Centigrade and Kelvin scales and are equal to 1.8 degrees in the Fahrenheit and Rankine scales.

the most careful attention in design to minimize losses. Most engines will operate initially at one half or less of the above efficiency but can often be developed to approach the 30 per cent value *if high efficiency is the prime interest.*

It must be emphasized that such calculation methods are suitable only for crude design approximations. Nevertheless, they provide a ready means for 'back-of-envelope' estimations which may be useful in technical meetings or at the concept stages of new, previously unexplored, projects.

Compression ratio

Whatever type of engine configuration adopted, it is difficult to increase the *volume* compression ratio much above the value $V_{max}/V_{min} = 2.5$. Attempts to increase this compression ratio will almost certainly result in there being inadequate void volume in the internal heat exchangers. This will result in either inadequate surface area for heat transfer or high pressure drops because of excessive aerodynamic friction pressure.

As a consequence of the low volume compression ratio the pressure ratio (p_{max}/p_{min}) in Stirling engines is very low compared with internal combustion engines. It rarely exceeds a value greater than 2. Furthermore, the *rates* of pressure change are very low for the pressure characteristic is virtually sinusoidal in form. This has important consequences to engine design particularly with regard to bearings and shafts.

Consider now the pressure level in the engine. If the minimum cycle pressure is 0.1 MN/m^2 (1 atmosphere) and the pressure compression ratio is 2, then the maximum pressure will be 0.2 MN/m^2 (2 atmospheres) and the *mean* pressure will be 0.15 MN/m^2 (1.5 atmospheres). If now the minimum pressure is elevated to 10 MN/m^2 (100 atmospheres) the maximum pressure will, to a first approximation, increase to 20 MN/m^2 (200 atmospheres) with a mean pressure of 15 MN/m^2 (150 atmospheres). The *range* of the pressure excursion (p_{max}/p_{min}) has increased from $0.2-0.1 = 0.1$ MN/m^2 ($2-1=1$ atmosphere) to $20-10 = 10$ MN/m^2 ($200-100 = 100$ atmospheres). This elevation of the mean pressure level from 0.15 to 15 MN/m^2 (1.5 to 150 atmospheres) resulted in an increase in the *range* of the pressure excursion from 0.1 to 10 MN/m^2 (1 to 100 atmospheres). The *work* produced is directly proportional to the range of the pressure excursion and, therefore, to the general pressure level in the engine. Increase in the pressure level will increase the engine output directly.

The work of the engine may be estimated from the mean pressure of the engine. However the cylinders must be designed to withstand the *maximum* cylinder pressure. Given the pressure ratio ($p_{max}/p_{min} = 2$, then, approximately, $p_{max} = (\frac{4}{3})p_{mean}$

CYLINDER MATERIALS

The necessary wall thickness of a cylinder to contain a given pressure may be estimated from the simple equation:

$$\sigma = \frac{pd}{2t} \tag{5.7}$$

where σ = maximum permissible stress
p = maximum internal pressure
d = cylinder internal diameter
t = wall thickness.

The maximum stress to be used in the calculation depends on the material to be used for the cylinder wall. It should be some fraction, say 0.8, of the *yield* or *proof* stress rather than the *ultimate* stress. The yield stress for metals ranges typically from 138 to 1034 MN/m^2 (20 000 to 150 000 lbs per sq in) and varies widely with temperature. The safe stress declines markedly and progressively with elevated temperature. Therefore cylinder design should not be undertaken without a good knowledge and understanding of the materials to be used and the temperatures likely to be attained. Frequently, the superior strength characteristics of alloy steels are achieved by particular forms of heat treatment or mechanical processing and can be profoundly affected by operation at high or even moderately elevated temperatures.

A complete discussion of material properties is beyond this text. However it should be clearly understood that selection of proper materials is the key to success and due regard should be paid at the design and development stage.

As a general rule the hot parts will likely be constructed from materials having superior strength characteristics at high temperatures. They will most likely be alloys containing an appreciable proportion of nickel and chromium. Stainless steels, typically an alloy of 18 per cent chromium and 8 per cent nickel, are readily available in a wide variety of sizes and forms and can be used for most early prototype development work.

In design of the engine care must be taken to avoid the creation of 'stress raisers'. These are areas where sudden changes in section occur such as corners, keyways, slots etc. where the local stress levels may be several times greater than the mean stress. This can lead to the formation and development of cracks that will eventually result in mechanical failure of the component. 'Stress raisers' can be avoided by careful attention to design to avoid sudden changes in section by the use of well rounded corners and fillets, large diameter holes, etc. There are many excellent texts on the design of mechanical machine elements which

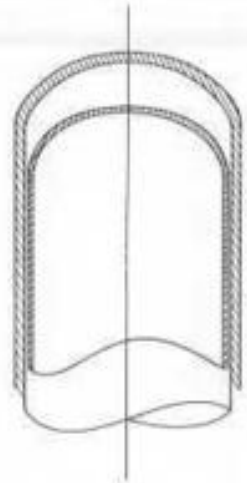


FIG. 5.2. Application of domed cylinder heads and displacer to avoid local stress concentration.

The significance of local stress concentration is particularly important at high temperatures and for this reason many cylinders have curved or domed ends rather than a simple plane end. This is illustrated in Fig. 5.2. The reciprocating element operating in the cylinder, the piston or displacer, should of course have a shape at the end corresponding to the internal form of the cylinder. This is necessary to reduce the 'clearance space', to a minimum—that is, the volume in the cylinder above the displacer at the top dead centre position.

Dome-ended metal forms for displacers and cylinders may be acquired from many unlikely sources as an alternative to 'one-off' manufacturing which tends to be expensive. Many model engines have been constructed with the cylinder and displacer made from aluminum cigar cases, one fitting inside the other. At a somewhat higher level, excellent stainless steel hemispherical forms in a reasonable range of sizes and wall thicknesses may be found among the soup ladles in the better class kitchen or commercial food equipment stores. These may be readily welded to stainless steel tubes to form the required shape.

Opportunities for such innovation abound and should not be overlooked by the designer anxious to minimize the time and cost of production.

THERMAL EFFECTS

Thermal conduction

All materials conduct heat to a greater or lesser extent. Metals are

of heat conducted by a given material is characterized by the physical property called the thermal conductivity, k. The units of thermal conductivity are watts per square metre of cross section in the direction of heat flow per metre of thickness of the material per degree centigrade temperature difference across the thickness of the material, i.e. kW/m °C.

The amount of heat conducted along a cylinder from the hot end to the cold end may be estimated from the equation:

$$Q_c = \frac{kA\,\Delta T}{l} \tag{5.8}$$

where Q_c = rate of heat transfer (watts) by conduction along the cylinder walls
k = thermal conductivity of the material
A = area of cross section for heat flow = πdt
ΔT = temperature difference between the hot and cold ends of the cylinder
l = length of cylinder.

Therefore

$$Q_c = \frac{\pi k dt\,\Delta T}{l}. \tag{5.9}$$

It will be recalled from the above discussion of the Beale number, that approximately,

$$P = (c\Delta T)pf\frac{\pi}{4}d^2L \tag{5.10}$$

where L = piston stroke. We know from the stress eqn (5.7) that $p = (2t\sigma)/d$ so that

$$P = (c\Delta T)\frac{2t\sigma}{d}f\frac{\pi}{4}d^2L = \frac{\pi}{2}(c\Delta T)\sigma tfdL. \tag{5.11}$$

Therefore, the ratio,

$$\frac{P}{Q_c} = \frac{c\sigma fLl}{2k} = \left(\frac{c\sigma}{2k}\right)fLl = KfLl, \tag{5.12}$$

where K is a constant $K = (c\sigma/2k)$. This illustrates the interesting fact that the ratio of power to heat transfer by conduction, is (1) *Independent of cylinder diameter, wall thickness and temperature ratio*, (2) strongly dependent on the length of the cylinder, the piston stroke and cyclic frequency.

The ratio P/Q_c, which clearly, should be as high as possible, is increased by the use of a cylinder material having a high strength (high σ) and a low conductivity (low k). Fortunately, stainless steel is a material

which combines the characteristics of high strength at elevated temperatures with thermal conductivity. Furthermore, it is available at moderate cost and can be readily formed and joined by welding or brazing. It is therefore the preferred choice of material for most serious prototype engines. It is often replaced by more exotic materials in the 'second-round' sophisticated advanced designs of prototype engines.

Shuttle heat transfer

An important thermal effect found in Stirling engines called '*shuttle heat transfer*' has the effect of increasing the apparent thermal conductance loss. The effect comes from the reciprocating action of the displacer in the cylinder. There is a temperature difference along the walls of the cylinder and the displacer, from the hot end to the cold end. With the displacer in the top dead centre position the temperature profile of the displacer along the length may be very similar to that of the adjacent cylinder wall as shown in Fig. 5.3. Then the displacer is moved to a region where the temperature of the cylinder wall is much less. Consequently heat will be transferred (by gaseous conduction and radiation) from the hot displacer wall to the cooler cylinder wall.

A similar process will occur if the cylinder wall temperature is above the displacer wall temperature when the displacer is returned to the top dead centre position. In this case of course the direction of energy flow would be from cylinder wall *to* displacer. Thus a progressive stepwise heat

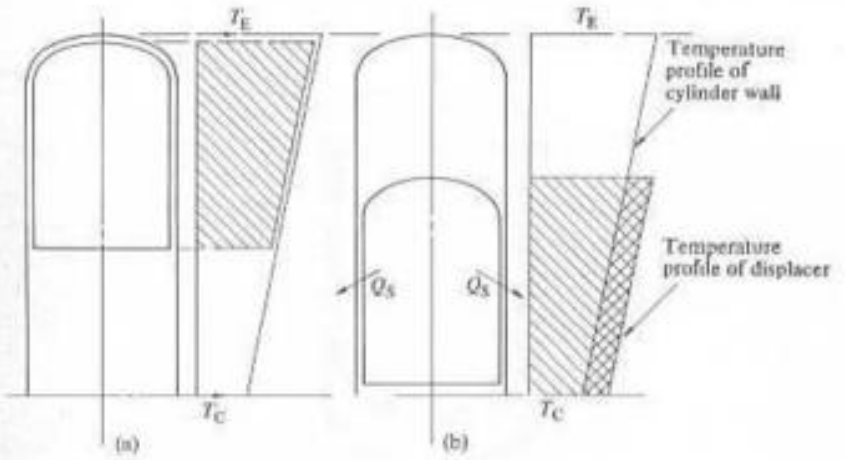


FIG. 5.3. Diagram illustrating shuttle heat transfer in Stirling engines. As the displacer moves from the top to bottom dead centre the hot parts of the displacer are adjacent to the cooler cylinder walls. The effect is to increase the heat transfer from the displacer to the cylinder walls, so increasing the effective thermal conduction loss.

transfer occurs, evocatively christened by Finkelstein as the 'bucket-brigade loss' but which has become known, more mundanely, as shuttle heat transfer.

The effect can be very important, particularly in small cryogenic cooling engines and most studies of the effect have been directed to that application (Zimmerman and Longsworth 1971). It corresponds to an effective increase in thermal conductance but is difficult to estimate until the design of the engine is complete.

It is self-evident that the shuttle heat transfer loss can be minimized by the use of a small stroke. However, given a particular piston swept-volume, the use of a small stroke implies the use of a large diameter. This in turn requires a thick cylinder wall and large cylinder circumference which combine to give a large cross-section area for simple conduction heat transfer. This is true for both the cylinder and displacer wall conduction. Therefore, with use of a shorter stroke, the gain in reduced shuttle heat transfer may be largely offset by an increase in the conduction heat transfer. Martini (1978d) has summarized available data on shuttle heat transfer. Judicious generalization of his recommendations lead to the following approximate equation which may be used, with caution, for estimation of the shuttle heat transfer:

$$Q_{sh} = 0.4L^2 \cdot k \cdot D \cdot (T_E - T_C)/(S \cdot Z)$$

where Q_{sh} = shuttle heat transfer, watts
L = displacer stroke, cm
k = gaseous conductivity of working fluid W m^{-1} K^{-1}
D = displacer diameter, cm
T_E = heater temperature, K,
T_C = cooler temperature, K,
S = annular gap between displacer and cylinder, cm,
Z = length of displacer, cm.

RECIPROCATING ELEMENT

In Chapter 6 different mechanical configurations that may be used for Stirling engines are discussed. In all of these systems, various reciprocating members are included. In general, these may be divided into two categories: pistons or displacers. The difference is illustrated in Fig. 5.4 which shows a diagrammatic representation of an engine arrangement known as the 'piston and displacer in same cylinder' engine.

The cylinder contains two reciprocating parts, a piston and, above it, a displacer. The upper part of the cylinder above the displacer is the expansion space at high temperature. The space between the piston and displacer is the compression space and is at ambient temperature. The two spaces are in mutual communication through the heater, cooler, and

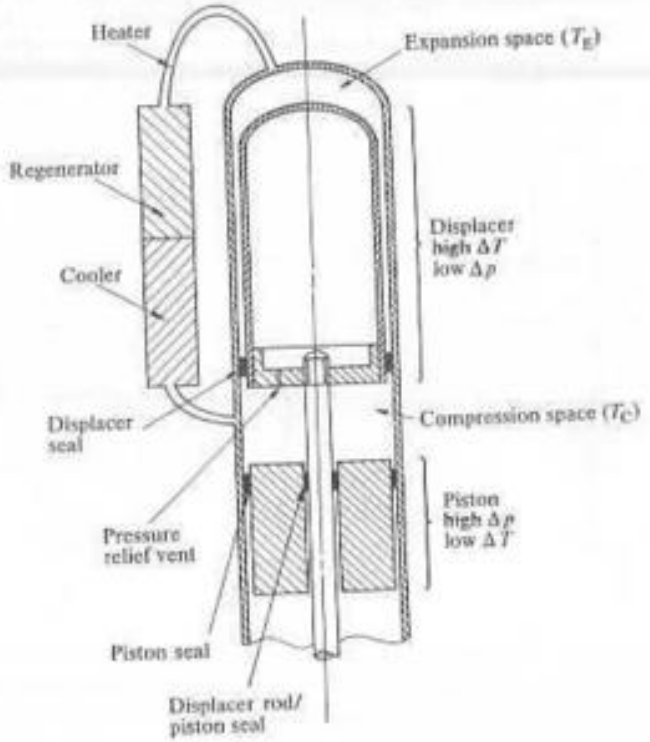


FIG. 5.4. Stirling engine arrangement known as the 'piston and displacer in same cylinder' illustrating the distinction between a piston and a displacer.

The *piston* is the component used to convert the gaseous energy of the working fluid to mechanical work for driving the load. This may be done directly, in the free-piston engine, or through a crank mechanism in the case of a 'kinematic' engine. This energy transfer requires that the piston be a structural member sufficiently robust to withstand all the gaseous, mechanical, and inertia forces imposed on it.

There is likely to be a large pressure difference across the piston between the working space and the crankcase. Therefore the piston also carries a seal to isolate the working space. The seal serves both to prevent the egress of working fluid and the ingress of lubricant. This latter is a very important function of the seal, to prevent contamination and blockage of the regenerator.

The temperature of the environment above and below the piston is approximately the same, so the piston may be designed without regard to thermal effects apart from thermal expansion in the radial direction.

A piston may therefore be characterized briefly as a component having a *high delta p and zero delta T*, in other words, a high pressure difference and a zero temperature difference across the upper and lower transverse faces.

A displacer is quite the reverse and may be characterized as a component having a *zero delta p and high delta T*, in other words, no pressure difference but a large temperature difference between its ends. A small pressure difference does in fact exist, equal to the pressure drop across the regenerator and associated heat exchangers.

A displacer must therefore be constructed as a lightweight structural element required to sustain only low pressure and inertia forces but with minimum thermal conduction loss. This leads automatically to the classical displacer construction shown in Fig. 5.4 of a light, hollow, thin-wall envelope having a dome end and a length two or three times the diameter. The shell may be reinforced at intervals along the length by horizontal plates, spot welded, brazed or simply pushed into the displacer shell. These serve as radiation shields and, principally, to reduce internal convective heat transfer.

Engines of moderate to high specific output are pressurized to very high pressures of the working fluid. In Stirling engines used for automotive applications the pressure of the hydrogen working fluid may approach 200 MN/m^2 (3000 lb per sq in). In such cases an important decision must be made about the internal volume of the displacer. It may be either pressurized or unpressurized. If unpressurized, the walls of the displacer will need to be sufficiently substantial to prevent crumpling of the shell. When exposed to the maximum cycle pressure this will increase the mass of the displacer and hence the inertia loading and also the thermal conduction loss along the walls of the displacer shell. The alternative is to pressurize the displacer internally but if the unit is pressurized and sealed during construction then the walls must be sufficiently thick to contain the high internal pressure against zero external pressure when the engine is being constructed or maintained.

A better alternative is to provide a very small hole (less than 0.0254 cm or 0.010 in diameter) in the displacer so that some of the working fluid may pass into or out of the shell as the pressure of the working fluid changes. The hole must be made very small so that the rate of fluid flow in normal operation will be small and the internal pressure will be approximately constant and equal to the mean pressure of the working fluid. It is vital that the flow to and from the displacer be restricted otherwise the internal volume of the displacer will become part of the internal void volume, the dead space, of the engine with consequent deleterious effects on both the power output and thermal efficiency.

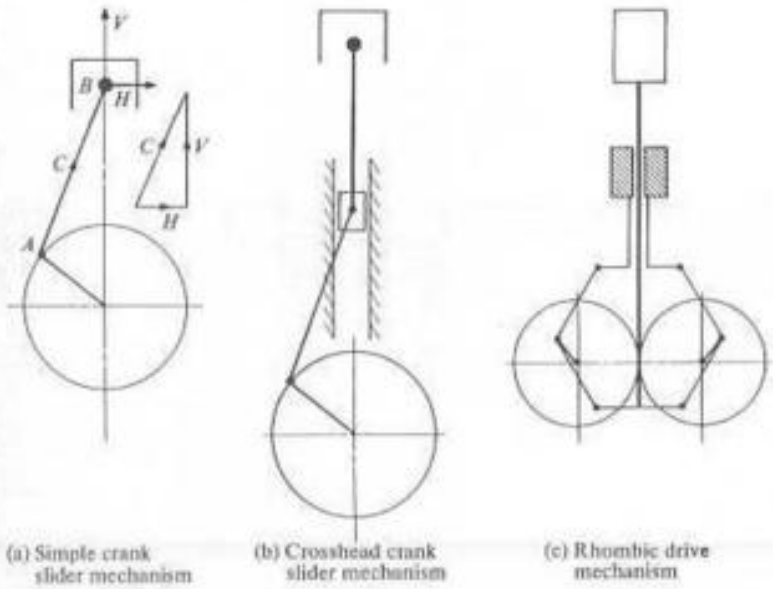
In every case it is likely to be advantageous to fill the displacer void volume with thermal insulation to minimize convective heat transfer. Solid cellular or fibrous materials suitable for high temperatures may be used but powder insulations are best avoided.

Double-acting engines

In Siemens double-acting engines a single reciprocating element per cycle is used. Successive cylinders are coupled through the regenerator and the associated heat exchangers so that the hot expansion space of one cylinder (above the reciprocating element) is connected to the cold compression space (below the reciprocating element) of the adjacent cylinder. In this case the reciprocator serves both as a displacer and a piston for there is both a significant pressure and temperature difference between the ends and work is transmitted from the displacer/piston to the load.

Piston side forces

A variety of kinematic mechanisms may be used to couple the reciprocating pistons or displacers to drive shafts. Many of these mechanisms result in side forces of variable magnitude acting to push the reciprocator against the side wall. Consider for example the simple crank connecting rod systems shown in Fig. 5.5(a). The force C acting along the line of the



(a) Simple crank slider mechanism (b) Crosshead crank slider mechanism (c) Rhombic drive mechanism

FIG. 5.5. Kinematic mechanisms for reciprocating drives. The simple crank slider mechanism in (a) causes a side thrust of the piston against the cylinder wall

connecting rod A–B may be resolved at the small end bearing B into vertical and horizontal components V and H respectively.

The horizontal component H, the piston side-force, causes a vertical drag force between the piston and cylinder $D = \mu H$ where μ is the coefficient of friction and acts in a direction opposite to the direction of piston motion. The side force also results in increased wear of the rubbing surfaces.

And wear, whether due to side forces or not is particularly unwelcome where a very high pressure difference has to be contained by seals in the piston. Efforts must be made therefore to separate the components responsible for sealing and for absorbing the side forces arising from the kinematic mechanism. One possibility is to use the cross-head arrangement shown in Fig. 5.5(b). This type of arrangement is used in the MAN/MWM and United Stirling engines discussed in Chapters 14 and 15 respectively.

An alternative mechanism to eliminate piston or displacer side thrust is the rhombic drive shown in Fig. 5.5(c). This type of arrangement is well-suited for use on single-cylinder, single-cycle engines and is discussed in Chapter 12 in relation to Philips engines.

SEALS

Sealing is undoubtedly the most difficult recurring problem in Stirling engines. The best way to increase the output of a Stirling engine of given size is to increase the pressure of the working fluid. Pressures therefore tend to be high. Further, high-output engines use hydrogen or helium as the working fluid rather than the heavier gases such as air, or carbon dioxide. Seals are therefore required in the piston to contain the fluid in the working space and prevent its leaking to the crankcase.

Small engines may be designed so that the crankcase is pressurized to the minimum cycle pressure. In that case the piston seal is required to seal only the pressure difference between the working space and crankcase pressures ranging from $(p_{min} - p_{min}) = 0$ to $(p_{max} - p_{min})$ where p_{max} and p_{min} are the maximum and minimum cycle pressures of working fluid.

Engines with pressurized crankcases will have a further seal, a dynamic rotary seal, where the crankshaft leaves the crankcase to contain the crankcase pressure. Alternatively, if the engine is to be used for producing electricity, the generator can be built integrally with the crankcase so that a *static* seal only is required where the power lead passes through the crankcase.

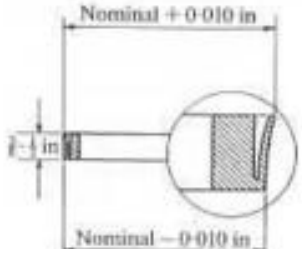
As engines become larger, use of the pressurized crankcase becomes less attractive because of the size and weight of the structure necessary to contain the pressure. In these cases the piston seal must contain the cyclic

Much work has been done by Philips on seals, initially close tolerance seals, and more recently, on the rolling diaphragm seals discussed in Chapter 12. Other companies have developed alternatives to the rolling seal. A great deal of effort has been invested by United Stirling on the multipart sliding seals discussed in Chapter 15. A similar cartridge seal system is under development by the Ford Motor Co. for use in Siemens type double-acting engines with a swash-plate drive. Seal development is an important part of the program sponsored by the NASA Lewis Research Centre for the Stirling automotive engine.

For sliding seals that are dry lubricated (or unlubricated) fluorocarbon materials are preferred, particularly polytetrafluoroethylene, commonly known as PTFE or the trade name 'Teflon'. This is characterized by a very low friction coefficient. A specially compounded PTFE material known as 'Rulon A' has a wear resistance that is orders of magnitude greater than Teflon and retains many of the other attractive properties. 'Rulon A' is therefore the preferred choice of material for Stirling engine seals. Percival (1974) has provided some interesting data on the use of Rulon seals in the General Motors Stirling engines discussed in Chapter 13.

One form of piston seal developed by William Beale and used with moderate success by the author is shown in Fig. 5.6. The seal is turned from Rulon bar or sheet stock, and is machined, so that the seal lip is about 0.0254 cm (0.010 in) over size on the nominal diameter (with a 5.08 cm (2 in) bore). The seal is tapered to a diameter about 0.0254 cm (0.010 in) under size at the lower face. A parting tool is used to cut the seal away around the periphery, to leave a thin flexible lip about 0.95 cm ($\frac{3}{8}$ in) long. The seal is mounted on the piston by means of epoxy-cement or a holding plate, and, when inserted into the cylinder, will be very tight. Standing overnight, the seal will become relatively free, and will become loose after working a few times up and down the cylinder. The cylinder should be made with a hard metal surface honed and polished.

Another form of piston seal is shown in Fig. 5.7. This embodies Rulon piston-ring grooves cut into the piston similar to conventional internal



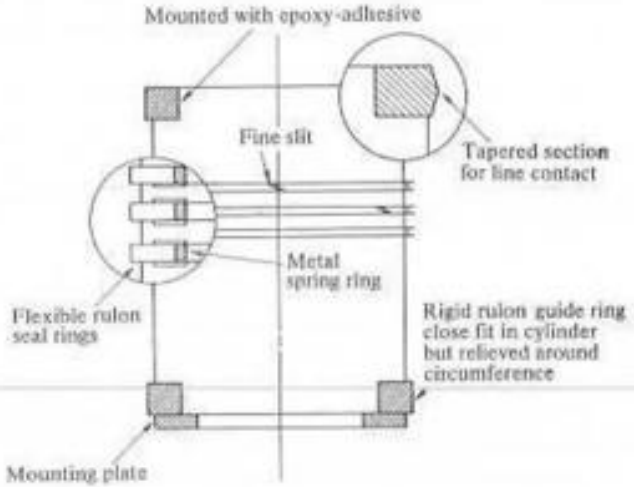


FIG. 5.7. Rulon seal and guide rings.

combustion engine practice. Two thin rings of rectangular cross section about 0.24 cm ($\frac{3}{32}$ in) to 0.32 cm ($\frac{1}{8}$ in) thick will be found more effective than a single thick ring. The rings should be a 'sloppy' fit in the piston ring grooves and should be supported or backed by some kind of metal spring ring lightly pushing the ring radially outwards. The ring should be split by a radial slit cut into the ring when mounted on a mandrel.

Neither the ring nor the lip seal described above offer guidance to the piston and so it will most likely be found necessary to provide guide rings in addition to the seal rings. In Fig. 5.7 the guide rings are shown at the axial extremities of the piston. They may be solid rings mounted by adhesive on a mounting plate to the piston. Alternatively, they may be split to facilitate mounting in a ring groove. In either case the ring should be a snug sliding fit in the cylinder and a close fit in the ring groove. The guide ring should be relieved at intervals around the circumference and shaped so the actual circumferential contact of the ring with the cylinder is a line or a small surface area.

In other engines, particularly free-piston machines, good results can often be obtained with close tolerance seals. A typical design is shown in Fig. 5.8. It consists, simply, of a well-fitting piston sliding within a cylinder. For best results the piston should be of a length that is at least twice the diameter and relieved at intervals along the length by circumferential grooves about 0.16 to 0.24 cm ($\frac{1}{16}$ to $\frac{3}{32}$ in) long and deep. The grooves should be spaced at regular intervals about one half inch (1 cm)

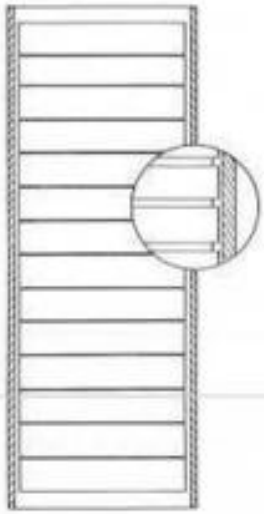


FIG. 5.8. Close tolerance seal.

The leakage past such a seal will be found to be proportional to the cube of the clearance between the piston and the cylinder. Therefore, it is essential that the clearance be reduced to the absolute minimum that will allow relatively free sliding of the piston in the cylinder. In producing the seal finish, grinding will be used to bring the unit close to nominal dimensions followed by multi-stage lapping and honing to final size. Thermal expansion effects will require the use of similar materials for the cylinder and the piston. Best results are likely to be attained with hard rubbing surfaces attained by hard anodized aluminum surfaces or by hard chrome plate or carburized steel surfaces.

In all the above cases, particularly where very high pressures are to be used, it will most likely be advantageous to construct the cylinder as a composite assembly of two components: (a) a thin-walled inner liner of high internal concentricity, with the appropriate internal plated or anodized finish and supported at the ends of (b) an enclosure serving to contain the pressure. Standard hydraulic cylinders are very suitable for adaptation as the internal high concentricity liner in Stirling engine cylinder assemblies.

Whatever the design of surface treatment adopted it will often be helpful to apply molybdenum disulphide powder to the rubbing surfaces before their operation.

All of the seals discussed above may be used in experimental prototype engines and are likely to provide a variety of experiences. Seals that are apparently identical will behave in different ways and, furthermore, will

behave differently from one day to another in ways that can be exceedingly tedious and frustrating.

The absence of a design that will give reproducible friction and leakage characteristics, combined with high cost and practical skill in manufacture and use of the above seals, means that none of these are feasible for use in production engines. A variety of commercial seal designs have been evaluated in rig tests with disappointing results. In some cases the sliding friction was high or in others the leakage was not tolerable. Furthermore, the same order of uncertainty was noted in the predictability of the results. There is a real and urgent need for further work in seal design for Stirling engines.

Whatever the design of seal developed, it is probable that the friction power absorbed and the seal life will be a strong function of the rubbing velocity, u. For a given engine displacement V_0, the length of stroke is:

$$L = \frac{4V_0}{\pi D^2} \tag{5.13}$$

The mean rubbing velocity is:

$$u = 2Ln \tag{5.14}$$

where n = engine speed in cycles per second. From the above it can be seen the rubbing velocity is therefore inversely proportional to the square of the cylinder diameter. The rubbing velocity can evidently be reduced by enlarging the cylinder diameter but of course this has the undesirable effect of increasing the circumference of the seal (in direct proportion to the diameter) which increases both the friction force and the length of the leakage path. A good compromise is a design where the stroke is about half the cylinder diameter.

Percival (1974) has provided extensive discussion of the experience at General Motors with seals for Stirling engines and is recommended for close study.

OIL CONTAINMENT

All of the above has been directed to the problem of minimizing leakage of the working fluid from the cylinder. Another sealing function, equally important, is to prevent egress of oil lubricant *into* the working space. Leakage of oil into the engine cylinder is bad because, eventually, it goes into suspension as fine droplets in the working fluid and is thereupon 'filtered' out by the finely divided metallic matrix of the regenerator. This partial blockage of the matrix pore space increases the pressure drop through the regenerator which reduces the power output of the engine. The temperature of the hot parts then increases because less heat is absorbed in expansion. Perhaps increased fuel may be supplied to

ture of the hot parts. The increase in temperature causes the oil trapped in the regenerator to carbonize which further increases the pressure drop. This progressive and cumulative deterioration in performance continues until eventually the engine becomes overheated and either the heater tubes or the regenerator matrix 'burn out'.

It is tempting to avoid these problems by using pre-packed, sealed, grease-lubricated bearings, or 'Oilite' phosphor-bronze, oil-impregnated bearings, or the many proprietary Teflon or Rulon Bearings. These are perfectly acceptable approaches in experimental or prototype engines and for short operating periods. For long-term operation, none of these has proved satisfactory so far, and it is likely that lubricating oil will have to be used for the bearings in the crank mechanism. In that case the problem of preventing egress of oil to the working space will need to be confronted and will most likely be found a particularly difficult problem to overcome.

Carbonaceous materials, frequently impregnated with bronze, silver, or other metal powders are used as oil-free bearing materials in many applications. Unfortunately carbon loses its favourable lubrication properties in very dry atmospheres. Little published data is known about the use of carbon-based materials as bearings with hydrogen or helium working fluids at high pressure. Attempts by the author to incorporate carbon-based materials in the seals and bearings of small cooling engines met with a mixed and on the whole, unfortunate experience.

There is a real need for fundamental friction and wear studies of compound materials in non-oxygenated atmospheres. If such work has already been done it is not sufficiently well-known to those who could profit from its application.

One of the many attractions of free-piston Stirling engines is the opportunity they provide to operate with gas lubricated bearings using the working fluid itself as the lubricant. No oil or grease is involved. Another significant advantage of free piston machines is the elimination of wear due to side thrust arising from the crank mechanism.

BEARINGS

In Stirling engines the pressure ratio p_{max}/p_{min} rarely exceeds 2, and the rate of change of the pressure is very low compared with internal combustion engines. This means that loads on the crankshaft bearings are likely to be much below the loadings that are commonplace in other piston engines. The bearings can therefore be reduced substantially in size compared with those that 'look to be about right' using judgment based on internal-combustion experience.

Learning to design and build Stirling engines for their own sake rather

lesson. This is particularly true of bearing and crankshaft design. The standard texts on internal combustion engine design and on the design of machine elements all contain sections dealing with the analysis of inertia and pressure forces in reciprocating engine systems. Careful attention to these procedures is necessary to ensure proper selection of bearings, crankpins and connecting rods of appropriate dimension. Newcomers should be warned that the dimensions and proportions calculated will probably appear to be too small and the calculated design dimensions will leave a feeling of disquiet until one becomes familiar and confident with the operation of the engine. This will be found particularly applicable for designers and engineers experienced in internal combustion engine practice.

PRINCIPAL DESIGN PARAMETERS

The principal independently-chosen design parameters of a Stirling engine are:

(1) The temperature ratio $\tau = T_C/T_E$, the ratio of temperatures in the compression and expansion spaces.
(2) The swept-volume ratio $\kappa = V_C/V_E$, the ratio of swept volume in the compression and expansion spaces.
(3) The dead-volume ratio $X = V_D/V_E$, the total internal volume of heat exchangers (and associated ducts and ports) expressed as a multiple of the swept volume in the expansion space.
(4) The phase angle α by which volume variations in the expansion space *lead* those in the compression space.
(5) The pressure of the working fluid, expressed as the maximum or mean pressure.
(6) The speed of the engine N.
(7) The bore and stroke of the reciprocating member in the expansion space.

Summary of Schmidt-cycle design equations

(8) Instantaneous volume of expansion space $V_e = \frac{1}{2}V_E(1+\cos\phi)$.
(9) Instantaneous volume of compression space $V_c = \frac{1}{2}\kappa V_E[1+\cos(\phi-\alpha)]$.
(10) Instantaneous volume of total working space $= V_e + V_c + V_D$.
(11) Instantaneous pressure $p = p_{max}(1-\delta)/[1+\delta\cos(\phi-\theta)]$,

$$\text{where } \delta = (\tau^2 + 2\tau\kappa\cos\alpha + \kappa^2)^{\frac{1}{2}}/(\tau+\kappa+2S)$$
$$\tan\theta = \kappa\sin\alpha/(\tau+\kappa\cos\alpha)$$
$$S = 2X\tau/(\tau+1).$$

(12) Pressure ratio $p_{max}/p_{min} = (1+\delta)/(1-\delta)$.
(13) Mean pressure $p_{mean} = p_{max}[(1-\delta)/(1+\delta)]^{\frac{1}{2}}$.

For a prime mover, $T_E > T_C$.

(14) Net cycle-power per cycle

$$P = (p_{max}V_T)\pi\frac{(1-\tau)}{(\kappa+1)}\left(\frac{1-\delta}{1+\delta}\right)^{\frac{1}{2}}\frac{\delta\sin\theta}{[1+(1-\delta^2)^{\frac{1}{2}}]}$$

where $V_T = (V_E + V_C) = (1+\kappa)V_E$.

(15) Power per unit mass of working fluid

$$P_{mass} = \frac{\pi(1-\tau)(1+\delta\cos\theta)(\delta\sin\theta)}{(1-\delta^2)^{\frac{1}{2}}[1+(1-\delta^2)^{\frac{1}{2}}]\left[\tau+\frac{\kappa}{2}(1+\cos\alpha)+S\right]}$$

and thermal efficiency

$$\eta = \frac{T_E - T_C}{T_E} = (1-\tau).$$

(16) Heat transferred in expansion space, per cycle,

$$Q_E = \pi p_{mean}V_E\frac{\delta\sin\theta}{1+(1-\delta^2)^{\frac{1}{2}}}.$$

(17) Heat transferred in the compression space, per cycle,

$$Q_C = -\tau Q_E.$$

For a refrigerating machine, $T_E < T_C$.

(18) Heat lifted in cold expansion space, per cycle,

$$Q_E = (p_{max}V_T)\frac{\pi}{(\kappa+1)}\left(\frac{1-\delta}{1+\delta}\right)^{\frac{1}{2}}\frac{\delta\sin\theta}{[1+(1-\delta^2)^{\frac{1}{2}}]}.$$

(19) Heat lifted per unit mass of working fluid

$$Q_{mass} = \frac{\pi(1+\delta\cos\theta)\delta\sin\theta}{(1-\delta^2)^{\frac{1}{2}}(1+(1-\delta^2)^{\frac{1}{2}})\left[\tau+\frac{\kappa}{2}(1+\cos\alpha)+S\right]}$$

(20) Coefficient of performance (COP)

$$= \frac{T_E}{(T_E - T_C)} = \frac{1}{(1-\tau)}.$$

(21) Heat transferred from the compression space (to cooling medium)

$$Q_C = \tau Q_E.$$

For a heat pump, $T_E < T_C$.

(23) Heat transferred from the hot (compression) space per cycle,

$$Q_C = (p_{max} V_T) \frac{\pi\tau}{(\kappa+1)} \left(\frac{1-\delta}{1+\delta}\right)^{\frac{1}{2}} \frac{\delta \sin\theta}{[1+(1-\delta^2)^{\frac{1}{2}}]}$$

(24) Heat transferred from the hot (compression) space, per unit mass of working fluid,

$$Q_{mass} = \frac{RT_c \pi\tau(1+\delta\cos\theta)\delta\sin\theta}{(1-\delta^2)^{\frac{1}{2}}(1+(1-\delta^2)^{\frac{1}{2}})\left(\tau+\frac{\kappa}{2}(1+\cos\alpha)+S\right)}$$

(25) Coefficient of performance (COP) $= T_C/(T_E - T_C) = \tau/(1-\tau)$.

(26) Heat transferred from the expansion space (heat source),

$$Q_E = Q_C/\tau.$$

(27) Power required to drive the heat pump,

$$P = Q_C(1-\tau)/\tau.$$

Optimization of design parameters

It is obvious from the Schmidt-cycle equations summarized above that the net cycle power and the thermal loads on the heat exchangers are direct linear functions of engine *speed* (N), *pressure* of the working fluid (p_{max}), and *size* of the engine, expressed in terms of the total swept volume V_T. The effect which the four principal design parameters (τ, κ, α and X) have on performance is less obvious. In particular, it is not clear what combination of these should be used to achieve optimum performance. This is an important consideration, since these parameters must be determined at the design stage and, except for the temperature ratio τ, cannot readily be varied afterwards, except by structural changes in the machine.

Figs. 5.9 to 5.12 show the effect on the cycle-power parameter $P/(p_{max} V_T)$ of independent variation of one of the four parameters τ, κ, α and X, with the other three maintained constant. In Fig. 5.9 the effect of the temperature ratio on cycle power is explored for expansion-space temperatures (T_E) both above and below the assumed compression-space temperature of 300 K, thus embracing both refrigerating machines and prime movers. With $T_E > T_C$, the power parameter is positive, and progressively increases as the expansion-space temperature increases. When $T_E < T_C$, the machine is acting as a refrigerator and, as the expansion-space temperature decreases, there is a progressive increase in

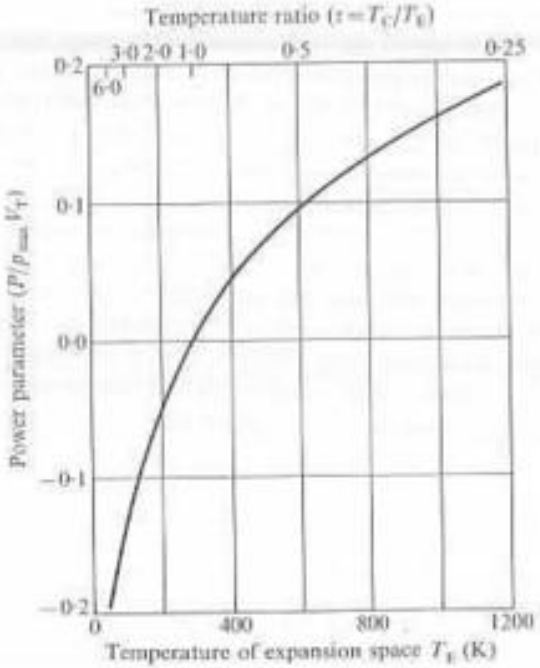


FIG. 5.9. Effect of temperature on cycle power. The figure shows the effect on the non-dimensional power parameter $P/(p_{max} V_T)$ of variation in the expansion space temperature T_E, with T_C constant at 300 K, phase angle $\alpha = 90°$, swept-volume ratio $\kappa = 0.8$ and dead-volume ratio $X = 1.0$. At expansion-space temperatures below 300 K, $\tau > 1.0$ and the power parameter is negative, because the cycle is acting as a cooling cycle, requiring a net input of work.

available power may be increased by the use of high-temperature materials for the expansion-space cylinder and heat exchanger, and for *refrigerators* the temperature of refrigeration should be as high as possible.

Fig. 5.10 shows the effect of the swept-volume ratio κ on the power parameter. The curves show clearly that, for given values of τ, α, and X, there is a definite optimum value of κ at which the power parameter is a maximum. Comparison of the two curves for $\tau = 0.25$ and 0.5 shows, however, that the optimum value of κ is not constant, but changes from about 0.75, when $\tau = 0.25$, to about 1.0, when $\tau = 0.5$. Changes in α and X also produce an adjustment of the optimum value of κ. Thus, there is no single 'best' value of κ.

Fig. 5.11 shows the effect of the dead-volume ratio $X = V_D/V_E$ on the power parameter. The message of this figure is very clear: increase in the dead space above the absolute minimum required reduces the power

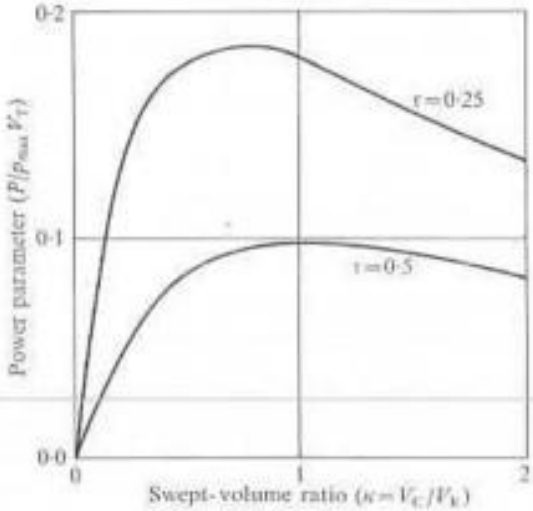


FIG. 5.10. Effect of the swept-volume ratio κ on cycle power. The figure shows the effect on the non-dimensional power parameter $P/(p_{max}V_T)$ of variation in the swept-volume ratio κ, with constant values of the temperature $\tau = 0.25$ and 0.5, phase angle $\alpha = 90°$, and dead-volume ratio $X = 1{\cdot}0$. On the two curves shown, for different values of τ, the maxima do not occur at the same value of κ. There is no single 'best' value for κ since the optimum value depends on the combinations α, τ, and X.

shows the effect of the phase angle α on cycle power. The power parameter is remarkably insensitive to changes in the phase angle over an extended range from 60° to 120° of crank rotation. For the particular assumed conditions the optimum value was between 90° and 115°.

A three-dimensional presentation is given in Fig. 5.13 of the variations in engine-power parameter with change in both phase angle α and swept-volume ratio κ, for constant values of τ and X. This presentation results in the generation of a solid surface. Any changes in τ or X cause the generation of a series of similar, but different, overlaying surfaces. The apex of the surface represents the maximum value of the power parameter, and occurs at the optimum combination of swept-volume ratio and phase angle, for the given values of τ and X. Fig. 5.13 shows two surfaces, generated with the different power parameters $P/(p_{max}V_T)$ and P_{mass}. For these two surfaces, the apex occurs at different combinations of phase angle and swept-volume ratio. In the case of the surface drawn for the power parameter $P_{mass} = P/MRT_C$, $\alpha_{opt} = 0.45\ \pi$ rad and $\kappa_{opt} = 2.9$. In the case of the surface drawn for the power parameter $P_{mass} = P/(p_{max}V_T)$, $\alpha_{opt} = 0.54\ \pi$ rad and $\kappa_{opt} = 0.74$. A simple explanation for this strange phenomenon is that optimization of design on the basis of the power parameter P_{mass} results in a machine configuration that makes the best possible use of a *limited mass of working fluid*. Optimization of

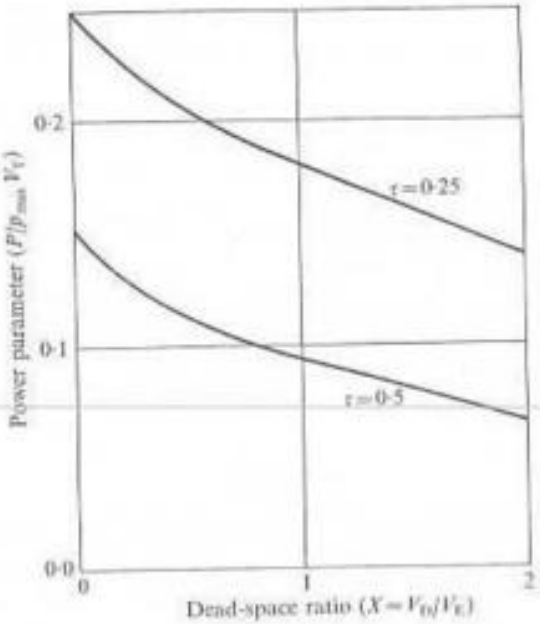


FIG. 5.11. Effect of the dead-space ratio X on cycle power. The figure shows the effect on the non-dimensional power parameter $P/(p_{max}V_T)$ of variation in the dead-space ratio X, with constant values of τ, κ, and α. The dead space is the porous volume of the regenerator and other heat exchangers, 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.

design on the basis of parameter $(P/p_{max}V_T)$ results in a machine configuration of the maximum possible power within limits of the *maximum pressure and combined swept volume*. The maximum pressure of the working fluid is an important design criterion, since this affects the strength, and hence the *weight*, of the machine structure, whereas the combined swept-volume V_T is indicative of the *size* of the structure. Clearly then, the power parameter $P/(p_{max}V_T)$ should be used for optimization purposes. However, once the basic machine configuration has been determined in terms of α, κ, τ and X, the power parameter P_{mass} may be used thereafter. Both it and $P/(p_{max}V_T)$ are equally applicable, and will return the same numerical values for the engine power per cycle P.

By way of example, Fig. 5.14 is a comparison of the work diagram obtained for the optimum combination of phase angle α and swept-volume ratio κ, with constant values of τ and X. In all three cases, the pressures are adjusted to be fractions of the maximum cycle-pressure, and so are comparable. Similarly, in all three cases, the maximum value of the

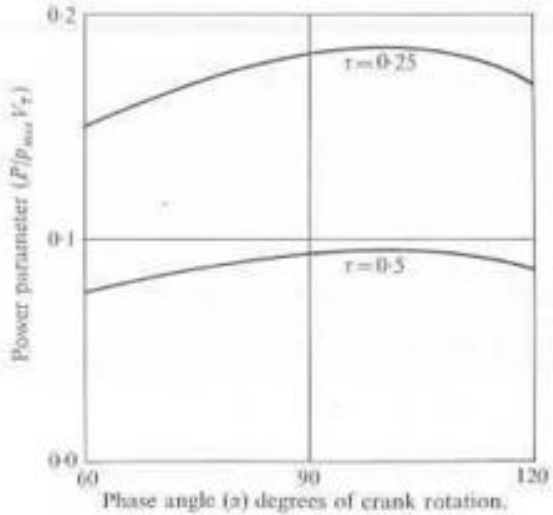
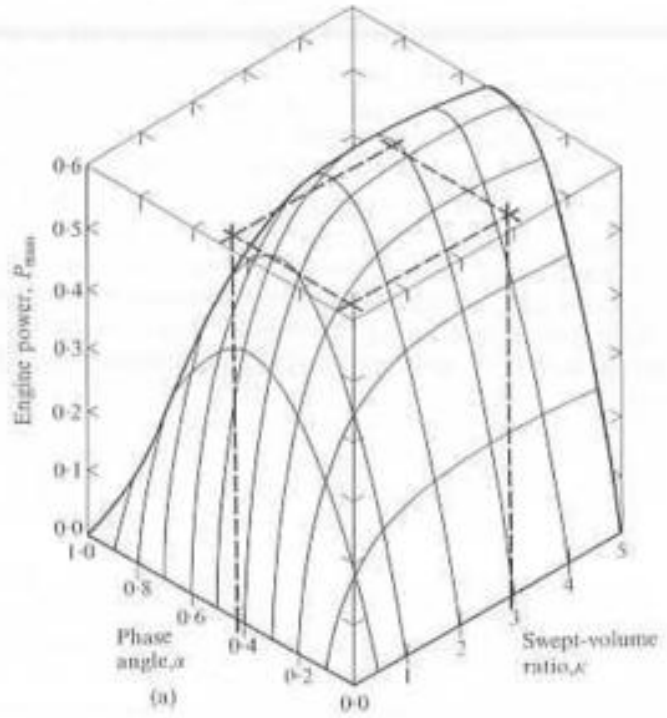
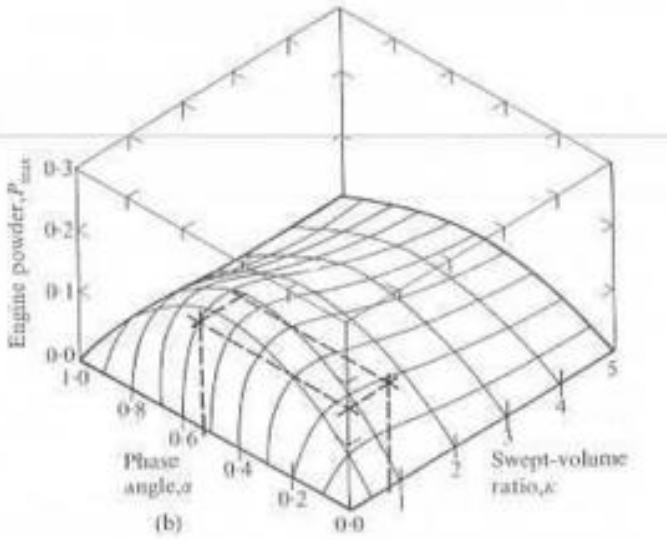


FIG. 5.12. Effect of the phase angle α on cycle power. The figure shows the effect on the non-dimensional power parameter $P/(p_{max}V_T)$ of variation in the phase angle α, with constant values of τ, κ and X. The power parameter is remarkably insensitive to variation in α. This permits considerable flexibility in the geometrical design of Stirling-engine drive mechanisms.

total enclosed volume has been made identical at an arbitrary value of 4.6. In each case, the diagram at the extreme left is the work diagram for the expansion space, the one in the middle is for the compression space, and the diagram at the right is the total enclosed volume. Fig. 5.14(a) is the diagram resulting from the combination $\alpha = 0{,}45\pi$ rad, $\kappa = 2.9$ $\tau = 0.3$, and $X = 1.0$, the optimum configuration based on P_{mass}. Fig. 5.14(b) is the diagram resulting from a combination $\alpha = 0.54\pi$ rad, $\kappa = 0.74$, $\tau = 0.3$, and $X = 1.0$, the optimum combination based on $P/(p_{max}V_T)$. Fig. 5.14(c) is the diagram resulting from a modification to the configuration of Fig. 5.14(b) to adjust the absolute value of the dead space volume V_D to be the same as in the configuration of Fig. 5.14(a).

FIG. 5.13. Effect of swept-volume ratio κ and phase angle α on cycle power. The figures are a three-dimensional representation of non-dimensional power parameters as a function of the swept-volume ratio κ and the phase angle α, with constant values of the temperature ratio $\tau = 0.3$, and the dead-space ratio $X = 1.0$. Fig. 5.13(a) is a representation for the power parameter P_{mass} (power per unit mass of working fluid) and Fig. 5.13(b) is a representation for the power parameter $P/(p_{max}V_T)$ (power for limited size and weight). The apex of the surfaces (the maximum value of the power parameter) occurs at the *optimum* values of phase angle α and swept-volume ratio κ. In case (a) $\alpha_{opt} = 0.45\ \pi$ radians and





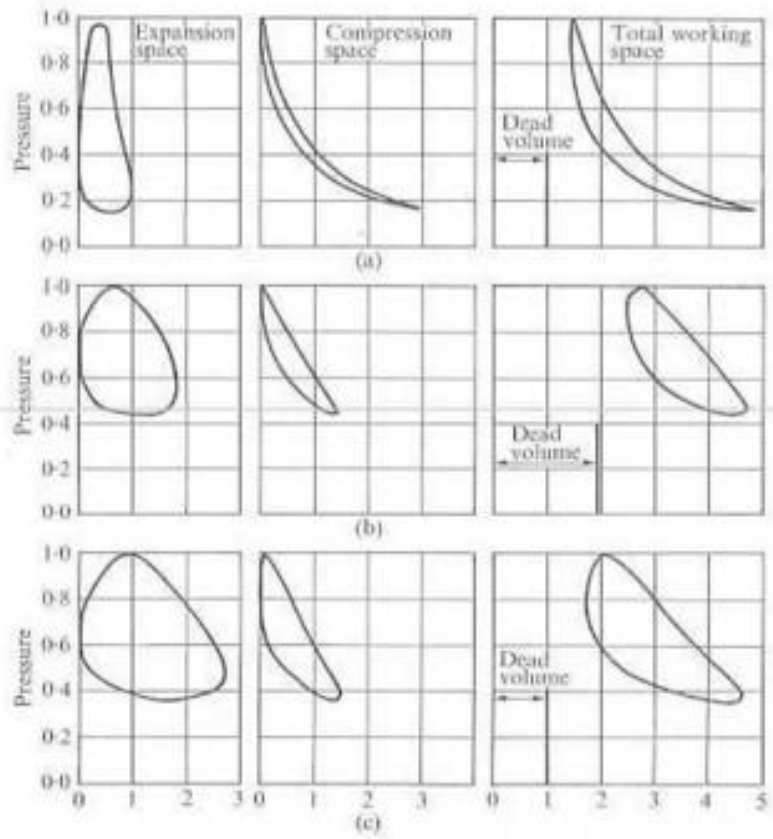


FIG. 5.14. Work diagrams for engines having the optimum combination of design parameters. The figure shows work diagrams for engines having optimum combinations of the design parameters α, λ τ, and X, as determined by reference to the three-dimensional representation shown in Fig. 5.13. In all cases, the diagram at left is for the expansion space, the centre diagram is for the compression space, and the diagram at right is for the total working-space. Fig. 5.14(a) shows work diagrams for the cycle, optimized using the power parameter P_{exmax}, with $\alpha_{\text{opt}} = 0.45\ \pi$ radians, $\kappa_{\text{opt}} = 2.9$, $\tau = 0.3$, and $X = 1.0$, as determined in Fig. 5.13(a). Fig. 5.14(b) shows work diagrams for the cycle, optimized using the power parameter $P/(p_{\max} V_T)$, with $\alpha_{\text{opt}} = 0.54\ \pi$ radians, $\kappa_{\text{opt}} = 0.74$, $\tau = 0.3$, and $X = 1.0$ as determined in Fig. 5.13(b).† Fig. 5.14(c) shows work diagrams for a cycle having a combination similar to Fig. 5.14(b), but with the dead space reduced to the same overall value as in Fig. 5.14(a) (same size of regenerator and heat exchangers). The net cycle output of this case is superior to case (a), by an even greater margin than case (b).

† For purposes of comparison, all the work diagrams are drawn with pressures as fractions of the maximum pressure, and with the same maximum value of the total working-space.

The two machines represented in Figs. 5.14(a) and 5.14(b) are comparable therefore in terms of size and weight. The maximum pressure and the maximum total enclosed volume are the same, although the dead volume in one machine is nearly twice as great as in the other. Despite this, the net work output of the machine in Fig. 5.14(b) is 1.38 times that of the machine in Fig. 5.14(a). When the dead volume of Fig. 5.14(b) is reduced to an amount comparable with that of Fig. 5.14(a) (as in machine represented by Fig. 5.14(c)) the ratio of net work output increases to 2.24 in favour of the machine in Fig. 5.14(c). This example is a convincing justification for optimization on the basis of the power parameter $P/(p_{\max} V_T)$.

Walker (1962) has drawn a corresponding comparison for refrigerating machines. Use of the cooling parameter $Q_E/(p_{\max} V_T)$ for optimization studies is preferred, because it leads to a machine configuration having the maximum cooling capacity for a given size and weight.

CONSOLIDATED DESIGN CHARTS

Despite the interest and attraction of diagrams such as Fig. 5.13, it can readily be understood that there exists a virtually infinite array of possible permutations of engine design parameters. It would be a tedious matter to search through the variations for an optimum combination. To overcome the difficulty, consolidated design charts have been prepared, and are presented in Fig. 5.15 (for prime movers) and Fig. 5.16 (for refrigerating machines).

Fig. 5.15 was prepared using the power parameter $P/(p_{\max} V_T)$ as the basis for optimization. Surfaces, similar to that shown in Fig. 5.13, were generated for the value of $P/(p_{\max} V_T)$ with different values of α and κ and constant values of τ and X. The apex of the surface was established, and the apex values of $P/(p_{\max} V_T)$, α_{opt}, and κ_{opt} were plotted on appropriate charts, constituting Fig. 5.15. These were all drawn on the common basis of expansion-space temperature T_E, with the compression-space temperature *always* maintained constant at 300 K. The apexes of other surfaces, with different values of τ and X, were determined, and plotted to obtain the complete consolidated chart. A similar technique was used to obtain the consolidated chart for the refrigerating machines, with optimization based on the cooling parameter $Q_E/(p_{\max} V_T)$ (Fig. 5.16).

The work was carried out, using a self-optimizing digital computer program (with automatic use of recognized hill-climbing techniques), to locate the apex of the surface from any given fixed values of τ and X and starting values of α and κ, as described by Walker (1962).

Use of the consolidated chart for design

The design charts of Figs. 5.15 and 5.16 are recommended for use in

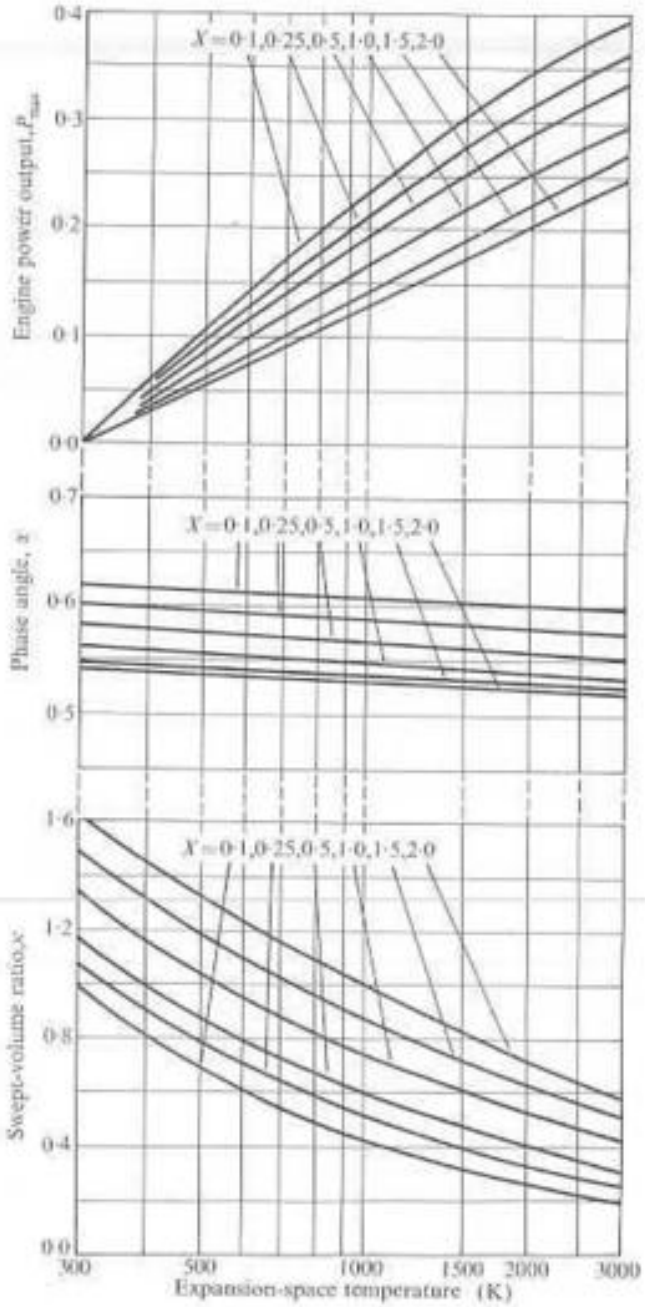


FIG. 5.15. Design chart for Stirling engines. The optimum combination of swept-volume ratio κ and phase angle α (in π radians) may be determined for given values of the dead-space ratio X and the temperature ratio τ. Assuming $T_C = 300$ K, and with the metallurgical limit controlling the expansion space temperature T_E, draw a vertical line through T_E. Intersection of this line with the selected value for X occurs at the optimum values for κ and α. The appropriate value of the non-dimensional power parameter $P/p_{max} V_T$ is determined from the upper diagram. Practical engines may produce 0.3 to 0.4

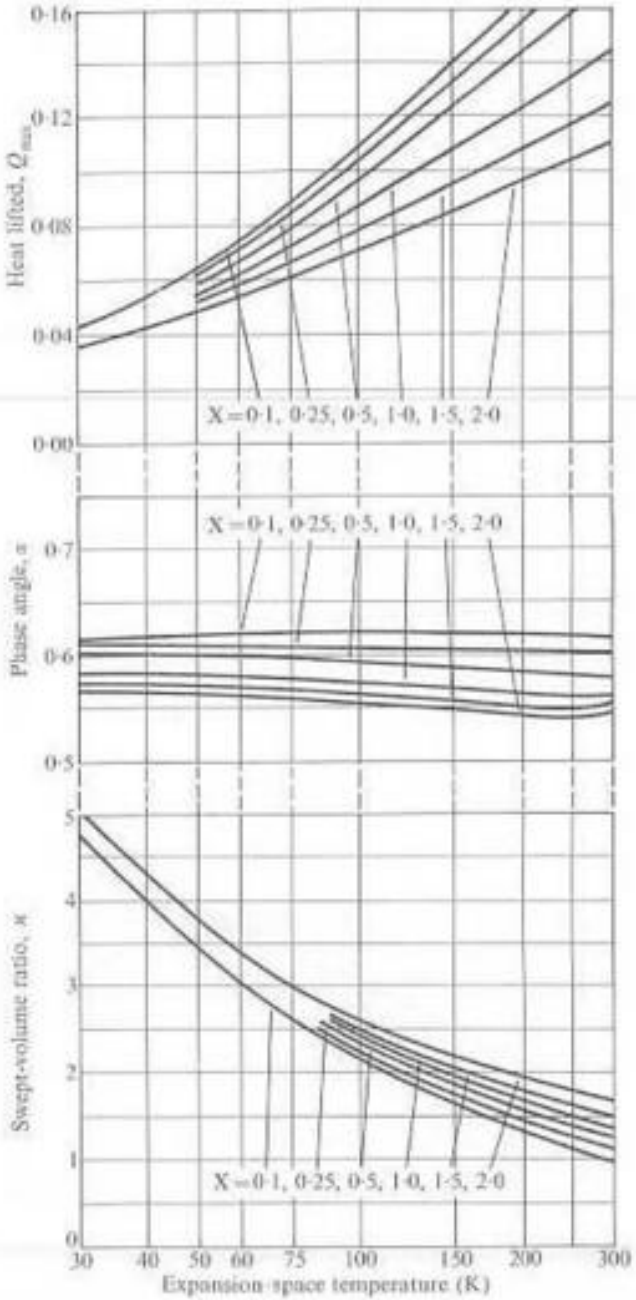
In the case of a prime mover, it is necessary first to establish the permissible temperature in the expansion space T_E. This is dictated by the nature of the thermal source, and by the materials to be used for the expansion-space heat exchanger and expansion cylinder. At the chosen temperature, a vertical line is drawn through the three charts. Where this line intersects the sets of curves, and at an appropriate value of the dead-space ratio X, the corresponding optimum value of both phase angle α and swept-volume ratio κ may be determined from the scales. The value of the engine-power parameter $(P/p_{max} V_T)$ may be determined also. With T_E known, T_C may be estimated (it is about 300 K with a water-cooled engine), so that τ can be calculated. With τ, X, α and κ thus established it is possible to proceed to the detailed design of the engine, utilizing the summary of design equations given earlier.

It cannot be over-emphasized that the predictions of Schmidt-cycle calculations are highly optimistic. Experience suggests that it is unwise to expect from a practical engine more than 30 to 60 per cent of the power and efficiency predicted by Schmidt-type analyses.

WORKING FLUID

In the Schmidt theory no explicit account is taken of the physical characteristics of the working fluid, except its behaviour as a perfect gas (i.e. it obeys the characteristic gas equation $PV = RT$). However, the assumptions on which the theory is based imply the use of an idealized working fluid, having properties not found in nature. The assumption that there are no aerodynamic-friction losses could only be true if the working fluid were to have zero viscosity. Similarly, the assumptions of perfect regeneration and isothermal compression and expansion can only be attained if the working fluid were to have unreal values for specific heat and thermal conductivity.

In practice there appear to be only three working fluids of significant interest: air, helium, and hydrogen. Air is of interest because it is so freely available. Helium and hydrogen are of interest because their thermophysical characteristics are such as to permit high rates of heat transfer and flow to occur, with relatively low aerodynamic-flow losses. In terms of engine performance, hydrogen is better than helium, and is also



very much cheaper, but is highly combustible in the presence of air or oxygen.

Engines of high specific output and high thermal efficiency, operating at high pressures and high speeds (i.e. greater than 2000 rev/min), must use hydrogen or helium as the working fluid, in order to achieve the rates of heat and mass transfer necessary, with tolerable flow losses. The sealing problems are very severe, however. Furthermore, the control systems needed to vary engine output are complicated, since they must incorporate reservoirs, valves, and, perhaps, a compressor to vary the pressure level, while conserving the working fluid. The cost of machines of this type is high, and applications are likely to be limited to relatively large engines, where the advantages of low noise (and pollution) levels justify increased cost, compared with internal-combustion engines. Cooling engines of high output (or those intended for refrigeration at cryogenic temperatures) must also use helium or hydrogen as the working fluid.

Engines using air as working fluid cannot achieve the high rates of heat and mass transfer found in hydrogen or helium engines. Such machines are, typically, large heavy engines of low specific output and low thermal efficiency. However, the working fluid can readily be replenished from atmospheric air, so that the sealing and materials problems are substantially eliminated, and the machines can be simple, cheap, and reliable. Air engines have such a poor performance that they offer no serious competition to internal-combustion engines, in either automotive or general-purpose applications. There is, however, an urgent and increasing need for low-power (less than one horsepower) engines of high reliability and moderate efficiency, capable of operating unattended for long periods (in excess of one year) and utilizing fossil, or radioisotope, fuels. The engines are required to drive electric-power generators for navigational, meteorological, and telecommunications purposes. Stirling-cycle air engines appear admirably suited for this purpose.

The comparative performance of Stirling engines with air, hydrogen, and helium is shown in Fig. 5.17. This is a reproduction of material presented by Meijer (1970a), based on advanced simulation calculations for a single-cylinder Stirling engine of 165.5 kW (225 hp). The figure shows how the engine's thermal efficiency is related to power density (in terms of kilowatt per litre of cylinder swept-volume) at different speeds and with three different working fluids (air, hydrogen, and helium). At high power densities and high speeds, hydrogen is appreciably better than helium,

FIG. 5.16. Design chart for Stirling-cycle cooling engines. Optimum values for swept-volume ratio κ and phase angle α may be determined for given values of the dead-space ratio X and the expansion-space temperature T_E, with $T_c = 300$ K, by drawing a vertical line on the chart through T_E. The cooling capacity in terms of heat lifted in the expansion space, $Q_{exp} = X$, will be optimized by a factor of about κ

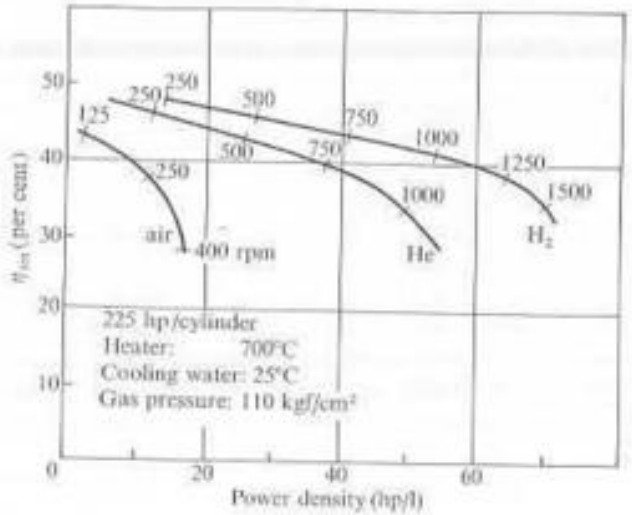


FIG. 5.17. Comparative performance of Stirling engines with air, hydrogen, and helium as the working fluid (after Meijer 1970). The figure shows that, at high speeds and high power-densities, hydrogen is the preferred working fluid, with helium the second choice. At low powers and low speeds, air may be used in small engines with little loss of performance and appreciable practical advantages in terms of fluid sealing, replenishment, and simplicity of design.

and the curve for air is not able to reach that part of the diagram. However, the important point to note is that, at *low speeds* and *low power densities*, there is no really significant difference between air, helium, and hydrogen. The selection of the working fluid is an important decision which has to be taken at the design stage, and requires the intended application of the engine to be clearly defined. If the intention is to make a high-speed high-performance engine then hydrogen or helium must be used, but if modest speed and performance can be tolerated then the use of air as the working fluid has considerable attractions, on grounds of simplicity and cost.

It is possible that other working fluids may be used in the future. Of interest, at present, are reacting fluids and two-phase two-component fluids. A more extensive discussion of working fluids will be found in Chapter 8.

6 MECHANICAL ARRANGEMENTS

INTRODUCTION

THE elements of a Stirling engine include two spaces at different temperatures having volumes that can be varied cyclically and which are connected through a regenerative heat exchanger and auxiliary heat exchangers. These simple elements can be combined in a surprisingly wide range of mechanical arrangements. Some have been identified (Finkelstein 1959) by the name of the inventor or original user. Many variations were used in the nineteenth century and have been adopted or re-invented for application to modern engines. In other cases novel mechanisms or embodiments previously unknown are used. New arrangements are still being devised, some are good, some bad; only time will tell which will attain commercial application.

The key identifying feature of any regenerative system is the manner in which the flow of working fluid is controlled. There are two possibilities; flow is controlled either by valves or by volume changes. In many respects the two types of machines are similar but in details of construction, operation, and fields of application, they are quite different.

The use of valves for flow regulation has the advantage of increased flexibility in flow control and timing with the possibility of pressure ratios (p_{max}/p_{min}) that are virtually unrestricted. On the other hand the presence of valves adds to the mechanical complexity of the system, provides sources of noise, and additional points of wear, so the possibility for long life with low maintenance is likely to be reduced.

In this work the name Stirling engine is limited to regenerative engines where the flow is controlled by volume changes. Machines where the flow is controlled by valves are called Ericsson engines. These names are chosen somewhat arbitrarily in an attempt to rationalize a very confused situation. No agreed distinctions have been established in the general literature and 'Stirling engine' is used as a generic title for all types of regenerative engines.

The names Stirling and Ericsson are themselves not entirely satisfactory for they suggest operation on ideal cycles having isothermal compression and expansion with regenerative transfer processes that are either constant pressure or constant volume. Practical engines operate with conditions far removed from those represented by the ideal cycle.

So far as is known, all the engines devised by Robert Stirling were of the closed-cycle type where the flow is controlled by volume changes,

Design variants of Stirling engines

Stirling engines can be broadly classified into two distinct families which may be identified as: (a) single-acting or (b) double-acting engines.

Single-acting engines are an ensemble of expansion space, compression space and associated heat exchangers in one or two cylinders with two reciprocating elements, one of which must be a piston. The other may be a piston or a displacer (see Chapter 5 for the distinction). Every ensemble constitutes a complete system which operates independently of any other single-acting systems that may be coupled on a common crankshaft or other kinematic mechanism.

The celebrated Philips rhombic-drive engine, invented by R. Meijer, is an example of a single-acting Stirling engine. It may be more completely described as a single-acting, single-cylinder, piston-and-displacer Stirling engine with rhombic-drive mechanism. Rhombic engines have been made in sizes ranging from miniature engines having output powers of a few watts to large engines of 100 horsepower per cylinder. They may operate as single-cylinder engines or in multiple units arranged on a common crankcase and connected to a common crankshaft. A well known multiple-cylinder, rhombic-drive engine was the Philips Type 4-235, four-cylinder, in-line engine, of about 100 horsepower installed as the propulsion motor in a DAF bus (see Chapter 12) and another was installed in a motor launch.

DESIGN VARIANTS OF DOUBLE-ACTING ENGINES

Double-acting engines, shown in Fig. 6.1, are ensembles of multiple cylinders arranged such that the expansion space of one cylinder is connected through the associated heat exchangers to the compression space located in an adjacent cylinder. There is one reciprocating element per cylinder, a piston-displacer. The number of Stirling-engine systems is equal to the number of cylinders. The great advantage of double-acting engines is that the number of reciprocating elements is *half* the number required in multiple arrangements of single-acting machines. This can lead to major simplification in the kinematic drive arrangements and so to reduced cost. The principal disadvantage is the limited flexibility in design and, to a lesser extent, operating conditions. Furthermore, in prototype development it is necessary to proceed with the whole multiple-cylinder engine rather than an experimental single-cylinder unit which can then be reproduced in multiple versions. Nevertheless the elimination of half the moving parts is so compelling that all significant development of large engines (over 20 horsepower) is currently limited to double-acting en-

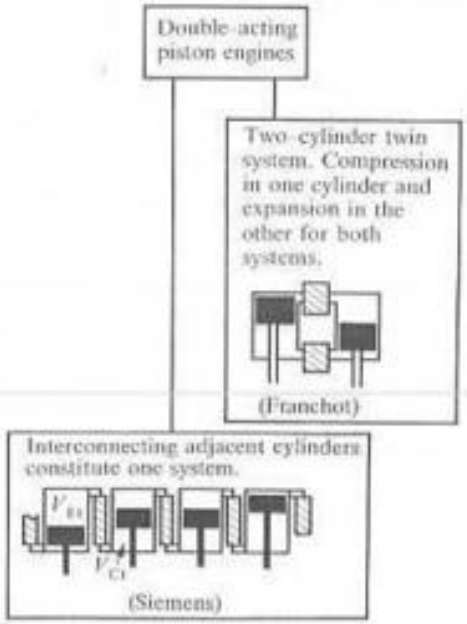


FIG. 6.1. Alternative arrangements of double-acting multiple cylinder engines.

gines. Single-acting engines will doubtless continue to dominate the small-engine field.

Single-acting Stirling engines were invented early in the nineteenth century, and the application in 1815 by Robert Stirling for his first engine patent may perhaps be reckoned an appropriate date. Other variations and arrangements of single-acting Stirling engines have been conceived at intervals since.

Double-acting Stirling engines were also invented in the nineteenth century. Babcock (1885) ascribes the invention of the two-cylinder, twin-system, double-acting engine to the French engineer, Charles Louis Franchot in 1853. This was subsequently re-invented a century later by Finkelstein and Polanski (1959) and was later incorporated in a multiple-cylinder, free-piston arrangement by Finkelstein (1963b). The general form of double-acting engines with three or more cylinders is ascribed by Babcock to the celebrated British scientist/engineer Sir William Siemens, better known for his work in steel making. Siemens designed the double-acting Stirling engine shown in Fig. 6.2, having four cylinders each containing a single piston and connected to a wobble plate drive. So far as is known, Sir William's concept was never reduced to practice, and it

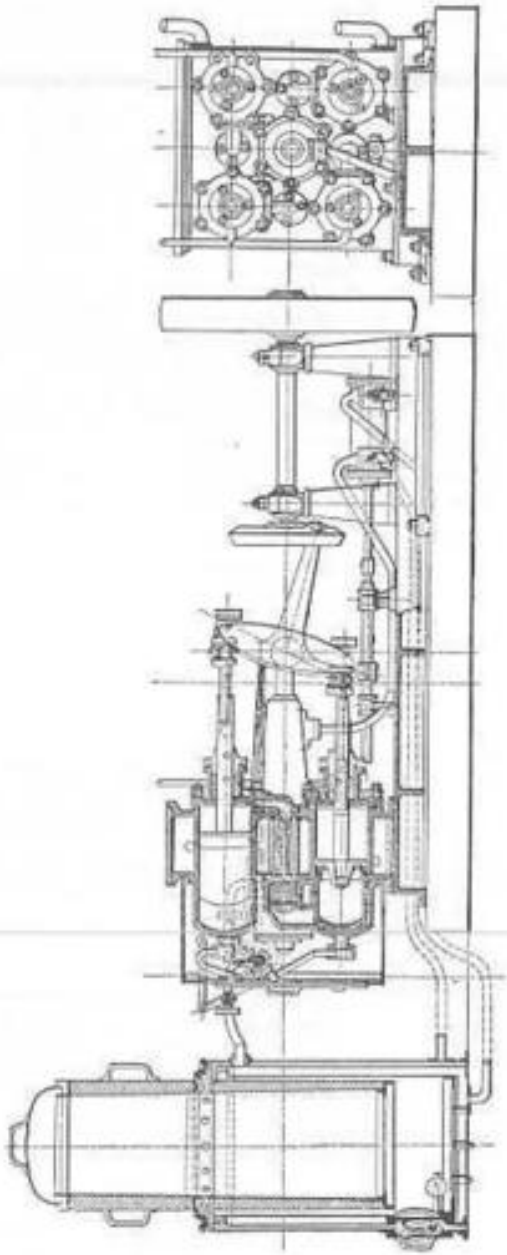


FIG. 6.2. Four-cylinder double-acting engine invented by Sir William Siemens in 1863 (after Babcock 1975).

invented 80 years later by van Weenan (1949) in the early days of the Philips program.†

Different versions of Siemens engines are currently the focus of development effort in Europe and the United States. The Philips/Ford automotive engine is a four-cylinder Siemens engine with a swashplate drive. The Swedish United Stirling engines are of the Siemens variety using crank-connecting rod drives with four cylinders arranged in a 'square' configuration. The MAN/MWM engines of West Germany utilize in-line or Vee arrangements of Siemens double-acting engines.

Walker (1978b) conceived the free-piston double-acting Siemens engine actually during the course of preparation of this book. Finkelstein (1978a) has described a number of new concepts and arrangements for multiple-cylinder, free-piston, 'cyclic-compounded' Siemens engines.

DESIGN VARIANTS OF SINGLE-ACTING STIRLING ENGINES

All existing arrangements for the single-acting Stirling engine may be broadly classified into two groups: (a) two-piston machines, and (b) piston-displacer machines. A further subdivision can be made in the latter group between machines in which the piston and displacer operate in a single cylinder, and those in which separate cylinders are provided. A representative example of each of these three arrangements is shown in Fig. 6.3. The principal distinction between a piston and a displacer is that there is a large pressure difference between the upper and lower faces of a piston so that a fluid seal is required to prevent the passage of gas from one side to the other. A displacer on the other hand has the same pressure of fluid both above and below, apart from aerodynamic-flow losses. When reciprocating, the displacer does no work on the gas, but merely displaces it from one side of the displacer to the other.

In the case of a piston, the pressure of fluid, above and below it, is not the same, except perhaps momentarily at some point in the cycle. Work is done on the gas by the piston, or on the piston by the gas, as the piston moves in the cylinder.

In some machines, the displacer is made up (either partly or wholly) by a porous metallic matrix, which itself constitutes the regenerative heat exchanger. Such an element is called a regenerative displacer.

Single-cylinder piston-displacer machines

Some of the possible alternative arrangements of single-cylinder piston-displacer machines are shown in Fig. 6.4. This is a particularly favourable

† In my earlier work, *Stirling Cycle Machines*, (O.U.P. 1973), I inadvertently attributed the re-invention of the Siemens engine to H. Rinia of the Philips Co. I now understand the invention was made, in fact, by van Weenan and offer my apologies to Ir. van Weenan herewith. G.W.

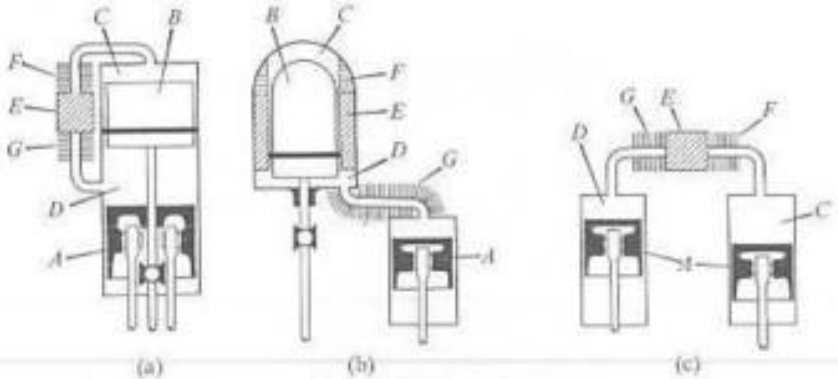
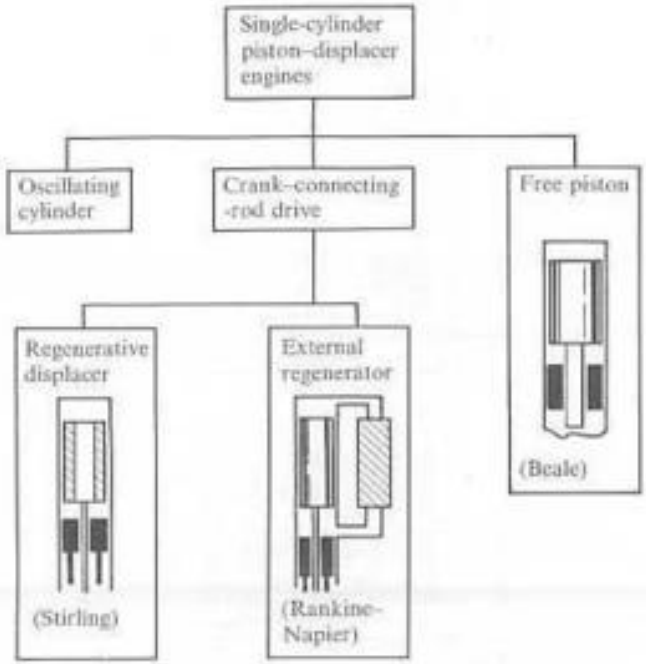


FIG. 6.3. Three basic arrangements by which most types of Stirling engines may be classified.
(a) Piston-displacer in the same cylinder.
(b) Piston-displacer in separate cylinders.
(c) Two-piston machine.



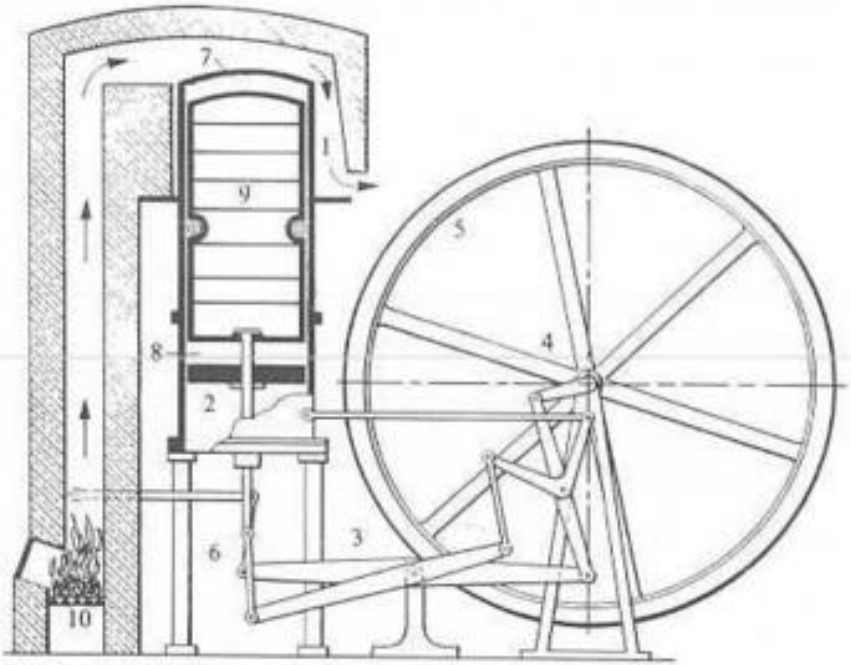


FIG. 6.5. The original Stirling engine. Reproduction of a drawing showing the first Stirling engine, from the original patent specifications of 1816. Such an engine was used in 1818 for pumping water from a quarry (after Finkelstein 1959).

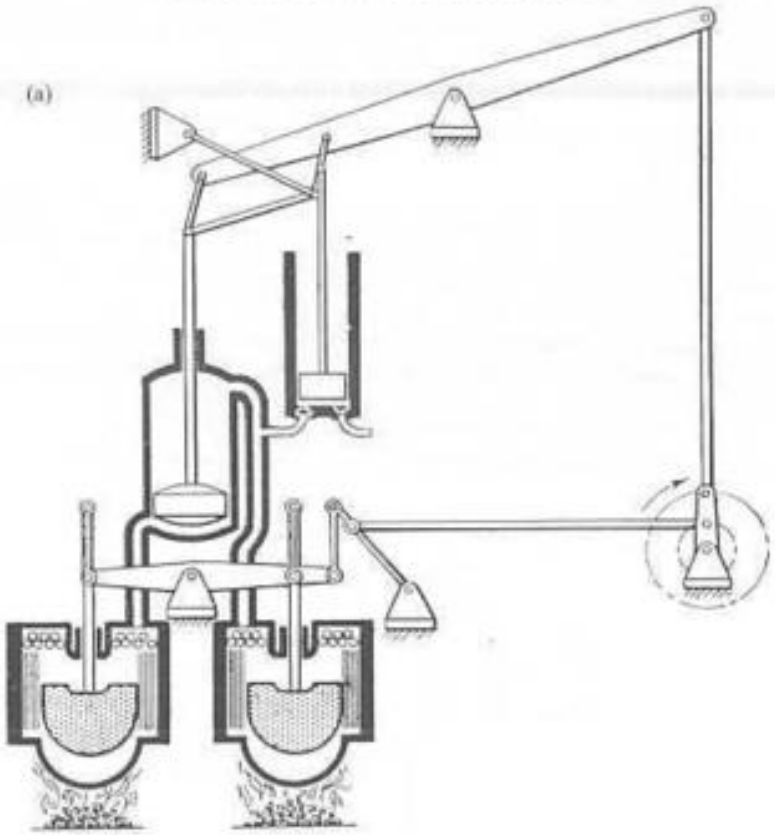
configuration; it was first used by Robert Stirling, in 1816, for the engine shown in Fig. 6.5. It has been used, also, for most of the machines developed by Philips, both prime movers and cooling engines.

Crank-driven engines can be of the type used by Stirling, with a regenerative displacer, or may have a separate external regenerator of the Rankine-Napier type. The possibility exists for the necessary volume variations to be gained by an oscillating-cylinder mechanism, but, so far as is known, machines of this type have not been developed. The free-piston engine is another interesting configuration. Such machines have been brought to an operational stage by William Beale of Sunpower Inc., Athens, Ohio, and seem promising for wide future application. Machines of this type are being developed for solar power conversion, air-conditioning, and heat pumps, as described in Chapter 11.

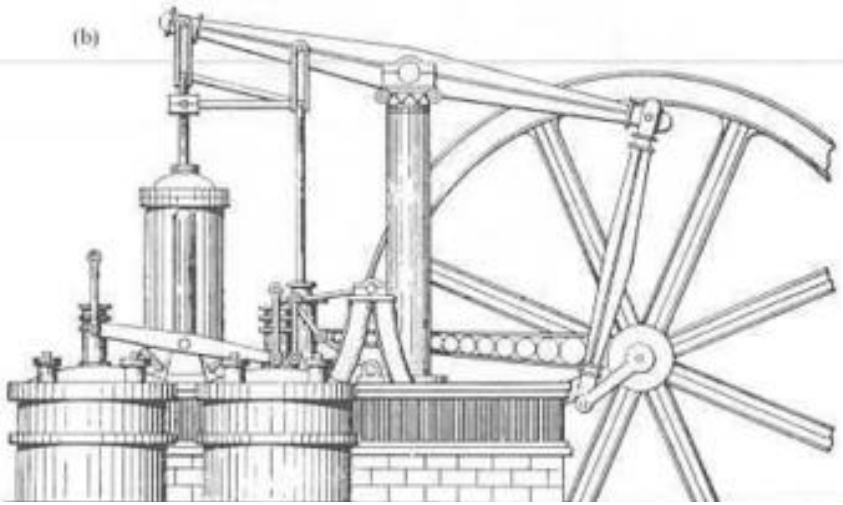
Two-cylinder-per-cycle piston-displacer machines

The first two-cylinder piston-displacer machine was, almost certainly,

(a)



(b)



Dundee foundry, but was subsequently abandoned because of repeated failure of the displacer cylinders, caused by overheating of the poor iron then available.

Single-acting versions of the machine, in various arrangements, are shown in Fig. 6.7. The version having a regenerative displacer may be identified as a Laubereau-Schwartzkopff engine, and, with a separate regenerator, as a Heinrici engine. An arrangement where the cylinder axes were at 90° was made commercially in the last century in fairly large numbers, and was known as the Robinson engine. A machine with interesting possibilities, proposed by H. Rainbow of Bristol, England, in 1971, is shown in Fig. 6.7. This machine has two pistons and a single displacer. This arrangement allows considerable flexibility in the drive mechanism, and facilitates the solution of both sealing and cooling problems.

Single-acting multiple-piston arrangements

Single-acting Stirling engines with multiple-piston arrangements can be classified broadly into four groups:

(a) piston-cylinder combinations,
(b) rotary assemblies,
(c) bellows-and-diaphragm types,
(d) free-piston devices.

Piston-cylinder combinations are the best known of all these groups. Fig. 6.8 shows a variety of single-acting two-piston machines, three with stationary cylinders and one rotary-cylinder machine. Of these possibilities only the Rider arrangement of parallel cylinders with the pistons coupled to a crankshaft was produced in any quantity in the last century.

The possible configurations embracing rotary assemblies, or bellows-and-diaphragm systems, is virtually endless. Most represent attempts to overcome difficulties of imbalance or seal problems, arising from reciprocating elements and the associated linkage, but, so far as is known, none of these has been brought to a commercial stage. One interesting rotary machine, proposed by Zwiauer,† is shown in Fig. 6.9. Two Wankel-type rotary engines are coupled on the same shaft, and two (or three) regenerators are arranged symmetrically around the axis. One Wankel unit constitutes the compression unit, and the other constitutes

† Unpublished project proposal, University of Calgary.

FIG. 6.6. Early twin single-acting Stirling engine with piston and displacer in separate cylinders.
(a) Diagrammatic cross-section, showing arrangement of the drive mechanism.
(b) Copy of an old engraving of a beam engine dating back to 1827 (after Finkelstein 1959).

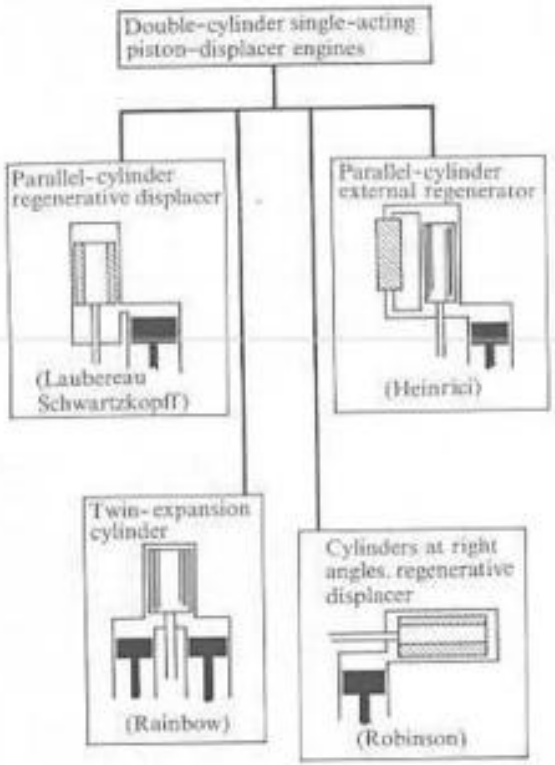


FIG. 6.7. Alternative arrangements of the double-cylinder single-acting piston-displacer engine.

the expansion unit. Each unit comprises three distinct spaces, and each space experiences two separate expansion or compression processes per revolution. Thus, a combination of the two engines embraces three separate systems, each undergoing two complete cycles per revolution. It is thought that this arrangement could provide a compact high-specific-output machine, but this has not been reduced to practice at the present time.

PISTON-DISPLACER VERSUS MULTIPLE-PISTON ENGINES

It has been shown above that many different arrangements of piston-displacer machines and multiple-piston machines are possible, some of which have been developed to a commercial degree. There is no one arrangement that excels above all others in every case, but there are a

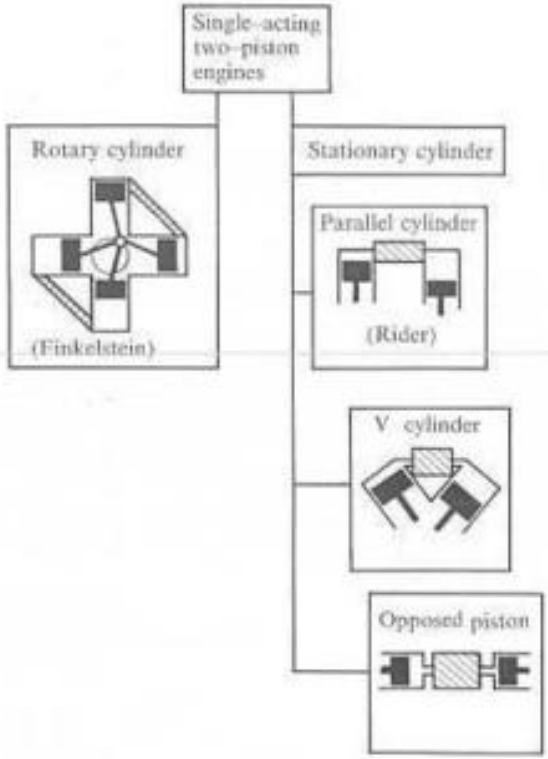


FIG. 6.8. Alternative arrangement of single-acting two-piston engine.

larger machines, the choice may lie between multiple, single-cylinder, single-acting, piston-displacer machines on a common crankshaft, or multi-cylinder, double-acting Siemens machines. There is every possibility that the simplicity of the Siemens arrangement will prove compelling. The reduced number of reciprocating elements can result in substantial economies in manufacture.

An important reason for the preference of piston-displacer machines over multiple-piston machines is that, in the former, it is somewhat easier to deal with the problem of reciprocating seals. On all machines, at least two dynamic-fluid seals are required. In the case of the three machines shown in Fig. 6.3, fluid seals are required on all the pistons, two in the case of the two-piston machine, and one each in the two other piston-displacer machines. An additional fluid-seal is necessary on the displacer

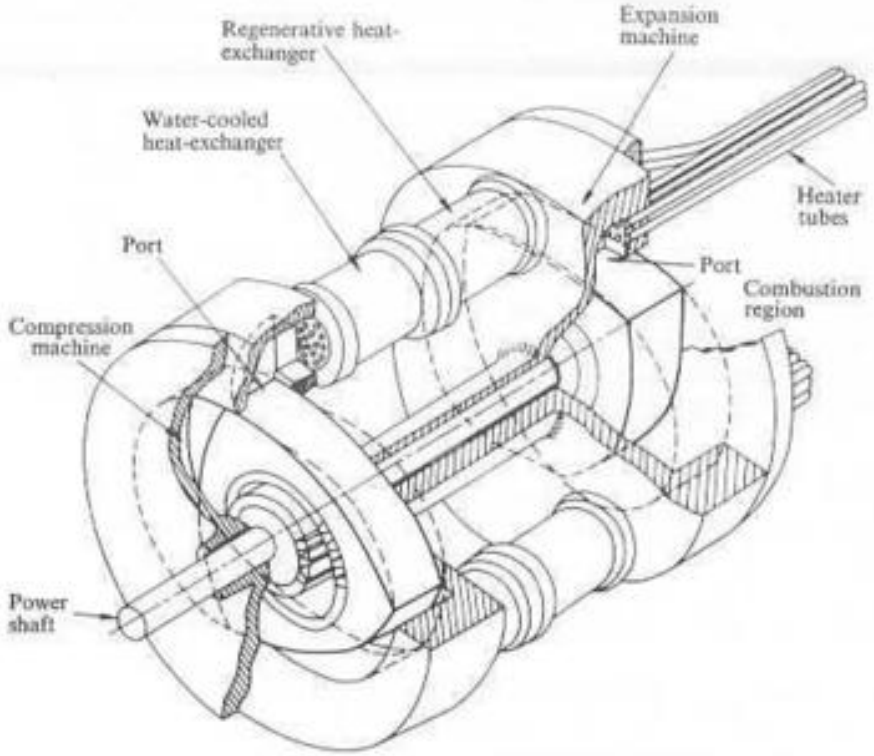


FIG. 6.9. Zwinner-Wankel configuration of rotary Stirling engine.

displacer rod is much smaller than the seal around the piston, with proportionally less leakage and friction. This is, perhaps, the most singular advantage of displacer engines, since the problem of reciprocating seals is particularly difficult, especially when working fluids other than air are used. A further advantage of piston-displacer machines is that the total reciprocating mass can be less than in multiple-piston machines. This facilitates balancing, and reduces vibration problems. The displacer does no work, and has to withstand merely the gas forces (arising from aerodynamic-flow losses) and its own inertia forces. Therefore, it can be structurally light, and requires correspondingly smaller rods, links, and bearings, so that appreciable savings in weight and mechanical-friction losses can be made.

The power output of a Stirling engine is, to a first approximation, a linear function of the pressure of the working fluid. Thus, one way to immediately increase the specific output is to pressurize the engine. On small engines, it is advantageous to pressurize the crankcase: this not only

reduces the duty on the reciprocating fluid-seals, but also reduces the structural strength requirement of the piston and connecting rod assembly, including bearings. This arises from the fact that, with a pressurized crankcase, the pressure difference across the piston is reduced to $(p_{\text{cylinder}} - p_{\text{crankcase}})$, instead of $(p_{\text{cylinder}} - p_{\text{atmosphere}})$ with an unpressurized crankcase. Savings in weight, mechanical-bearing friction, and seal friction may be gained thereby. These are offset by the fact that (a) the crankcase is now a pressure vessel with an increased strength requirement, and (b) that at least one dynamic seal is involved if the crankshaft is required to exit from the crankcase.

The problem of a *rotating* crankshaft-seal is less rigorous than that of a *reciprocating* piston-seal, and further, may be eliminated by combining the electric motor, or generator driven by the engine, into the crankcase; however, this may cause appreciable 'windage loss' in a highly pressurized engine.

The use of a pressurized crankcase is limited to small engines. On large engines the weight of the crankcase becomes excessive if it is pressurized to the minimum cycle pressure.

SINGLE-CYLINDER VERSUS TWO-CYLINDER PISTON-DISPLACER MACHINES

It is almost intuitively obvious that, for a pressurized engine, the single-cylinder piston-displacer configuration leads to a crankcase of minimum size and weight. As the engine power rating increases, the crankcase becomes a dominant fraction of the total engine weight, and for large engines, the simple expedient of a pressurized crankcase must be abandoned.

There are two other advantages of the single-cylinder piston-displacer engine over the machine with separate cylinders. In the two-cylinder machine, (Fig. 6.3(b)), the compression space is divided between the displacer cylinder and the piston cylinder, and includes the port connecting the two. This space can never be reduced to zero, so that (in effect), the compression space has a large clearance volume. This clearance volume must be included with the dead space, X, and as we have seen earlier, any increase in X results in a decrease in power output.

The second advantage of the single-cylinder piston-displacer engine is that, in every revolution, the displacer and piston both sweep the same part of the cylinder, although at different times. This overlap of strokes is shown clearly in Fig. 6.10 and represents a most efficient utilization of the available engine-cylinder volume.

The advantages of the separate-cylinder piston-displacer machine are:

(a) the increased flexibility for production-engineering design of the crankshaft and connecting-rod system

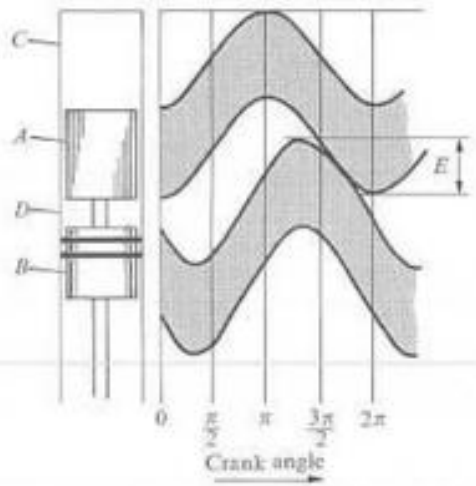


FIG. 6.10. Piston and displacer motion in a single-cylinder engine.

(b) the separation of the displacer-rod seal to a fixed location in the displacer cylinder, rather than the more limited environment of the piston crown.

In practice, these are very important advantages. An attractive arrangement of a piston-displacer in separate cylinder machine is shown in Fig. 6.11. The engine is arranged in Vee form. Engines of this configuration in a power range of about 10 kW are at an advanced stage of development at the Swedish company, F.F.V. (a part-owner of United Stirling). These engines are soon to be introduced for commercial use (Johannson 1978).

DESIGN VARIANTS OF ERICSSON ENGINES

Regenerative engines of the type where the flow is controlled by valves (called here Ericsson engines) are found, like their Stirling cousins, in a wide variety of types, shapes, and sizes. Sometimes engine arrangement for both types can be very similar; the only distinction between them is the existence (Ericsson) or non-existence (Stirling) of valves which allow the passage of fluid through the working space in a cyclic manner, and generally control the flow of the working fluid. In this distinction it is important to note that we exclude the gas valves used on an intermittent basis as part of the control system on Stirling engines to vary the working fluid pressure.

The families of Ericsson engines are not considered here in detail, but a brief guide to the principal types might be in order, so as to assist in identification. The degree to which any of the theoretical material or accumulated engineering experience discussed here might be applied to

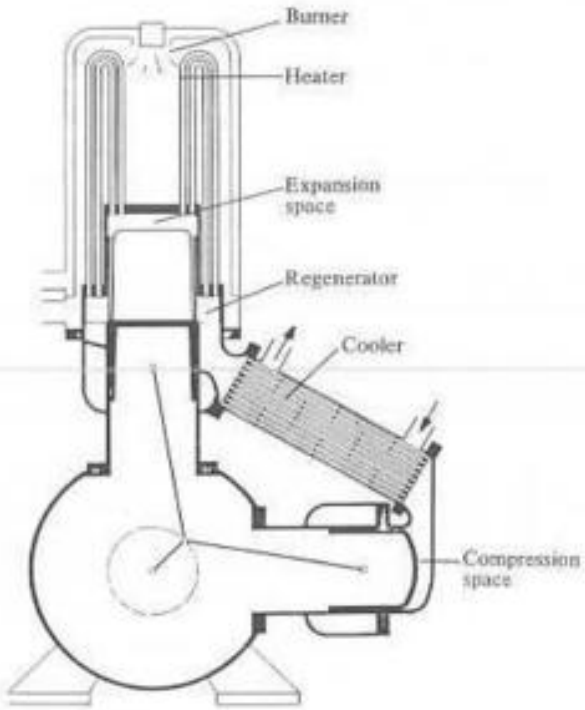


FIG. 6.11. Vee arrangement of the piston-displacer in separate cylinder arrangement.

Fig. 6.12 is a 'family tree' of Ericsson engines. In most cases, they can be classified either as displacer machines or as piston machines. Each of these principal groups can be further subdivided. Displacer machines may have either a constant working-volume or a variable working-volume. Piston machines may be classified into single-piston machines or two-piston machines.

Fig. 6.13 shows some of the design variants of displacer machines, and identifies some of the better-known arrangements by the names of their inventors. Of the variable working-volume type, only one example is shown. This was first used by John Ericsson, and contains both a piston and a regenerative displacer, coupled together (but moving out of phase) by means of a crank-connecting-rod system. The arrangement may be equipped with gas-operated (or mechanically-operated) valves. This arrangement is potentially attractive for very large nuclear-reactor installations, where the working fluid could be passed as the coolant through the reactor core.

There is a larger range of possibilities in machines of constant working-

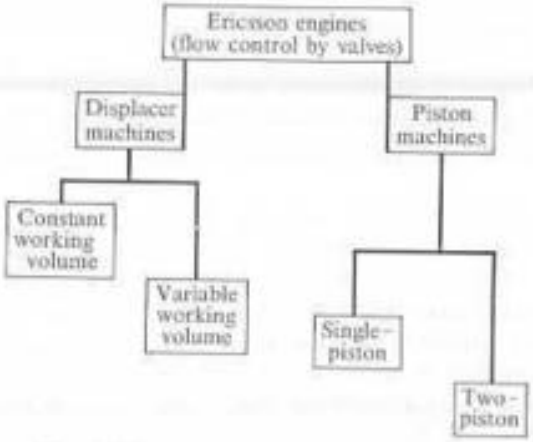


FIG. 6.12. Design variants of Ericsson engines.

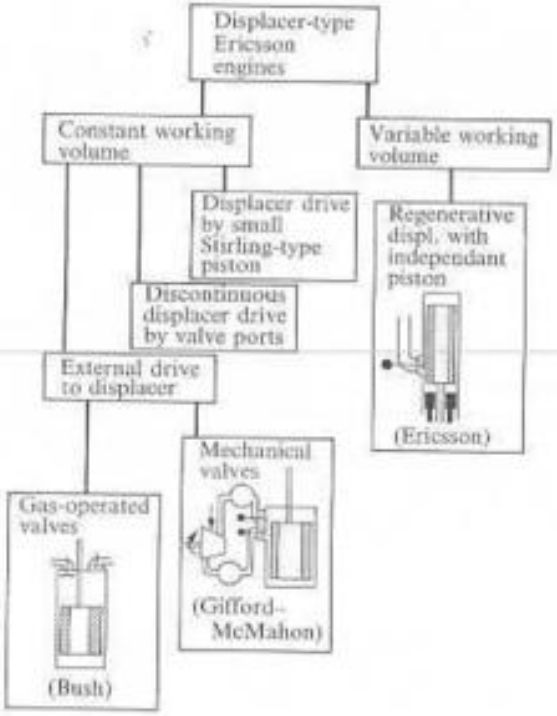


FIG. 6.13. Alternative arrangements of displacer-type Ericsson engines.

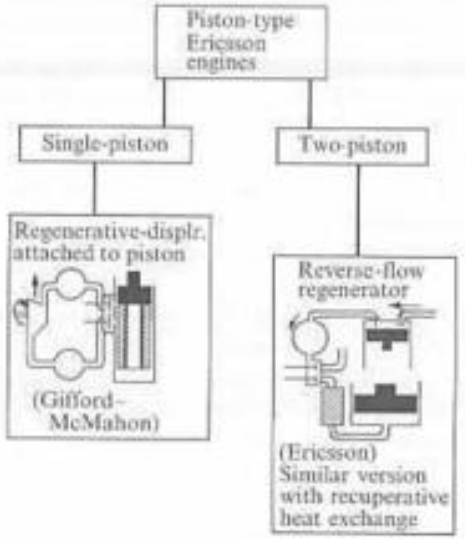


FIG. 6.14. Alternative arrangements of piston-type Ericsson engine.

the manner in which the displacer is driven. Of the type with an external drive to the displacer, an arrangement with gas-operated valves was patented by Bush for use as a pressure generator, whereas the type with mechanically operated valves has been successfully developed commercially as a Gifford-McMahon cryogenic cooling engine. A displacer-type free-piston engine with pneumatic drive designated as a Buck engine is used as the Aerojet artificial heart engine described in detail in Chapter 17. In the literature this is frequently called a Stirling engine.

The Bush engine has been the subject of development effort by Smith and Lee (1978) at the Massachusetts Institute of Technology and is called the 'ported hot gas engine'.

7 HEAT EXCHANGERS IN STIRLING ENGINES

INTRODUCTION

HEAT exchangers are key components in Stirling engines: their significance cannot be overemphasized. No engine can work properly with poor heat exchangers although, conversely, the best heat exchangers will not, of themselves, make good an otherwise poor engine.

There may be four separate heat exchangers in a Stirling engine system. These are illustrated in Fig. 7.1 and include, for the prime mover converting heat to work:

(a) heater
(b) regenerator
(c) cooler
(d) exhaust/inlet-air preheater.

For the refrigerator, utilizing work to produce cold at low temperatures, a different terminology must be used. The heater becomes the 'freezer' and the exhaust/inlet-air preheater becomes the 'precooler'.

For a Stirling engine operating as a heat pump, a yet different terminology is appropriate. The heat pump utilizes work to elevate the temperature of heat supplied at near-atmosphere temperatures. In this application the heater of the prime mover becomes the 'absorber' of the heat pump and the cooler of the prime mover becomes the 'heater' of the heat pump.

In all cases whether the engine operates as prime mover, refrigerator or heat pump the *direction* of heat flow is the same: *to* the expansion space and *from* the compression space. Only the temperatures are different:

(a) In the engine the heater operates at a high temperature, near the metallurgical limit, and the cooler operates at ambient temperature.
(b) In the refrigerator the freezer is at the low refrigerating temperature and the cooler operates at ambient temperature.
(c) In the heat pump the absorber operates at ambient or heat supply temperature and the 'heater' delivers heat at an elevated temperature.

In principle, therefore, the primary considerations for the heat exchangers are similar for all applications of Stirling engines. With this preamble we shall hereafter confine our attention to the prime mover application.

FUNCTION OF THE HEAT EXCHANGERS

The function of the heat exchangers is very simply to exchange heat.

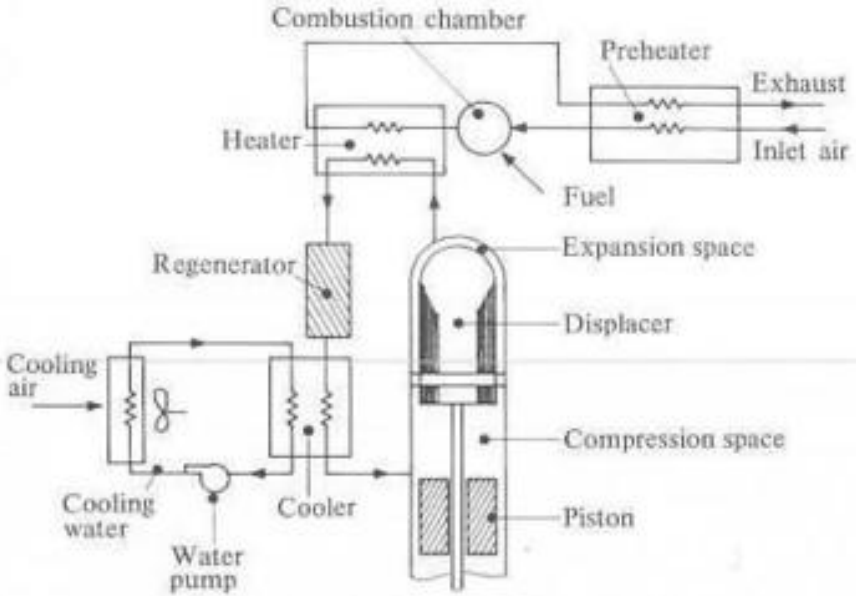


FIG. 7.1. Location of heat exchangers in Stirling engines

and supplies heat to the engine working fluid contained within the working space of the engine adjacent to the expansion space.

The *cooler* does just the reverse. It absorbs heat from the working fluid adjacent to the compression space and rejects this to the engine cooling medium (air or water) at the sink temperature.

The *regenerator* functions as a thermodynamic sponge alternately accepting heat from or rejecting heat to the working fluid. It can be visualized as a kind of 'storage lagoon' of energy but of a rather special kind, for the temperature ranges progressively over the whole range from cooler to heater temperature. Energy flows in the regenerator are very high, usually three or four times the energy flow in the heater.

The *exhaust-gas/inlet-air preheater* serves to increase the temperature of fresh air entering the engine *en route* to the combustion space. This can be achieved utilizing the thermal energy contained in the combustion products leaving the combustion space at a high temperature. In this way fuel is saved. The amount of fuel necessary to raise the inlet-air to combustion temperature is reduced and so the engine efficiency is increased.

The exhaust-gas/inlet-air preheater is not essential to the operation of the engine in the same way the heater, cooler, and regenerator are. The engine will work well enough without a preheater but less efficiently, and

If the engine's thermal source is a radioisotope, or solar or stored heat, instead of fossil fuel combustion, no inlet-air is necessary and so of course the preheater can be eliminated. Presently most Stirling engine applications use heat derived from the combustion of fuel in air and in most cases a preheater is essential.

TEMPERATURE AND ENERGY DISTRIBUTION

Fig. 7.2 shows a typical temperature distribution in a Stirling engine of advanced design. Inlet-air enters the engine at the atmosphere temperature and is heated in the preheater before passing to the combustion space. Fuel is added and combustion occurs, heating the products of combustion to a very high temperature. The combustion products then pass through the heater, where heat is transferred to the working fluid, and through the preheater, where heat is transferred to the inlet-air. The cooled products finally leave the engine. In many applications where air pollution is an important consideration, a fraction (up to one half) of the exhaust products are recirculated back through the combustion chamber. This increased mass flow of relatively inert fluid moderates the maximum temperature attained in combustion and so reduces the amount of oxides of nitrogen (NO_x) produced. Some extra work is then required to cause the air to flow through the system and therefore a fan will be necessary.

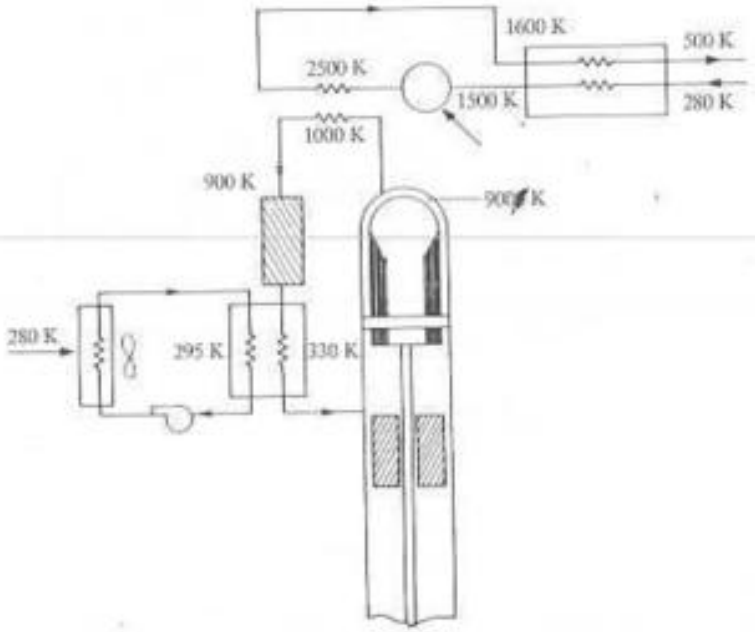


FIG. 7.2. Typical temperature distribution in Stirling engines

This will most likely be found on the 'cold' inlet-air side. The combustion chamber and preheater therefore operate at a pressure slightly above atmospheric. A fuel pump will also be required to supply and, perhaps, atomize the fuel.

Working fluid in the engine passing from the regenerator to the expansion space is heated in the heater by energy transferred from the products of combustion. The working fluid, probably hydrogen or helium at high pressure, has excellent heat-transfer characteristics compared with the combustion products at atmospheric pressure. For this reason a very high temperature difference will most likely exist between the temperature of the combustion products and the heater-tube walls compared with the temperature difference between the tube walls and working gas.

At lower temperatures heat is transferred *from* the working fluid in the cooler near the compression space. The heat is rejected from the cooler to water circulated through the engine. Water is a dense fluid with excellent heat-transfer properties and so only a small temperature difference in the cooler is required to effect the energy flow.

The heat rejected from the working fluid to the cooler is eventually dissipated to the atmosphere in an air-cooled radiator. A fan is required to draw air through the radiator and a pump is necessary to circulate the water.

In stationary power applications the water may be drawn from a virtually infinite source such as a lake or river. Even here, however, an intermediate cooling system will usually be preferred to avoid contamination of the engine cooler by sediments drawn in by the cooling stream.

Energy flows in a Stirling engine are shown in Fig. 7.3. This Sankey diagram is reproduced from the important and fundamental paper by Zacharias (1971a). Starting from the top of the diagram there is a given energy input to the system (designated as 100 per cent). Recirculation of a fraction of the exhaust gas is shown to an energy equivalent of about 43 per cent. Power and other losses for recirculation consumes about three per cent of the input energy supply.

For the particular case shown the energy loss from the system carried off in the heated exhaust gas is about 14 per cent.

At the bottom of the diagram the heat loss to the cooling water is 45 per cent, mechanical friction consumes a further 5 per cent and approximately 32 per cent of the energy supplied is transformed to work available at the engine shaft. An important point to note from Fig. 7.3 is the very high levels of energy flow in the regenerator, amounting to over four times the energy input to the heater, or alternatively nine times the energy flow in the cooler, and twelve times the energy flow to work. The 'recycling' of this remarkable energy flow is the significant attraction of the Stirling engine.

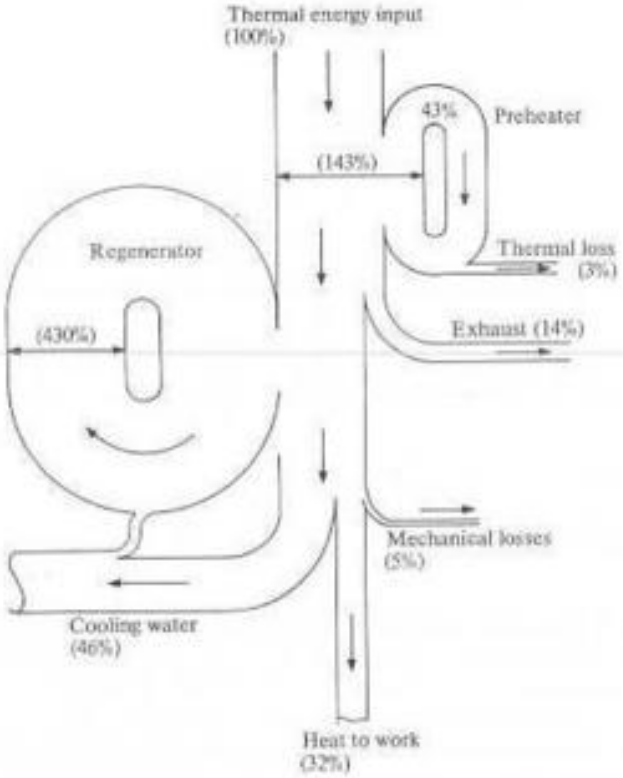


FIG. 7.3. Energy flow in a Stirling engine

Fig. 7.3 illustrates well the important differences between an internal combustion engine and a Stirling engine. To a first approximation the energy flows in a diesel engine may be thought of as three equal streams comprising 33 per cent each to work, to cooling, and to exhaust. Thus the energy flow to exhaust is twice that of the Stirling engine and the cooling system load is much less.

In a diesel engine, a high energy flow to exhaust is permissible because combustion of the fuel occurs *inside* the engine cylinder, so that no heater is necessary and the heat not converted to work must be dissipated either to exhaust or the cooling system. This is not the case in the Stirling engine where combustion of the fuel occurs *outside* the engine cylinder. Any heat passing to the exhaust represents a direct loss of energy that has

through the heater and not converted to work must be dissipated by the cooling system.

Returning therefore to the example given by Zacharias and reproduced in Fig. 7.3, every effort should be directed to reducing the exhaust stack loss from 15 to 10 or even 5 per cent. This would most likely increase the fraction converted to work from 32 to 35 per cent leaving 55 to 60 per cent to be handled by the cooling system and by extraneous convective and radiative heat transfers from the engine. This heavy load in the cooler and the sensitivity of the engine efficiency to increase in the cooler operating temperature is one of the major stumbling blocks to the use of Stirling engines in automotive applications.

TRANSIENT-FLOW EFFECTS

Difficulties in the design of heat exchangers for Stirling engines arise from the cyclic transient-flow effects. Most industrial heat exchangers are subject to steady constant-flow conditions with relatively slow rates of change in the flow conditions. This is by no means the case for the heat exchangers used in Stirling engines where the flow conditions change continuously and experience wide variations in pressure, density, and velocity, to the extent of *reversing* the flow direction twice per cycle. All this complicates the situation considerably and makes the design of the regenerator and other heat exchangers a difficult art.

Initial contemplation of Stirling engines leads one to believe that when the engine is operating, the working fluid flows from the expansion space through the heater, regenera.or, and cooler to the compression space, and then retraces its step in returning to the expansion space.

Such a view is oversimplified and applicable only to the ideal Stirling engine. In practice none of the fluid ever moves all the way from the expansion to the compression space. Instead, a given hypothetical particle of fluid simply oscillates cyclically in a limited region of the engine in similar fashion to ocean driftwood repeatedly washed up on the shore and swept away once more by the receding waves.

This is illustrated in Fig. 7.4 which shows the cyclic trajectory as a function of crank angle of particular particles of the working fluid in a Stirling cycle cooling engine. The cyclic trajectories were calculated by Walker (1960) using Schmidt isothermal theory. For the particular case considered no particle passed into more than two regions of the engine per cycle i.e., expansion space/freezer, freezer/regenerator, regenerator/cooler and cooler/compression space. As can be seen from Fig. 7.4 no particle ever passed right through the regenerator.

Walker (1960) appears to have initially recognized the phenomenon of

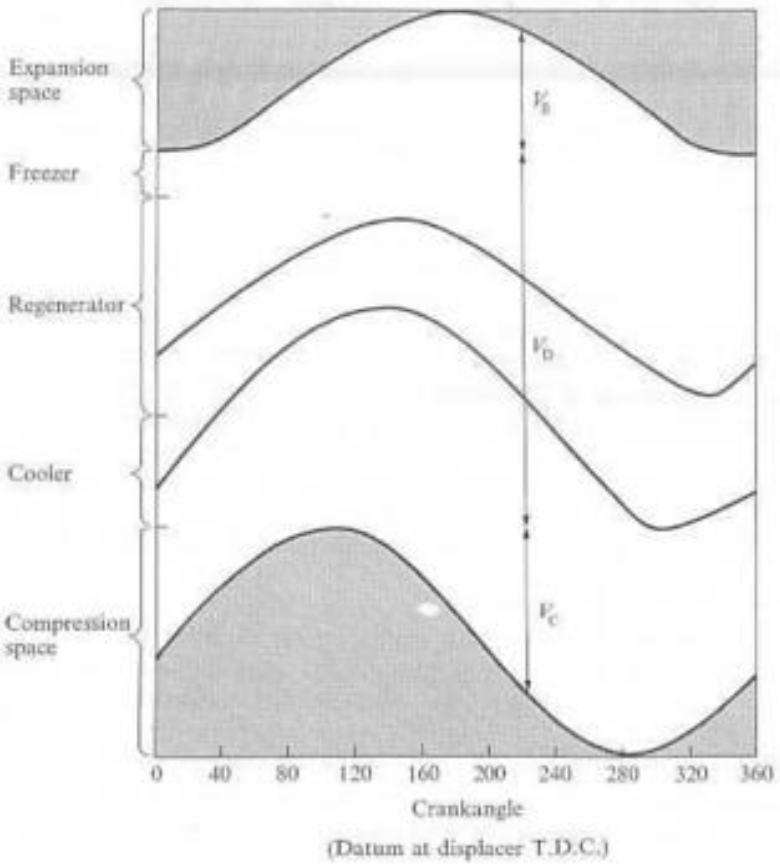


FIG. 7.4. Fluid particle displacement. The figure shows the cyclic trajectory of individual particles of the working fluid in a Stirling-cycle cooling engine, calculated using the Schmidt isothermal theory. The significant point of interest is that no particle of the working fluid ever passes from the expansion space to the compression space, or indeed ever passes right through the regenerator.

However the topic has received considerable recent attention (Organ 1976) and the development of sophisticated computer simulation programs has elevated the display of particle trajectories almost to a *nouveau art* form (Schock 1978b).

In the course of calculating the particle trajectories mentioned above, Walker (1960) determined the mass rates of flow of working fluid into and out of the compression and expansion spaces of the subject cooling engine. The characteristics obtained thereby are reproduced in Fig. 7.5. Examples are given for two different mean pressures of the hydrogen working fluid.

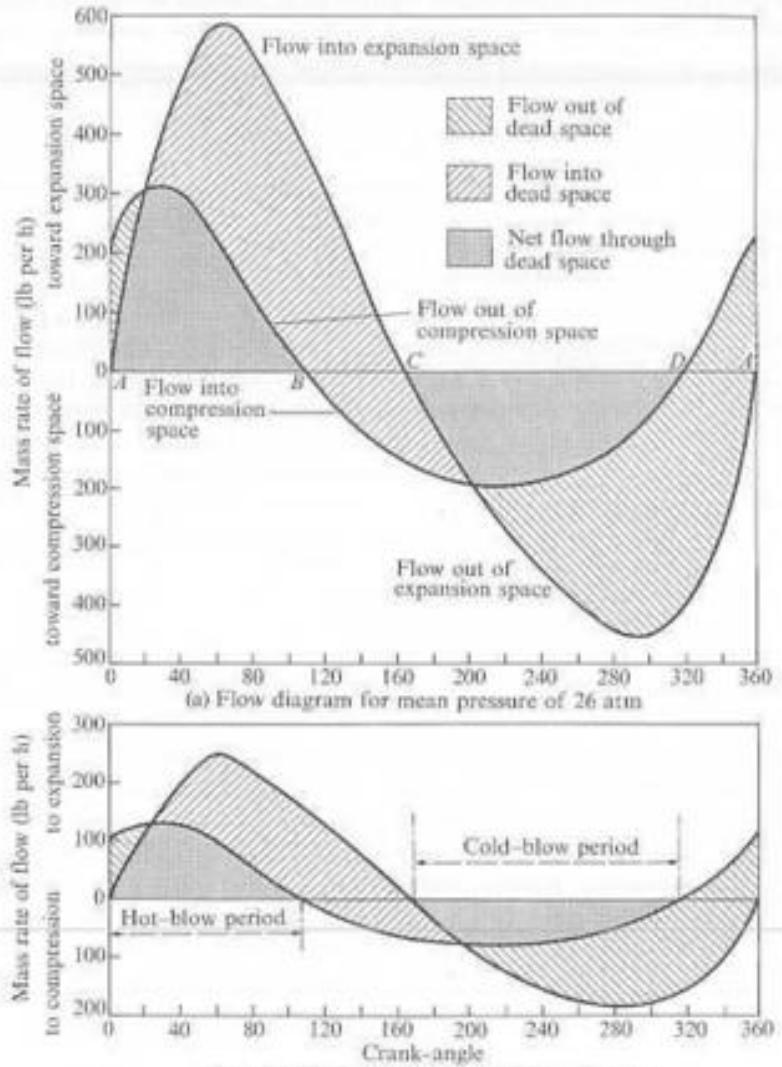


FIG. 7.5. Mass flow in a Stirling-cycle cooling engine. The figure shows the cyclic mass-flow rate for a Stirling-cycle cooling engine, calculated using the Schmidt isothermal theory, for mean pressures of (a) 26 and (b) 11 atm.

The diagrams given in Fig. 7.5 are somewhat complicated but worthy of close attention to appreciate a fundamental aspect of Stirling engine operation. Each diagram contains two curves, superimposed. One curve represents the mass-flow rates into and out of the expansion space; the other represents mass-flow rates into and out of the compression space. Curves *above* the zero datum line represent flow *into* the expansion space and *out* of the compression space. Curves *below* the zero datum line represent flow *out* of the expansion space and *into* the compression space. When these are superimposed as in Fig. 7.5 the areas where the curves overlap represent the period of *net flow through the dead space*, that is through the heat exchangers. Referring to Fig. 7.5, the period *A*—*B* represents the flow of fluid through the heat exchangers towards the expansion space with fluid flowing from the compression space into the dead space and from the dead space into the expansion space.

In period *B*—*C* fluid flows from the dead space into both the compression space and the expansion space. This can be thought of as the phase of emptying the dead space. In period *C*—*D* the fluid flows through the dead space towards the compression space with fluid leaving the expansion space and entering the compression space. Finally the period *D*—*A* is the phase of filling the dead space when fluid is flowing into the dead space from both the compression and expansion spaces.

The important point to appreciate is that net flow through the dead space, in effect, the regenerator, heater, and cooler, does not occur for much more than half the total cycle time, and further, the mean rate of the net flow is substantially below the mean rates of flow into and out of the two compression and expansion spaces.

Most of the relations found in the literature for heat transfer and fluid friction depend on equations based on groupings of dimensionless groups with appropriate coefficients and power indices i.e., the Dittus-Boelter equation for the heat transfer in turbulent flow in smooth circular pipes is:

$$(Nu) = 0.23\,(Re)^{0.33} \tag{7.1}$$

where Nu = Nusselt number $= \dfrac{hd}{k}$

Re = Reynolds number $= \dfrac{\rho v d}{\mu}$

and h = heat-transfer coefficient
k = thermal conductivity
ρ = gas density
v = gas velocity

The heat transferred:

$$Q = hA\,\Delta T \tag{7.2}$$

where Q = rate of heat transfer
A = area for heat transfer = πdLn
ΔT = temperature difference between the gas and wall temperature
L = length of tube
n = number of tubes.

Given the continuously variable flow rates suggested in Fig. 7.5 plus the cyclic pressure variation over a ratio of $p_{max}/p_{min} = 2$ approximately, it is clear that the velocity v and the density ρ will vary continuously, with consequent profound effects on the heat-transfer coefficient and the actual heat transfer. Variation in the rate of heat transfer will reflect on the temperature difference ΔT with consequent variations in temperature which in turn will cause variations, albeit minor, in the thermophysical properties of the gas, thermal conductivity, specific heat, and viscosity.

It is evident from the above that the thermofluid processes in a Stirling engine are entirely and continuously transitory in nature. Thus far, no simple design procedures for handling this type of flow have become generally available. It is necessary therefore in preliminary design work to assume a reasonable value for mean flow rates and to calculate the heat transfer and friction effects for that assumed flow. This will allow an initial determination of the tube diameters and lengths or of fin widths and depths for the heater and cooler and also for estimation of the matrix diameter and length for the regenerator.

It must be recognized that the use of a mean rate of flow is a crude approximation to the real situation. Modifications of the design are to be anticipated for the optimum engine performance either in the light of more sophisticated analysis once the design is established or as the result of actual testbed operating experience.

Nevertheless a start has to be made somewhere and curves such as those given in Fig. 7.5 will be found useful in arriving at representative mean flow rates for use in conjunction with the steady-flow heat-transfer and friction data found in the literature.

GENERAL ASPECTS OF DESIGN

The best compilation of design data for the heat exchangers in Stirling engines is undoubtedly the classic work by Kays and London (1964) entitled *Compact Heat Exchangers*. The book is, quite simply, required reading for anyone wishing to undertake the design or analysis of the heat

nature and wide availability of the book no effort will be made to reproduce the straightforward design procedures for compact tubular, finned and regenerative heat exchangers that are so well treated therein. Rather, the space available will be devoted to brief discussion of some of the aspects of design peculiar to Stirling engines that are not found or not stressed in *Compact Heat Exchangers.*

The principal consideration particular to Stirling engines is the compelling need to make effective use of the internal void volume of the heat exchangers and connecting ports. We have seen earlier how an increase in the dead volume results in a reduced volume compression ratio V_{max}/V_{min} and in a reduced pressure ratio p_{max}/p_{min}. The result is a small pressure-volume or work diagram for the engine and so the output declines progressively as the dead space increases. The thermal efficiency on the other hand may be unaffected or may even increase if the increase in dead volume has been wisely applied to increasing the effectiveness of the regenerative or other heat exchangers.

If the dead space is simply increased by insertion of a spacer, an oversized header, or 'wasted' in some other way the result will likely be decreases not only in the power but also in efficiency.

Overzealous pursuit of the minimum possible dead space is just as deleterious. It will result in insufficient area for heat transfer so the effectiveness of the heat exchangers will be reduced and, further, excessive frictional pressure drops will result with substantial decrease in output and efficiency. A proper balance of areas for heat transfer, with tolerable fluid-friction effects arising from flow, sudden changes in section, entrance effects, and flow reversals will be achieved by careful, thoughtful usage of the internal dead space. This is worthy of the most careful attention and instant, positive rewards await the judicious scrutiny of a design or of the fabricated prototype. The insertion of instrumentation, particularly displacement or pressure transducers can result in a profound increase in the dead volume of a really 'tight' engine design. One should always be alert to the unfortunate consequences that can result in the course of instrumenting a good engine.

HEATER DESIGN

In general most heater designs can be divided into two classes: tubular or finned. An example of each is shown in Fig. 7.6. Both systems combine three separate heat transfer processes:

(a) convective heat transfer from the external heating medium to the walls of the tube or fin,
(b) conductive heat transfer through the tube wall to the inner surface or to the root of the fin and hence to the internal fin,
(c) convective heat transfer from the internal walls of the tube or fin to

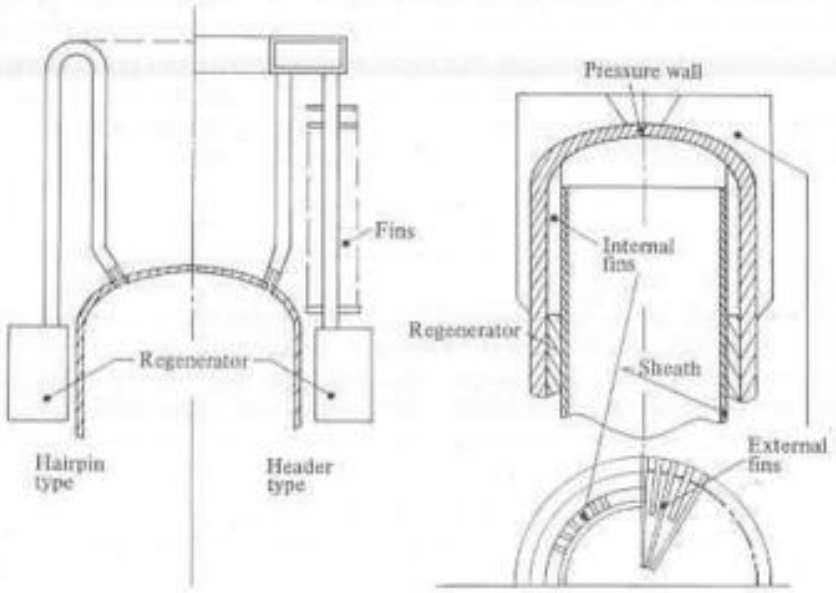


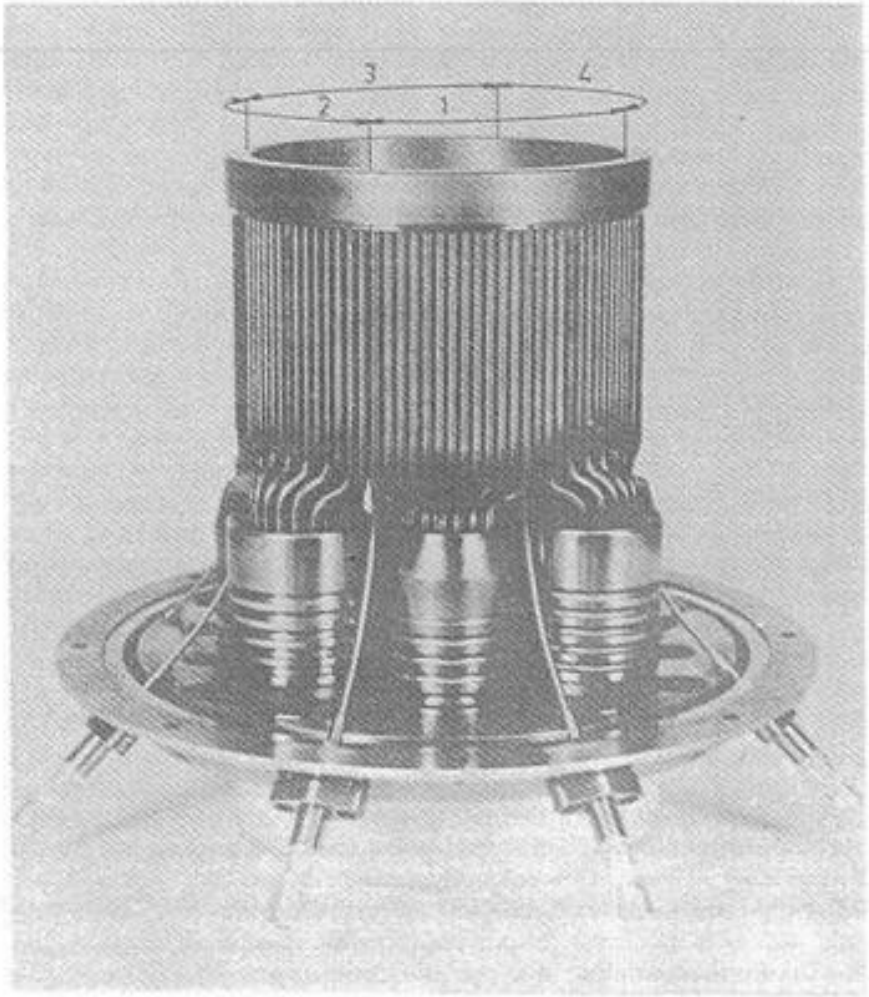
FIG. 7.6. Types of heater head design

In most engines the working fluid is pressurized and so is relatively dense and moving with a relatively high velocity so the internal heat transfer process is well developed. Similarly most metals are relatively good conductors of heat so that only a small temperature difference across the tube or cylinder wall is necessary to accomplish the desired heat transfer. In a combustion system at atmospheric pressure the limiting heat-transfer process is likely to be the external convective heat-transfer process. The products of combustion are not dense and may be moving relatively slowly so that a large temperature difference will be necessary to accomplish the transfer of heat.

Many designs feature a combustion space with the burner located along the axis of the cylinder. This should result in a radial distribution of heat with uniform temperature distribution around the circumference of the fins or tubes. However it is rarely possible to achieve a completely uniform temperature distribution and thermal distortion may occur. 'Hot spots' will exist and since, to prevent burnout, these must be at or less than the maximum sustainable operating temperature of the metal (the metallurgical limit) the *mean* temperature of the head will be substantially (up to 100 °C) less than the metallurgical limit. This low mean

The fin design shown in Fig. 7.6 lends itself well to application with an annular form of regenerator. It appears to be best suited for use with small engines (less than 1 kW) where the advantages of simplicity and compact design are self-evident.

For larger engines tubular heat exchangers have been favoured and several interesting variations for tubular heaters are discussed in Chapters 12–16 dealing with engines manufactured by the Philips and licensees of Philips companies.



Tubular heaters of the simple double hairpin tubes shown in Fig. 7.6 offer severe and relatively expensive production problems in brazing the tubes to the cylinder head. A 'dip brazing' or vacuum furnace brazing technique may be used for the stainless or high nickel steel tubes in concert with a stainless steel cylinder head. Tubular exchangers are well suited to designs where multiple regenerators can be used with groups of three to six tubes per regenerator. This provides a flexible design allowing for relatively unrestrained movement of the heater tubes during initial heating and subsequent cool down of the tubes after use. Cracking of heater tubes resulting from restraints imposed on thermal movements are a common feature of designs at an early stage.

The simple hairpin heaters of Fig. 7.6 have now largely been supplanted by more advanced designs using some form of manifold header at the upper end as shown in Fig. 7.7. This facilitates construction and permits the use of different tubes for the internal and external rows. It is not uncommon to find the outer row finned to compensate for the reduced heat transfer because of the temperature of the lower combustion products passing over the outer tubes.

The inequality in temperature between the inner and outer tubes, and also around the periphery of the tubes themselves, imposes a severe design limitation when combined with the very highest pressure stress levels found in advanced engines where thin wall tubes are used to minimize wall conduction effects. This is discussed briefly in Chapter 14 following an excellent presentation on the subject by Zacharias (1973).

Recent developments at Philips have resulted in tubular heaters having manifold headers at both the top and bottom of the tubes as shown in Fig. 7.8. This permits a single axial penetration to the expansion space. This, when coupled with the use of glass ceramic inserts acting as thermal insulators in the hot cylinder ends, allows the use of water-cooled expansion space cylinders of relatively low-cost steels. This interesting development at Philips (Meijer 1978) offers much hope for the future substantial reduction in cost that is necessary if Stirling engines are to move out into a wider stage of application.

INDIRECT HEATING

All the above was concerned with directly heated tubes or fins in contact with the combustion products or receiving direct radiant energy from a solar concentration or isotope emitter.

Substantial advantages accrue from the use of an intermediate liquid metal heat-transfer loop. Usually the medium is a sodium/potassium (Na K) eutectic mixture or a condensing sodium vapour. The advantage of using liquid metal indirect heating is that the same high rates of heat

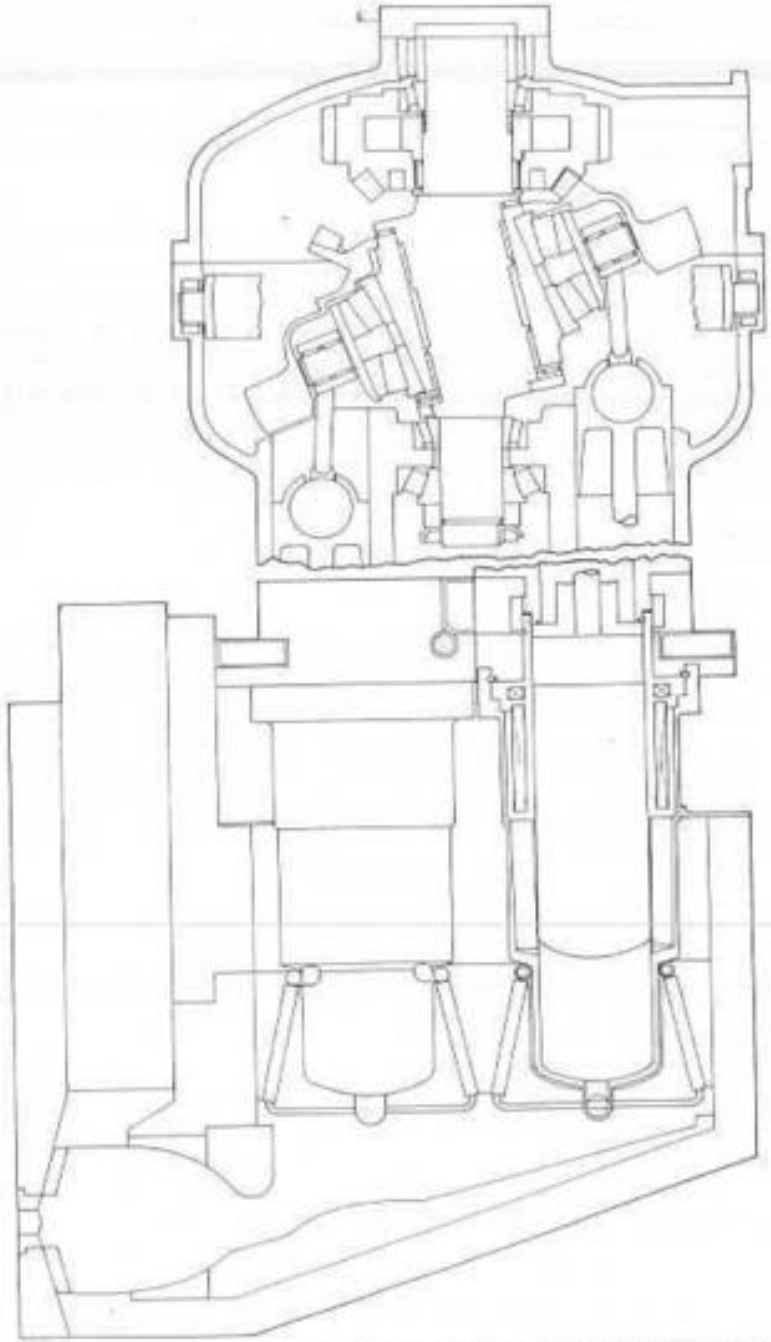


FIG. 7.8. Design layout of an advanced concept study engine for automotive propulsion

internally. As a consequence the tubes can be shortened and a substantial gain in dead space achieved thereby. Furthermore the tubes or fins attain a uniform temperature, hot spots are eliminated and so the *mean* heater temperature which is also now the maximum temperature can be elevated to the metallurgical limit.

Use of a sodium heat pipe or eutectic liquid metal loop allows the energy source to be located remotely from the engine. This is attractive for solar heated engines, and for cogeneration applications using municipal, agricultural, and industrial wastes in incinerators (probably with fluidized bed combustors). Indirect heating is also well suited for combination with Stirling engines in conjunction with thermal storage systems (thermal battery) such as may be used in automotive propulsion or underwater power systems.

A brief discussion of the Philip and General Motors experience with indirectly heated Stirling engines may be found in Chapters 12 and 13 respectively, with references to the source documents.

COOLER DESIGN

In principle Stirling engines may be air-cooled or water-cooled just as internal combustion engines are. However we have seen earlier that because the exhaust stack loss must be low, the cooling system of a Stirling engine must handle up to twice the load imposed on the cooling system of an internal combustion engine of similar power output. In addition to this, the efficiency of a Stirling engine falls markedly as the cooler temperature increases and is accompanied by deterioration in the mechanical properties of the polymer materials commonly used for sliding seals. Therefore it is desirable to have the cooler temperature at the minimum possible value.

This combination of factors makes direct air cooling of Stirling engines virtually impossible except in small model engines or larger but unpressurized low power, slow-running engines intended for long unattended operation.

Indirect air cooling is therefore obligatory for engines of moderate to high specific output using an intermediate liquid cooling loop connecting the engine and air/liquid 'radiator' (actually an atmospheric forced-convective heat exchanger). There is no particular advantage to be had by using liquids other than water except in cold climes where a mixture of water and ethylene glycol or alcohol may be necessary.

Water has excellent heat transfer properties and unbelievably high rates of heat transfer can be achieved with very moderate temperature potentials between the fluid and the wall. As a consequence the limiting heat transfer process will most likely prove to be the internal convective heat transfer between the working fluid and the cooler walls of the tubes

or fins. Just as much careful attention must be given to the design of the cooler as to the heater despite the initial appearance of the heater as the challenging unit using exotic materials and operating at the technological limit. As an example Kitzner (1977b), in discussing the design of the Philips/Ford Type 4–98 double acting engine with four cylinders and a swashplate drive, indicated that 72 heater tubes were sufficient but that 2440 tubes of 0.9 mm bore were considered necessary in the cooler. Of course the fact that the cooler operates near ambient temperature allows the use of lower cost metals (e.g. aluminium) and joining techniques (adhesives) not appropriate for high-temperature use.

REGENERATIVE HEAT EXCHANGERS

Ideal regenerator

Ideal regeneration was assumed in our previous discussion of both the Stirling cycle and Schmidt cycle of operation. Ideal regeneration is achieved when the fluid entering and leaving the matrix does so at one of two constant temperatures, T_E at the expansion end and T_C at the compression end of the matrix. This is possible only if operations are carried out infinitely slowly, if the heat-transfer coefficient or the area for heat transfer is infinite, or if the heat capacity of the fluid or matrix is zero or infinite, respectively.

In both the Stirling and Schmidt cycles there is no difference in the instantaneous pressure across the matrix, so that the ideal regenerator has no fluid friction. Further, in the case of the Stirling cycle, the void volume of the matrix is zero. In the Schmidt cycle, the void volume is an independently-chosen parameter, and is considered part of the total void volume of the system.

The form of the temperature field in the regenerative matrix is not significant for either the Stirling or Schmidt cycle, but is usually represented as a linear, or transitional, function along the length of the matrix. It is important in the Schmidt cycle, because the effective temperature of the dead space T_D is always taken as the arithmetic mean of the constant temperatures T_E and T_C.

Practical regenerator

The regenerator in a practical engine operates under conditions far removed from those assumed for the ideal case, discussed above. The temperatures of the working fluid at the inlet to the matrix are not constant, but vary with cyclic periodicity, because the processes of compression and expansion are not isothermal. The temperatures at the exit from the matrix are also variable, not only because of the inlet periodic-

FIG. 7.9. Thermal regenerator in counter-flow operations. (a) Hot fluid A enters matrix at constant inlet temperature during the hot blow. (b) Fluid A leaves matrix at a variable temperature always below the inlet valve, but increasing with time asymptotically to the inlet valve. (c) After the flow of fluid A ceases, cold fluid C enters the matrix at constant inlet temperature during the cold blow. (d) Fluid C leaves the matrix with a variable temperature always above the inlet valve, but decreasing with time asymptotically to the inlet valve.

flow conditions at the inlet to (or exit from) the matrix are not constant, but vary continually. The pressure, density, and velocity vary over an appreciable range, and the temperature varies over a more limited range.

Theory of regenerator operation

The most comprehensive treatment of thermal regenerators is that given by Jakob† comprising a distillation of the classical work of Hausen (1929, 1931), Nusselt (1927), Schumann (1929), and Anzelius (1926). Elsewhere, Iliffe (1948) has reviewed and extended the work of Hausen and others. Coppage and London (1953) have summarized and compared the various results presented in the literature, and Kays and London (1958) have established a rational basis for the design of regenerators, correlated with the design of other forms of compact heat-exchanger. Valuable contributions have been made also by Johnson (1952) and Tipler (1948). None of the work was directed specifically to the application of regenerators in Stirling engines, but was either of a fundamental nature or specific to gas-turbine applications.

Operating conditions

Various modes of regenerator operation may be postulated, but that which is generally of most interest is called the *state of cyclic operation*. This is the state obtained when, after repeated heating and cooling for a fixed time-cycle consisting of one heating and one cooling period, the temperature at any one point in the fluid (or the matrix) is then the same as it was a full cycle earlier.

Fig. 7.9 is a representation of a thermal regenerator in counterflow operation. In the state of cyclic operation the regenerator is assumed to function as follows. Hot fluid at a constant *inlet temperature*, entering from the left-hand end, passes through the matrix, gives up part of its heat, and leaves the right-hand end with a variable temperature, lower

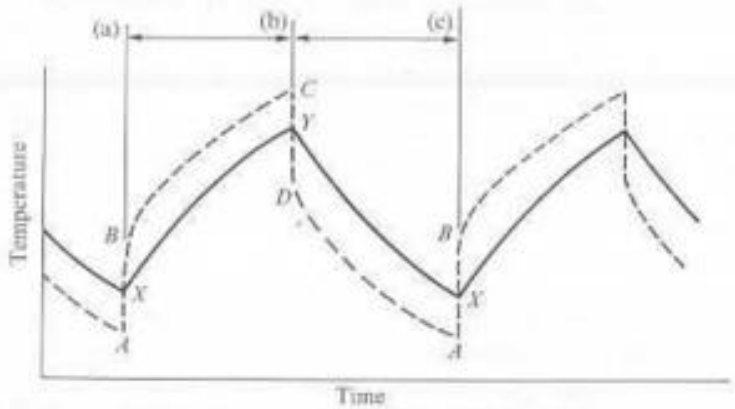


FIG. 7.10. Time-temperature variation of fluid and matrix in a thermal regenerator. The figure shows the possible form of the time–temperature variation at some interim point in a thermal regenerator in the state of cyclic operation. (a) to (b) is the hot-blow period. The fluid temperature increases from A to B as the flow is switched from the cold to hot fluid and thereafter increases towards the hot-fluid inlet temperature. The matrix temperature increases from X to Y during the hot-blow period due to heat transferred from the hot fluid to the matrix. At (b) the flow is switched to cold fluid and the period (b) to (c) is the cold-blow period. As the flow is switched the fluid temperature decreases from C to D and thereafter decreases asymptotically towards the constant temperature. During the cold blow the matrix temperature decreases from Y to X as heat is transferred from the matrix to the fluid.

than the inlet temperature. The supply of hot fluid is discontinued, and all the fluid is ejected from the matrix through the exit at the right. Cold fluid now enters at a constant inlet temperature from the right, passes through the matrix, is heated by absorbing heat from the matrix, and leaves at the left-hand end with a variable temperature above the inlet temperature. The cold fluid supply is discontinued, and all of the fluid is ejected from the cold end, to complete the cycle of operations.

Figure 7.10 shows the possible form of variation with time of the matrix temperature and fluid temperature, at one particular station in the matrix, with the regenerator in a state of cyclic operation. Fig. 7.11 shows the temperature field in a regenerator, for both fluid and matrix, at the instant of flow-reversal. The upper curves represent the temperature of the fluid and matrix at the end of heating-blow and the start of the cooling-blow. The lower curves represent temperature conditions at the end of cooling-blow and the start of heating-blow. At any particular station along the length of the matrix, the temperatures may fluctuate between the upper and lower curves, in a time-dependent relationship similar to that shown in Fig. 7.10. There are four periods in the cycle. Considering the passage of the hot fluid, the 'blow period' is the time taken for the total quantity of fluid to pass any joint in the regenerator; the 'reversal period' is the time which elapses between the entry of one

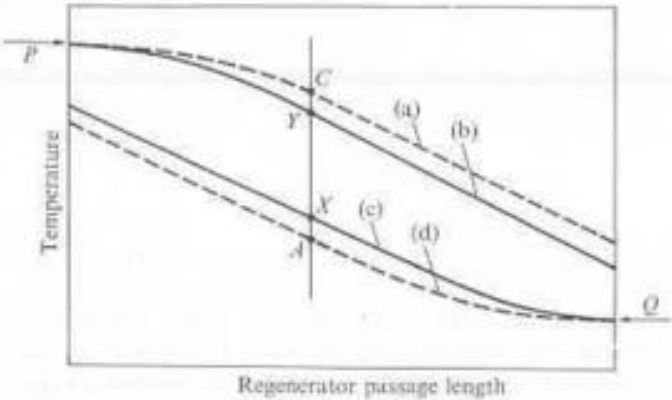


FIG. 7.11. Spatial temperature variation of fluid and matrix in a thermal regenerator. The figure shows the spatial temperature variation in a thermal regenerator, at the instant of flow-switching in the state of cyclic operation, with hot and cold fluids having the constant inlet-temperatures P and Q, respectively.
(a) Fluid temperature at end of the hot blow.
(b) Matrix temperature at end of the hot blow, and start of the cold blow.
(c) Matrix temperature at end of the cold blow.
(d) Fluid temperature at the end of the cold blow.
Points A, C, X, and Y correspond to the conditions represented by A, C, X, and Y in Fig. 7.10.

fluid and the entry of the other. Similar blow and reversal periods exist for the passage of the cold fluid. As Iliffe (1948) has pointed out, in practical regenerators the blow period is the same as the reversal period, since the last portion of fluid to enter is driven out by the other fluid through the port by which it came in. In the hypothetical ideal regenerator, the blow period is always less than the reversal period by the time taken for a gas particle to travel from one end of the regenerator to the other. Therefore, if this effect is ignored, we are assuming that the *time for a particle to pass through the regenerator is small compared with the total blow-time.*

Other significant simplifying assumptions have been found necessary to render the analysis of operation tractable. Some of these are summarized below.

(a) The thermal conductivity of the matrix must be simple. Nusselt considered four cases:
  (i) The thermal conductivity of the matrix is infinitely large. This means there would be no temperature difference in the matrix, and Nusselt's calculation shows that this type would have a poor performance.
  (ii) The thermal conductivity of the matrix is infinitely large, parallel to the fluid flow, and finite, normal to the fluid flow. In

practice, this may be approached by a very short regenerator, with a matrix composed of thick walls.

(iii) The thermal conductivity of the matrix is zero, parallel to the fluid flow, and infinitely large, normal to the fluid flow.

(iv) The thermal conductivity of the matrix is zero, parallel to the fluid flow, and finite, normal to the fluid flow.

Cases (iii) and (iv) correspond closest to the practical regenerator, but it is unfortunate that the analyses of these two cases are the most complicated. Schultz (1951), Tipler (1947), and Hahnemann (1948) have examined the effect of longitudinal heat conduction in the walls of regenerator passages, and have demonstrated this to have a negligible effect in certain cases. Saunders and Smoleniec (1948) state that 'for matrices built up in layers, such as gauzes, or matrices made of refractory, the conduction effect is almost certainly negligible'.

(b) The specific heats of the fluids and of the matrix material do not change with the temperature.

(c) The fluids flow in opposite directions, and have *inlet temperatures that are constant both over the flow-section and with time.*

(d) The *heat-transfer coefficients and fluid velocities are constant with time and space,* even though they may be different for the two fluids.

(e) The *rate of mass flow of either fluid is constant during the blow period,* even though it may be different for the two fluids, and the blow periods may be different.

Very little theoretical work appears to have been done on regenerators operating under conditions not fulfilling assumptions (b), (c), and (d), and most results are available for operation with equal blow-times and equal mass flow. However, Johnson (1952) and Saunders and Smoleniec (1948) have investigated this latter effect. Saunders and Smoleniec also considered the effect of variation in the specific heats of the fluid and matrix, for a particular case. They found the assumption of constant values, made in (b), resulted in less than one per cent error in the effectiveness.

Another interesting (but impractical) case, considered by Nusselt (1927), was for a regenerator with an infinitely-small reversal-period, and in which the fluids had been switched infinitely often. The theory for the case is simple, and corresponds to that for a 'recuperator', or normal continuous-counterflow heat/exchanger, in which the two fluids flow continuously, and are separated by metal walls.

Presentation of results

The performance results calculated for regenerators, assumed to be operating under the conditions discussed above, have been presented in a

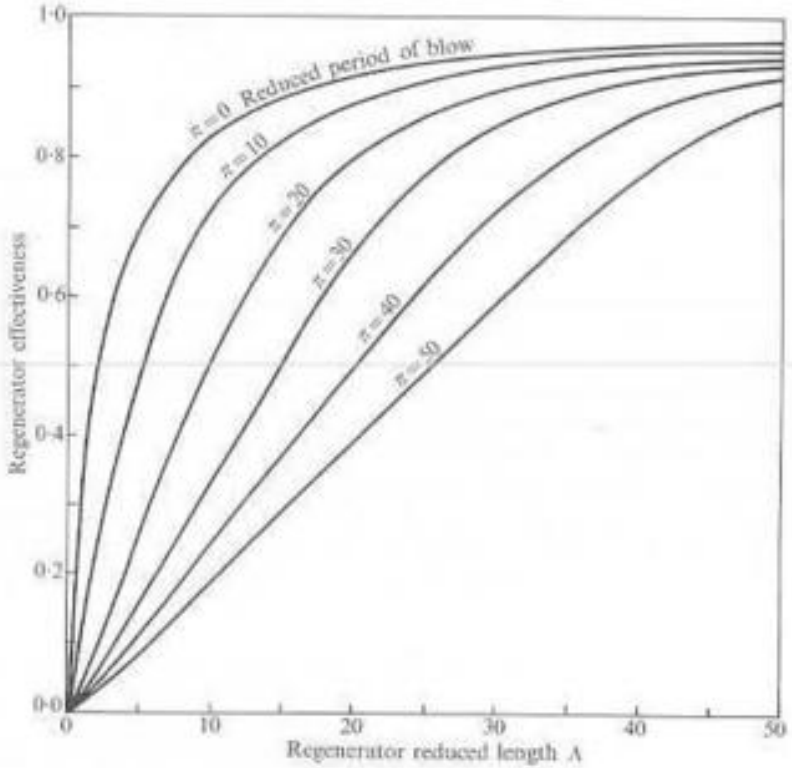


FIG. 7.12. Regenerator effectiveness as a function of the reduced length Λ and reduced period π (after Hausen).

Hausen, and reproduced in Fig. 7.12. These have been supplemented by similar curves calculated by Johnson and Saunders and Smoleniec. The curves show that the effect on regenerator effectiveness of variation in two dimensionless parameters called (after Hausen), the 'reduced length' (Λ) and the 'reduced period' (Π). The reduced length (in the flow direction) is defined by

$$\Lambda = hAL/VC_p$$

where

h = heat-transfer coefficient between fluid and matrix, per unit surface area,

A = matrix surface area per unit length,

V = fluid-volume flow rate,

C_p = specific heat of the fluid

The reduced period is defined by

$$\Pi = hAZ/MC,$$

where h and A are as defined above,

M = mass of matrix material,
C = specific heat of matrix material,
Z = blow-time.

Frequently Λ and Π are combined by the quotient

$$\Pi/\Lambda = U = (VC_p/MC)(Z/L),$$

and called the 'utilization factor', representing the ratio of the sensible-heat capacity of the fluid per blow to the heat-storage capacity of the matrix.

In practice, regenerators may have different reduced periods and reduced lengths for the hot and cold blows, so that there are four factors to be considered. In these cases, Saunders and Smoleniec recommend that average values be used, suggesting (on the evidence of calculations carried out by Johnson), that the error is small. This is probably because, even when the actual blow-times are unequal, the reduced periods are much nearer equality, since a reduction in the actual blow-time Z is usually accompanied by an increase in the rate of fluid flow V.

The usefulness of the concept of two reduced dimensionless parameters and the curves of regenerator effectiveness is limited by the accuracy of the heat-transfer data. This is generally measured experimentally using the 'single-blow' transient technique, first described by Furnas (1932) and, later, by Saunders and Ford (1940), Johnson (1952), Saunders and Smoleniec (1948), Coppage (1952), Rapley (1960), Vasishta (1969), and Wan (1971). In this technique, the matrix is subjected to a flow of hot fluid, entering with a constant inlet-temperature, and the change in the exit-temperature is measured against time. The theory for 'single-blow' operation was first given by Schumann (1929), and may be used to extract, from the measured data, the heat-transfer coefficient relevant to the particular tested matrix. Very careful measurements are required, and there is, in fact, some doubt as to whether this data can be applied to regenerators operating cyclically. A reasonable amount of heat-transfer data is contained in the references given above, but comparison is difficult because several slightly different forms of presentation have been used.

Application of theory to regeneration in Stirling engines

Theories of regenerator operation, discussed above, were developed initially for air-liquefaction and gas-separation plants and for air preheaters for boilers. These plants are large and, in general, two regenerators

per unit are used, one being heated and the other being cooled. The blow-times are very long, ranging from ten minutes to several hours.

Later the theory was adapted and extended, during application of regenerative heat exchangers to gas turbines. Here the blow-times are much shorter. Coppage and London (1953) refer to 'a reversal time of a quarter of a second (two complete cycles per second) which is near the maximum permissible frequency without undue "carry-over loss"' and, again, 'the idealization of no flow-mixing is closely met when the flow-passage length is short, and such shortness of length appears to be good design procedure for the most suitable types of surface'. Most regenerators in gas-turbine engines have a relatively large frontal area and a short flow length, so that, although the blow-time is short, the residence-time of the particle in the matrix is also very short.

The above theory seems applicable, in a reasonably realistic way, to regenerators used in gas-turbine engines and air preheaters, but not applicable to regenerators used in Stirling-cycle engines. The theory is based on assumptions which, clearly, do not apply in the Stirling engine. The most important of these is, perhaps, that the time for a particle to pass through the matrix is small compared to the total blow-time. In a Stirling engine the blow-times are exceedingly short. For example, at the moderate engine speed of 1200 rev/min, or 20 c/s, the blow-time is ten times less than the permissible minimum in a gas turbine. We saw earlier (Fig. 7.4) that the blow-times are so short that no particle ever passes right through the matrix. From Fig. 7.5 we saw the actual net flow time through the matrix was about half the complete cycle time, the remaining time being occupied in either filling, or emptying, the dead space. The heat-transfer process that occurs must be very complex, involving a repetitive fluid to matrix, matrix to fluid, fluid to matrix, cyclic relationship, rather like the water bucket passed from hand to hand in a fire-fighting operation. Other important assumptions of the theory are that the inlet conditions, temperature, rates of mass flow, and fluid velocity remain constant with time. Clearly, this is not true in *any* Stirling-cycle regenerator for the inlet conditions vary constantly, and Fig. 7.5 shows extreme variation in rate of mass flow. The maximum rate of *net* flow through the matrix is only about half the maximum flow into and out of the expansion space.

Attempts to analyse the regenerators of Stirling engines by any of the recommended procedures require the adoption of 'average' conditions for the flow. Such gross approximation is required to determine these 'average' values that the value of the ultimate result is thought to be highly questionable. No recommendation can be made, at this stage, for the application of any theories of regenerator operation as aids to regenerator design.

Although the situation is unsatisfactory at present, there is reason to hope for improvements. Smith and co-workers at the Massachusetts Institute of Technology have made a promising start (Qvale and Smith, 1968, 1969). They discuss an approximate solution for the thermal performance of a Stirling-cycle regenerator, in which there is provision for non-steady pressure (and mass-flow) conditions, including the possibility of sinusoidal variation, with a phase difference in the peak values. By assuming a second-order polynomial form for the temperature field in the regenerator, a closed solution was obtained for the net enthalpy flux. The theory remains highly idealized, with the assumption that the gas temperature and matrix temperature, at one location, are practically constant with time, and that there are no wall (or fluid-friction) effects. However, at present, the theory does not appear to be sufficiently well developed to be of direct use in regenerator design. Köhler and co-workers at the Philips Laboratories, Eindhoven, have done more research on regenerators in Stirling engines than anybody else, but, unfortunately, little of this work has been published. Although Dr. Köhler presented a series of lectures on regenerators at the Technische Hochschule, Delft, in 1969, these were never published in the open literature.

Nodal analysis of Stirling engines, exemplified by Schock (1978b), Urielli (1977), Finkelstein (1975a), and Tew (1978)†, attempts to simulate exactly the operation of all components in a Stirling system including the regenerator. This is accomplished by dividing the system into a number of cells and applying rigorous thermofluid analysis to each node in turn.

Nodal analysis requires access to large high-speed computers and so is not suitable for general use. However there is little doubt that the time is near at hand when sufficient experience will be accumulated for the preparation of a series of consolidated design charts for general use in the selection of an optimum regenerator configuration.

Experimental performance

Little appears to have been published about the effect of imperfect regeneration on the performance of Stirling-cycle machines, or about experimental work on regenerators, tested under conditions approximating to those present in a Stirling engine.

Davies and Singham (1951) carried out some experiments on a small thermal regenerator, composed of brass and copper wire gauzes, subjected to the oscillating flow of a constant volume of air, at atmospheric pressure and at a frequency of five cycles per second. The air was heated on one side of the regenerator, and cooled on the other. Continuous

records of the temperature of the air were taken on both sides of the matrix. It was concluded from these experiments that:

(1) for a given gauze matrix, the regenerator efficiency increases with the matrix weight, but the improvement takes place at a progressively diminishing rate,
(2) for a given matrix weight, the regenerator efficiency increases with decreases in the diameter of the gauze wire.

Tests, with equal weights of brass and copper gauze, gave approximately the same values for regenerator efficiency. Thus, although the copper had a thermal conductivity about three times greater than that of the brass, this appears to have had little effect. It was concluded that, with fine wires of these materials, the conductivity lag is extremely small. In these tests, the regenerator efficiency was obtained by analysis of the continuous fluid-temperature records, measured at each end of the regenerator matrix.

Experiments by Walker (1961a), with a series of different regenerators on the Philips gas refrigerating machine, have confirmed the second conclusion reached by Davies and Singham, namely, that reduction in wire-diameter increases the effectiveness of the regenerator. The criterion of performance was taken to be the quantity of liquid air produced by the machine, operating at a constant speed and the mean pressure of the working fluid. A reduction in the wire-diameter, with approximately constant matrix weight and porosity, resulted in an increase in the surface area for heat transfer.

Work by Murray, Martin, Bayley and Rapley (1961) has shed some light on the performance of regenerators under sinusoidal flow conditions. It was found that frequency appeared to have little effect on the heat-transfer process, but the shape of the wave has a significant effect. With pulsating flow, the effectiveness of the tested gauze matrices was appreciably below that obtained under steady flow conditions. With flame-trap matrices, an improvement in the heat-transfer rate, in unsteady flow, was noticed.

Regenerator design—a practical guide

In the absence of adequate theoretical assistance in regenerator design, a few helpful suggestions are offered below. They are not intended to be fundamental rules.

The regenerator designer must attempt to solve the problem of satisfying a number of conflicting requirements. To minimize the temperature excursion of the matrix, and thus improve the overall effectiveness of the regenerator, the ratio of the heat capacity of the matrix to that of the gas ($M_m C_m / M_g C_g$) should be a maximum. This

On the other hand, the fluid-friction loss must be limited. We saw, in Chapter 3, that the effect of the pressure drop across the matrix is to reduce the range of the pressure excursion in the expansion space, thereby adversely affecting the area of the expansion-space P–V diagram. This reduces the net work-output and thermal efficiency of a prime mover, and the amount of heat lifted and coefficient of performance of a cooling engine. The fluid-friction loss is minimized by a *small, highly porous matrix.*

A third, and most important, consideration is that of dead space. The size of the dead space influences the ratio of maximum to minimum volume of working space, and this directly affects the ratio of maximum to minimum pressure. For maximum specific output, both ratios should be as high as possible, and, for this to be achieved, the dead space should be made as small as possible. This can be achieved by a *small, dense matrix.*

To improve the heat-transfer performance, and establish the minimum temperature difference between the matrix and the fluid, it is necessary to expose the maximum surface area for heat transfer between the fluid and matrix. Therefore, the matrix should be *finely divided*, with preferential thermal conduction at a maximum normal to the flow, and minimum in the direction of the flow.

Finally, it is important to appreciate that the regenerator acts as an exceedingly effective filter of the working fluid, so that any oil, or grease, particles are retained in the fine flow-passages. In the case of a cooling engine, any impurities in the working fluid that condense in the low-temperature region of the expansion space will accumulate in the regenerator. This build-up is cumulative, and has the effect of increasing the fluid-friction losses, so that the pressure excursion in the expansion space is decreased, and the performance of the cooling engine progressively diminishes. In the case of the prime mover, any accumulation of oil particles in the regenerator inhibits the flow of working fluid, and increases the pressure loss. The temperature in the expansion space thereby increases, and may be even further increased, because more fuel is supplied in an attempt to restore the lost power. This increase in temperature carbonizes the fuel, thereby further blocking the flow passage, and the process continues in cumulative fashion, until catastrophic overheating of the engine occurs. From this aspect, the regenerator should offer minimal obstruction to the flow.

Thus, we have the following desirable characteristics for a regenerative matrix:

for maximum heat capacity—a large, solid matrix,
for minimum flow losses—a small, highly porous matrix,
for minimum dead space—a small, dense matrix,

for maximum heat transfer—a large, finely-divided matrix,
for minimum contamination—a matrix with no obstruction.

Clearly, it is impossible to satisfy all these conflicting requirements. With our present understanding of the cycle, it is not possible to quantify the relative significance of the various aspects.

Prime movers

In most engine designs, considerable attention is given to the regenerator, and comparatively little to the problem of the heater and cooler. As a consequence, heat transfer to and from the engine is poor, and the engine fails to operate satisfactorily. This stimulates yet further interest in the regenerator, with the investment of much experimental effort in trying different regenerator arrangements. Frequently, surprise is expressed when this produces absolutely no effect on engine performance except that a reduction in size results in improved operation. Surprise becomes confusion when the experiments are extended to the point of diminishing the regenerator to such an extent that it has, in effect, been completely removed from the engine. It is a matter of experience that, in small low-pressure engines, removal of the regenerator nearly always results in improved performance. This is because the gains due to a reduction in the dead space, and, to a lesser extent, a reduction in the conducting path of the regenerator enclosure and in fluid-friction losses, more than offset the loss of thermal capacity and area for heat transfer of the regenerative matrix.

In most small, low-speed machines (up to, say, 5 cm bore, with less than 5–6 atm pressure and operating at below 1000 rev/min), it is adequate (for a start, at least), not to incorporate a formal regenerator in the engine design, but, rather, to depend on the action of a regenerative annulus around the displacer.

One type of displacer system with a regenerative annular duct, used with success by William Beale in small free-piston Stirling engines and also by the writer, is shown in Fig. 7.13. The displacer is made of a thin-walled, low conductivity stainless-steel tube, closed at the hot end by an inverted 'top-hat' section which is machined from a solid bar so as to be a close fit in the tube. After assembly, the seam may be gas-welded, and the joint section trimmed and trued by grinding. Inside the displacer, a series of radiation shields may be provided, as shown, either cut from solid material or fabricated. The lower end of the displacer is closed by another closely fitting plate. Since this end operates in the cooled zone, the end plate can be of light alloy or stainless steel. An epoxy-cement joint has been found adequate for fixing. Good results have been obtained with displacers about three diameters long. The displacer operates

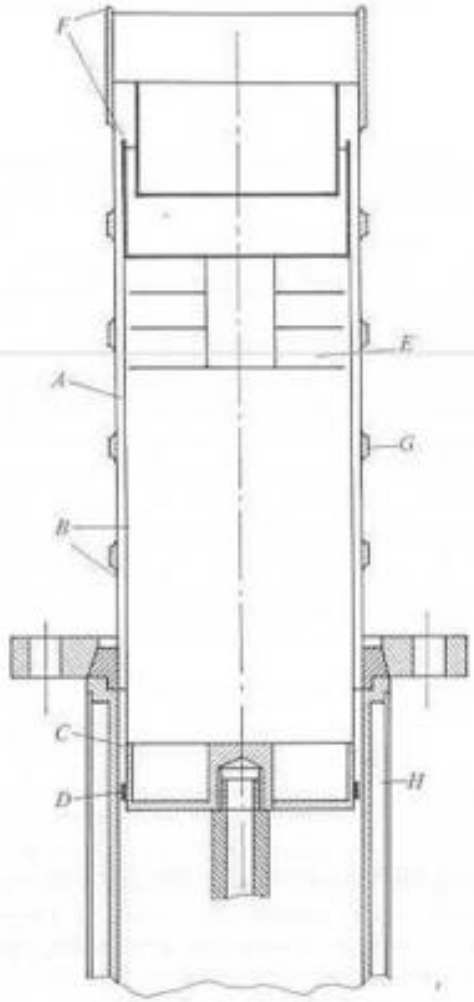


FIG. 7.13. Detail of regenerative annulus.
A, regenerative annular gap 0.015–0.030 in.
B, thin-wall sections to minimize thermal conduction.
C, epoxy joint.
D, rulon guide-ring.
E, radiation shield.
F, welded seams.
G, stiffening rings.
H, cooling jacket.

in a cylinder, also of low-conductivity stainless steel and having a thin cross-section, except for occasional circumferential stiffening rings, left during manufacture. The top of the cylinder is closed by another inverted 'top-hat' section with an external welded joint. The lower end of the cylinder is, of course, attached by a flange to the cooled compression-space cylinder. The cylinder of the displacer may actually be shorter than the displacer, so that the bottom cooled end of the displacer operates within the compression-space cylinder. This makes it possible to mount a guide ring of P.T.F.E.-based material around the lower end of the displacer, and have it operate on a cooled wall-section.

The annulus formed between the displacer and the cylinder is then the flow passage, connecting the expansion and compression spaces. It acts as a regenerator, since the top end is always in the heated section, and the bottom end always in the cooled section. It is a simple device, but remarkably effective if the displacer and cylinder wall are reduced to very thin sections, to minimize thermal conduction losses. The gap between the displacer and the cylinder wall is a critical dimension with regard to heat transfer, and should be between 0.015 and 0.030 inches. It is important, also, that a regular annulus be established with a uniform circumferential gap, to equalize heat-transfer and fluid-flow effects. The problem of heat transfer in an annular duct, with an axial temperature gradient and a reciprocating internal member, does not appear to have been studied, and might be a topic of considerable appeal for a university research program.

The limits of applicability of the regenerative annular duct are not known, but it is likely that the system would become less and less effective as the cylinder bore, cylinder pressure, or engine speed were increased. The initial inadequacy would appear, perhaps, in the heater section, and some improvement might be gained by providing an extended surface for heat transfer by using internal finning: but this would be difficult to accomplish without substantially increasing the dead space. Eventually, it would become necessary to resort to increasingly complicated heaters, probably of external-tubular form, and it is at this point that a regenerative matrix becomes worthwhile. By this time, however, one is developing an engine of advanced form that would probably evolve with close similarities to machines of the Philips type.

Cooling engines

The regenerator in a cooling engine appears to be much more important than in a prime mover, but by happy coincidence, the materials problem is less severe.

low-conductivity compressed-paper sleeve, and made up from the random packing of short lengths of copper wire 0.001 inches in diameter, mounted in annular form around the displacer. The author has found woven wire mesh of copper and phosphor bronze to be effective packing for regenerative matrices. These can be had in a wide variety of mesh densities and wire sizes. As the mesh density increases and the wire diameter decreases, the price per unit area increases very steeply to the point where it is doubtful that the material could be used for production machines. An annular regenerator is very expensive, because the centre section, punched from the screen, is 'wasted'. Wire screens can be 'sintered' easily, to form a stable semi-rigid block. One way is to pack the screen in some form that can be loaded with a weight. Then, the wire screen is cleaned by immersion in nitric (or hydrochloric) acid, and the loaded assembly is heated for a short period in a furnace with a reducing atmosphere. On removal, it will be found that the screen has 'sintered' to a solid assembly that can be lightly machined. It is important to arrange the screen so that the wires are normal to the axes of flow, otherwise the axial conduction may be too high. Sintering with light loading does not appear to significantly increase the axial conduction of the screens, and because of a considerably reduced porosity, it does improve the pack.

It is not possible to make specific recommendations for design, although the following points merit consideration. The wire used should be fine (0.001 to 0.002 inches in diameter), closely packed, and compressed, to minimize voids. A dead-space ratio of one is a good target, but difficult to achieve, and at least half the dead space should be regenerator void-volume. As a rule, the regenerator arrangement should be of such proportions that the total cross-section of the duct is equal to that of a right normal cylinder, having a diameter equivalent to its length.

Heat-transfer and fluid-friction characteristics of dense-mesh wire screens

The heat-transfer and fluid-friction characteristics of a variety of porous media were given by Coppage and London (1956) and have been supplemented by later data. However, little data on the flow in *dense* wire screens has been published for the size ranges of interest when considering the regenerators of Stirling-cycle cooling engines. Values measured at the University of Calgary by Vasishta (1969), and by Wan (1971), are included here, but no other values are known with which these results may be compared. To validate the experimental apparatus, Vasishta did obtain some results for stainless-steel mesh, in sizes comparable with those studied by Coppage, and found the results to be in close agreement.

The heat-transfer and fluid-friction data for two sizes of screen are given in Figs. 7.14 and 7.15 respectively, both sizes of screen were woven

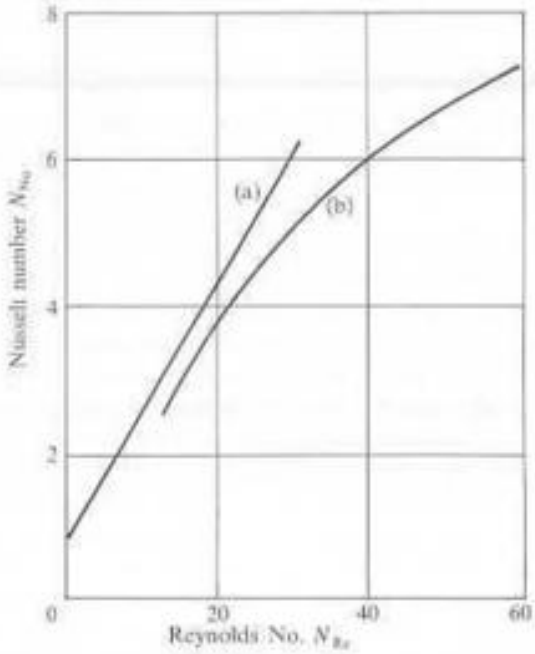


FIG. 7.14. Heat-transfer characteristics of dense-mesh wire screen (after Walker 1972).
(a) 400 × 400 strands per inch, 0.001 inch wire diameter.
(b) 200 × 200 strands per inch, 0.002 inch wire diameter.

from phosphor-bronze wire, having the following composition:

Tin 3.5–3.8 per cent; Phosphorous 0.3–0.35 per cent; Iron 0.1 per cent; Lead 0.05 per cent; Zinc 0.3 per cent; Copper rem.
Density 8875 kg m^{-3} (554 lb ft^{-3}); Thermal conductivity 81.3 Wm^{-1} K^{-1} (47 Btu h^{-1} ft^{-1} °F^{-1})
Specific heat 0.435 kJ kg^{-1}K^{-1} (0.104 Btu lb^{-1} °F^{-1})

The two screen-sizes investigated were

(a) 200 × 200 strands per inch, 0.0021 in wire diameter,
(b) 400 × 400 strands per inch, 0.001 in wire diameter.

The heat-transfer characteristics are presented as the Nusselt number N_{Nu}, as a function of the Reynolds number N_{Re}, defined as follows:

$$N_{Nu} = (4r_h/k)(h/f),\ N_{Re} = \rho_f V d/\mu_f = (4r_h/\mu_f p)(W_f/A_f),$$

where r_h = calculated hydraulic radius of the screen, h = heat-transfer coefficient, k_f = thermal conductivity of the fluid, ρ_f = density of fluid, V = volume flow rate of fluid in matrix, W_f = mass flow rate of fluid in

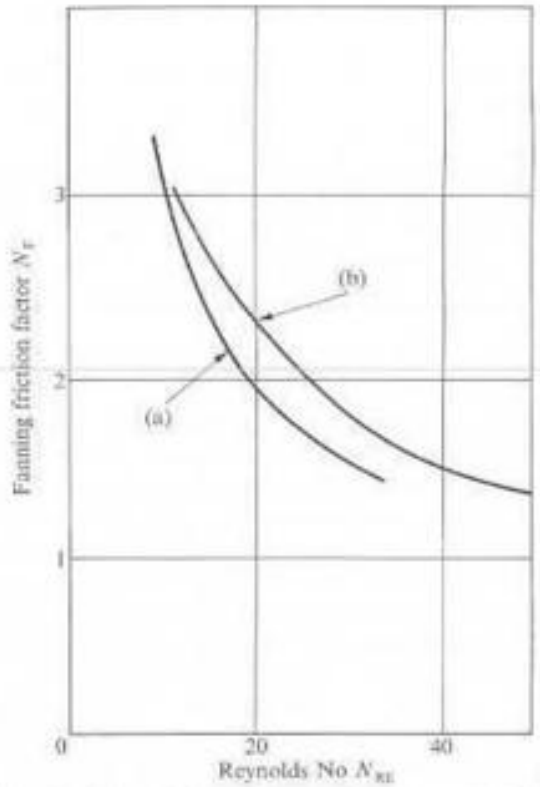


FIG. 7.15. Fluid-friction characteristics of dense-mesh wire screen (after Walker 1972).
(a) 400×400 strands per inch, 0.001 inch wire diameter.
(b) 200×200 strands per inch, 0.002 inch wire diameter.

matrix, A_f = frontal area, p = calculated porosity, μ_f = dynamic viscosity of fluid, where p = (volume of matrix − volume of metal)/volume of matrix and r_h = total volume of connected void spaces/total surface area = volume of matrix × porosity/total surface area.

The fluid-friction characteristics are presented as the Fanning friction factor N_F, as a function of the Reynolds number N_{Re}, defined as follows:

$$N_F = 2\rho_f \cdot \Delta P \cdot r_h \cdot p^2/nLG_A^2,$$

where ΔP = pressure drop, n = number of layers of screen, L = length of matrix, G_A = mass flow per unit area, and the remainder are as defined above.

PREHEATERS

A preheater is not necessary to the operation of a Stirling engine. It is a desirable accessory for all but small primitive engines and it is virtually

The function of the preheater is to heat incoming air *en route* to the combustion chamber using thermal energy in the departing stream of hot combustion products. This saves fuel and also endows the engine with the advantageous characteristic of a 'cool' exhaust. Exhaust-gas/inlet-air preheaters are sometimes also used on steam (Rankine cycle) and gas turbine (Brayton cycle) engines. When thus equipped they are said to operate on a 'regenerative cycle'.

Therefore, complying with customary engineering usage, a Stirling engine equipped with a preheater may also be said to be operating on a regenerative cycle. Thus in the Stirling engine there are two regenerators to consider: (a) the *internal* regenerator interposed between the cooler and the heater through which flows the working fluid at high pressure and (b) the *external* regenerator which is the exhaust-gas/inlet-air preheat heat exchanger operating at, substantially, atmospheric pressure.

To add yet further confusion the Stirling engine external regenerator, or the exhaust-gas/inlet-air preheater, may be a heat exchanger that is of the *regenerative* or alternatively of the *recuperative* type.

A regenerative heat exchanger consists of a porous matrix of finely divided material through which flows *alternate* streams of hot and cold fluid. A recuperative heat exchanger is one where separate flow channels are provided for the hot and cold fluid streams and through which, in most cases, the fluid flows continuously.

Both recuperative and regenerative preheaters have been applied to Stirling engines. Early Philips engines all used recuperative preheaters, one of which is shown in detail in Fig. 12.7. This led to the characteristic large cylinder head enclosure that can be seen on the engine illustrated in Figs. 12.12 and 12.18. Recuperative exchangers can be made that are highly effective from the aspect of heat transfer. They are limited in effectiveness by the surface area for heat transfer that can be designed into a given size of envelope, a given mass of metal, or manufactured to a given cost. Units of the high heat transfer effectiveness required in Stirling engines tend to be relatively large, heavy, and very expensive assemblies of crimped or pre-formed thin-wall stainless-steel plates with inlet and exhaust headers or manifolds.

The MAN/MWM accordion type air preheater shown in Fig. 14.4 is a promising approach to a low-cost effective recuperative heat exchanger. The core of the exchanger is a single strip of thin-wall aluminium folded repeatedly and contained in a metal case so that exhaust gas flows on one side of the folded sheet and inlet air on the other.

Cleaning is a particular difficulty with recuperative heat exchangers of compact form with fine flow passages. Condensed combustion products or

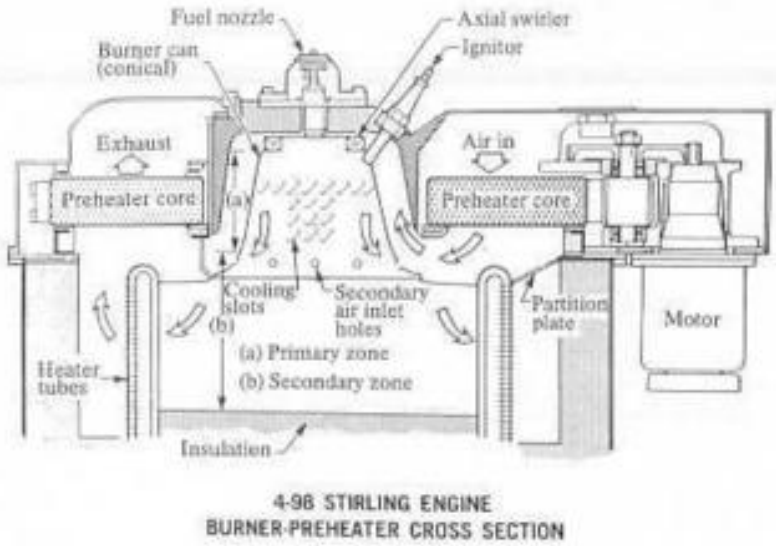


FIG. 7.16. Cross-section of Philips-Ford 4-98 Stirling engine burner preheater.

Regenerative heat exchangers offer the prospect of air preheaters that are lighter in weight, less bulky, cheaper, and with reduced tendency to fouling than recuperative exchangers.

Regenerative exchangers are used in the Philips/Ford multicylinder engines with swashplate drive which are under development for automotive use. Fig. 7.16 is a cross-section of the hot parts assembly of a Philips/Ford automotive Stirling engine showing the regenerative heat exchanger (called on the drawing the preheater core). Similar units are incorporated in Stirling engines for automotive use being developed by United Stirling of Sweden.

In all these cases the regenerative exchanger is of the form of the 'thermal wheel' shown in Fig. 7.17. A flat disc of porous matrix material, usually ceramic but sometimes metal, is contained within two adjacent ducts through which the hot and cold fluids are passing. The axis of the disc is parallel to the fluid motion. The disc is caused to rotate slowly so that a given element of the disc moves alternately through the exhaust gas and inlet air streams and a regenerative heat exchanger is thus created. If the fluid flows are in opposite directions an appreciable self-cleaning effect is obtained.

The 'thermal-wheel' regenerative heat exchanger has been studied extensively in connection with the development of gas-turbine engines for vehicular applications, and the technology is directly applicable to the Stirling-engine air preheater. The disc may be fabricated from any

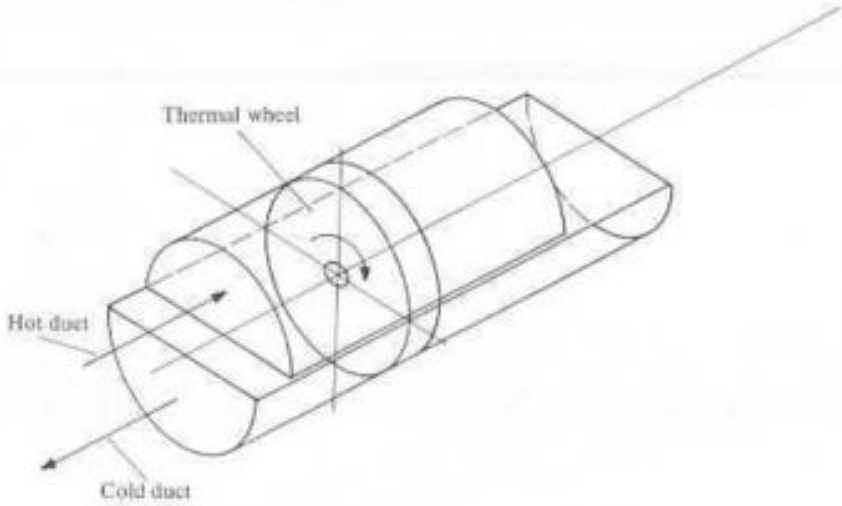


FIG. 7.17. 'Thermal wheel' regenerative heat exchanger

suitable porous material but ceramic materials are preferred. This is because their extremely low coefficient of thermal expansion endows the ceramic thermal wheel with the potential capacity to sustain extended use while experiencing repeated thermal and cooling cycles when rotated in the hot and cold fluid streams. In many metallic materials, the repeated heating and cooling causes cracks to appear and distortion to occur, a condition described generally as thermal fatigue.

The vehicular gas turbine literature contains many papers relating to thermal-wheel regenerative heat exchangers and progress is reported regularly at the semi-annual Vehicle Contractor Coordination meetings of the U.S. Department of Energy, the American Society of Mechanical Engineers International Gas Turbine Conferences, and the Annual Intersociety Energy Conversion Engineering Conferences.

8 WORKING FLUIDS IN STIRLING ENGINES

INTRODUCTION

VARIOUS working fluids may be used in Stirling engines. With few exceptions, all the regenerative engines of the nineteenth century used air. Indeed, they were called 'air engines' or 'hot-air engines'. Most of them operated at near-atmospheric pressure, although in some, the pressure level was increased to a few atmospheres above the ambient. Air was cheap, and readily available, so absolute sealing was not imperative to their operation. The engines were large, slow-running, low-power machines having a thermal efficiency of a few per cent at the most. Above all else they were safe compared with the boilers of the steam engines then also widely used. Parenthetically it can be noted that water for the steam engines was the other fluid as readily available and as cheap as air.

A century later the early Philips engines also used air as the working fluid. These were small machines in the main but at least one 22 kW (30 hp) double-acting Vee engine was developed (van Beukering 1973). The Philips air engines were radically different to the machines of the earlier time. They were fast (2000 revolutions per minute) and highly pressured (1.37 MN/m^2 or 200 lbs per sq in) so that the power levels and efficiencies were high compared with the nineteenth century machines.

Later on, in experiments with air engines working as refrigerators, Philips used hydrogen and helium as working fluids. They achieved remarkable results, attaining temperatures sufficiently low to liquefy air (80 K) by condensation on the cold cylinder head of the expansion space of a single-stage machine. Thereafter, hydrogen or helium was used as the working fluid, with minor exceptions, for all subsequent Philips engine developments (from 1954 onwards) (Meijer 1959a).

Hydrogen was mainly used, for in those days helium was more expensive and less abundant than it is today. The author recalls that in England, in 1956, prices for hydrogen and helium were respectively 4 pence and 21 shillings (252 pence) per cu ft—a factor of 63. In Canada, in 1978, the price for gas in cylinders of 200 cu ft capacity was 7.4 cents and 13.2 cents per cu ft for hydrogen and helium respectively—a factor of less than two.

Other heavier gases may be used as working fluids in Stirling engines but, for the reasons discussed later, would be less efficacious than the light gases, hydrogen or helium. Air is still used in small model engines but no other gases are widely used except in experimental units. For example, argon was the working fluid of the miniature Stirling engine in the Westinghouse/Philips artificial heart system (see Chapter 17).

one-component, gaseous working fluids discussed above. John Malone (1931) in England achieved remarkable results using liquid working fluids including water, gasoline, glycerin, etc. Water was also used as the working fluid in the Thermoelectron tidal regenerator engine incorporated in another artificial heart system (see Chapter 17).

Again, some early studies are in progress of Stirling engine systems with dissociating working fluids or with compound working fluids having a component that periodically changes phase from liquid to gas. These are described below.

GASEOUS WORKING FLUIDS

Theoretical comparisons

The first numerical comparison of different working fluids was given by Meijer (1970a) with the presentation of the characteristics reproduced in Fig. 8.1. This figure was a summary of extensive computer optimization studies carried out by Philips using their Stirling engine simulation computer program. All the results referred to large engines of 165 kW (225 brake hp) per cylinder having a heater temperature of 700 °C (1295 °F), cooler temperature of 25 °C (77 °F) and maximum gas pressure of 110 MN/m^2 (15 954 lbs per sq in). The overall efficiency of the

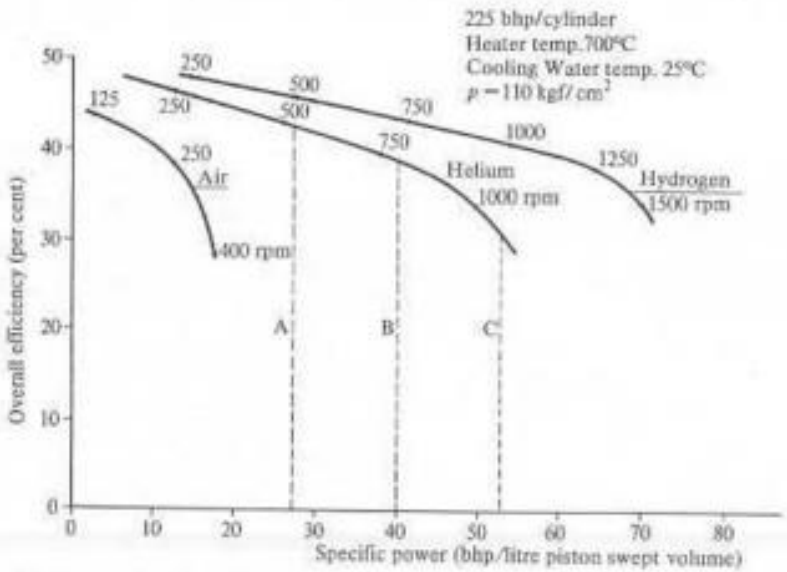


FIG. 8.1. Comparison of calculated performance for Stirling engines with different working fluids. Engines have the same output of 165 kW (225 brake hp) per cylinder and have been optimized for the maximum possible efficiency with heater and cooler temperatures of

engine, η_{bsth}, was shown as a function of the power output per unit displacement, in horsepower per litre. The *size* of the engine decreased as the curve moved from left to right of the figure.

Three curves were given, for air, for helium, and for hydrogen. At points on each curve the engine speed was quoted corresponding with the maximum efficiency and engine output. The engine speed increased as the curve moved to the right of the figure.

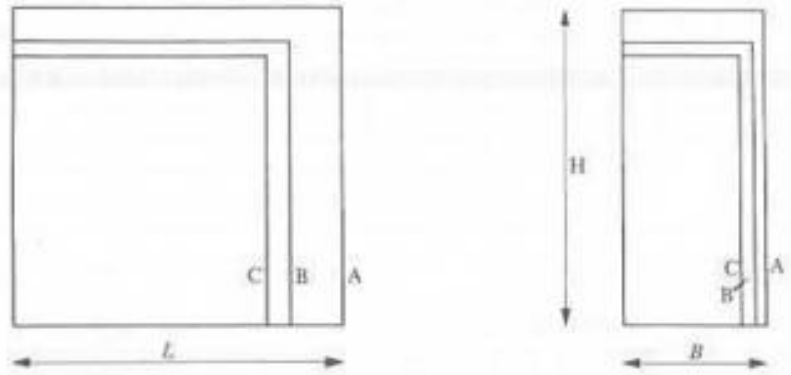
Near the extreme left of the figure there was little difference between the three curves. At a speed of 250 revolutions per minute the air engine had a somewhat lower efficiency, 38 per cent compared with 47 for helium and perhaps 49 with hydrogen. However, the power density was not markedly different, at about 8.9 W/cm³ (12 hp per litre). This suggested that in low-power, slow-running engines there was little or no advantage in thermodynamic operation to be gained using hydrogen or helium compared with air. However, with air as the working fluid the requirements for fluid sealing are greatly relaxed and less reservoir storage of surplus fluid is necessary. The air can be simply replaced as required by a small compressor. Thus, small, low-power, stationary electric generators with extended life requirements and unsophisticated design are as likely to be using air for the working fluid as hydrogen or helium.

Moving to the right of Fig. 8.1 it becomes clear that air cannot be used for high speed engines of high specific output. Further, at the highest speeds and power levels, hydrogen becomes significantly superior to helium. Therefore, in automotive applications where power density is vitally important, it is likely that hydrogen will be the preferred working fluid. A contributory and increasingly significant benefit is that the thermal efficiency with hydrogen can be appreciably higher.

Helium would likely be selected on considerations of safety for use in confined situations: ships, underwater power systems, total energy plants, heat pumps, or stationary generators in buildings. Hydrogen is highly reactive with oxygen with extremely wide flammability limits. Helium is inert.

Outline designs for a large 660 kW (900 hp) 4-cylinder engine were presented by Meijer (1970a) using the results given in Fig. 8.1. The resultant engine design envelopes are shown in Fig. 8.2 and are presumed to be rhombic drive engines. The subsequent development of double-acting machines would allow further reduction to about half the sizes given in Fig. 8.2.

Curves to show the effects of heater temperature and maximum pressure with hydrogen as the working fluid were also given by Meijer (1970a) and are reproduced in Fig. 8.3. As before, the efficiency was shown as a function of the brake power output per litre of piston swept



Stirling engine		Speed	Calc overall eff.	L	H	B
(900 bhp 4 cylinders)	A	525 rpm	43·5%	2·8	2·7	1·1 m
Helium p_m=110 atm	B	810 rpm	39 %	2·3	2·4	1·0 m
Heater temp 700°C	C	1260 rpm	31 %	2·1	2·3	0·95 m
Cooler temp. 25°C						

FIG. 8.2. Comparative size envelopes for a 660 kW (900 brake hp) 4-cylinder engine with helium as the working fluid at the three different operating states shown on Fig. 1. Engines are rhombic-drive machines. Double-acting Siemens engines would be about half this size (after Meijer 1970a).

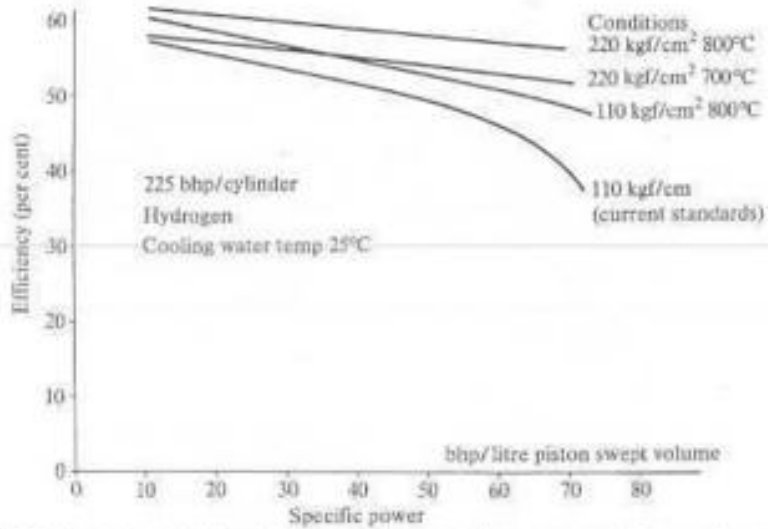


FIG. 8.3. Comparison of the effects of maximum cycle pressure and temperature on the maximum possible efficiency versus power-density characteristic for engines using hydrogen as the working fluid with a brake power output of 165 kW (224 brake hp) per cylinder and a cooler temperature of 25 °C (77 °F). Results were calculated using the Philips simulation program. No allowance made in computation of efficiency for losses in the preheater (after Meijer 1970a).

volume. The engines were optimized for maximum efficiency with an output of 170 kW (225 brake hp) per cylinder.

A total of four curves was included, two at maximum pressures of 110 MN/m^2 (15 954 lbs per sq in) and two at 220 MN/m^2 (31 908 lbs per sq in). For each pressure, two heater temperatures of 700 and 800 °C (1292 and 1472 °F) were represented. The curves indicate that substantial gains in engine efficiency would result from the application of heat-resistant steels permitting an increase in either or both the pressure and temperature levels. The gains are more pronounced at the higher power-density levels.

Later, Michels (1976) reported a study of the effect on efficiency of operating temperatures for the heater and cooler with different working fluids. Michels used the Philips Stirling cycle simulation computer program to calculate the optimum efficiencies attainable with:

(a) three different heater tube temperatures, 850, 400, and 250 °C (1562, 752 and 482 °F)
(b) two different cooler temperatures, 100 and 0 °C (212 and 32 °F)
(c) three different working fluids, hydrogen, helium, and nitrogen.

Michels based his calculations for reference purposes on the Philips Type 1-98 single-cylinder Stirling engine. This has a swept volume of 98 cm^3 (5.98 in^3) and is capable of delivering about 15 kW (20.4 hp) at 3000 revolutions per minute with a maximum cycle pressure of 220 MN/m^2 (31 908 lbs per sq in) using hydrogen. For his study Michels maintained the basic engine configuration but allowed changes in the dimensions of the heater, cooler, and regenerator within the overall limitation that the revised designs were capable of fitting onto the existing engine geometry. The heat exchangers were optimized for maximum indicated efficiency, this being defined as the ratio of 'the engine power assuming no mechanical friction . . . . to the heat delivered to the heater'.

The results given by Michels are reproduced in Fig. 8.4. They show the optimized indicated efficiency of the Type 1-98 engine as a function of power output for the different temperatures and working fluids specified above.

Unfortunately, Michels failed to include on his curves the engine speed although he noted 'the speed increased from left to right along the curves'. Similarly, he failed to provide any information on the pressure levels of the working fluid except to note that '. . . the pressures and dimensions were determined such that maximum efficiency was obtained'. The curves he presented were therefore much less useful than would appear at first sight.

Estimates of mechanical losses (principally friction) in the engine were included in the calculations to obtain the broken lines given in Fig. 8.4 of

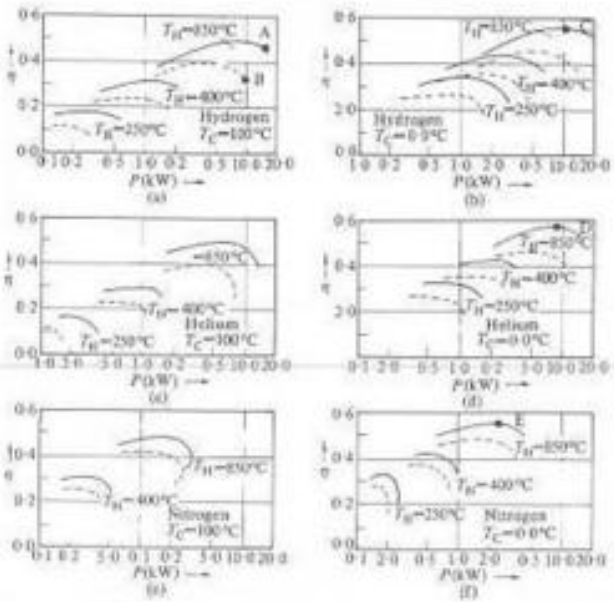


FIG. 8.4. Maximum possible efficiency versus power output for a Philips Type 1-98 rhombic-drive, single-cylinder engine with different heater and cooler temperatures and three different working fluids. Results were calculated using the Philips simulation program for maximum possible efficiency for an engine with 98 cm^3 (5.98 in^3) piston displacement. Speed and cycle pressure were not disclosed. Speed increases from left to right. It is thought that pressures also increase from left to right (after Michels 1976).

carefully noted that a logarithmic scale was used for the horizontal power axis, so that very significant differences in power existed for the relatively short distances between such points as, for example, A and B in Fig. 8.4(a).

Comparison of the characteristics given in Figs. 8.1 and 8.4 show that the *shapes* of the curves are different. In Fig. 8.1 the efficiency declined as the power per unit displacement increased, whereas in Fig. 8.4 the reverse is true, for as the specific power increased, the efficiency first increased to a maximum and then fell away. This difference is thought to have arisen because in Fig. 8.1 the maximum pressure was maintained constant whereas no limitation was imposed on the pressure to calculate the results in Fig. 8.4. Now increase in power output can be obtained by increase in either or both the engine speed and engine pressure level. On Fig. 8.4 it is likely, therefore, that the pressure increased from left to right along the curves just as the speed did in both Figs. 8.1 and 8.4.

An interesting feature of Michels' results was that, for a given tempera-

example, the peak efficiency at a heater temperature of 850 °C (1562 °F) and a cooler temperature of 0 °C (32 °F). For hydrogen, point C, Fig. 8.4(b) the maximum indicated efficiency is 56 per cent at 12 kW (16.3 hp) power output. For helium, point D, Fig. 8.4(d) the peak efficiency is 56 per cent at 9 kW (12.2 hp) power output. For nitrogen, point E, Fig. 8.4(f) the peak efficiency is 55 per cent at 2.5 kW (3.4 hp) power output. This correspondence of the same maximum indicated efficiency but at significantly different power levels was found at other heater and cooler temperatures. It is presumed that the pressures and speeds for helium would be somewhat less than for hydrogen and very much less for nitrogen. What a pity it is Michels did not see fit to include this essential additional data in his paper.

Parenthetically, it will be recalled that air is composed of 79 per cent nitrogen and 21 per cent oxygen so that Michels' results for nitrogen can generally be interpreted as applicable to 'air' without serious error.

Experimental comparisons

Very little experimental data about the effects of different working fluids have been published. Dros (1965a) published the comparison reproduced in Fig. 8.5, of the performance of a large Stirling cycle cooling

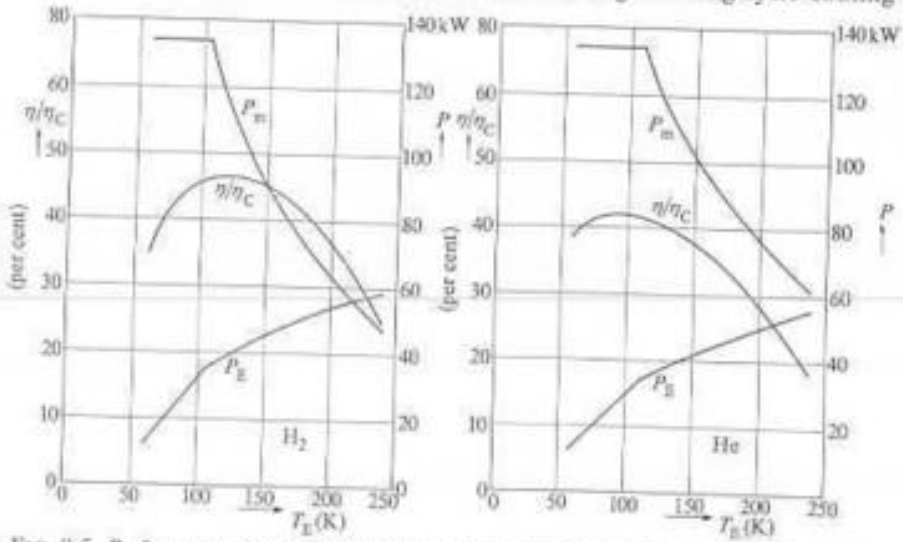


FIG. 8.5. Performance characteristics of large opposed-piston Stirling-cycle cooling engine using hydrogen and helium as the working fluid. The machine was supplied with water cooling at a temperature of 15 °C (59 °F) and flow rate 20 m³/h. (706 ft³/h) Measured values of refrigerating capacity P_E and shaft power P_m and also the relative efficiency η/η_c calculated from these two quantities are shown as functions of the cold-side temperature. At temperatures above 110 K (198 °R) the maximum cycle pressure was limited to 60 MPa (870 lb per sq in). At lower temperatures the pressure was reduced to maintain the shaft power constant at 134 kW (182 hp), the maximum permissible value (after Dros 1965a).

engine with hydrogen and helium as the working fluid. This shows the cooling effect generated and the power input required as a function of refrigerating temperature with the same engine speed and maximum cycle pressure. Comparison of the data, presented separately side by side, for hydrogen and helium, shows that with hydrogen as the working fluid the engine produced a higher refrigerating effect and consumed less power than with helium. In this paper Dros points out that at low cryogenic temperatures helium deviated less from ideal gas behaviour than hydrogen and so the advantage of hydrogen becomes less marked than in power engines.

Loftus (1964) evaluated the performance characteristics of a large four-cylinder 265 kW (360 hp) Stirling engine with hydrogen and helium as the working fluid. The engine was made by Philips in 1963 at the

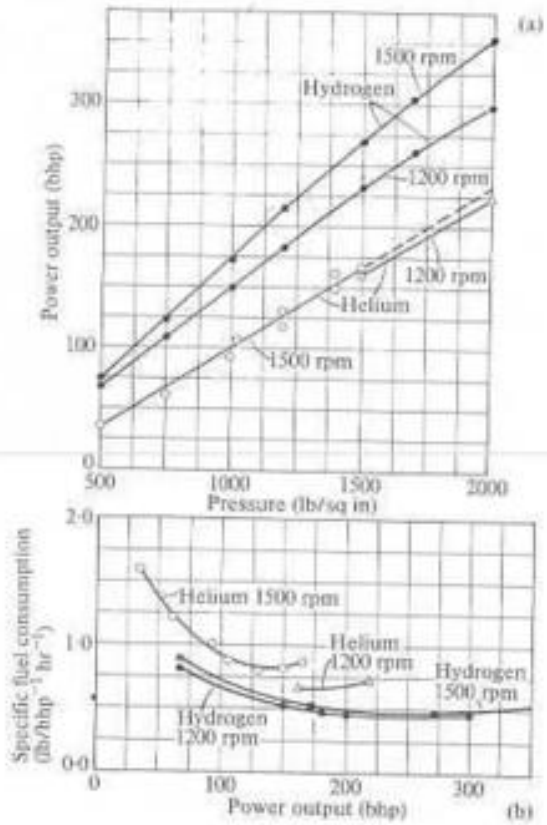


FIG. 8.6. Performance characteristics of 480 kW (360 hp) 4-cylinder Philips Stirling engine, type 4-S1210 with hydrogen and helium as the working fluids. Engine speeds were 1200 and

request of the Electromotive Division of General Motors for evaluation by the U.S. Navy. The engine, designated the Philips Type 4-S1210, is described in Chapters 12 and 13. Loftus ran the engine first with hydrogen as the working fluid and then with helium. No significant changes were made in the engine parts or experimental set-up. Some of his results with the two working fluids are reproduced in Fig. 8.6. He found that the engine produced more work at a higher efficiency when running with hydrogen as the working fluid than when running with helium.

Thermophysical properties

What then are the reasons for one gas to be better than another for the working fluid in Stirling engines? The answer lies in what are called the 'transport properties' of the fluid including the viscosity, thermal conductivity, specific heat, and density. Table 8.1 contains comparative data of these properties for hydrogen, helium, air, and carbon dioxide.

The properties are important in two ways as regards heat transfer and aerodynamic-friction losses. Heat capacity, also called the specific heat, and the thermal conductivity are parameters which principally control the

Table 8.1. Thermophysical data for air, hydrogen, helium, carbon dioxide, and water vapour (from *Thermodynamic and transport properties of fluids (SI units)*, Y. R. Mayhew and G. F. C. Rogers, Basil Blackwell, Oxford).

Fluid	Property	Temperature (K) 250	500	1000
Air	C_p (kJ/kgK)	1.003	1.029	1.141
1 atm	ρ (kg/m^3)	1.412	0.706	0.353
$M = 28.9$	$\mu \times 10^6$ (kg—ms)	15.99	26.70	41.53
	$k \times 10^6$ (kW/mK)	22.27	40.41	67.54
Hydrogen	C_p (kJ/kgK)	14.05	14.51	14.98
1 atm	ρ (kg/m^3)	0.098	0.0491	0.0246
$M = 2$	$\mu \times 10^6$ (kg/ms)	7.92	12.64	20.13
	$k \times 10^6$ (kW/mK)	156.1	271.8	452.2
Helium	C_p (kJ/kgK)	5.19	5.19	5.19
1 atm	ρ (kg/m^3)	0.195	0.097	0.048
$M = 4$	$\mu \times 10^6$ (kg/ms)	18.40	29.30	46.70
	$k \times 10^6$ (kW/mK)	134.0	202.6	—
Carbon dioxide	C_p (kJ/kgK)	0.791	1.014	1.234
1 atm	ρ (kg/m^3)	2.145	1.073	0.536
$M = 44$	$\mu \times 10^6$ (kg/ms)	12.60	23.67	39.51
	$k \times 10^6$ (kW/mK)	12.90	32.74	67.52
Water vapour	C_p (kJ/kgK)	1.855	1.954	2.288
low pressure	$\mu \times 10^6$ (kg/ms)	9.42	17.3	37.6

processes of heat transfer to and from the fluid in the cooler, heater, and regenerator. Density and viscosity are important in relation to the flow friction losses, which control the pump work necessary to move the fluid about the engine to perform the desired heat transfers from the heater or to the cooler. These flow losses are directly proportional to $\rho u^2/2$ where ρ is the gas density and u the gas velocity.

The density of a gas may be calculated from the characteristic gas equation as:

$$\rho = \frac{pM}{RT} \tag{8.1}$$

where p is the pressure
M is the molecular weight
R is the universal gas constant and
T is the absolute temperature.

Therefore, for a given pressure and temperature the density ρ is directly proportional to the molecular weight, M.

The heat transfer processes occurring may be characterized by the equation:

$$Q = hA\,\Delta T \tag{8.2}$$

where Q = heat transferred
h = heat transfer coefficient
A = area for heat transfer
ΔT = temperature difference between the fluid and solid wall.

The heat transfer coefficient h is one component of a dimensionless group called the Nusselt number:

$$N_u = hk/c \tag{8.3}$$

where N_u = Nusselt number
h = heat transfer coefficient
k = thermal conductivity
c = heat capacity.

Another important dimensionless group of parameters involved in convective heat transfer processes is the Reynolds number:

$$R_e = \rho u d/\mu \tag{8.4}$$

where R_e = Reynolds number
ρ = density
u = gas velocity
d = a characteristic dimension of the flow
μ = gas viscosity

the form:

$$N_u = BR_e^q \tag{8.5}$$

and B and q are constants which depend on the flow conditions. Therefore:

$$h \cdot k/c = B(\rho u d/\mu)^q. \tag{8.6}$$

The best working fluid is the gas which combines a high heat transfer coefficient (large h) with low friction or pumping losses (low ρu^2).

In general, hydrogen has the best combination of transport properties. It will result in less friction losses than helium or air for a given heat transfer rate at a given pressure or temperature situation. Alternatively, for a given flow loss in an engine, at a particular pressure and temperature level, the engine can run faster with hydrogen than with helium or air so that it has a higher specific output.

Steady-flow analysis

The flow situation in a Stirling engine is so complex that it is not easy to compare numerically the advantages of one working fluid and another without becoming involved in the intricacies of advanced computer simulation studies. A better approach is to consider a steady-flow situation where an analogous combination of good heat transfer and low pumping losses is important. This exists in many engineering situations but is particularly important in the gas-cooled nuclear reactor. Hall (1958) has given an excellent model for reactor heat transfer and the following treatment is an abbreviated version of his comparison of coolants.

The Reynolds analogy between fluid friction and heat transfer may be expressed in the form:

$$f/2 = St = h/\rho uc \tag{8.7}$$

where f = Fanning friction factor
St = Stanton number
h = heat transfer coefficient
ρ = density of fluid
u = fluid velocity
c = heat capacity of fluid.

The pressure drop with fluid flowing in a channel:

$$\Delta p = \frac{1}{\rho}\left(\frac{W}{A}\right)^2 \frac{2fL}{d} = \frac{1}{\rho}\left(\frac{W}{A}\right)^2 \frac{fsL}{2A} \tag{8.8}$$

where W = mass flow of fluid
A = cross section area of flow
s = perimeter of duct
L = length of duct

The pumping power P is:

$$P = \frac{1}{\eta}\frac{W}{\rho^2}\left(\frac{W}{A}\right)^2 \frac{fsL}{2A} = \frac{1}{\rho^2}\left(\frac{W}{A}\right)^3 \frac{fsL}{2\eta} \tag{8.9}$$

where η = pump efficiency.
The heat transferred in the channel is

$$Q = h(sL)\,\Delta T \tag{8.10}$$

so that:

$$\frac{P}{Q} = \frac{1}{\eta}\frac{1}{\rho^2}\frac{(W/A)^3(f/2)sL}{h(sL)\Delta T} \tag{8.11}$$

and since $St = h/\rho uc = hA/Wc = f/2$
then

$$\frac{P}{Q} = \frac{1}{\eta}\frac{1}{\rho^2}\left(\frac{W}{A}\right)^2 \frac{1}{c\,\Delta T} \tag{8.12}$$

but

$$W/A = Q/c(T_2 - T_1)A \tag{8.13}$$

where T_1 and T_2 are the temperatures of the fluid, at inlet and outlet from the duct.

Therefore, substituting eqn (8.13) in eqn (8.12)

$$\frac{P}{Q} = \frac{Q^2}{\eta\rho^2 c^3\,\Delta T(T_2 - T_1)^2 A^2} \tag{8.14}$$

or

$$Q = (\eta\rho^2 c^3\,\Delta T(T_2 - T_1)^2 A^2 (P/Q))^{\frac{1}{3}}. \tag{8.15}$$

Therefore, in a given situation with specified temperatures and (P/Q) ratio:

$$Q \propto \sqrt{(\rho^2 c^3)} \tag{8.16}$$

and since we have seen above that $\rho \propto M$ then

$$Q \propto \sqrt{(M^2 c^3)} \tag{8.17}$$

Hall (1958) is at pains to point out the severe limitations of the analysis particularly with regard to finned surfaces. Despite this, the author finds it a useful measure for comparing the relative merits of working fluids for Stirling engines.

Consider, for example, the five fluids whose properties are given in Table 1: hydrogen, helium, air, carbon dioxide, and water vapour. The

The specific heats are 14, 5, 1, 0.8, and 1.8 kJ/kg-K (3.35, 1.2, 0.24, 0.192, and 0.43 Btu/lb$_m$ °R) respectively. Therefore, from eqn (8.17) above the comparative heat transfer Q for a given P/Q ratio is:

Hydrogen	Q_h:	104
Helium	Q_{he}:	44
Air	Q_A:	29
Carbon dioxide	Q_{CO_2}:	31
Water vapour	Q_{H_2O}:	44.

On this basis of comparison, hydrogen is clearly the preferred heat transfer fluid with helium or water vapour a second choice and air or carbon dioxide a rather poor third choice.

Water vapour has not been used extensively as the working fluid in a Stirling engine. It can exist at temperatures of 50 to 60 °C (122 to 140 °F) (characteristic of automotive engine cooler temperatures) as a vapour only if the pressure is very low, whereas high pressures are necessary to attain a high power density. Water vapour is a candidate for use in a Stirling engine with a compound or multiphase working fluid as described later. The fluid then changes phase from liquid in the compression space to vapour in the hot expansion space. It may or may not operate in association with a gaseous carrier component such as hydrogen or helium.

SAFETY

Despite its many attractions as the best working fluid for Stirling engines, hydrogen has the disadvantage of extremely wide flammability limits in air ranging from 5 to 75 per cent mixtures of hydrogen and air. Other gases, methane, for example, have much closer flammability limits of 6 to 14 per cent methane in air by volume. Moreover, hydrogen has a high affinity to oxygen and the enthalpy of reaction (heat released in burning) 129 000 kJ/kg (30 960 Btu/lb$_m$) of hydrogen, compares with 50 143 kJ/kg (12 034 Btu/lb$_m$) for methane.

Automotive engines

It is therefore difficult to escape a feeling of apprehension when contemplating Stirling-cycle vehicle engines pressurized to 100 to 200 MN/m^2 (29 007 lbs per sq in). From the material presented above there is no doubt that the automotive engine must use hydrogen; there really is no alternative for engines of high power density and efficiency. Yet, the prospect of high pressure hydrogen systems, generally available for public use, is a daunting prospect to those concerned with matters of safety.

It is inevitable that some leakage will occur from the working space to

favoured by Philips it is inevitable that sooner or later material failures will occur with consequent leakage of hydrogen from the working space.

Porosity

Hydrogen is, moreover, a difficult gas to contain. It is so light and fluid that it passes readily through materials that are apparently solid. At high temperatures and gas pressures the effect is enhanced and most materials including metals are porous to some degree to hydrogen. In an engine with a direct-heating fossil-fuel combustion system any leakage through the hot parts of engine pose no safety hazards; hydrogen will simply burn (to water vapour). For indirectly heated systems with a liquid-metal heat-transfer loop, some means of venting the hydrogen gas must be incorporated.

Percival (1974) cited hydrogen permeation of materials as one of the principal unresolved problems in Stirling engines. He indicated that one avenue of approach adopted by Philips was to provide an impermeable ceramic liner in the heater tubes but provided no details and the matter has never been mentioned in any of the several papers published by Philips.

Hydrogen embrittlement

Another important effect to be considered in the use of hydrogen as a working fluid is the embrittlement which metals experience when exposed to hydrogen, particularly at high temperatures and pressures. The effect of hydrogen on the mechanical properties of metals and other materials is extraordinarily complex and profound. An excellent compilation of material on the subject of hydrogen effects was assembled by Beachem (1977).

These important matters have not been sufficiently addressed in the public literature. They are expected to receive due attention in the Stirling engine research program supported by public funds now being developed in the United States by the Department of Energy.

General Motors Corporation's research

A significant statement regarding safety in the use of hydrogen as the working fluid in Stirling engines was contained in Percival's (1974) historical record of the General Motors development program. In that work Percival reproduced a memorandum to the G.M. laboratory management in which he investigated possible restrictions on the use of hydrogen working fluid in the 110 kW (150 hp) 4L23 General Motors double-acting Stirling engine for bus installation. He examined the possible restrictions on public highways, and in tunnels, determined the

use of hydrogen with various regulatory authorities who might have been responsible for restrictions on the use of hydrogen but no existing regulations or limitations on its use were found.

The total amount of hydrogen contained in the GM 4L23 bus engine and reservoir was estimated, with the engine pressurized to the normal value of 10.3 MN/m^2 (1500 lbs per sq in), to be a total of 0.022 kg (0.050 lbs), equivalent to a free volume at 21.11 °C (70 °F) of 0.2724 m^3 (9.62 cu ft) This volume of gas escaping from the engine and mixing with air to a 5 per cent mixture would need to be confined to a volume of 5.435 m^3 (1.52 m × 1.52 m × 2.44 m) (192 cu ft: 5 ft × 5 ft × 8 ft) to assure flame propagation on ignition.

Assuming ignition and the reaction of at least one half of the hydrogen the heat released would be 1370 kJ (1300 Btu). This was said by Percival to be equivalent to 0.032 kg ($\frac{1}{12}$ lb) of gasoline or $\frac{1}{6}$ of the smallest available can of propane. Another factor favourable to hydrogen cited by Percival was the relatively low flammability limits for hydrocarbon gases: 2.2 per cent for propane, 1.6–2 per cent for butane, 1.4 per cent for gasoline, 5–6 per cent for methane. For hydrogen, the lower flame propagation limit ranged from 4 per cent, for combustion vertically upward, to 9 per cent, for combustion vertically downward. For hydrocarbon gases, on the other hand, the flammability limit was more or less independent of the direction of propagation so they were more likely to be entirely consumed. The extended upper flammability limit for hydrogen, 74 per cent compared with 6 to 14 per cent for hydrocarbon gases, was dismissed by Percival rather summarily as being of serious concern only in the case of non-habitable spaces. He failed to address the question of explosion of a rich hydrogen mixture in a closed engine compartment, thought by many to be a serious problem.

Percival pointed out that hydrogen diffused at a rate 4 to 8 times that of hydrocarbon gases. In the open air or in a large room an instantaneous leakage of all the hydrogen from a 110 kW (150 hp) engine would be dissipated to a non-flammable mixture in a few seconds. On the other hand, hydrocarbon gases disperse more slowly and, being heavier than air, tend to accumulate or remain concentrated in low lying areas.

In summary, Percival presented a convincing case that the potential danger of fire or explosion from hydrogen leakage in the bus installation was small and substantially less than the possibility of fire from a hydrocarbon fuel source.

Ford Motor Company's research

Further work on the hazards of hydrogen as the working fluid in automotive Stirling engines was carried out by Goodale and Walter (1976) at Stanford Research Institute on behalf of the Ford Motor Company. Goodale's report is contained as an appendix in the comprehensive report by Kitzner (1977a).

No other firm data is available, but it is said that United Stirling have approached the Swedish regulatory authorities and have gained approval for the use of hydrogen as the working fluid in their automotive engine. It is further understood that their development work has embraced the experimental study of fire or explosion hazards with hydrogen engines in vehicles in garages or confined spaces. No details have been published.

No doubt, all the companies presently involved are pursuing similar work and are engaged in discussions with the regulatory authorities. The development of the so-called 'hydrogen economy' in the future, particularly in the United States, will result in hydrogen becoming far more familiar than it is today. Safety measures and handling procedures will be developed that will, no doubt, do much to allay the apprehensions expressed in the early part of this section. There are no reports of accidents involving Stirling engines in the literature, except a brief reference, made almost in passing by Percival (1974) in discussing experiences with an experimental GPU-3 engine. Hearsay has it that one fatal accident occurred at Philips in the early days of their work on hydrogen engines (mid-1950s) but no details of this are known.

COMPOUND WORKING FLUIDS

Introduction

Compound working fluids in Stirling engines were investigated in a preliminary way by Walker and Agbi (1974). They assumed a compound working fluid to have two components: the gaseous carrier and the phase change component. The phase change component experienced a change of phase from liquid to vapour in moving from the cold space, through the regenerator to the hot space.

The principal attraction of the compound working fluid was the possibility of achieving a high specific output at a moderate mean pressure level. Secondary advantages were anticipated in terms of improved heat transfer arising from the boiling and condensing processes as well as some relief in the critical problem of reciprocating seals.

Walker and Agbi (1974) carried out the preliminary study of compound working fluids by comparison of a series of idealized Schimidt-type thermodynamic cycles. The gaseous working fluid had the characteristics of air, and mixtures of air and water having some mass ratio $\beta = m_v/m_a$ particular to the cycle were prescribed.

It was found that for the same limits of maximum and minimum temperature (cost) and volume (size) and for the same maximum pressure (weight) the area of the work diagram for the cycle having a compound working fluid was substantially larger than that for the simple gaseous

working fluid. In other words, in an engine of the same size, cost and weight, the use of a compound working fluid increased the work output.

A physical explanation for this improvement was found in the recognition that the phase change from liquid to vapour of one component of the working fluid caused, in effect, an increase in the volume compression ratio of the cycle with consequent benefit to the pressure ratio and net cycle work.

Improvement in the volume compression ratio was a highly desirable effect. In the design of practical Stirling engines, it was rarely possible to attain a volume compression ratio greater than 2.5 without undue sacrifice of heat transfer surface area or introducing unacceptably high fluid friction effects. By increasing the effective volume compression ratio, an increase in the amplitude of the cyclic pressure excursion was obtained with consequent beneficial results on the area of the work diagrams for the various spaces in the engine.

Isothermal analysis with compound working fluid

The Schmidt analysis (see Chapter 4) was found amenable, with relatively minor modification, for application with a two-phase, two-component working fluid.

The compound working fluid was assumed to consist of two components, one behaving at all times as a perfect gas, and the other in the compression space existing in the liquid state at low temperatures and, in the expansion space in the vapour state at high temperatures where it was further assumed to behave as a perfect gas. The specified mass ratio $\beta = m_v/m_g$ was assumed to prevail throughout the system and to be unaffected by the phase change from liquid to vapour of one component.

In the conventional Schmidt analysis, a linear temperature profile was assumed for the regenerator. With a compound working fluid the arbitrary assumption was made that the regenerator dead space was divided into two volumes, one maintained at temperature T_E, the other at temperature T_C and of such size that the total mass of working fluid *in the regenerator* was divided equally between the two spaces. At the interface it was assumed that a step change in temperature occurred, accompanied by a change in phase of one component, and that the interface moved as required to maintain the equality of mass in each regime. There is no physical justification for this assumption. It was adopted simply for the sake of computational convenience and will probably need to be improved for future work.

The total pressure p was taken to be the sum of the partial pressures of the two components and, further, to be instantaneously constant throughout the system, i.e.

Then p_{vc}, the partial pressure of one component in the liquid state, was taken to be always zero, and the volume of the liquid was assumed to be negligible, so that in the compression space $p = p_{gc}$.

It was recognized that the assumption of a constant mass ratio β was probably not attainable in practice because the vapour component would tend to migrate from the hot expansion space to the cold compression space. This could perhaps be compensated by gravitational pumping, i.e. putting the cold space above the hot space. A constant mass ratio, β, might represent one extreme limit achieved by an engine running fast. The contrasting limit achieved in an engine running slowly might occur in the case where the vapour pressure $p_{ve} = p_{vc}$ has a negligible value so that all the phase change component exists in the liquid state in the compression space. In this case, β in the expansion space is effectively zero and the results correspond to that of the cycle with a single component gaseous working fluid.

Principal assumptions

1. The regenerative process was perfect.
2. The process of compression and expansion occurred isothermally at temperatures T_C and T_E, respectively.
3. Volume variations in the compression and expansion spaces occurred sinusoidally.
4. The vapour component at temperature T_C existed in the liquid state with negligible volume and vapour pressure.
5. The vapour component at temperature T_E existed in the superheated vapour state and behaved as a perfect gas according to the ideal gas equation $p_v V = m_v R_v T$.
6. The gaseous component behaved as a perfect gas at all times and obeyed the equation $p_g V = m_g R_g T$.
7. The total mass of the working fluid distributed between the compression, expansion, and dead spaces remained constant.
8. The ratio of mass of vapour component to gaseous component was constant and uniform throughout the machine.
9. The instantaneous total pressure of the working fluid was uniform throughout the working space.
10. The mass of working fluid in the dead space was divided equally into two parts, one at temperature T_C, and the other at temperature T_E. At the interface of these two masses there was a phase change of the vapour component from superheated vapour to liquid.
11. The rotational speed of the engine was constant.
12. Steady-state conditions for the overall operation of the engine

variation only and equilibrium existed between components and phases at all times.

Summary of analysis

The process of development of the analysis was precisely similar to that given in Chapter 4 and yielded the following equations:

1. *Volume*

(a) Expansion space

$$V_e = \tfrac{1}{2}V_E(1+\cos\phi) \tag{8.19}$$

(b) Compression space

$$V_c = \tfrac{1}{2}V_E(1+\cos(\phi-\alpha)) \tag{8.20}$$

(c) Dead space

$$V_D = XV_E = V_{DE} + V_{DC}. \tag{8.21}$$

2. *Pressure*

(a) Instantaneous pressure

$$\frac{p}{p_{max}} = \frac{1-\delta}{1+\delta\cos(\phi-\theta)} \tag{8.22}$$

(b) Pressure ratio

$$\frac{p_{max}}{p_{min}} = \frac{1+\delta}{1-\delta} \tag{8.23}$$

where $A = (K^2+\kappa^2+2\kappa K\cos\alpha)^{\frac{1}{2}}$

$B = K+\kappa+2S$

$\delta = A/B$

$S = \dfrac{2KX}{1+K}$

$K = \left(\dfrac{\tau}{1+\beta N}\right)$ $\quad N = M_a/M_c$ and $\phi = \tan^{-1}\left(\dfrac{\kappa\sin\alpha}{K+\kappa\cos\alpha}\right)$

3. *Work Done*

(a) Expansion space

$$\frac{P_E}{p_{max}V_T} = \frac{\pi}{2}\left(\frac{1}{(1+\kappa)}\right)\left(\frac{1-\delta}{1+\delta}\right)^{\frac{1}{2}}(\Delta\sin\phi) \tag{8.24}$$

(b) Compression space:

$$\frac{P_C}{p_{max}V_T} = -\frac{\pi}{2}\left(\frac{\kappa}{1+\kappa}\right)\left(\frac{1-\delta}{1+\delta}\right)^{\frac{1}{2}}(\Delta\sin(\theta-\alpha)) \tag{8.25}$$

(c) Net cycle work

$$P = P_E - P_C = (1-K)P_E \tag{8.26}$$

4. *Heat Transferred*

Adequate expressions were not derived for the heat transfer and the enthalpy and entropy changes expressed in terms of the principal design parameters. In the simple Schmidt cycle the equality $[dQ = \int(p\,dV)_T]$ holds because the fluid is considered a perfect gas and the processes of compression and expansion are assumed isothermal. With a compound working fluid the equality does not hold, for the fluid is not a perfect gas in the compression space. The heat transferred may perhaps best be estimated by consideration of the enthalpy or entropy changes for the unsteady flow two-fluid systems represented by the expansion and compression spaces, considered separately. This has not yet been done. It is suggested here that the various assumptions (particularly those of isothermal compression and expansion) would lead ultimately to a cycle efficiency equivalent to the Carnot value as is the case with the Ideal Stirling and Schmidt cycle. In that case

$$\eta = \frac{P}{Q_E} = (1-\tau) \tag{8.27}$$

5. *Mass Distribution*

(a) Expansion Space

$$\frac{m_e}{m^*} = (1+\beta)\frac{K}{2}\left[\frac{1-\delta}{1+\kappa}\right]\cdot\left[\frac{(1+\cos\phi)}{(1+\delta\cos(\phi-\theta))}\right] \tag{8.28}$$

and the mass velocity

$$\frac{d(m_e/m^*)}{d\phi} = (1+\beta)\frac{K}{2}\left[\frac{1-\delta}{1+\kappa}\right]\cdot\left[\frac{\delta\sin(\phi-\theta)-\delta\sin\theta-\sin\phi}{(1+\delta\cos(\phi-\theta))^2}\right] \tag{8.29}$$

where the characteristic mass

$$m^* = \frac{p_{max}V_T}{R_aT_C}$$

(b) Compression Space

$$\frac{m_c}{m^*} = (1+\beta)^\kappa\frac{K}{2}\left[\frac{1-\delta}{1+\kappa}\right]\left[\frac{(1+\cos(\phi-\alpha))}{(1+\delta\cos(\phi-\theta))}\right] \tag{8.30}$$

$$\frac{d(m_c/m^*)}{d\phi} = (1+\beta)^\kappa\frac{K}{2}\left[\frac{1-\delta}{1+\kappa}\right]\cdot\left[\frac{\delta(\sin(\phi-\theta)-\sin(\theta-\alpha))-\sin(\phi-\alpha)}{(1+\delta\cos(\phi-\theta))^2}\right] \tag{8.31}$$

(c) Dead Space

$$\frac{m_{DE}}{m^*} = XK\frac{(1+\beta)}{(1+\kappa)}\left(\frac{1-\delta}{1+K}\right)\cdot\frac{1}{(1+\delta\cos(\phi-\theta))} \tag{8.32}$$

Since

$$\frac{m_D}{m^*} = \frac{2m_{DE}}{m^*}$$

the mass velocity

$$\frac{d(m_D/m^*)}{d\phi} = 2\left(\frac{1+\beta}{1+\kappa}\right)\left(\frac{KX}{1+K}\right)\left(\frac{(1-\delta)(-\delta\sin(\phi-\theta))}{(1+\delta\cos(\phi-\theta))^2}\right) \tag{8.33}$$

Nomenclature

The nomenclature for the above is precisely that used for the Schmidt cycle analysis in Chapter 4 with the addition or substitution of:

A defined by $(K^2+\kappa^2+2\kappa K\cos\alpha)^{\frac{1}{2}}$
B defined by $K+\kappa+2S$
K defined by $\dfrac{\tau}{1+\beta N}$
M molecular weight
m^* = characteristic mass defined above (eqn 8.29).
m_c = mass of working fluid in the compression space
m_D = mass of working fluid in the dead space
m_e = mass of working fluid in the expansion space
m_a = mass of air in the working space
m_v = mass of vapour in the working space
m_w = total mass of working fluid (m_a+m_v)
N = ratio of characteristic gas constants M_a/M_v
P = engine output
P_C = engine output in the compression space
P_E = engine output in the expansion space
p = instantaneous cycle pressure
p_{ac} = instantaneous partial pressure of the air component in the compression space
p_{ae} = instantaneous partial pressure of the air component in the expansion space
p_{vc} = instantaneous partial pressure of the vapour component in the compression space
p_{ve} = instantaneous partial pressure of the vapour component in the expansion space
Q_E = heat transferred in the expansion space

R_v = characteristic gas constant of vapour
T_C = absolute temperature of the compression space
T_E = absolute temperature of the expansion space
t = time
V_C = swept volume of compression space
V_c = instantaneous volume of the compression space
V_C = dead volume
V_{DC} = volume of dead space at temperature T_C
V_{DE} = volume of dead space at temperature T_E
V_E = swept volume of the expansion space
V_e = instantaneous volume of the expansion space
V_T = combined swept volume of the working space V_E+V_C
X = dead volume ratio V_D/V_E
α = angle by which volume variations in the expansion space lead those in the compression space
β = component mass ratio m_v/m_a
Δ defined by $\dfrac{2}{\delta}[(1-\delta^2)^{\frac{1}{2}}-1]$
δ defined by A/B (see eqn 8.23)
η = thermal efficiency $\dfrac{P}{Q_E}$
θ defined by $\tan^{-1}\left[\dfrac{\kappa\sin\alpha}{K+\kappa\cos\alpha}\right]$
κ = swept volume ratio V_C/V_E
ξ = ratio of the dimensionless engine output with compound working fluid to the output with a gaseous working fluid
τ = temperature ratio T_C/T_E
ϕ = crank angle (ωt)
ω = angular speed

Note: The suffix a or v refers to the air or vapour component. An upper case suffix refers to a constant or maximum value. A lower case suffix refers to an instantaneous value.

Principal design parameters

The principal independent design parameters of a regenerative Stirling engine, according to the Schmidt analysis, are:

1. the temperature ratio $\tau = T_C/T_E$, ratio of temperatures in the compression and expansion spaces
2. the swept volume ratio $\kappa = V_C/V_E$, ratio of swept volumes in the compression and expansion spaces

4. some characteristic pressure of the working fluid of which the maximum pressure was the preferred criterion
5. the dead space ratio $X = V_D/V_E$ the porous volume in the regenerator and associated heat exchangers expressed in multiples of the swept volume in the expansion space
6. engine speed.

In the case of the compound working fluid there was an additional independent design parameter, the mass ratio $\beta = m_c/m_e$ and freedom of choice of the selection of fluids having different thermodynamic characteristics.

Design charts for Stirling engines acting as prime movers and refrigerating machines were given in Chapter 5, prepared from an optimization study based on the Schmidt analysis. For a particular case with $T_E = 1000$ K (1800 °R) and $T_C = 300$ K (540 °R) so that $\tau = 0.3$ and with the dead space ratio X specified as 1.0 the optimum values of phase angle and swept volume ratio were found to be 0.54π radians (97°) and 0.74 respectively. These values were taken as the *standard design configuration* for most of the cases reported here.

Effect of compound working fluid, work diagrams

In Fig. 8.7 a comparison is drawn between the work diagrams obtained for three Schmidt cycles having mass ratios of $\beta = 0$, 1, and 2, and the standard design configuration of $\tau = 0.3$, $X = 1.0$, $\alpha = 0.54$ and $\kappa = 0.74$. Three work diagrams for each case were obtained. The diagrams show the simultaneous variation of pressure and volume in the expansion and compression space and in the total working space.

The area of these diagrams represents the expansion and compression work of the expansion and compression spaces. The area of the diagram

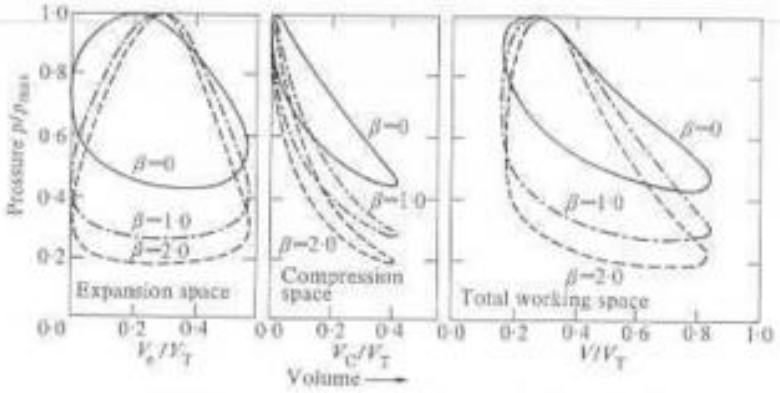


FIG. 8.7. Work diagrams for Schmidt-cycle system with compound working fluid (after Walker and Agbi 1974).

for the total working space represents the indicated power output of the engine and is in fact exactly equal to the difference in the areas of the diagrams for the expansion and compression spaces.

For the purpose of comparison the diagrams are rationalized so that pressures and volumes are expressed as fractions of the maximum value, thereby permitting comparison of machines similar to a first approximation in size, weight, and cost. The area of the diagram for the total working space was thought to be the principal criterion: the bigger the area, the better the engine. It was clear from Fig. 8.7, that the cycles with a compound working fluid, $\beta = 1.0$ and 2.0, had work areas for the total working space much greater than that for the single fluid system $\beta = 0$. For the particular cases considered, values of the work ratio ζ (or ratio of the diagram areas for $\beta = 1.0$ and 2.0 compared with the diagram area for $\beta = 0$) were 1.6 and 1.76, respectively.

Effect of mass ratio

Further study of the effect of the mass ratio β resulted in the generation of the data shown in Fig. 8.8. This figure shows the work ratio ζ as a

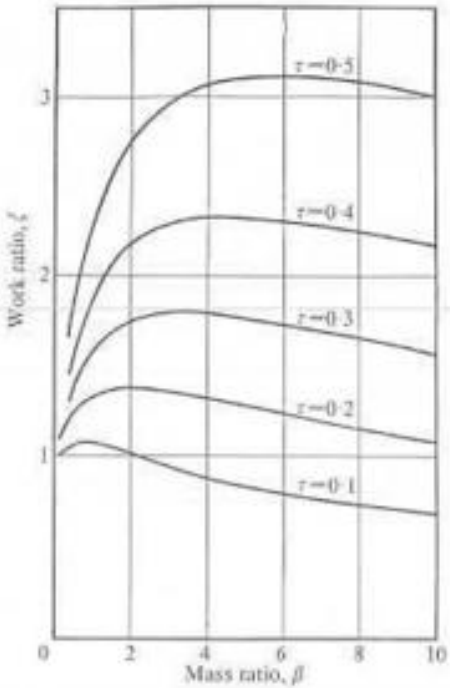


FIG. 8.8. Degree of improvement in the specific output (work ratio) of Schmidt-cycle systems with a compound working fluid (after Walker and Agbi 1974).

function of the mass ratio β. A family of curves was presented for a series of temperature ratios ranging from $\tau = 0.1$ ($T_E = 3000$ K, (5400 °R), $T_C = 300$ K (540 °R)) to $\tau = 0.5$ ($T_E = 600$ K (1080 °R), $T_C = 300$ K (540 °R)). It was interesting to note from this figure that at very high temperatures ($\tau = 0.1$) the addition of a phase change component above a mass ratio of 2 was actually detrimental and resulted in a work ratio ζ less than 1. In every other case an improvement in the work ratio was obtained by the addition of a phase change component, with marked benefits at the lower temperatures. According to Fig. 8.8 an improvement in specific output of over three times was gained with a temperature ratio $\tau = 0.5$ ($T_E = 600$ K (1080 °R), $T_C = 300$ K (540 °R)). This may have particular significance for future developments of low temperature 'bottoming cycle' power systems utilizing engine-exhaust energy, or flat-plate solar collectors, as the thermal source.

System variables

The effect of the other system design variables τ, κ, α, and X were investigated in systematic fashion by the independent variation of one parameter, while maintaining the others constant at the arbitrarily selected standard design configuration $\tau = 0.3$, $\alpha = 0.54\pi$ (97°), $\kappa = 0.74$, and $X = 1.0$ for a range of β values 0, 1.0, and 2.0. The results of the study are presented in Fig. 8.9. The engine output was represented in dimensionless terms as $(P/p_{max}V_T)$.

Fig. 8.9(a) shows the effect of variation in the phase angle α on the power output for three different mass ratios. Increase in β causes substantial increase in the power output and a slight change in the angle at which the maximum power occurs.

Fig. 8.9(b) shows the effect of variation in the swept volume ratio κ on the power output for three different mass ratios. Increase in β causes substantial increase in the power output and a marked reduction in the swept volume ratio at which the maximum power occurs.

Fig. 8.9(c) shows the effect of variation in temperature ratio τ on engine output. A decrease in τ corresponds to an increase in the expansion space temperature T_E with the compression space temperature held constant. With a gaseous working fluid the engine output increases progressively with increase in expansion space temperature. This is not true for the compound working fluid. For a mass ratio of $\beta = 1.0$ there is a progressive increase in the output power towards an apparent maximum. For a mass ratio of $\beta = 2.0$ the engine output increases slightly to a maximum value at $\tau = 0.2$ and then actually decreases with further

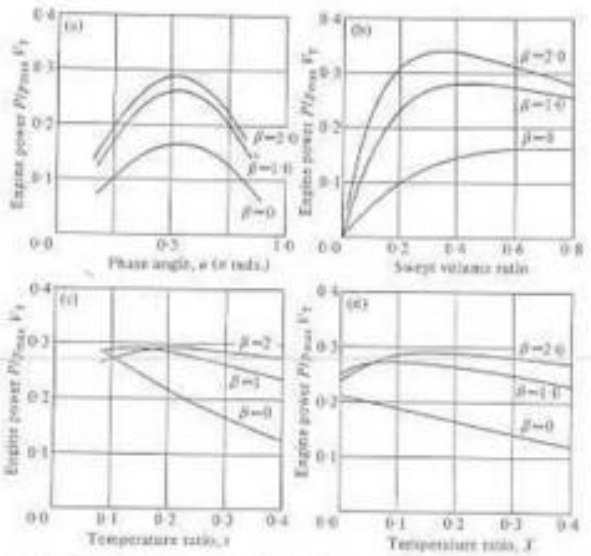


FIG. 8.9. Composite diagram illustrating the effects on the power output of Schmidt-cycle systems with a compound working fluid as a consequence of changes in the temperature ratio, dead volume ratio, phase angle and swept volume ratio (after Walker and Agbi 1974).

characteristic for a power system which may find application in some control and monitoring functions.

Fig 8.9(d) shows the effect of variation in the dead space ratio X on the power output of the engine. It has long been recognized that for gaseous working fluids the dead space is an important design criterion and should be reduced to the minimum in order to maximize power output. It is therefore of substantial interest that the power output with a compound working fluid is much less sensitive to variation in the dead space. Indeed, as shown in Fig. 8.9(d), the power output actually increases with increase in dead volume ratio up to $X = 1.0$ and thereafter declines along a shallow curve. This virtual independence of power output to the dead space is important for it will permit the use of heat exchange components of enhanced design with large internal surface area.

Distribution and mass flow rate

It was of interest to study the mass distribution of working fluid in a Stirling engine with a compound working fluid. The mass distribution characteristics obtained for the standard design configuration with $\beta = 0$

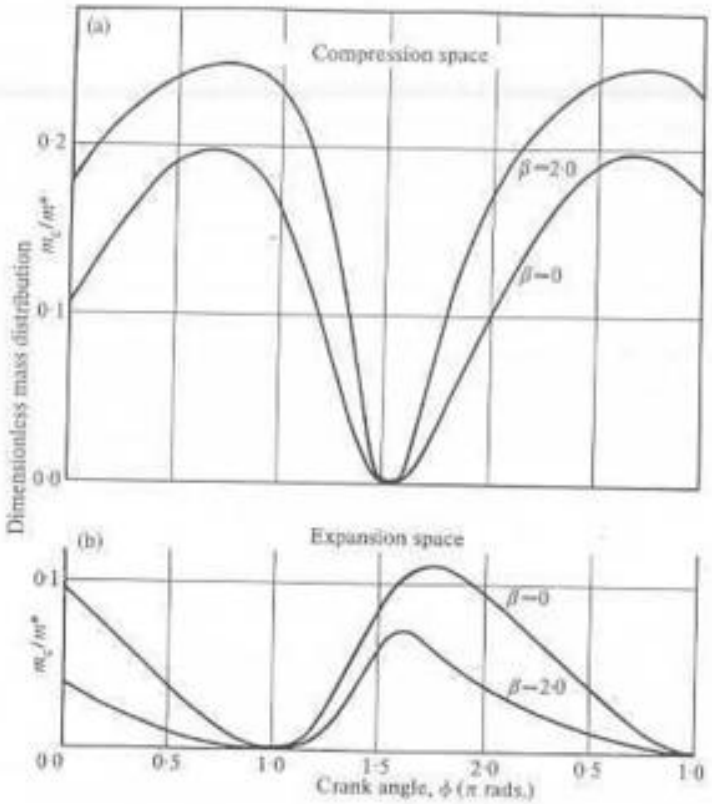


FIG. 8.10. Mass distribution in the compression and expansion spaces of Schmidt-cycle systems with a gaseous and compound working fluid (after Walker and Agbi 1974).

compression space. Fluid in the dead space is not included. The fluid masses are expressed in dimensionless terms by the arbitrary definition of a reference mass m^*.

It was of interest to note from Fig. 8.10 that with a compound working fluid there was a concentration of mass in the compression space and a reduced mass in the expansion space, compared with a gaseous working fluid. The concentration of mass in the compression space was explained by the assumption that the vapour component at low temperatures was assumed to have zero vapour pressure and volume so that the gas component pressure in the compression space was greater than in the expansion space. This effect was additional to the normal densification of the working fluid due to the temperature difference. Concentration of mass in the compression space was thought to explain

the reduction observed in Fig. 8.9(b) in the optimum value of the swept volume ratio κ with increasing β.

Cyclic mass velocity characteristics for the flow of working fluid to and from the expansion and compression spaces were derived for the standard design configuration. These curves are presented in Fig. 8.11(a) for the case $\beta = 0$ and Fig. 8.11(b) for the case $\beta = 2.0$.

In both Figs. 8.11(a) and (b) the curve G–C represents the rate of fluid flow *out* of the expansion space, the curve C–K–E represents fluid flow rates *into* the expansion space, and the curve E–M represents fluid flow *out* of the expansion space. The curve H–B represents fluid flow *into* the compression space, the curve B–J–D represents flow *out* of the compression space, and the curve D–L–N represents flow *into* the compression space.

Plotted in this inverse fashion the curves are of interest when overlaid as in Figs. 8.11(a) and (b), for the shaded areas then revealed the rate and duration of flow through the regenerator. Thus, the shaded areas A–G–B plus E–M–F represents the net flow of fluid through the regenerator in the direction *from* the expansion space *towards* the compression space. Similarly, the shaded area C–K–D represents net flow through the regenerator in the direction *from* the compression space *towards* the expansion space. In the part of cycle, B–C, fluid was flowing *into* the regenerator from both expansion and compression spaces and for the cyclic fraction D–E, fluid was flowing *out* of the regenerator into both the compression and expansion spaces.

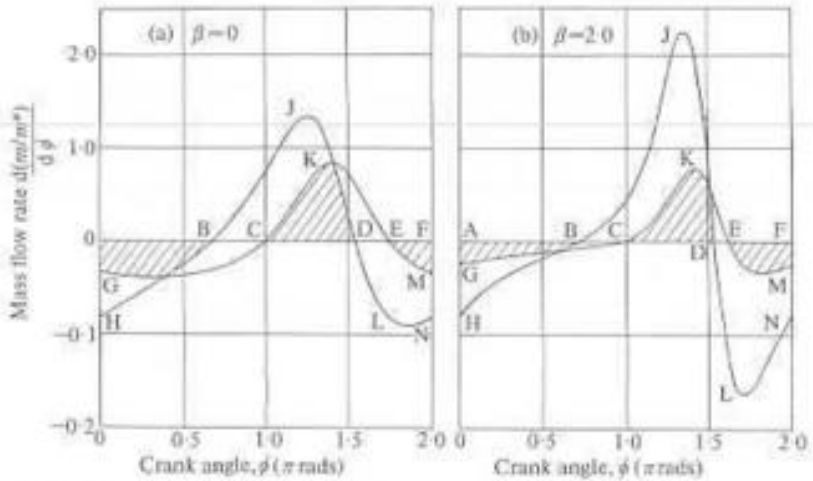


FIG. 8.11. Mass velocity characteristics for the compression and expansion spaces of a Schmidt-cycle system with a gaseous and compound working fluid (after Walker and Agbi 1974).

Comparison of Figs. 8.10(a) and (b) shows that the rates of mass flow for the compression space were substantially increased with a compound working fluid and the rates of flow for the expansion space marginally decreased, a result that could be deduced from Fig. 8.10. Increase in the fluid mass velocities could be expected to increase the fluid friction effects thereby increasing the difference in pressure between the expansion and compression spaces causing a deleterious effect on the engine power output.

Optimum design

Subsequently Walker and Agbi (1973) found that the optimum combination of swept volume ratio κ and phase angle α for given values of temperature ratio τ and dead volume X varied with the mass ratio β. With a compound working fluid the optimum ratio of swept volumes in the compression and expansion space was appreciably less than the optimum value with a gaseous working fluid. This arose out of the difference in mass distribution in the engine such as that discussed above with reference to Fig. 8.10.

The effect of the modification of optimum swept volume ratio was beneficial. When the change in optimum swept volume ratio was made and the proportions were adjusted such that the same maximum working space volumes were compared, the result was as shown in Fig. 8.12. This figure shows work diagrams for the expansion, compression, and expansion spaces for three cases:

(a) the optimum design configuration for a gaseous working fluid, $\tau = 0.3$, $X = 1$, $\alpha = 0.54\pi$ radians (97°) and $\kappa = 0.74$. In this case $\beta = 0$.

(b) same parameter configuration but with a compound working fluid, $\beta = 2$.

(c) optimum design configuration for a compound working fluid with $\beta = 2$, $\tau = 0.3$ and $X = 1$. In this case $\alpha = 0.53\pi$ radians (95°) and $\kappa = 0.35$.

The *net* work-diagram area for case (b) was 1.76 times that for case (a), and was given previously in Fig. 8.7. The *net* work-diagram area for case (c) (same size machine as case (b) but with the proportions adjusted for maximum power) was 2.0 times that for case (a).

Walker and Agbi (1973) presented optimum design charts similar to those for the gaseous working fluid given in Chapter 5. For the compound working fluid separate charts were developed for different values of the mass ratio β.

Fluid combinations

The theory available was so highly idealized that the only characteristic

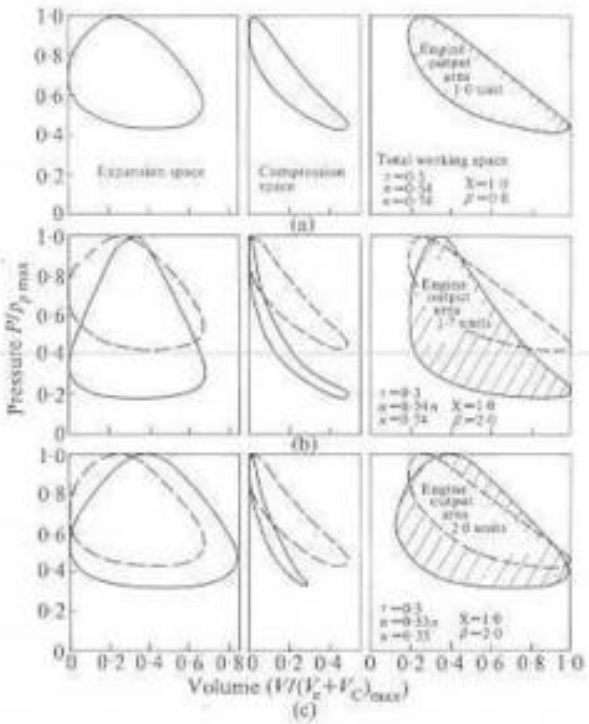


FIG. 8.12. Optimum work output for Schmidt-cycle systems with gaseous and compound working fluids (after Walker *et al.* 1973).

weights M_e and M_v. Studies were made of fluid combinations having a range of values of the ratio $N = M_e/M_v$. Light gases (helium and hydrogen, $M_e = 4$ and 2) were combined with heavy liquids (Freons $M_v > 100$); heavy gases (carbon dioxide $M_e = 44$) were combined with light liquids (water $M_v = 18$) and intermediate combinations were investigated.

For the particular configuration studied, $\tau = 0.3$, $X = 1.0$, $\kappa = 0.74$, $\alpha = 0.54\pi$ (97°), $\beta = 2$ the best results in terms of the maximum work ratio, ζ, were gained with a molecular weight ratio N corresponding to that for air and water i.e. $N = 29/18 = 1.6$. Insufficient work was done for this result to be generally applicable to all machine configurations. Furthermore, the idealizations of the theory are such that any conclusions drawn must remain tentative.

The elements of the theory thought to be particularly suspect are the assumptions of isothermal processes, constant mixing of the fluids for uniform mass-ratio β and neglect of aerodynamic-flow loss. We have seen

of working fluid. There is every reason to believe that the compound working fluid would be similarly dependent but there is insufficient data presently available for rigorous analysis.

Refrigeration applications

All the work outlined above was carried out with reference to prime movers, power systems converting heat to work. The expansion space was heated and the compression space was cooled.

The applicability of the concept of compound working fluids to refrigeration applications was investigated by Walker and Agbi (1973). It was found that a compound working fluid had the same beneficial effect resulting in an increase in the effective volume compression ratio. This in turn caused an increase in the range of the pressure excursion and hence an increase in the refrigerating capacity of the machine with no increase in engine size, weight, or cost compared with a gaseous working fluid. The degree of improvement of the compound working fluid was significantly greater at relatively high refrigeration temperatures and declined progressively until at cryogenic temperature levels it was not markedly different to the conventional gaseous cycle. Moreover, at the lower temperatures it was necessary to resort to a hypothetical fluid as the phase change component was required to have properties not possessed by any known real fluid.

Potential applications for compound working fluids

The preliminary investigations outlined above indicated that the specific output of Stirling engines, acting as prime movers or cooling engines, may be significantly improved by the use of compound working fluids. The degree of improvement (twice as much) was sufficient to justify the next level of investigation.

With a compound working fluid and consequent high specific output it may be possible to decrease the very high pressure levels of gaseous working fluids that are characteristic of automotive engines. Moreover, the boiling and condensing processes occurring to the phase change component during the cycle are associated with very high rates of heat transfer, and the phase change component may have a high enthalpy of evaporation (water has a very high value). Consequently, the processes of compression and expansion in the engine are likely to approximate to isothermal conditions more closely than in the gaseous machine.

The presence of liquid as droplets in suspension or as a film, could have profoundly beneficial effects on the effectiveness and life of fluid seals. The presence of water or light hydrocarbons on PTFE filled plastic (Rulon) seals operating in helium has been found by the author to greatly enhance both the operation and life. The effects are extremely variable and difficult to reproduce, and insufficient experimental work has been done

to arrive at any other than the most general conclusions. Because of the difficulty in reproducing results, no account of the work has been published.

Improvement to seal life, friction, and wear is significant in double-acting engines for the dry seal on the piston separating the two Stirling systems (at different pressure levels) found in each cylinder. The seal is invariably located at the cold (ambient temperature) end of the piston and thus adjacent to a compression space where the phase change component would exist as a liquid.

In engines with a mechanical drive, crank, rhombic, or swash-plate, another pressure seal is found where the piston or displacer-rod passes through the cylinder. This can either be a rolling seal (Philips) or a Rulon rubbing seal. Invariably the seal is located at the ambient-temperature end of the cylinder where the phase change component would exist as a liquid. It would be a simple matter indeed to design the seal locations as shown in Fig. 8.13 such that the liquid phase change component accumulates in the seal well. This transforms the seal from a gas seal to a liquid seal and thereby relaxes the seal requirements by several orders of magnitude.

In the studies of compound working fluids described above the

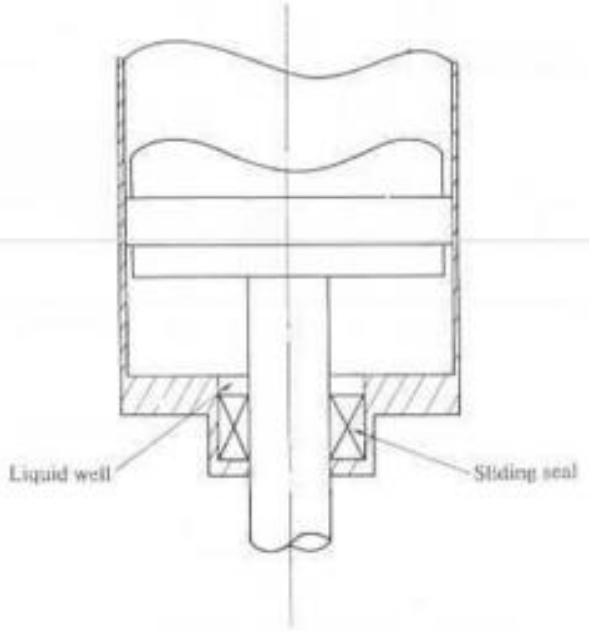


FIG. 8.13. Sliding liquid seal in engine cylinder with compound working fluids.

greatest degree of improvement in specific output was found at the lower expansion-space temperatures (super-ambient) in the case of prime-mover engines and at the higher refrigeration temperatures (sub-ambient) for cooling engines. This suggests that compound working fluids might well be applied in engines operating on low grade thermal sources, i.e. exhaust-heat bottoming-cycle systems for gas turbine or internal combustion engines, or relatively unsophisticated, and therefore low-cost, solar collectors. For cooling engines the best applications, at high refrigeration temperatures, may be air conditioning units, heat pumps, and food preservation or processing.

Intriguing combinations of Stirling-cycle prime movers driving Stirling-cycle cooling engines are at an outline design stage. Both units use the same compound working fluid, one of the Freons in combination with hydrogen, as thermally-activated cooling engines driven by solar energy, exhaust heat, natural gas, fossil fuel, municipal waste, and biomass combustion.

Experimental work with compound working fluids

Little experimental work with compound working fluids has been reported. William Beale added a small quantity of water to the cylinder of a small demonstration free-piston engine (see Fig. 8.11) to improve sealing in the working space. He found the resultant increase in the cyclic pressure excursion was sufficient for the output to improve dramatically until, after a few cycles, the displacer collapsed with the increased pressure.

This experience prompted the author to undertake the studies described above and to construct the apparatus shown in Fig. 8.14. It consists of a long cylinder closed at both ends and containing a hollow displacer. The displacer was actuated by a crank connecting-rod mechanism driven by a motor. Various speeds and strokes of the displacer may be obtained. One end of the cylinder was heated (by internal electric resistance heaters) and the other end was cooled by a water jacket. The cylinder was fitted with a Kistler quartz piezoelectric pressure transducer and thermocouples in the hot and cold spaces in the cylinder to measure mean temperatures. The apparatus was mounted within a structural steel frame so that it could be operated with the cylinder axis at any orientation. In the cylinder, a gas charging valve was provided that may be coupled to a compressed gas bottle. Similarly, a liquid charging valve was provided so that a measured volume of liquid could be injected into the cylinder by hypodermic syringe (before the cylinder was pressurized with gas).

A clearance space between the cylinder and the displacer provided a

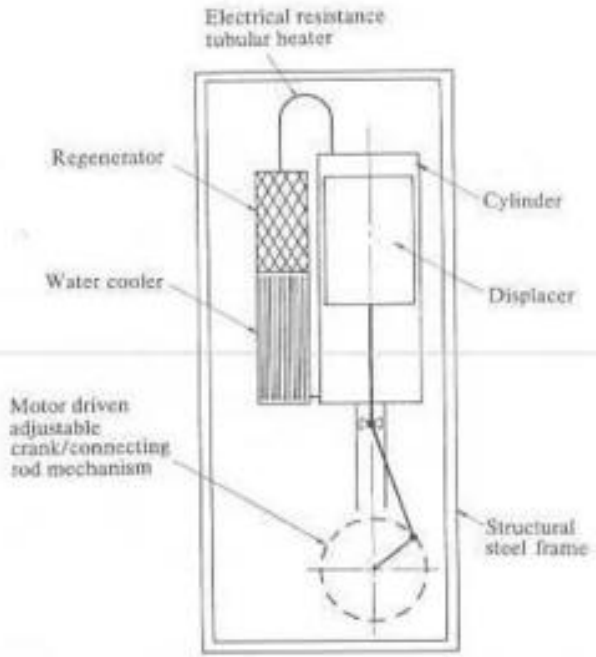


FIG. 8.14. Experimental apparatus for the study of compound working fluids.

hot end of the cylinder. The other end operated always at the cold end so that along the length of the displacer there was an appreciable temperature difference. The displacer and cylinder walls were made of thin wall stainless steel to reduce the conduction losses.

A family of displacers with a range of length to diameter ratios has been made. Similarly, the cylinder is made up from several elements of a length to suit the various displacers and the several displacer strokes that may be used. The cylinder elements are constructed in several internal diameters to provide for a range of clearances in the regenerative annulus. The 'gap' or diametral clearance has a critical effect on regenerative annulus performance. For a given pressure drop the mass-flow rate is a function of the cube of the gap dimension. It is vital that the displacer be maintained central (to provide a uniform gap) and that the gap be minimized.

An alternative arrangement, as yet unused, is available where a seal is provided at the cold end of the displacer and a flow path for the working fluid is provided from the hot to cold end through:

(b) a regenerator case, which, in turn, is connected to
(c) a water-cooled tubular heat exchanger connected to the cold end of the cylinder.

The apparatus, in either form, corresponds to a single-cylinder Stirling engine of the piston-displacer type in which the piston is held stationary, so the volume of the system remains constant. As the displacer moves in the cylinder, fluid is displaced from the hot space to the cold space and vice versa. This causes a cyclic variation in the pressure of the working fluid that is, of course, exactly in phase with the motion of the displacer, or the volume variation in the hot 'expansion' space. Maintaining the total volume constant by, in effect, holding the piston fixed, introduces considerable simplification in the analysis of a Stirling engine system. The intent of the apparatus was, first, to support the development of a procedure for predicting the range of pressure variation in the constant total volume system that conformed well with the measured values for different compound fluids. It was anticipated that a similar procedure could be adapted for real Stirling engines with a variable total volume system. The same apparatus was intended for subsequent studies of regenerative heat exchangers in which the fluid experienced a phase change. Virtually no information in the literature has been located about the operation of regenerative heat exchangers operating with a phase change of the fluid flow.

Initial experience with the apparatus using air and water has been most disappointing for no way has been found to maintain a uniform distribution of the two components in the cylinder. The water either collects in the cold space, or if the apparatus is operated upside down, with the hot space below the cold space, there are intermittent, massive pressure excursions when large droplets of water are suddenly vaporized. The operation is reminiscent of those unfortunate early aviators whose spectacular catastrophes are regularly screened on television to the huge amusement of one's children.

Notwithstanding the author's own lack of success, the principle of the apparatus is commended to other potential researchers as a good place to start the study of this exciting field of research.

CHEMICALLY REACTIVE WORKING FLUIDS

Introduction

Improvement in the power density of a Stirling engine can be obtained by the use of a chemically-reactive working fluid operating as a condensing, dissociating gas. It may be used alone or in combination with the inert gaseous carrier component.

The effect of a condensing, dissociating working fluid is similar to the two-phase two-component working fluid discussed above. The chemically reactive working fluid, liquid in the low temperature region of the cycle, evaporates to a vapour as the temperature increases with an accompanying increase in the specific volume. At higher temperatures the vapour or gas may dissociate into less complex compounds or to elemental species in the gaseous state. Dissociation may be accompanied by a stoichiometric mole change and the reaction may be endothermic or exothermic.

Nitrogen tetroxide

The fluid that has received most attention is nitrogen tetroxide. The dissociation reactions of interest are:

(1) $N_2O_4 \rightleftharpoons 2NO_2 - 623$ kJ/kg (268 Btu/lb_m)
(2) $2NO_2 \rightleftharpoons 2NO + O_2 - 1230$ kJ/kg (529 Btu/lb_m).

Both reactions are rapid and equilibrium conditions are swiftly approached at all temperatures (Russer and Wise 1956). The further reaction involving dissociation of NO to elemental species is, by comparison, a slow reaction that may be disregarded.

Nitrogen tetroxide is of interest because of the large increase in the number of moles as the equilibrium is shifted from left to right, and because the reactions are fast and are endothermic from left to right. This has the effect of increasing the apparent heat capacity of the fluid and assists in a closer approach to isothermal expansion. Finally, and most importantly, the reaction has been studied extensively and tables of thermodynamic and transport data are available (Baker *et al.* 1964, Krasin and Nesterenko 1967, Krasin 1971).

Nitrogen tetroxide is highly corrosive and very toxic, an unpleasant fluid altogether to contemplate for use in Stirling engines. Such considerations are not important in theoretical investigations of the mole change effect, though for practical engines or for laboratory investigations they are all important.

Walker and Metwally (1977) studied the use of nitrogen tetroxide as the working fluid in a Stirling engine. The study was an elementary theoretical analysis, based on the Schmidt cycle, with nitrogen tetroxide in combination with an inert gaseous carrier component. The study was designed to complement the earlier work of Walker and Agbi (1974) with a two-phase two-component condensing, non-reactive working fluid (water and air). The results of the two studies were presented in the literature in exactly similar format for direct comparison. With the partially-reactive, condensing, working fluid there was a very substantial increase in the specific output compared with the use of a gaseous working fluid. Comparisons were made on the basis of similar maximum pressures, maximum volumes, and temperature ratios so that, at least to first order approximation, the engines were the same weight, size, and cost. The comparison was made for the standard design case for

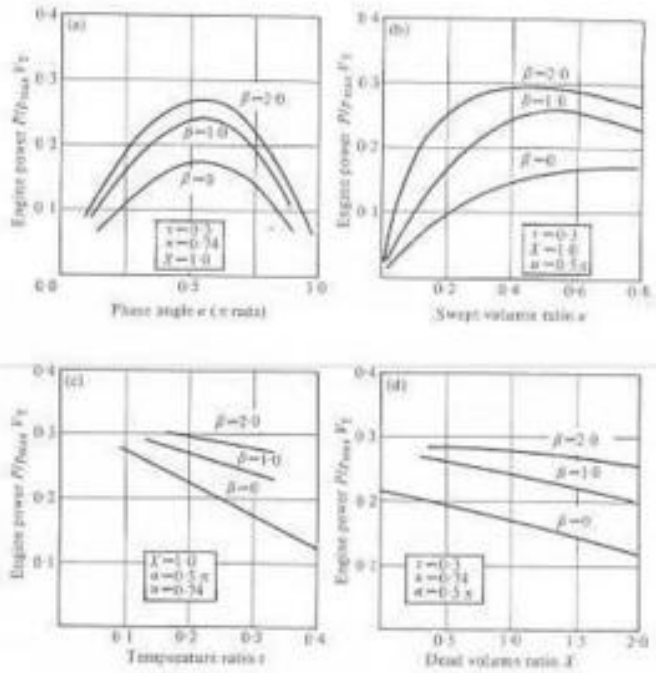


FIG. 8.15. Effect of system variables in the power parameter of a Schmidt-cycle Stirling engine system with a partially reactive, condensing working fluid; β is the mass ratio (m_r/m_a) of reactive component to inert gaseous carrier (after Walker and Metwally 1977).

maximum power output with a gaseous working fluid given by Walker (1962), i.e. temperature ratio $\tau = 0.3$, dead volume ratio $X = 1.0$, swept volume ratio $\kappa = 0.74$ and the phase angle $\alpha = 0.54\pi$ (97°).

The results of a parametric study about the standard design case are presented in Fig. 8.15. In this figure, the mass ratio β is m_r/m_a where m_r and m_a are the masses of reactive component and inert gaseous carrier respectively in the working space. This figure may be compared with Fig. 8.9, a similar diagram for the two-phase two-component working fluid.

Work diagrams are given in Fig. 8.16 for the standard-design case with a gaseous working fluid, $\beta = 0$, and with a partially reactive condensing working fluid. This figure may be compared directly with Fig. 8.7 for the two-phase two-component working fluid.

The work ratio ζ, or degree of improvement compared with a gaseous working fluid, as a function of the mass ratio β is as shown in Fig. 8.17. This figure for a partially-reactive working fluid may be compared directly

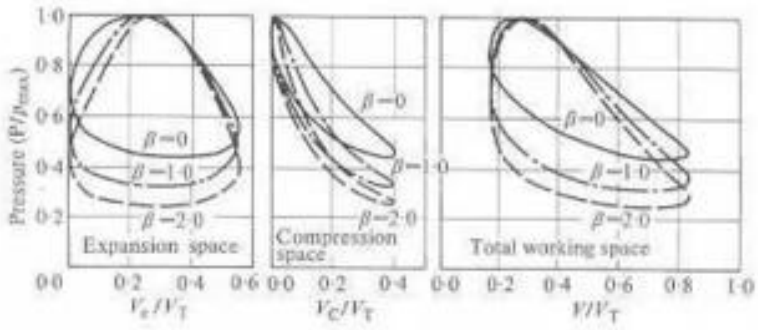


FIG. 8.16. Work diagrams for a Schmidt-cycle Stirling engine system with a partially reactive condensing working fluid. Diagrams are drawn for the standard design case $\tau = 0.3$, $X = 1$, $\alpha = 0.54\pi$ rads, $\kappa = 0.72$; β is the mass ratio (m_r/m_a) of reactive component to inert gaseous carrier (after Walker and Metwally 1977).

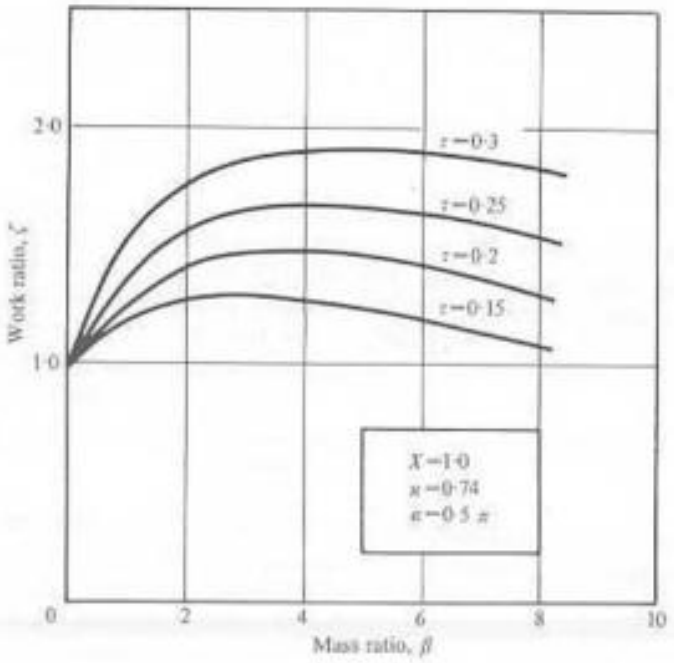


FIG. 8.17. Work ratio for a Schmidt-cycle Stirling engine system with a partially reactive

The general conclusion drawn from this very limited and highly idealized study was that the chemically reactive working fluid offered no significant advantages over the more simple two-phase, two-component working fluid. However, it is not satisfactory for the matter to be simply left there, for the work done so far is not of sufficient depth for definitive conclusions to be drawn. Others are therefore encouraged to extend these studies.

Single component multi-phase systems

There is the possibility that fluids could be found with the appropriate properties to allow their use as the *sole component* in a condensing, dissociating system. Removal of the inert gaseous carrier would obviate the problem of ensuring a uniform mass ratio β at the various temperature levels in the engine. Continuous uniform mixing has proved so far to be a particular difficulty in experimental work with two-phase two-component working fluids. There is no reason to believe the problem would be any less with condensing, dissociating fluids in combination with an inert carrier. There is much to be gained by the use of a single component working fluid, liquid in the cold space, and vapour, perhaps dissociated, in the hot region. Water was used in this way as the working fluid for the Thermoelectron 'tidal regenerator engine' of an artificial heart system described in Chapter 17.

Wolgemuth (1969a) studied the equilibrium performance of the theoretical Stirling cycle with nitrogen tetroxide and later Kovtun *et al* (1967) did similar work. Wolgemuth concluded that with a condensing chemically-reactive working fluid, a substantial gain in power density could be achieved. However, the gain was somewhat offset by a loss in thermal efficiency depending on the cycle pressure ratio and the regenerator effectiveness. On the other hand, Kovtun *et al.* concluded that use of a dissociating gas in the ideal Stirling engine would *increase* the efficiency. The example was given for an efficiency of 13 per cent with a non-dissociating gas, increasing to 24 per cent with a dissociating gas.

The reason for the discrepancy in the findings of Wolgemuth and Kovtun *et al.* lies simply in the fact that both assumed deviations from the ideal Stirling cycle and made different assumptions. In the simple idealized Stirling cycle (as in every thermodynamic cycle where all the external heat is added or rejected at constant temperatures) the thermal efficiency has to be the Carnot efficiency, whatever the working fluid.

The availability of large computers and the increasing level of sophistication in Stirling engine analysis combine to suggest that interesting studies in the field of complex working fluids lie ahead. There is an urgent need for fundamental experimental work in this area.

LIQUID WORKING FLUIDS

Closed-cycle regenerative engines with liquid working fluids were described by John Malone (1931). They conform in every way to the definition for a Stirling engine given in Chapter 1. However, liquid cycle regenerative engines are sufficiently different to be classified separately, perhaps as Malone-cycle engines.

So far as is known, Malone presented only one account of his work. His paper was tantilizingly vague and non-technical but claimed that indicated efficiencies of 27 per cent were obtained. The paper contained photographs of a large two-cylinder engine and a smaller single-cylinder machine. They operated at low speeds of from 24 to 250 revolutions per minute. Very high pressures were used. In the description given, the pressure varied from a low value of 6.8 MN/m^2 (1.5 tons per sq in) to a maximum of 27.4 MN/m^2 (6 tons per sq in).

The mechanical arrangement used by Malone to describe his engine operation was similar to that shown in Fig. 8.18. Two parallel cylinders contained reciprocating elements. One was a displacer including a regenerator and the other was a piston equipped with pressure seals. The arrangement corresponded in every way to the Heinrici/Stirling engine arrangement for a piston-displacer system in separate cylinders. The upper end of the displacer cylinder was heated and the lower end was cooled. The compression cylinder was cooled.

Malone gave the pressure of the working fluid with the elements disposed in the four positions shown in Fig. 8.18. A speculative set of

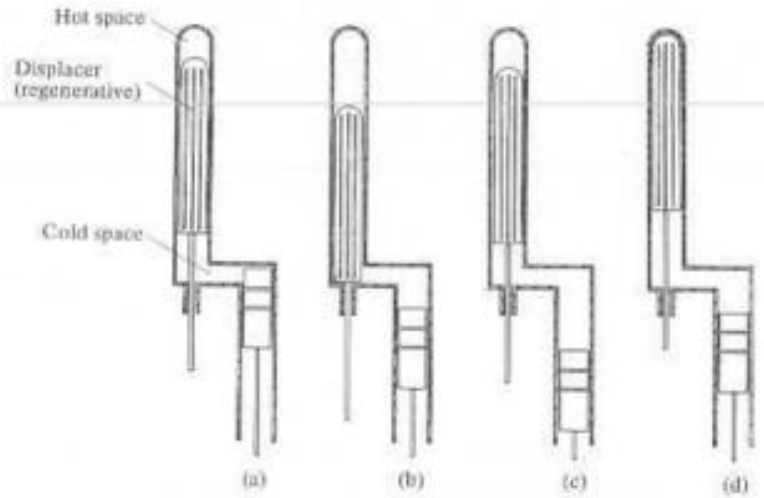


FIG. 8.18. General arrangement and operating sequence of Malone engine with liquid

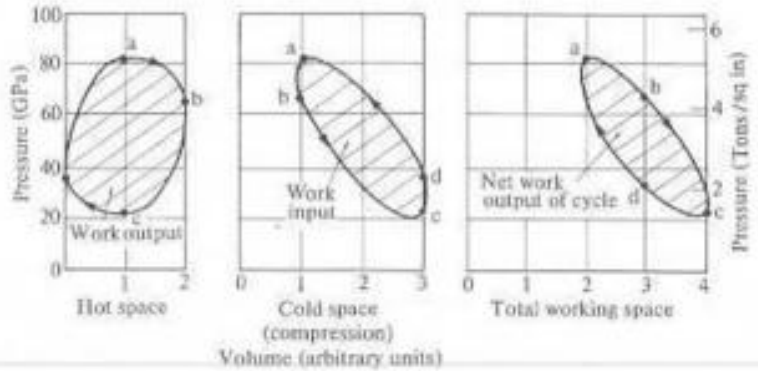


FIG. 8.19. Speculative work diagrams for Malone engine with liquid working fluid based on pressure data given by Malone (1931).

work diagrams for the system based on arbitrary volume units and the pressures quoted by Malone are shown in Fig. 8.19. The system is clearly capable of producing work in exactly the same way as a conventional Stirling engine.

Liquids are generally thought to be incompressible, that is, to have the same density regardless of pressure or temperature. For most applications this assumption causes no significant error. However, liquids are compressible although, of course, very much less so than gases. It is this change in volume with pressure and temperature that provides the means to operate a liquid-cycle engine.

By way of example consider Fig. 8.20. This shows isotherms for water presented on the pressure–volume plane. Isotherms for 0 °C (32 °F) through to 648 °C (1200 °F) are shown. The critical isotherm for 270 °C (700 °F) is shown emphasized and the critical point is marked at the bottom right-hand corner. The critical isotherm is important for it sets the upper limit to the temperature at which a liquid can exist. Above this temperature, whatever the pressure, the fluid is a dense gas with some, but not all, the characteristics of a liquid.

The operating range of pressures for the Malone engine is marked (at the left) in Fig. 8.20. The temperatures of the working fluid in the Malone engine are not known but the furnace gas temperature was given as 1371 °C (2500 °F). It is therefore likely that in the hottest parts of the cycle the temperature of the working fluid exceeded the critical temperature of 270 °C (700 °F) so the fluid was in fact not liquid. A better description for the cycle might therefore be the Malone *dense-phase cycle.*

Malone indicated that he had operated engines with mercury, oil, and

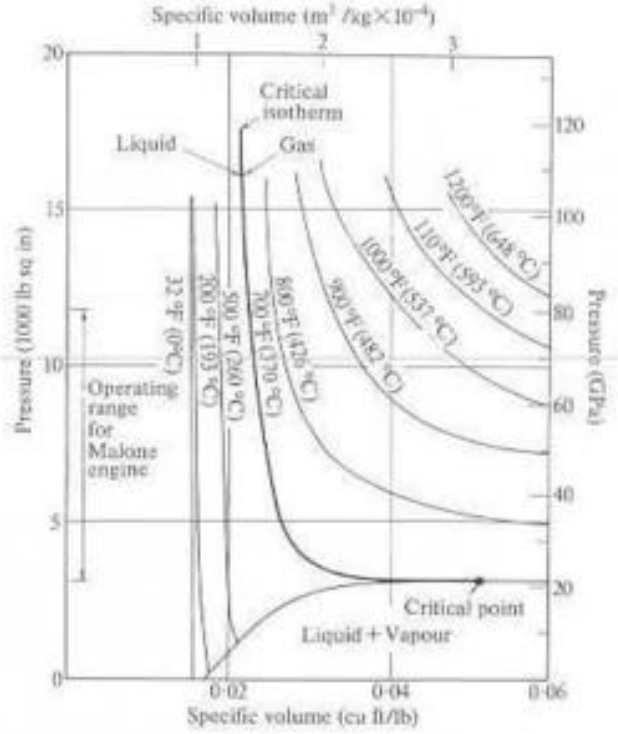


FIG. 8.20. Isotherms for water on pressure–volume plane (prepared from the Electrical Research Association's *Steam Tables*, E. J. Arnold Ltd., London, 1967).

fluids had been assessed including petrols (gasoline), spirits, liquid carbon dioxide, sulphur dioxide, but water was the preferred medium. In a discussion of the desirable fluid properties he mentioned, in passing, possibilities for compound working fluids of immiscible liquids, and for separate working fluids in the hot and cold regions (separated perhaps by buoyancy forces) raising intriguing prospects for future research.

The very high pressures in dense-phase engines require the use of thick cylinder walls. Therefore, to minimize thermal conduction losses the displacer cylinder is bound to be a long cylinder of small diameter. Furthermore, because of the extremely high rates of pressure change, the swept volume of the piston will be very small. To reduce cylinder wall thickness and hence weight, a small diameter and relatively long stroke would be used.

effective form of power control for it would vary the mass flow of fluid passing from the cold space to the hot space and hence the range of the pressure amplitude. The exact form of his power control system was not given but appeared to involve the use of a rack-and-pinion gear on the displacer rod.

The high density of the dense phase working fluid would almost certainly preclude operation at high speed. Therefore, the engines would likely be heavy, slow-running machines having thick cylinder walls and very high pressures. What then are the advantages of dense-phase machines that warrant further research attention? One advantage compared with gaseous machines is the very high rates of heat transfer attainable with liquids and their very high specific heats or heat capacities. Secondly, the seal problem for liquids are likely to be easier to handle despite the very high pressures involved. Thirdly, the fluid serves well as a lubricant. A unique and peculiar advantage of the liquid engine is the inherent self-pressurizing characteristic. This may be clearly seen by reference to Fig. 8.20. Consider, for example, an engine at ambient temperature containing liquid water at atmospheric pressure. If the hot parts are heated the pressure level will increase automatically because the heated medium in the hot zone will expand and compress the remaining fluid in the engine as a result of the very high (dp/dv) characteristic evident in Fig. 8.20.

One likely application for a Malone engine is a solar-powered water pump. A water pump using water as the working fluid has evident advantages. Moreover, it can be seen from Fig. 8.20 that appreciable changes in specific volume occur at moderate temperature changes. Therefore, relatively unsophisticated, low-cost solar collectors might be used although, of course, the power density and efficiency would be very low.

The attraction of such a system would be enhanced if the engine could be made self-starting. It is not evident how this might be done, although there is every reason to believe that novel and ingenious free-piston/free-displacer mechanisms could be made that would share with the Beale free-piston Stirling engine the characteristic of inherent self-starting capability (see Chapter 11). The field appears wide open for innovative research and novel developments in Malone dense-phase or liquid-cycle engines.

9 OPERATING CHARACTERISTICS OF STIRLING ENGINES

INTRODUCTION

A STIRLING engine consists essentially of two spaces of variable volume and different temperatures connected by a duct. The spaces are filled with the working fluid and the duct is provided with apparatus for adding heat (heater), abstracting heat (cooler) or storing heat (regenerator). The system can be arranged in any number of ways, in single or multiple combinations, with the volume variations caused by reciprocating or rotary motion. Whatever form of mechanical arrangement is adopted certain common factors and considerations prevail which affect the system performance. These are discussed below.

IDEAL MAXIMUM POWER AND EFFICIENCY

For any Stirling-engine system the maximum power and efficiency would be achieved if the ideal Stirling cycle described in Chapter 2 could be followed. This requires that *all* the working fluid in the system is, at any instant, in the same condition (thermodynamic equilibrium) and all the heat added to the cycle or rejected from it is transferred at constant temperature. Similarly, at any particular location in the matrix, all the heat transferred between the regenerator and the working fluid occurs at constant temperature.

Such conditions are not, of course, achievable in real engines. It is simply not practical to have all the working fluid in one space or the other at the same temperature. There would be no 'dead' space in the engine, that is, no volume in the connecting duct with all the heat transfer apparatus. Equally unrealistic is the requirement for constant temperature (isothermal) heat addition and rejection. It could only be achieved with infinite rates of heat transfer or with a working fluid having a heat capacity that was zero.

The ideal theoretical cycle does define the absolute upper limit of performance and might be represented, for example, by the lines 'A–A' drawn in Fig. 9.1. This shows the power output per unit mass of working fluid and thermal efficiency as a function of some parameter, let us say, the pressure of the working fluid or the speed of the engine.

THE REAL CYCLE

Dead volume

The first practical modification to the ideal cycle is to recognize the impossibility for all the working fluid to be in the same place at the same condition at the same time. Even with the discontinuous piston motion of

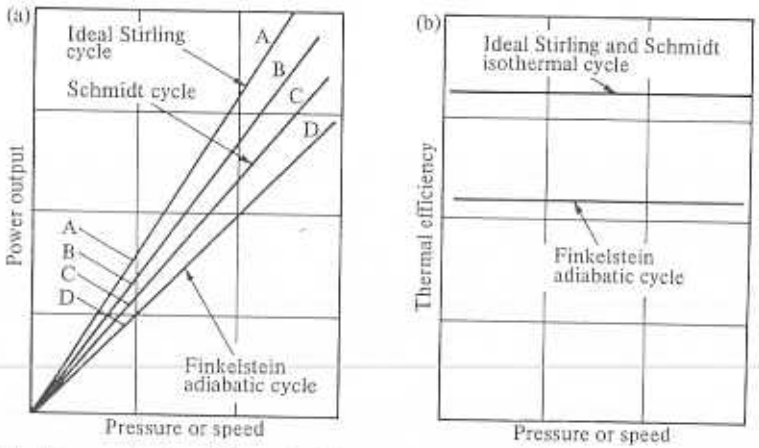


FIG. 9.1. Power output and thermal efficiency of Stirling engines as a function of engine speed or pressure level. Effects of dead volume, harmonic motion, and adiabatic compression and expansion on the ideal cycle.

the ideal Stirling engine there must be some void volume in the regenerator and other heat exchangers. Including this dead volume reduces the amplitude of the pressure excursion when the working fluid is moved from the cold to the hot space and results in a reduction in the power output per unit mass of working fluid, represented by the line 'B–B' in Fig. 9.1(a). The efficiency remains at the ideal cycle value.

A departure from discontinuous piston motion to the sinusoidal piston motion of the Schmidt isothermal cycle causes a redistribution of the working fluid between the compression, expansion and the dead space with consequent further reduction in the range of the pressure amplitude and power output. This is represented by the line 'C–C' in Fig. 9.1(a). As before, the efficiency remains at the ideal cycle value.

In most of the mechanisms used to effect the volume variations the motion is not exactly sinusoidal. As an example, consider the simple crank slider mechanism shown in Fig. 9.2. The motion of the piston P may be represented by the equation:

$$(x/r) = (1 - \cos\theta) + (l/r)(1 - \cos\phi) \qquad (9.1)$$

where x is the distance moved by the piston from the outer dead point A,
r is the crank radius
l is the length of the connecting rod
θ is the crank angle
ϕ is the connecting rod angle.

The second term of the equation is the distortion to true sinusoidal motion of the first term imposed by the crank connecting rod obliquity.

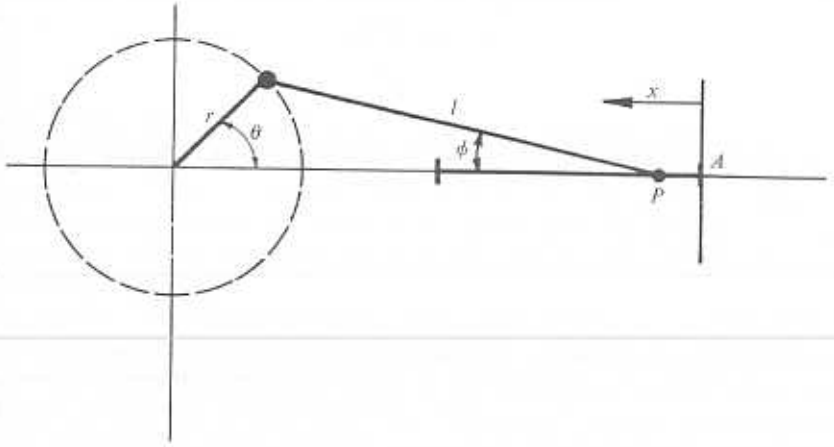


FIG. 9.2. Simple crank-slider mechanism.

The effect can be minimized by the use of a long connecting rod, but this cannot be taken too far as it leads to long, heavy machines.

The effect of non-sinusoidal piston motion is to further modify the mass distribution of working fluid in the engine and may or may not cause a reduction in the power output depending on the other engine parameters. In many cases the effect of non-sinusoidal piston motion is of secondary importance and often may be neglected, at least at the preliminary design stage.

Adiabatic cycle

None of the above factors affected the thermal efficiency. We have assumed so far that isothermal conditions exist in both the compression and expansion spaces so the ideal efficiency remains even though *the quantities* of heat transferred may decrease. However, real engines simply cannot achieve isothermal conditions. The heat transfer is always limited. A better model for high-speed (2000 revolutions per minute) engines may be the Finkelstein adiabatic cycle. This assumes that no heat transfer takes place in the compression and expansion cylinders, i.e. the heat transfer coefficient is zero. In the heater and cooler, however, the rates of heat transfer are assumed to be infinite so the working fluid in those spaces is always at the upper and lower cycle temperatures T_{max} and T_{min}. The cycle is discussed in more detail in Chapter 4.

A conversion from isothermal to adiabatic conditions in the engine cylinder causes a marked redistribution of the cyclic mass variation of the working fluid. The *mean temperature* in the expansion space is less than in the isothermal case, whereas the mean temperature in the compression

space is increased. The work output from the expansion space is therefore decreased, whereas the work input to the compression space is increased. The result is that *both* the net cycle work and cycle thermal efficiency are reduced as shown by line 'D–D' on Fig. 9.1. The conversion to adiabatic operation has a dramatic effect on the efficiency, sometimes decreasing it to half the isothermal value.

THERMAL LOSSES

Various other losses and operating effects may now be considered. For the sake of clarity the line 'D–D' of the adiabatic cycle has been transferred to a new diagram, Fig. 9.3.

Conduction

An important effect is the thermal conduction along the cylinder walls and other conduction paths from the hot zone to the cold zone. Some thermal conduction is inevitable and ways to reduce it to a minimum are discussed in Chapter 5.

Shuttle heat transfer

Shuttle heat transfer or the 'bucket brigade loss' is similar to conduction heat transfer. In the course of the reciprocating motion of the displacer adjacent parts of the cylinder wall and displacer are at different temperatures. This results in heat transfer by a 'dynamic conduction' process from the hot to cold parts.

Convection and radiation

Heat may be lost by convection and radiation from the engine. In an engine with a fossil-fuel/air-combustion system this loss is minimized by

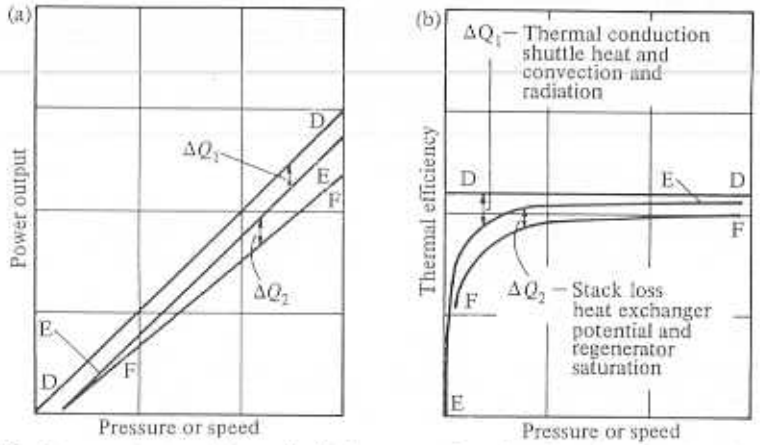


FIG. 9.3. Power output and thermal efficiency as a function of engine speed and pressure level. Effects of thermal conduction, shuttle heat transfer, convection and radiation losses, stack losses, heater and cooler thermal potential loss and the regenerator thermal saturation loss of effectiveness.

enveloping the hot parts by the air preheater. Thermal insulation shrouds may also be used.

All these thermal effects depend mainly on the temperatures in the engine and are more or less independent of the speed and pressure. They may therefore be represented by a constant reduction ΔQ_1 in the power output, represented by the line 'E–E', Fig. 9.3(a). The effect on the engine efficiency of this constant loss is more pronounced at low speeds or pressures where the engine power output is low. Therefore, the efficiency characteristic assumes the curved form represented by 'E–E' in Fig. 9.3(b), tending to zero at the lower powers.

Stack loss

Another thermal loss, the stack loss, arises from the heat carried off by the exhaust gas. In the Stirling engine, heat carried off by the exhaust is heat that has not passed into the engine and so represents a direct loss. It is partially recovered by the inlet air preheater but this is limited to a finite size and so some stack loss is inevitable. The magnitude of the stack loss will increase as the amount of heat supplied to the engine increases.

Heat exchanger temperature potential

In the heater and cooler a temperature difference between the hot-side and cold-side fluid provides the thermal potential to cause the heat transfer to occur. In the heater, the engine working-fluid temperature is *less* than the heater-wall temperature, whereas in the cooler the reverse is true and the working fluid is *warmer* than the cooler-wall temperature. As the quantities of heat transferred increase with increase in either speed or pressure, so the thermal potential must increase. This can be represented as another form of thermal loss, the magnitude of which increases as the power level rises.

Imperfect regeneration

The regenerator of the engine may be thought of as a thermodynamic sponge, which alternately accepts heat from and rejects heat to the working fluid. The effectiveness of the regenerator depends on the ratio of thermal capacities of the metal matrix and of the gas flowing through it. As the density of the gas increases (with increase in pressure) or as the frequency of flow reversal increases (with speed) the heat capacity of the gas flow increases. This tends to saturate the regenerator matrix and the result is a larger amplitude in the temperature swing of the metal. This causes a reduction in the effectiveness of the regenerator corresponding to a 'loss' which increases at high power levels.

The combined effects of stack loss, the heat exchanger thermal potential loss and the regenerator saturation loss 'ΔQ_2' may be represented on

Fig. 9.3 by the line 'F–F'. This is drawn schematically to show the reduction in power increasing linearly as the pressure and speed increases and results in a virtually constant reduction of the cycle thermal efficiency.

FRICTION EFFECTS

For the sake of clarity the line 'F–F' is transposed to a new diagram, Fig. 9.4.

Mechanical friction

There are two important friction effects to consider in Stirling engines. The first is the mechanical-friction effect arising from the action of piston rings, rubbing seals, bearings, oil pumping, and the like. With the engine idling and producing no useful output power there will still be considerable mechanical friction. This will increase as the engine speed and pressure increase. Inclusion of mechanical friction reduces engine output and efficiency as represented by the line 'G–G' in Fig. 9.4.

Aerodynamic friction

Aerodynamic friction is another important friction effect. This is manifest in the fluid pressure drop which occurs across the heater, regenerator and cooler. The difference in pressure between the compression and expansion spaces results in a reduction of the amplitude of the pressure variation in the expansion space. As a consequence the area of the work diagram for the expansion space is reduced and so the *net cycle work* is reduced. The pressure drop is a function of the density and the *square* of velocity of the fluid flow. It may be represented by the line 'H–H' in Fig. 9.4.

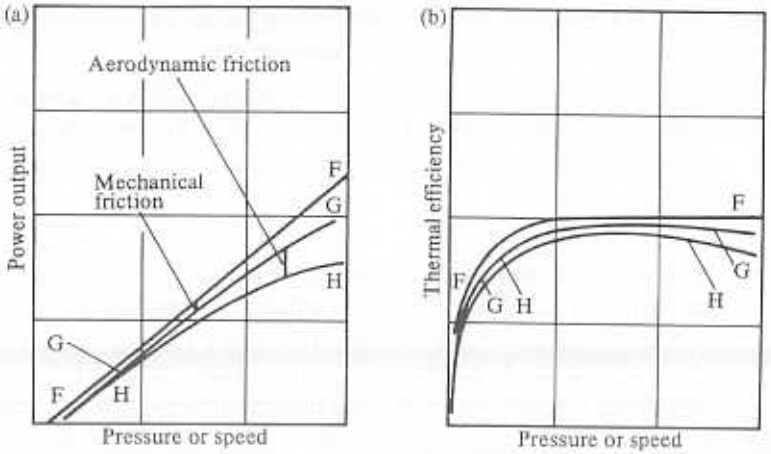


FIG. 9.4. Power output and thermal efficiency as a function of engine speed and pressure level. Effects of mechanical and aerodynamic friction.

ENGINE AUXILIARIES

Transposing the line 'H–H' onto a new diagram, Fig. 9.5, the only correction now left to make is to include some allowance for the power consumed by the engine auxiliary equipment. This may comprise a lubricating-oil pump, an electric alternator, or generator for battery charging, lighting and control functions, a fluid compressor, cooling water pump, radiator fan, air preheater drive and miscellaneous other devices. The power consumption of the auxiliaries is most likely to increase as the speed or pressure level increases.

The net power cycle efficiency for the engine, driving its auxiliaries, is represented in Fig. 9.5 by the line 'J–J'. This then is perhaps characteristic of the shape of performance curve that might be measured by dynamometer brake testing of the engine.

OPERATING CHARACTERISTICS OF REAL ENGINES

Power and efficiency

Confirmation of the above was provided by Meijer (1959a) with the publication of the brake power and efficiency characteristics shown in Fig. 9.6. These were measured on a 30 kW (40 hp) single-cylinder rhombic drive Philips Stirling engine of the piston-displacer type with hydrogen as the working gas. The engine cylinder bore was 88 mm (3.46 in) with a power piston stroke of 60 mm (2.36 in). The pressure compression ratio p_{max}/p_{min} was 2.0 with a maximum related pressure of 13.7 MN/m^2 (2000 lb per sq in). The nominal temperatures for the engine were 700 °C (1290 °F) in the heater and 15 °C (60 °F) in the cooler.

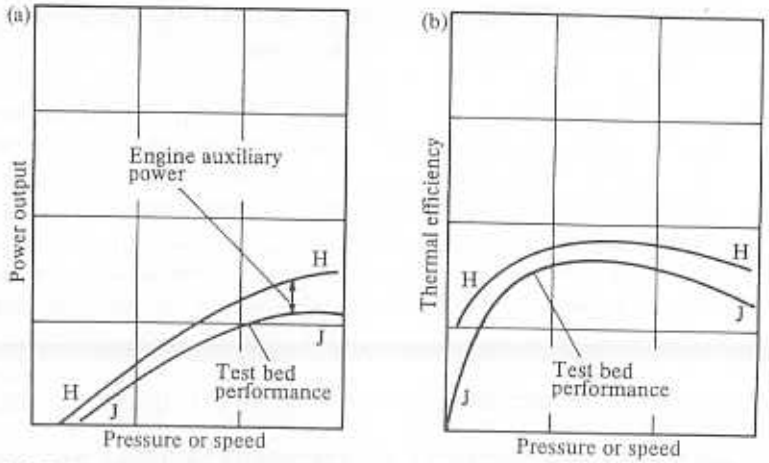


FIG. 9.5. Power output and thermal efficiency as a function of engine speed and pressure level. Effects of engine auxiliary equipment.

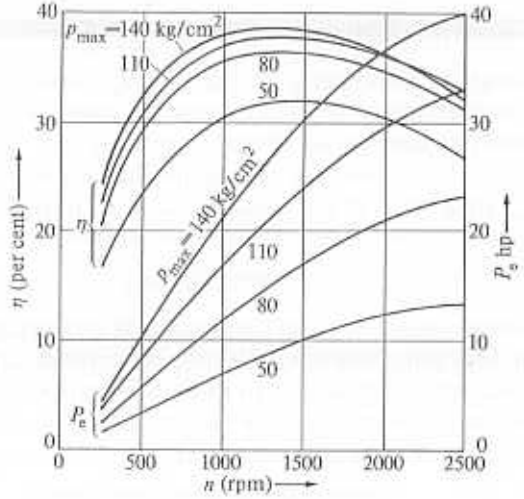


FIG. 9.6. Brake power and thermal efficiency of single-cylinder rhombic-drive Philips Stirling engine (after Meijer 1959a).

Torque/speed

Other data provided by Meijer included the engine torque curve as a function of speed reproduced in Fig. 9.7. The virtually flat torque/speed curve of the Stirling engine is particularly favourable for automotive applications. High torque at low speeds is desirable to achieve good acceleration. The inherent high torque at low speeds of Stirling engines allows the use of a relatively simple transmission system for vehicular use and helps offset the increased capital cost compared with internal combustion engines.

Cyclic torque

In addition to the flat torque/speed characteristic discussed above, the Stirling engine has a particularly favourable cyclic torque characteristic. This is the torque variation experienced, at the output shaft, during one revolution. The cylinder torque of a Stirling engine is much less variable than for an internal combustion engine of the same power.

Meijer (1970b) made the comparison shown in Fig. 9.8 for four-cylinder Stirling and spark-ignition engines of about 73 kW (100 hp). The reason for the virtually constant torque of the Stirling engine is that the range of the pressure variation in each cylinder is small, $p_{max}/p_{min} = 2$ approximately, and there is a complete cycle in each cylinder every revolution. In the internal combustion engine the pressures may range from 0.08 to 5 MN/m² (12 to 725 lb per sq in) per cycle and, in a

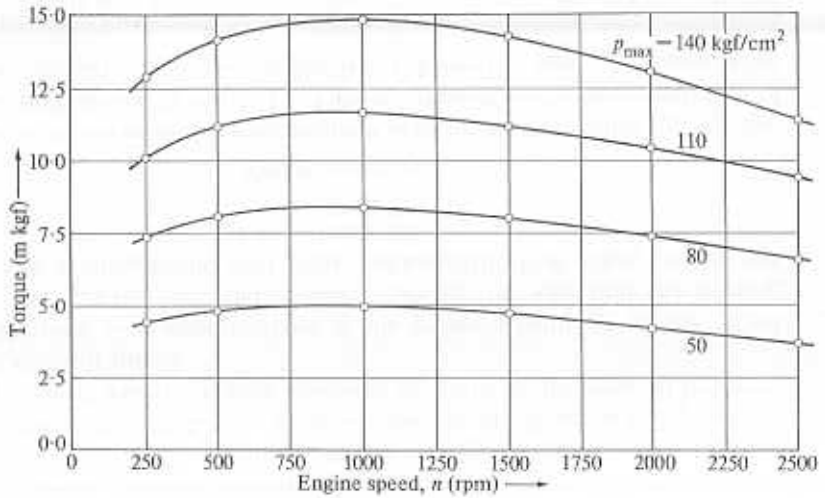


FIG. 9.7. Torque/speed characteristic of single-cylinder rhombic-drive Philips Stirling engine (after Meijer 1959a).

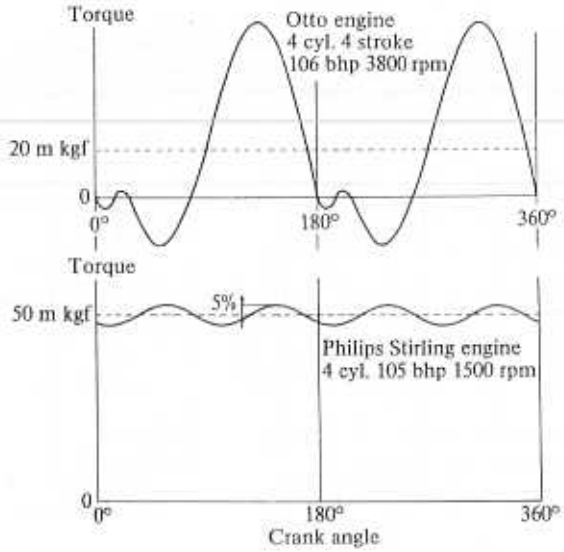


FIG. 9.8. Comparison of cyclic torque variation for four-cylinder, 74 kW (100 hp) Stirling and gasoline engines (after Meijer 1970b).

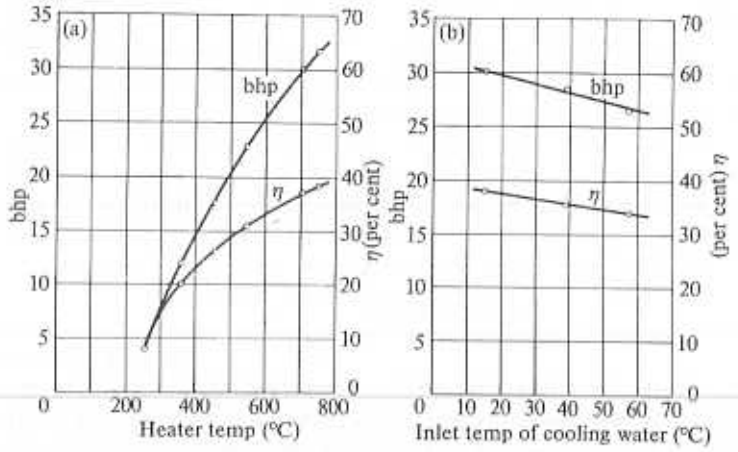


FIG. 9.9. Brake power and thermal efficiency of single-cylinder rhombic-drive Philips Stirling engine as (a) a function of the heater temperature and (b) the cooler temperature. (After Meijer 1959a.)

four-stroke engine, there is a complete cycle every two revolutions. The advantage of the reduced torque variation is that the size of flywheel necessary for the engine is greatly reduced.

Heater and cooler temperatures

Meijer (1959a) also gave the data reproduced in Fig. 9.9 showing the performance in terms of power output and efficiency of the engine as a function of the heater temperature and of the cooler temperature. These show very clearly the improvements in performance that may be gained from operation at high heater temperatures and low cooler temperatures.

Heat balance

Heat balances for the engine as a function of speed or power output were also given by Meijer and are reproduced in Fig. 9.10. These are important characteristics for they show that the heat balance of a Stirling engine is independent, to a surprising degree, of the engine power output or speed. Of the total heat supplied the useful heat (brake work) is between 30 and 40 per cent. This, of course, is the thermal efficiency and a measure of the useful output per unit cost of fuel. About 50 per cent of the heat supplied is rejected to the cooling water. This, of course, includes all the thermal conduction, shuttle heat transfer, and most of the mechanical and aerodynamic frictional losses discussed above.

In this particular engine the flue losses (stack losses) were about 10 per

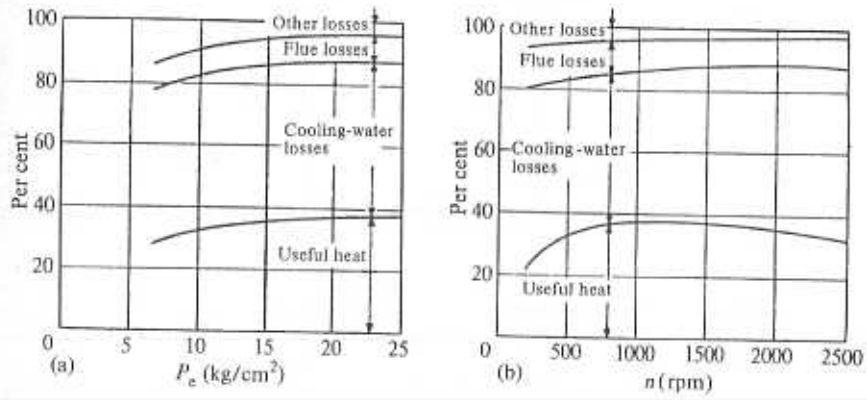


FIG. 9.10. Heat balance for a single-cylinder, rhombic-drive Philips Stirling engine (a) as a function of engine output at constant speed $n = 1500$ revolutions per minute, and (b) as a function of speed at constant maximum pressure = 13.7 MN/m² (2000 lb/sq in). (After Meijer 1959b.)

cent of the total heat supplied and the balance was attributed to 'other losses', presumably due to incomplete fuel combustion and to convective and radiative heat transfer from the engine. This engine was tested on a dynamometer but without the auxiliary equipment necessary for independent operation.

Part load performance

The relatively constant thermal efficiency of Stirling engines over a very wide range of power output or speed is a particularly advantageous characteristic for vehicle use. In most traction applications full power is rarely required and the engine operates for most of the time at between 0.2 and 0.4 full power.

A 'flat' part load characteristic is the prime advantage of the diesel engine over gas turbines and gasoline motors. The fact that the Stirling engine has a comparable part load characteristic is a prime reason for sanguinity when contemplating vehicle use.

Performance map

Meijer (1959a) gave the performance map of characteristics for Stirling engines, reproduced in Fig. 9.11. This is the most useful representation of performance data for a heat engine for it shows the 'best' operating point for the machine.

The vertical axis is the mean effective pressure of the engine derived from the equation:

$$\text{Power, } P = (p_e L A n)/K \qquad (9.2)$$

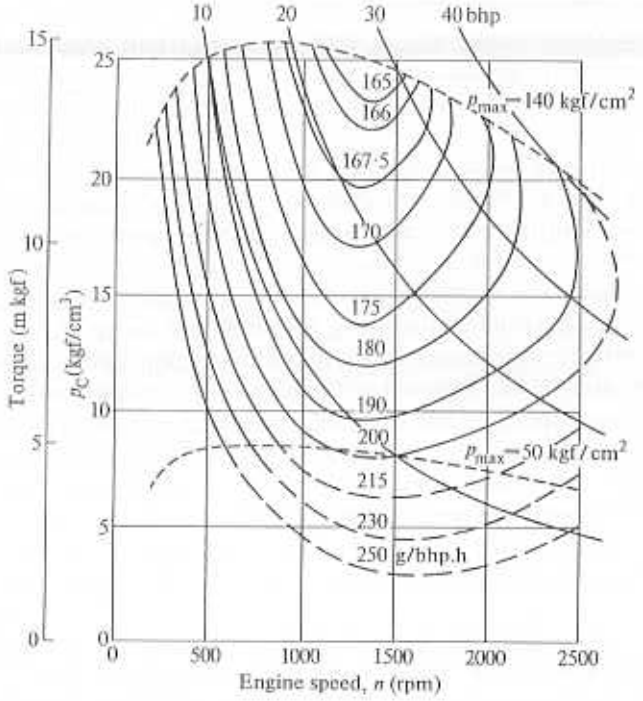


FIG. 9.11. Performance map for single-cylinder, rhombic-drive Philips Stirling engine. Curves of constant specific fuel consumption are drawn against the brake mean effective pressure (b.m.e.p.) p_e and engine speed n. Curves of constant power output are shown for $P = 10$, 20, and 30 hp (7.5, 15, and 22.5 kW). The upper and lower broken lines represent lines of constant maximum pressure of 140 and 50 kg/cm² (2000 and 735 lb/sq in). (After Meijer 1959a.)

where P = output of the engine
p_e = mean effective pressure
L = length of stroke of the piston
A = area of piston so that LA represents the volumetric displacement per cycle
n = speed of the engine
K = some constant.

Consideration of this equation will reveal that the mean effective pressure is simply a measure of the engine work output *per unit volume* of piston displacement per cycle. As such it is of exceptional value for it permits the comparison of engines of radically different size, speed and net power

output. Model aircraft diesel engines and huge marine diesel engines both have a similar mean effective pressure.

The horizontal axis on Fig. 9.11 is the engine speed and the contours are the lines of constant specific fuel consumption, grams of fuel per hour per horsepower. The lines for constant power output of 7, 15 and 22 kW (10, 20 and 30 hp) are also shown. The diagram is bounded at the top and bottom by the broken lines for the maximum (13.7 MN/m²) (1990 lb per sq in) and minimum (5 MN/m²) (725 lb per sq in) pressures at which dynamometer results were obtained.

The engine was unable to achieve its maximum efficiency (or minimum specific fuel consumption, the inverse of efficiency) because of the limitation of the maximum pressure. A more complete set of characteristics, but for a different engine, was published later by Meijer (1970b). These are shown in Fig. 9.12 and refer to an optimized four-cylinder Stirling

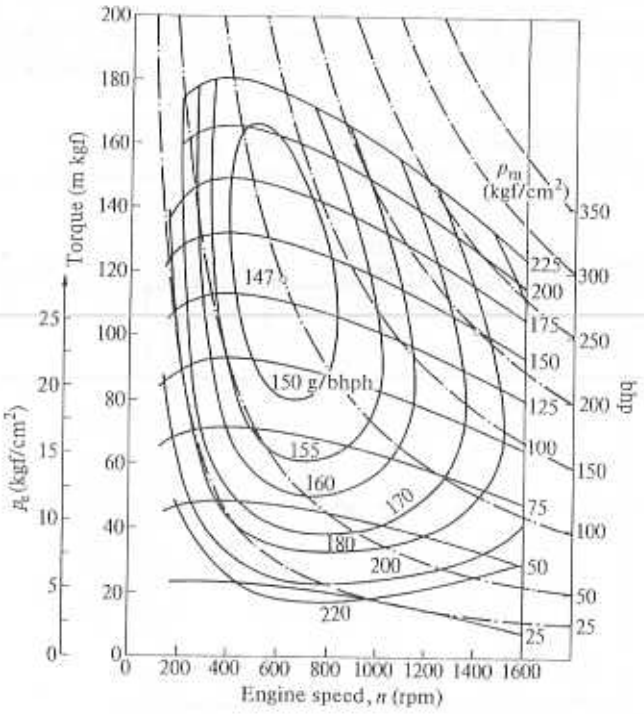


FIG. 9.12. Performance map for 4-cylinder, rhombic-drive Philips Stirling engine for traction applications. Characteristics are calculated for optimized engine with all auxiliaries. (After Meijer 1970b.)

engine for automotive use. These were calculated characteristics with all auxiliaries included.

EMISSION CHARACTERISTICS OF STIRLING ENGINES

Stirling engines fuelled with diesel oil or gasoline have superior exhaust emission characteristics compared with internal combustion engines. This is a particular advantage for the automotive application given the present level of public interest in environmental issues. In the United States various federal and state government agencies have introduced exhaust emission standards. In general, the California vehicle standards are the most demanding.

The constituents of principal interest in an engine exhaust are:

(a) the unburned hydrocarbons (HC)
(b) carbon monoxide (CO)
(c) the oxides of nitrogen (NO_x)

The reason for the favourable emission characteristics of Stirling engines is that combustion takes place continuously at constant temperature and at low, near-atmospheric, pressure in a combustion chamber with hot internal walls. The flow velocities are relatively low. A combustion process in such conditions is likely to result in the virtually complete combustion of the hydrocarbon fuel with minimal values of carbon monoxide.

Nitrogen, usually regarded as an inert gas, does react at high temperatures with oxygen to form nitrogen oxides. The reaction is strongly dependent on temperature and also on the time of association at the high temperature. The quantity of nitrogen oxides formed in combustion is minor and in most cases can be neglected. However, nitrogen oxides have been identified as a contributory factor to the formation of atmospheric pollution. This happens, for example, in the very special circumstances of Los Angeles where high concentrations of automobile exhaust emissions are exposed to strong sunlight in a basin with limited air circulation.

The favourable combustion environs leading to low hydrocarbon and carbon monoxide levels invariably provide the proper circumstances for maximizing the levels of nitrogen oxides. The Stirling engine is no exception. Meijer (1971) gave the data reproduced in Fig. 9.13 for the concentration of nitrogen oxides in a Stirling engine as a function of the inlet air temperature.

Formation of the oxides of nitrogen can be suppressed with the provision of excess air which, of course, depresses the temperature of the combustion space and hence the rate of the nitrogen/oxygen reaction. Recirculation of the engine exhaust can achieve the same result. An indication of the dramatic reduction in nitrogen oxides concentration achieved with partial exhaust gas recirculation was given by Meijer

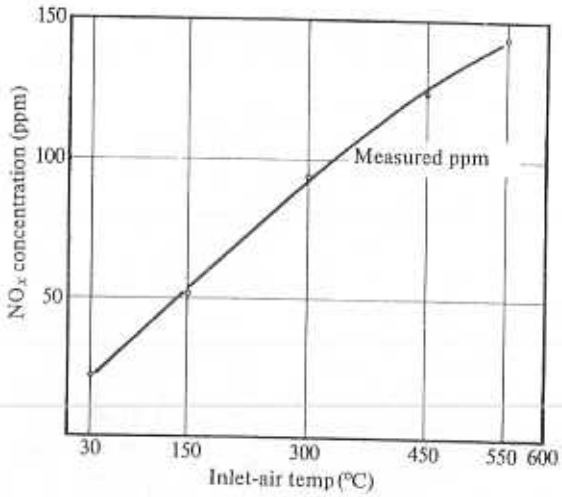


FIG. 9.13. Nitrogen oxides concentration in the exhaust of a Philips Stirling engine as a function of the inlet-air temperature (after Meijer 1970b).

(1970b) in presenting the data reproduced in Fig. 9.14. This shows that the nitrogen oxides concentration of 110 parts per million was measured in the exhaust of a Stirling engine with no recirculation. With one third of the exhaust gas recirculated through the combustion chamber the concentration of nitrogen oxides was reduced to about one third the original value. Little advantage was gained by increasing the recirculation fraction significantly above a third.

Recirculation of the exhaust gas does require some design modifications to allow for the increased mass flow in the combustion space and, of course, additional pump work is required to handle the increased flow. It is of interest to observe that if a significant fraction of the exhaust stream is to be passed back to the incoming air stream the requirements for stream separation in the exhaust/incoming air preheater may be greatly relaxed. The thermal load and hence size of the preheater may also be reduced by the proportion of exhaust recirculated provided the hot exhaust fraction is mixed with the heated air immediately prior to admission to the combustion chamber.

A number of important papers have been published about the emission characteristics of Stirling engines with particular reference to automotive applications. Michels (1972) carried out an extensive experimental simulation of a California Vehicle Standards test for an automobile engine of 128 kW (174 hp) (Philips Type 4-215 double-acting Stirling engine) in a Ford Torino. He found that the emission levels of the Stirling engine

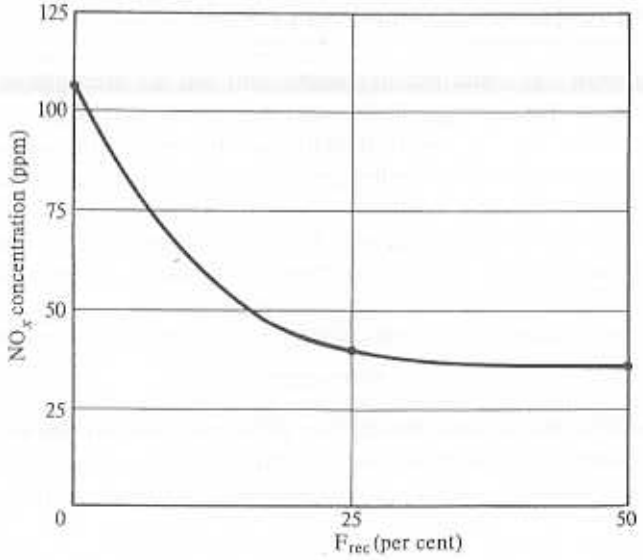


FIG. 9.14. Effect of exhaust gas recirculation on the concentration of oxides of nitrogen in the exhaust of a Philips Stirling engine (after Meijer 1970b).

could be comfortably maintained below the then contemporary regulatory proposals for 1975/76. The comparative values are summarized below:

Emissions	*Michels* (1972) mg/m (g/mile)	1975/76 *Regulatory proposals* mg/m (g/mile)
Hydrocarbons	0.062 (0.1)	0.255 (0.41)
Carbon Monoxide	0.192 (0.31)	2.113 (3.4)
Oxides of Nitrogen	0.109 (0.175)	0.249 (0.4)

Similar data confirming the favourable emission characteristics of Stirling engines with specific data for the United Stirling and MAN/MWM engines was given by Alm *et al.* (1973). Earlier Lienische and Wade (1968) reported on the Stirling engine emission studies carried out at General Motors.

NOISE

Stirling engines make little noise when running. This is perhaps the most advantageous characteristic compared with internal combustion engines. Silent power units have obvious attraction for the armed forces, but low noise is equally attractive in civil applications at all power levels from lawnmower and motorbike engines to large locomotive or heavy traction motors.

The reason why a Stirling engine runs, literally, like a sewing machine is that there are no valves or periodic explosions in the cylinder or the combustion space. Combustion proceeds as a steady continuous process at atmospheric pressure in a well-insulated and, therefore, muffled chamber. The rates of pressure change of working fluid in the cylinder are very low and there are no metal parts in intermittent or violent contact. The motion of the sliding and rotating parts is moderate with no sudden acceleration or deceleration. Furthermore, many designs of engine can be dynamically balanced, either perfectly, or partially, to eliminate or minimize vibration.

Frequently, where noise in a Stirling engine installation is a problem, the source can be identified as some auxiliary or ancillary unit. In many cases the principal noise comes from the fan used in the radiator cooling system. Other cases where noise was considered excessive have been traced to gears.

Postma *et al.* (1973) mention noise tests with the Philips bus which returned sound levels of 68 db, some 10 to 15 db lower than current passenger cars. Alm *et al.* (1973) have given other data equally favourable about the German and Swedish engines from the aspect of noise. Test results given by Schab (1964) in U.S. navy tests of the large 265 kW (360 hp) four-cylinder Philips Stirling engine confirmed their extremely favourable noise characteristics.

STARTING

Free-piston Stirling engines are self-starting, a unique characteristic among heat engines. This alone makes them excellent candidates for those applications where automatic starting is necessary or advantageous. They are, for example, well suited for solar applications with a fixed concentrator. When the sun is at the appropriate level in the morning, the concentrator will heat the engine cylinder enough to start operation which then continues until the late afternoon. Another possibility for automatic starting is in furnace applications where a thermally activated system is required to provide enough power to operate a fan or water pump to circulate the heated fluid.

Stirling engines with crank, rhombic, or swash-plate drive systems are not self-starting. Indeed engine starting is one of the less attractive aspects of these Stirling machines compared with internal combustion engines.

The heater head assembly of a Stirling engine must be hot for the engine to operate. Therefore the hot parts must be preheated before the engine can be started. In an automotive engine using gasoline or diesel fuel this requires a rather complicated starting system, perhaps similar to that shown in Fig. 9.15 (after Meijer 1970b). To start the engine a

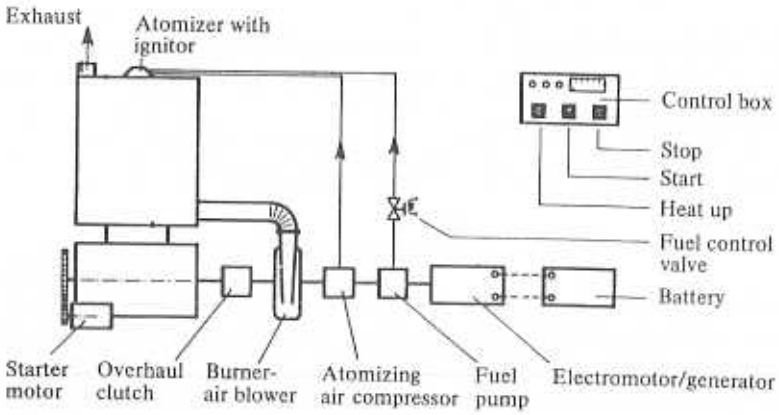


FIG. 9.15. Starting system for a Stirling engine (after Meijer 1970b).

battery-driven motor/generator first drives the fuel pump, an air compressor for fuel atomization, and a fan to provide the burner air to heat the cylinder head. After a suitable interval the engine is turned by a conventional starter motor and, providing the engine is hot, it cannot fail to start.

The preheat interval is a difficulty that may deter consumer acceptance of the engine in automotive applications. Postma *et al.* (1973) have projected that the warm-up time to drive-away conditions will be 15 seconds at ambient temperatures of 21 °C (70 °F) and longer at lower temperatures.

Stirling engines, of course, are not necessarily fuelled by combustion systems using fossil fuels. In the 'electric economy' that lies ahead, automotive Stirling engines may be driven from thermal batteries, charged overnight by electric heating. In this situation no preheat period will be required and instant starting of the Stirling engine will be possible.

ENGINE RESPONSE

Stirling engines can be made to respond rapidly to sudden changes in load and speed. This is necessary for engines in automotive applications where very substantial changes in load and speed occur continuously except for long-haul, motorway traction and locomotives. On the other hand, the ability to maintain constant speed with sudden load change, on or off, is an important characteristic of stationary engines used for power generation, total energy systems, and heat pumps.

An increase or decrease in load on the engine will, naturally, result in a concomitant adjustment of the fuel supply to maintain the heater tube temperatures at the specified temperature. Due to the large thermal mass of the engine hot parts the response of the fuel system to sudden changes

in the load would not be adequate with this single control. Instead, an additional power control system is used which can effect a virtually instantaneous response of the engine to sudden load changes. Control systems for Stirling engines are discussed in more detail in Chapter 10.

SPECIFIC OUTPUT

The size and weight of Stirling engines for a given power output is important in many applications, particularly in the automotive, locomotive, space, and under-water power fields. It is still important, but rather less critical in the stationary generator, heat pump and total energy, and surface marine fields.

Savings in size and weight result from elevation of the pressure of the working fluid and from ingenious design. The rhombic-drive single-action, piston-displacer engines of the 1960s were relatively big, heavy machines. However, the conversion to double-acting Siemens type Stirling engines (see Chapter 13) permitted the construction of machines that were half the size and weight of the rhombic-drive units. This brought Stirling engines to sizes and weights comparable with diesel and gasoline motors. Meijer (1970b) presented the data reproduced in Fig. 9.16 which shows the specific weight as a function of power output for Stirling, diesel, and gasoline engines.

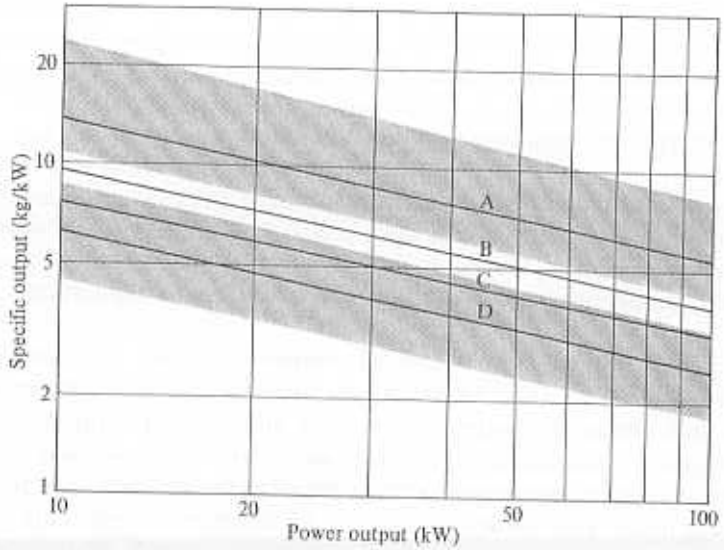


FIG. 9.16. Comparison of the specific output of Stirling, Diesel, and gasoline engines as a function of the power output of the engines. Diesel engines lie in the upper shaded band; gasoline engines in the lower shaded band. The Stirling engines represented by lines A, B, C, and D are machines at different stages of development (after Meijer 1970b).

A comparison of MAN/MWM Stirling traction motors with diesel engines of comparable power was given with considerable discussion by Zacharias (1974). The pictorial comparison given by Zacharias is reproduced in Chapter 14, Fig. 14.5.

United Stirling engines (see Chapter 15) have been installed in a variety of vehicles, cars, and trucks without modification to the engine compartment (Hallare and Rosenqvist 1977).

The principal disadvantage of the Stirling engine as regards size and weight is the large, heavy cooling system that must be provided. The cooling system for a Stirling engine has to handle about twice the load of an internal combustion engine of comparable power because the heat lost to the exhaust must be minimized. This unfortunate condition was well illustrated in the comparative heat balance for a Stirling engine and diesel engine presented by Meijer (1970b) and reproduced in Fig. 9.17.

COST

Economic factors are ultimately dominant in the success or otherwise of new technical developments. At present there is little reliable information about the overall economics of Stirling engines compared with other engines. The use of heat-resistant steels or ceramic components and the relatively sophisticated control systems necessary make it impossible to conceive the production of Stirling engines comparable in cost with conventional internal combustion engines.

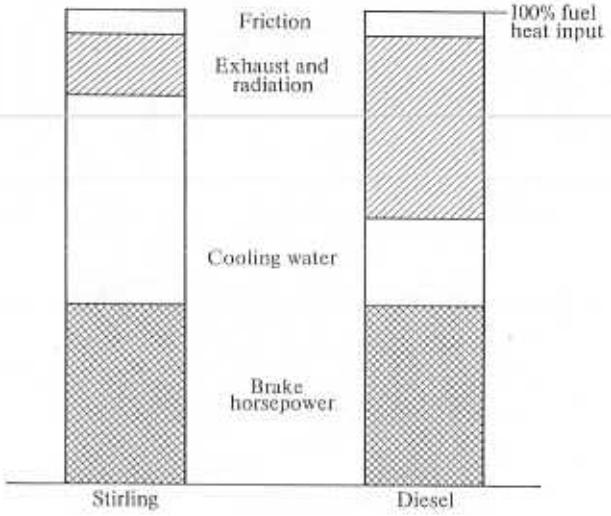


FIG. 9.17. Comparison of the heat balances for Stirling and Diesel engines of the same output (after Meijer 1970b).

A reasonable target might be for the Stirling engine to cost as little as twice the cost of an existing diesel motor. Recall here too that the diesel motor itself costs about twice the price of a gasoline engine. The Stirling engine may have economic advantages in terms of higher efficiency, lower maintenance, and reduced lubricant consumption and so have lower operating costs than internal combustion engines.

Overall economic assessments for Stirling and diesel traction motors have been published by Rosenqvist *et al.* (1977) and their results emerge in favour of Stirling engines. However, this is not surprising for the study was done by United Stirling of Sweden. Such comparisons always depend on a number of conjectural assumptions which may or may not be fulfilled so that various conclusions can be drawn. It is necessary to remember that the diesel engine itself is a moving target. Presently, experimental diesel engines with ceramic components are now running on test beds with thermal efficiencies over 50 per cent. The application of ceramics to Stirling engines will surely improve performance but equal improvements may also result in other types of traction motor.

AIR ENGINES

All the above discussion has been directed to advanced Stirling engines using hydrogen or helium as the working fluid at very high pressures with emphasis on high efficiency and specific output. At the present time there is also substantial interest in a quite different class of Stirling engine, the small (less than 1 kW (1.36 hp)) thermal engine required for a variety of power or pumping applications. Reliability is most important and specific power relatively unimportant from both the weight and volume aspect. Thermal efficiencies of 20 per cent or so would be readily accepted, for this would be twice or more the conversion efficiency of the principal competition, the thermoelectric generator.

With no critical requirement for high efficiency or specific output, the use of air as the working fluid is attractive as one approach to the seal problem. A minor leak of working fluid can be tolerated, for the air can be simply replenished by a small compressor driven by the engine. In many applications very simple engine controls might be used, for the engine would operate at constant speed and load using isotope or liquid petroleum gas as the heat source.

The best Stirling cycle air engines were undoubtedly those developed in the first phase of Philips' work on the Stirling engine. These engines are discussed in some detail in Chapter 12. One well-developed Philips air engine, similar to that shown in Fig. 12.1, was combined with a power generator into the motor generator set shown in Fig. 9.18. Several hundred of these were made in a pre-production series but no commercial applications could be defined and so work on the engine ceased and further work was concentrated on the larger engines. Many of the small

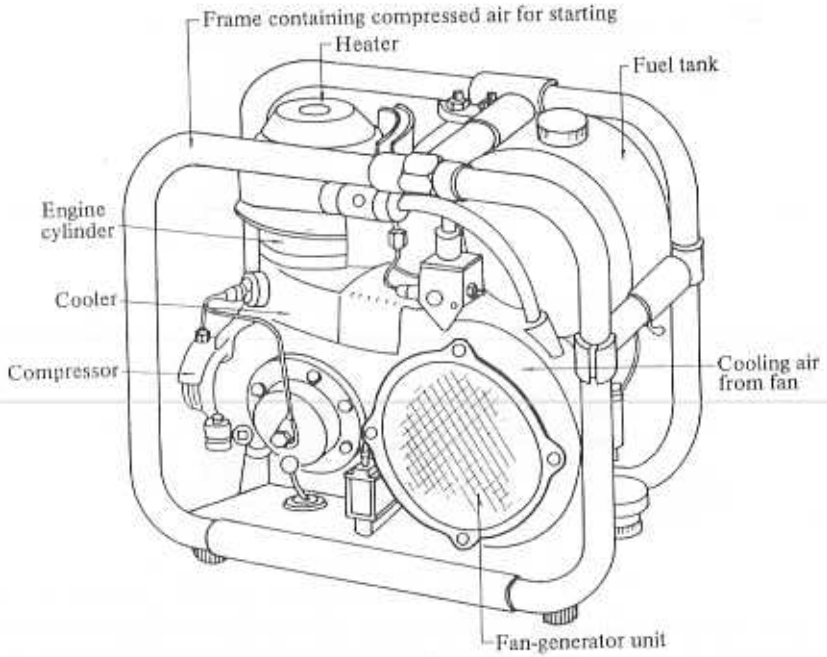


FIG. 9.18. Philips Stirling engine/generator set. Engine runs at 1500 revolutions per minute using air as the working fluid at a mean cycle pressure of 13.5 bar. Electrical output is 200 watts.

engines that had been made were dispersed to educational establishments in Europe. The operating characteristics of this small engine were never published by Philips or any of the recipients.

In the early 1970s the operating characteristics of the engine were carefully measured by Ward (1972) at the University of Bath. This was done as part of a program of development for small Stirling engines for navigation signal beacons, encouraged by the Trinity House Lighthouse Service, London, England, and by Atomic Energy of Canada Ltd. Ward's results are given below in some detail for they have not yet found their way into the open literature and they provide an important point of departure for new development in the area.

The engine used by Ward was removed from a Philips Type 102C motor generator set and modified to operate with water cooling and liquid petroleum gas (LPG) as heat source. A cross-sectional view of the engine is shown in Fig. 9.19†, and particulars of the engine are summarized in Table 9.1.

† This remarkably fine drawing was originally prepared by Dr. A. Organ of Kings College, London, and redrawn in modified form at the University of Calgary.

Table 9.1. Particulars of small Philips Stirling engine (after Ward 1972).

Type-Single cylinder piston/Displacer Engine.	
Cylinder bore	0.056 m
Stroke of piston	0.027 m
Stroke of displacer	0.025 m
Swept volume in expansion space	6.38×10^{-5} m^3
Swept volume in compression space	6.71×10^{-5} m^3
Total dead volume (estimated)	7.97×10^{-5} m^3
Phase Angle: Minimum compression space volume occurs 2.09 radians (120°) after minimum expansion space volume.	
Regenerator: Fine steel wool wound in annular space perpendicular to gas flow.	

The engine was incorporated into a test unit as shown in Fig. 9.20. Brake output power was measured by a swinging field dynamometer. LPG and air flows were measured by rotameter and engine cylinder head temperatures by chromel-alumel thermocouples brazed to the cylinder head.

The fuel used in these tests was the commercial mixture of butane (C_4H_{10}) and propane (C_3H_8) known as 'Calor gas'. The actual mixture ratio was unknown and probably varied but the typical mixture is 90 per cent propane and 10 per cent butane by weight with a lower calorific value of 46 500 kJ/kg (19 988 Btu/lb$_m$). For all series of tests the air flow was adjusted to obtain the maximum cylinder-head temperature for a given fuel flow. This ensured optimum utilization of the fuel. The measured air/fuel ratios varied from a minimum of 20.7 at maximum speed

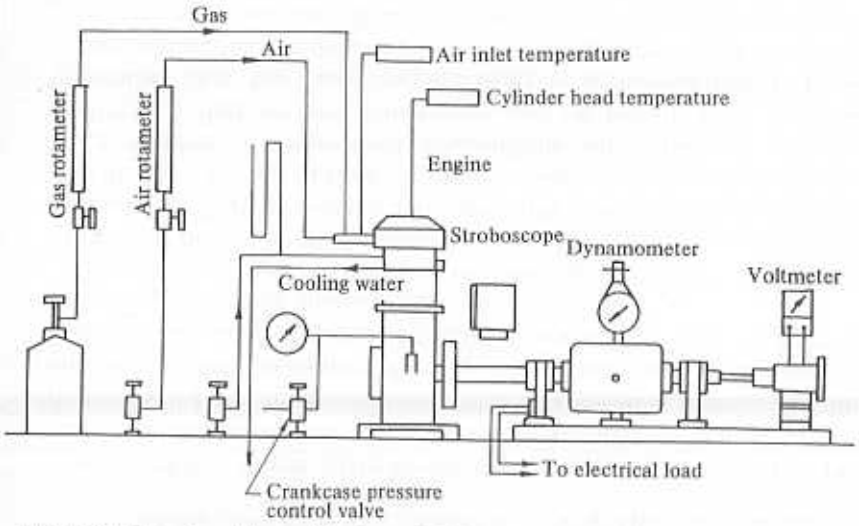


FIG. 9.20. Experimental assembly for small Stirling air engine tests (after Ward 1972).

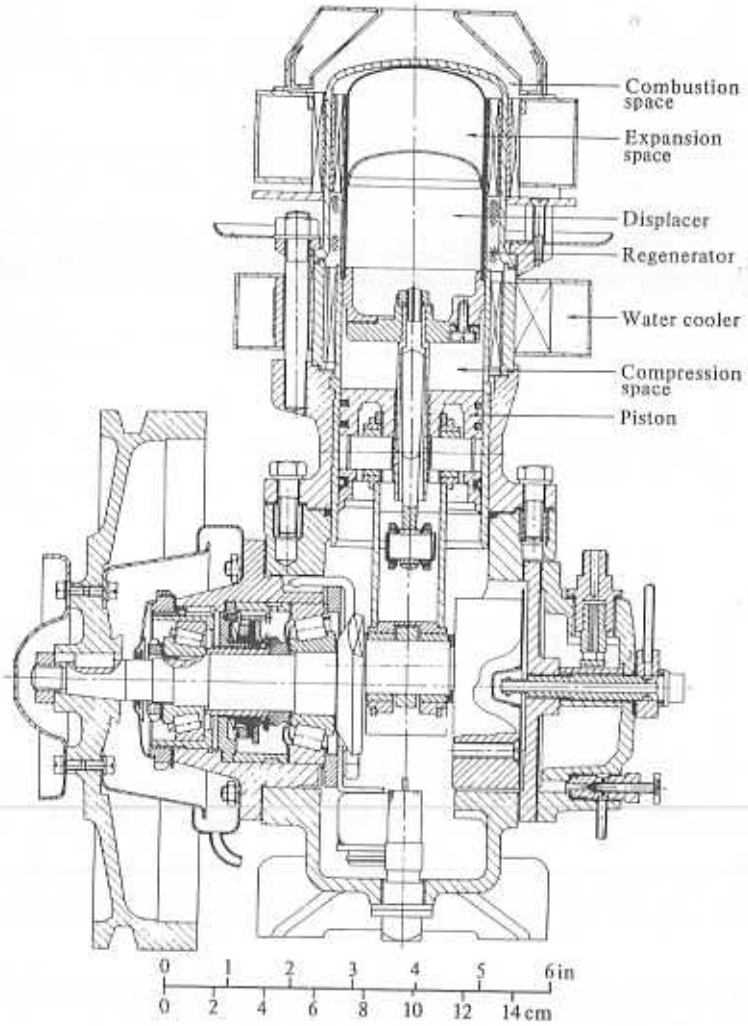


FIG. 9.19. Cross-sectional diagram of Philips type 102C Stirling engine modified for water cooling (after Organ)

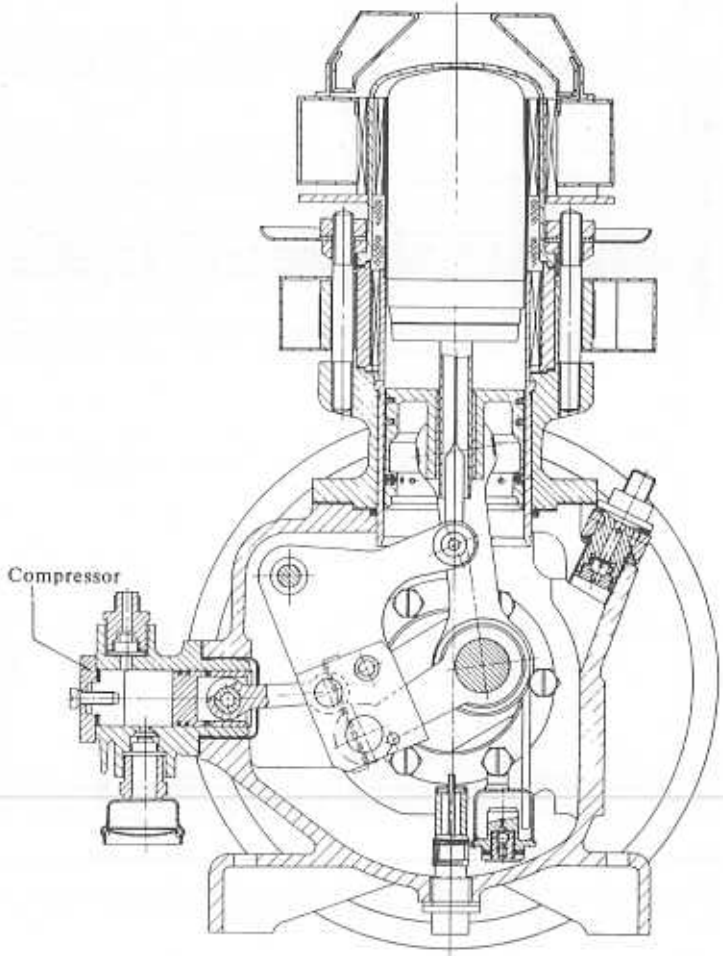


Table 9.2. Summary of reduced engine performance data.

Cylinder head temp °C	Operating pressure bar	Engine torque mN	Power watts	Fuel rate g/min	Brake specific fuel consumption kg/kW hr
900	12.41	2.47	465.5	7.7	0.99
	11.03	2.35	442.9	7.7	1.04
	9.66	2.35	442.9	7.8	1.06
	8.28	2.06	388.2	7.3	1.13
	6.90	1.72	324.2	6.5	1.20
	5.52	1.41	265.7	6.3	1.42
	4.14	0.91	171.5	6.0	2.10
800	12.41	1.94	365.6	7.0	1.15
	11.03	2.00	376.9	7.4	1.18
	9.66	1.86	350.5	6.9	1.18
	8.28	1.69	318.5	6.7	1.26
	6.90	1.43	269.5	6.0	1.34
	5.52	1.14	214.8	5.6	1.56
	4.14	0.73	137.6	5.4	2.35
700	12.41	1.22	229.9	5.9	1.54
	11.03	1.32	248.8	6.4	1.54
	9.66	1.25	235.6	6.0	1.53
	8.28	1.18	222.4	5.8	1.56
	6.90	1.02	292.2	5.3	1.65
	5.52	0.79	148.9	4.3	1.73
	4.14	0.49	92.3	4.3	2.80
600	12.41	0.20	37.7	5.4	8.59
	11.03	0.76	143.2	5.5	2.30
	9.66	0.82	154.5	5.3	2.06
	8.28	0.57	107.4	4.3	2.40
	6.90	0.54	101.8	3.7	2.18
	5.52	0.39	73.5	3.2	2.61
	4.14	0.19	35.8	2.8	4.69

(a) Engine speed 1800 RPM

Cylinder head temp °C	Operating pressure bar	Engine torque mN	Power watts	Fuel rate g/min	Brake specific fuel consumption kg/kW hr
900	12.41	2.86	479.1	7.7	0.96
	11.03	2.65	443.9	7.4	1.00
	9.66	2.72	455.7	7.0	0.92
	8.28	2.32	388.6	6.8	1.05
	6.90	2.09	350.1	6.2	1.06
	5.52	1.62	271.4	5.6	1.24
	4.14	1.14	191.0	5.2	1.63
800	12.41	2.34	392.0	6.7	1.03
	11.03	2.55	393.7	7.1	1.08
	9.66	2.13	356.8	6.4	1.08
	8.28	1.97	330.0	6.0	1.09
	6.90	1.65	276.4	5.5	1.19
	5.52	1.29	216.1	5.1	1.42
	4.14	0.91	152.4	4.8	1.89
700	12.41	1.50	251.3	6.1	1.46
	11.03	1.67	279.8	5.5	1.18
	9.66	1.52	254.6	5.3	1.25
	8.28	1.50	251.3	5.0	1.19
	6.90	1.09	182.6	4.6	1.51
	5.52	0.96	160.8	4.1	1.53
	4.14	0.57	95.5	3.7	2.32
600	12.41	0.85	142.4	4.7	1.98
	11.03	0.78	130.7	4.4	2.02
	9.66	0.85	142.4	4.3	1.81
	8.28	0.89	149.1	3.8	1.53
	6.90	0.74	124.0	3.7	1.79
	5.52	0.62	103.9	3.2	1.85
	4.14	0.30	50.3	2.9	3.46

(b) Engine speed 1600 RPM

Cylinder head temp °C	Operating pressure load	Engine torque mN	Power watts	Fuel rate g/min	Brake specific fuel consumption kg/kW hr
900	12.41	3.19	467.6	7.2	0.92
	11.03	3.16	463.2	6.9	0.89
	9.66	2.77	406.0	6.2	0.92
	8.28	2.51	367.9	5.7	0.93
	6.90	2.24	328.3	5.5	1.00
	5.52	1.69	247.7	5.3	1.28
	4.14	1.30	190.6	5.2	1.64
800	12.41	2.87	420.7	6.9	0.98
	11.03	2.49	365.0	6.2	1.02
	9.66	2.38	348.9	5.5	0.94
	8.28	2.26	331.3	5.1	0.92
	6.90	1.94	285.0	4.8	1.02
	5.52	1.56	228.7	4.6	1.21
	4.14	1.02	149.5	4.1	1.64
700	12.41	1.94	284.4	5.5	1.16
	11.03	1.97	288.8	4.9	1.02
	9.66	1.90	278.5	4.8	1.03
	8.28	1.69	247.7	4.4	1.07
	6.90	1.47	215.5	4.2	1.17
	5.52	1.14	167.1	3.8	1.36
	4.14	0.79	115.8	3.2	1.66
600	12.41	1.13	165.6	4.2	1.52
	11.03	1.36	199.3	4.2	1.26
	9.66	1.29	189.1	3.8	1.21
	8.28	1.13	165.6	3.8	1.38
	6.90	1.02	149.5	3.3	1.32
	5.52	0.80	117.3	3.0	1.53
	4.414	0.53	77.7	2.6	2.01

(c) Engine speed 1400 RPM

Cylinder head temp °C	Operating pressure bar	Engine torque mN	Power watts	Fuel rate g/min	Brake specific fuel consumption kg/kW hr
900	12.41	3.64	457.3	6.9	0.90
	11.03	3.56	447.3	6.7	0.90
	9.66	3.21	403.3	6.0	0.89
	8.28	2.74	344.3	5.8	1.01
	6.90	2.38	299.0	5.4	1.08
	5.52	1.91	240.0	5.2	1.30
	4.14	1.36	170.9	4.8	1.68
800	12.41	3.11	390.7	5.9	0.91
	11.03	2.89	363.1	5.4	0.89
	9.66	2.68	336.7	5.2	0.93
	8.28	2.42	304.0	5.1	1.01
	6.90	2.03	255.0	4.5	1.06
	5.52	1.59	199.8	4.6	1.38
	4.14	1.18	148.5	4.0	1.62
700	12.41	2.32	291.5	5.1	1.05
	11.03	2.24	281.4	4.7	1.00
	9.66	2.24	281.4	4.1	0.87
	8.28	2.03	255.0	4.0	0.94
	6.90	1.72	216.0	3.9	1.08
	5.52	1.39	174.6	3.4	1.17
	4.14	1.00	125.6	3.1	1.48
600	12.41	1.52	191.0	4.1	1.29
	11.03	1.51	190.0	3.7	1.20
	9.66	1.43	179.7	3.4	1.14
	8.28	1.30	163.3	3.1	1.14
	6.90	1.10	138.2	3.0	1.30
	5.52	0.82	103.0	2.7	1.57
	4.14	0.57	71.6	2.5	2.10

(d) Engine speed 1200 RPM

and power output at high cylinder-head temperatures to a maximum of 32.2 at the minimum power output at low cylinder-head temperatures.

The reduced engine performance data obtained by Ward are summarized in Table 9.2. The data are arranged in four sub-tables for constant speeds of 1800, 1600, 1400 and 1200 revolutions per minute. In all cases measurements were made at cylinder-head temperatures of 900, 800, 700 and 600 °C (1652, 1472, 1292 and 1112 °F), with mean cycle pressures in the range 0.4 to 1.2 MN/m^2 (4 to 12 bar).

The results are presented graphically in Figs. 9.21 and 9.22. Each figure shows the brake power and brake specific fuel consumption at constant engine speed or constant cylinder-head temperature as a function of mean pressure in one case or engine speed in the other.

Ward carried out a series of interesting 'motoring' tests with the engine. A motoring test involves driving the engine with the dynamometer acting as a motor and measuring the power input to the engine. Motoring tests are normally carried out in evaluation of internal combustion engines to establish the friction horsepower. However, if a Stirling engine is driven that way it acts like a refrigerator and so the input work is made up partly of the mechanical friction losses and partly of the cycle work to accomplish the refrigeration effect. Ward overcame this problem by including a large volume (compressed gas bottle) in the engine dead space. This ensured there was no significant cyclic pressure variation during operation of the engine and therefore no refrigeration effect was created.

He carried out motoring tests at different speeds and different mean pressures with the engine always at ambient temperature conditions and obtained the results presented in Fig. 9.23. This shows the motoring

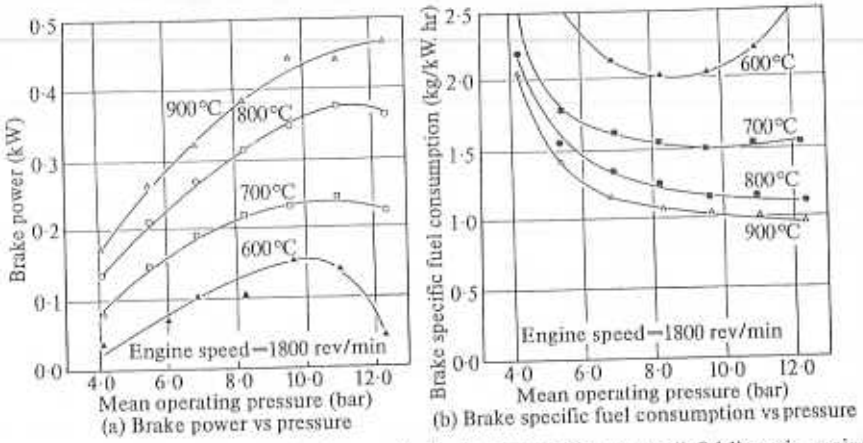


FIG. 9.21. Brake power and brake specific fuel consumption of small Stirling air engine as a function of mean pressure at four different cylinder head temperatures and a constant engine speed of 1800 revolutions per minute.

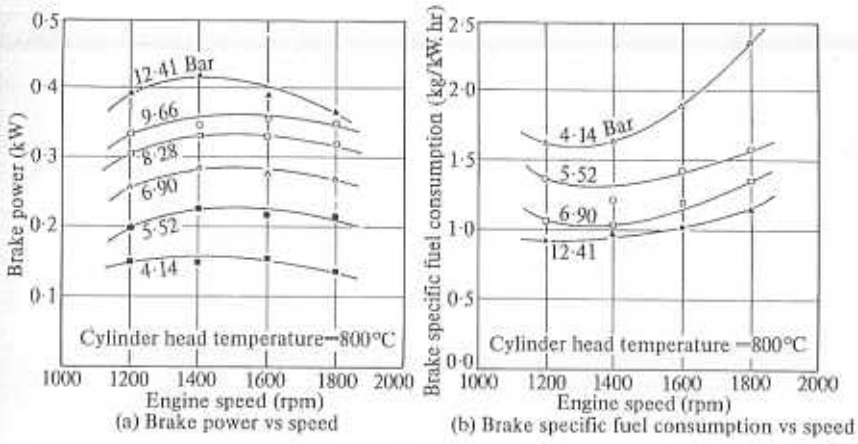


FIG. 9.22. Brake power and brake specific fuel consumption of small Stirling air engine as a function of engine speed at different mean operating pressures and a constant cylinder-head temperature of 800 °C.

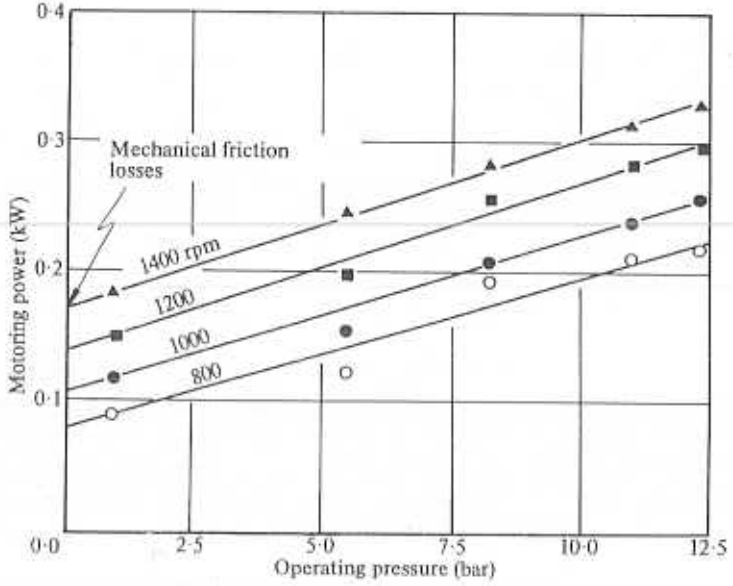
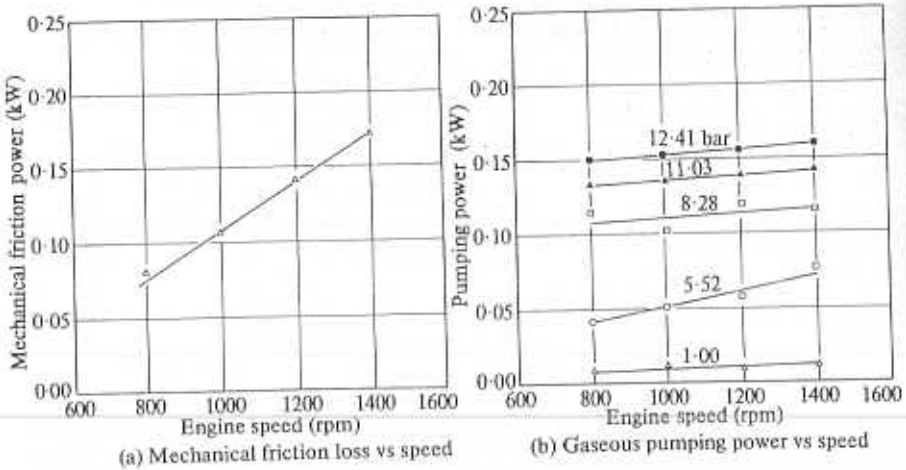


FIG. 9.23. Input motoring power to small Stirling air engine as a function of mean operating pressure at four different speeds and with engine cylinder at ambient temperatures.



(a) Mechanical friction loss vs speed

(b) Gaseous pumping power vs speed

FIG. 9.24. Mechanical friction and gaseous pumping power of small Stirling air engine as a function of engine speed and mean pressure.

power required to drive the engine as a function of operating pressure at four different speeds.

The motoring power shown in Fig. 9.23 is the combined total of the mechanical friction and the gaseous pumping work necessary to overcome the aerodynamic-friction and the positive-displacement work.

A possible interpretation of these results is as follows. For each constant speed condition the motoring power curve (Fig. 9.23) has been extrapolated to the zero mean pressure axis. The motoring power at zero pressure was then assumed to be the mechanical-friction work of the engine. This was plotted in Fig. 9.24(a) and was found to be a strong function of engine speed. The mechanical-friction work was assumed to be independent of the mean pressure, so the balance between total motoring power and mechanical friction was attributed to the gaseous pumping work. This balance is plotted in Fig. 9.24(b) and can be seen to be a function mainly of pressure (and hence density) and, to a lesser extent, engine speed.

By reference to Fig. 9.21 it can be seen that the brake output at 1400 revolutions per minute with an operating pressure of 1.2 MN/m^2 (12.41 bar) was 0.42 kW (0.57 hp). At a similar speed and pressure condition, the motoring power required, shown in Fig. 9.23, was 0.34 kW (0.46 hp). Therefore, the indicated power of the engine was approximately 0.42 + 0.34 = 0.76 kW (1.03 hp) and the 'mechanical' efficiency was $\frac{0.42}{0.76} = 0.55$. This, of course, includes the gaseous pumping work which is much more significant in Stirling engines than internal combustion engines.

More complete details of the engine, test procedure, and the results obtained may be found in the thesis by Ward (1972).

FREE-PISTON STIRLING ENGINES

None of the above material is directly applicable to free piston Stirling engines. Many trend effects are common, but free piston Stirling engines are emerging as a whole new genus of engines with their own special peculiarities, advantages, and problems. Readers with particular interest in free-piston engines are directed to Chapter 11.

10 CONTROL SYSTEMS FOR STIRLING ENGINES

INTRODUCTION

CONTROL systems are necessary to regulate the power output (torque) and speed of a Stirling engine. Sometimes the engine speed is held constant whatever the load, i.e. stationary constant-speed fixed-frequency electric-power generators. Sometimes, as in automotive applications, wide ranges of both speed and load are encountered.

A rapid response of the engine to sudden changes in load is important in many applications. The efficiency of the engine is also important. Engines operate most of the time at part-load, producing only a fraction of the maximum load of which they are capable. In many cases, therefore, the efficiency at part load is of more concern than the efficiency at maximum rated power.

By way of example, consider the hypothetical load/speed characteristics shown in Fig. 10.1. Diagram (a), representative of an electric-power generator, a pump, or a fan, shows the input power requirements as a function of speed for different levels of voltage or pressure. Diagram (b), representative of a Stirling engine, shows the power output as a function of speed at different levels of the mean pressure in the engine. If the engine be coupled directly to the load the characteristics will be superimposed as in diagram (c).

The engine output will be entirely absorbed to drive the load and the engine will thus assume a speed at which the output power is identical to the input required, say point A in Fig. 10.1(c). Increase in the engine mean cycle pressure will increase the power output from the engine. This will cause the engine to accelerate to a higher speed thereby increasing the load power requirement until a new balance is established at point B for example.

Engines in stationary applications are normally regulated to operate at constant speed and so would operate along the constant speed line C–D.

Automotive engines are not coupled directly to the driving wheels of the vehicle. A transmission system is interposed which invariably includes a gearbox or other device for changing the ratio of engine shaft speed to vehicle road wheel speed. The input power/speed requirement of the load may then be represented as shown in Fig. 10.2(a). Note that the horizontal axis is the speed of the *input shaft to the gearbox* (output shaft of the engine). Changing the gear ratio between input and output shafts of the gearbox causes step changes in the load power requirements.

The engine output and load input power/speed characteristics may be

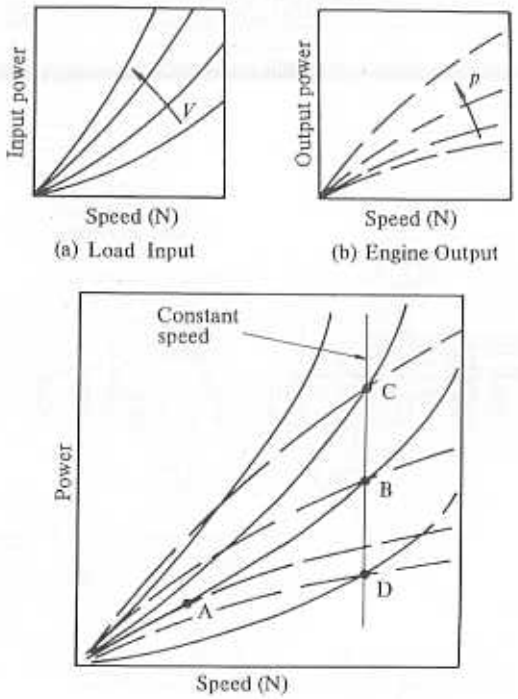


FIG. 10.1. Hypothetical power/speed characteristics for a Stirling engine and a load driven by the engine. Diagram (a) shows the power requirement as a function of speed for an electric power generator, hydraulic pump, or pneumatic fan as a function of different levels of voltage or pressure. Diagram (b) shows the engine power output as a function of speed at different levels of the mean pressure of the working fluid. Diagram (c) is the superimposed characteristics of the power requirements of the load and the power output of the engine. The operating speed of the engine occurs at the coincidence of power levels.

superimposed as in Fig. 10.2(b). The resultant map may be used to determine the appropriate engine operating conditions and gearbox ratios for the optimum vehicle handling characteristics.

Maximum temperature

The thermal efficiency of all heat engines depends principally on the maximum temperature achieved in the cycle. In the Stirling engine the maximum temperature of the working fluid occurs in the heater tubes and is made as high as the material of the heater tubes can withstand.

To maintain maximum engine efficiency, the temperature of the heater tubes should be kept constant at the highest possible value. As the load increases or decreases, the amount of fuel supplied to the burner should be proportionally adjusted so as to keep the heater tubes at a temperature just below the limiting value for safe operation.

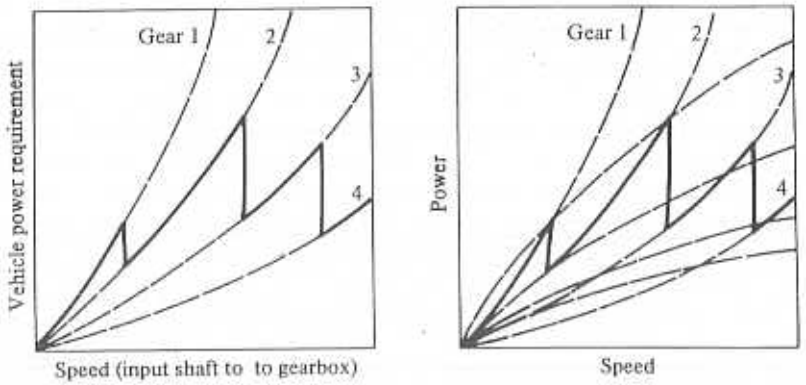


FIG. 10.2. Power/speed characteristics for an automotive Stirling engine. Diagram (a) shows the vehicle input power requirements as a function of the speed of the gearbox input shaft at three gearbox ratios. Diagram (b) shows superimposed vehicle power requirements and engine power output characteristics. These may be used to determine optimum operating conditions for the engine and vehicle transmission systems.

Unfortunately, adjustment of the fuel alone is not adequate. The ratio of air to fuel has an important effect on the efficiency of the combustion process and also on the emission products in the engine exhaust. Therefore, coordinated control of both the air and fuel supplied to the combustion chamber is necessary to ensure high efficiency with low pollution exhaust.

Engine response

The response of the engine to changes in the combustion conditions is slow. The change becomes manifest in the engine only as it affects the temperature of the heater tubes and thus the rate of heat transfer to the working fluid. For some applications this slow response is adequate, particularly where load and speed conditions are more or less constant, such as small power generators. For most applications, however, and in particular for the automotive application, a more positive and rapid response to changes in the load are necessary. A second control system is therefore required. This second system has come to be known as the *power control* system although it is in fact the *torque control* system.

Temperature and power control systems used on contemporary Stirling engines are reviewed below and other possibilities that have not yet found application are discussed.

TEMPERATURE CONTROL SYSTEMS

The temperature control systems used in Stirling engines are all basically similar. The principle is that the temperature of the heater tubes (or other critical part) is to be maintained constant at all operating conditions. The temperature is monitored by a thermocouple which generates a signal for some control device regulating the air flow, fuel flow or both to maintain the temperature constant.

The essential features of the United Stirling temperature control system are shown in Fig. 10.3. Hallare and Rosenqvist (1977) describe the air fuel control systems as follows:

'The temperature of the heater tube is measured by a thermocouple 1. The signal of the thermocouple is amplified and converted in the electronic control unit 2 to a signal controlling the position of the air throttle 3. Thus the right amount of air is delivered to the combustor via the burner air blower 4. In a slightly modified Bosch k-Jetronic unit, a sensor plate 6 installed inside a conical air passage provides a position indication of air flow rate.

The fuel from the tank 5 passes an electric pump 7 and a filter 8. The fuel pressure is held constant by a relief valve 9. The position of the sensor plate controls, via a plunger 10, the amount by which a fuel metering port is opened.

The differential pressure across the metering port is maintained at a constant value by a valve 11 so that the fuel flow to the atomizer depends only upon the amount the port is opened.

The air-fuel ratio depends upon the hydraulic counter pressure controlled by a pressure regulating valve 12. Adjustment of the ratio over the load range can be achieved by a modification of the shape of the conical air passage'.

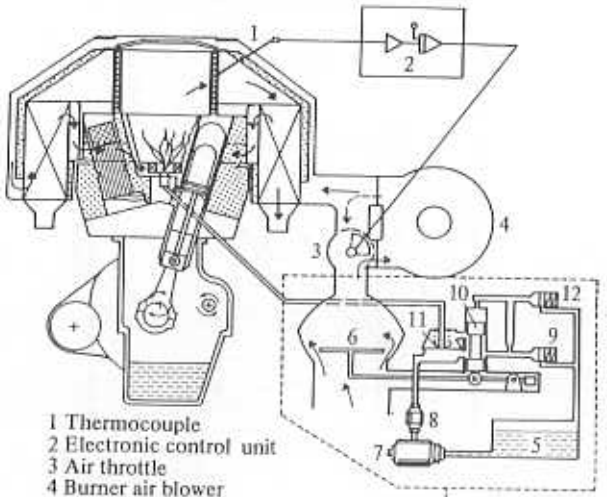


FIG. 10.3. Schematic diagram of United Stirling fuel/air control system (after Hallare and Rosenqvist 1977).

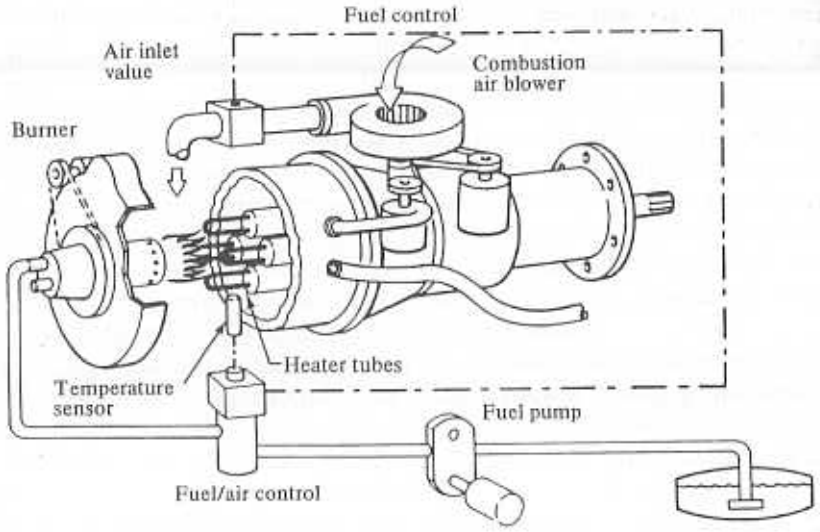


FIG. 10.4. Schematic diagram of Philips/Ford fuel/air control system (after Postma *et al.* 1973).

A system essentially similar was described briefly by Postma *et al.* (1973) for the Philips/Ford Type D.A. 4-215 engine. The system is shown schematically in Fig. 10.4 and was described by Postma as follows:

'The Stirling engine fuel control is designed so that the engine heater tubes are operated at a constant temperature of 1470 °F (799 °C). A temperature sensor mounted on one of the heater tubes is the primary control device in the air and fuel control circuit. The combustion blower is driven directly from the engine. The amount of air supplied is regulated by the temperature sensor through a throttle valve which is interconnected with the fuel/air control to maintain a constant A/F ratio (30 per cent excess air)'.

Earlier Neelen *et al.* (1971) briefly described the fuel control system used on the four-cylinder rhombic-drive Philips Type 4-235 traction motor developed for installation in buses.

No description of the fuel/air control system used on MAN/MWM engines was found in any of the references consulted but it is thought to be similar to those discussed above.

POWER CONTROL SYSTEMS

Pressure level modulation

Variation in the mean pressure level of the working fluid is the most widely used and best-known control system for power regulation in Stirling engines. It was used to some extent on the more sophisticated air

engines of the nineteenth century and was early adopted as the principal power control system for Philips engines (van Weenan 1947).

Constant speed system (Philips)

Meijer (1959a) gave an excellent description of the system used for power regulation on the first rhombic-drive engines using hydrogen or helium as the working fluid. The objective was to maintain a constant speed of operation of the engine and was achieved by increasing or decreasing the pressure level of the working fluid in the cylinder as the engine speed changed, causing the engine to accelerate or decelerate to the specified speed.

The system is shown schematically in Fig. 10.5. It operated as follows. The governor (1), driven by the engine shaft ensured that, at the nominal speed of operation, a certain oil pressure was maintained in pipes (2) and (3). If the load torque increased, causing the engine to slow down, the governor raised the oil pressure in the pipes with the result that valve (4) in the feed device (5) opened. High pressure hydrogen then passed from the storage reservoir (6) through the feed device (5) and valve (4) to the engine cylinder (8) via the non-return valve (7).

Increase in the pressure level of the working fluid caused an increase in the power output from the engine and consequently an acceleration in speed. Injection of additional working fluid to the engine continued until the engine speed had climbed back to the original value. At the set speed

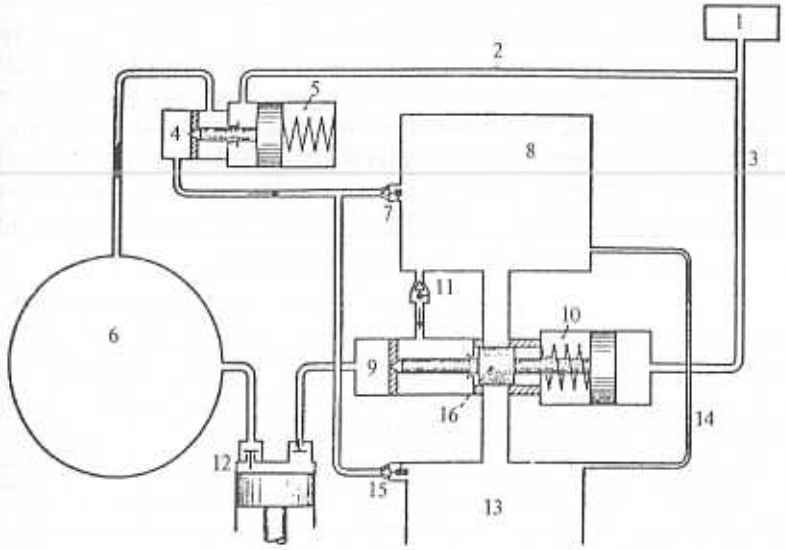


FIG. 10.5. Schematic diagram for power control in a Stirling engine by variation in the pressure level of the working fluid (after Meijer 1959a).

the oil pressure in pipe (2) controlled by governor (1) was sufficiently low for the valve (4) to close and admission of additional gas was therefore curtailed.

If a decrease in the load torque occurred it resulted in an increase in the speed of the engine. As a consequence the governor decreased the oil pressure in pipes (2) and (3) with the result that valve (9) of the regulator (10) opened. This permitted gas to flow from the engine cylinder via the non-return valve (11) through valve (9) to the auxiliary compressor (12) and hence back to the storage reservoir (6). If the maximum cycle pressure exceeded the pressure of the storage vessel sufficient gas would flow intermittently from (8) to (6). A sharp decrease in the load however would require the auxiliary compressor to function.

Release of gas from the cylinder reduced the power produced thereby resulting in a slow-down of the engine. This continued until the power output had declined to a compatible level with the new value of load torque at which time valve (9) closed and further release of gas from the engine ceased.

The need to compress the fluid back to the higher pressure of the storage reservoir restrained the flow of gas from the cylinder to the amount that could be handled by the compressor. As a consequence the response of the engine to sudden decreases in the load was substantially longer than the response to sudden increases in load.

Short-circuiting (Philips)

To improve the engine response to sudden load decrease a supplementary power control system was incorporated in the speed regulator. The new control was remarkably simple. Two or more spaces in the engine were interconnected so as to 'spoil' the pressure characteristic of the individual spaces operating separately. This supplementary system called 'loss regulation' or 'short-circuiting' caused both a change in the phasing and a reduction in the amplitude of the pressure variation in the engine cylinder and effected a reduction in power output.

In the unit described by Meijer, the two spaces interconnected were the single cylinder and the 'buffer space' below the piston. In a rhombic-drive engine, the space below the piston (the buffer space) is normally charged with working fluid to the same mean pressure as the engine. This relieves the gas dynamic forces acting on the piston and adjusts the hermetic seal problem to one of sealing the small diameter piston rod rather than the large diameter piston. The buffer space experiences a cyclic pressure variation that is approximately inverse to the cylinder pressure variation.

In addition to opening and closing valve (9) as described above the regulator (10) also actuated a slide throttle (16) which offered the gas a direct connection between the cylinder and the buffer space. This was

more or less equivalent to a leak past the piston and resulted in a virtually instantaneous reduction in the amplitude and a change in the phasing of the pressure variation and hence in the power output of the engine. The system resulted in a decrease in engine efficiency and for this reason was termed by Meijer (1959a) 'loss-regulation' but more recently has come to be called 'short-circuit control'. It was a useful device for it made the engine response virtually instantaneous to sudden changes in load. Moreover, it permitted the use of a very small auxiliary compressor to pump the gas back to the storage compartment.

Variable speed system (Philips)

A similar control system was described by Neelen *et al.* (1971) for the four-cylinder rhombic-drive Philips Type 4-235 traction motor. In this case the engine must operate at variable speeds as well as variable loads.

A simplified diagram of the system described by Neelen *et al.* (1971) is shown in Fig. 10.6. The valve *S* corresponds to the valve 4 in Fig. 10.5 and the valve *D* to valve 9. Similarly the short-circuit valve *SC* corresponds to the bypass valve 16. The complete functional diagram of the system described by Neelen *et al.* for an automotive engine is shown in Fig. 10.7.

Depression of the accelerator pedal caused the supply valve *S* to open and for the valves *D* and *SC* to remain closed. The cylinder pressure was

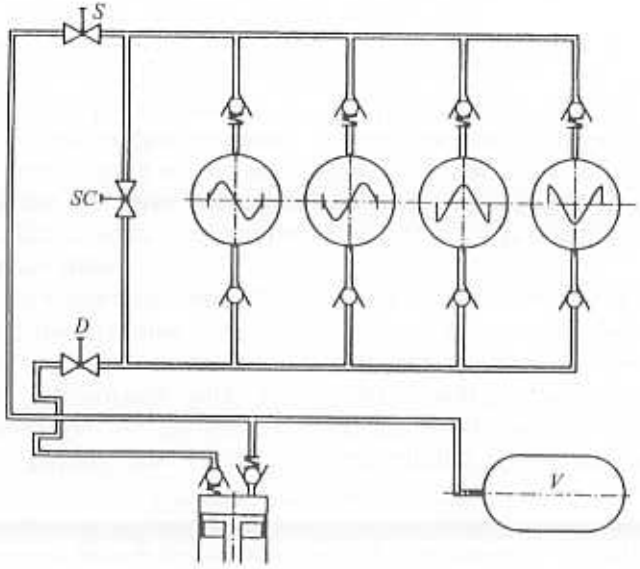


FIG. 10.6. Simplified diagram for power control in a multiple-cylinder Stirling engine by variation in the pressure level of the working fluid (after Neelen *et al.* 1971).

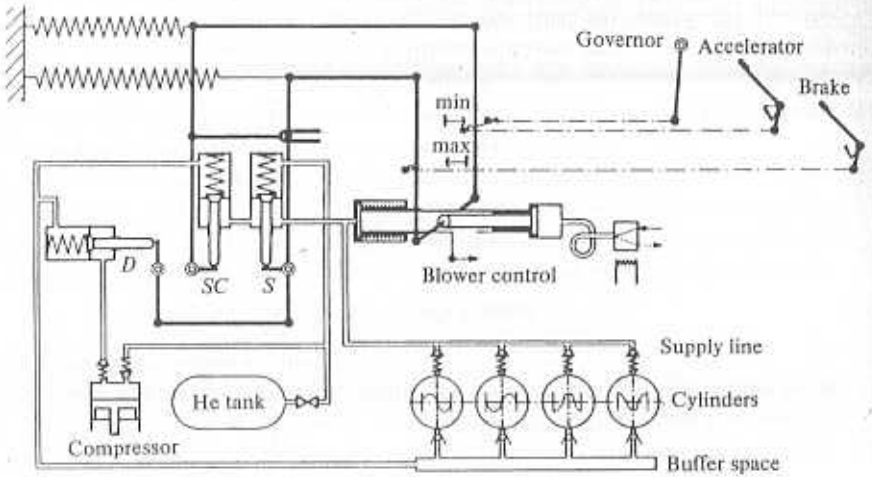


FIG. 10.7. Functional diagram for torque (pressure) control for Philips Type 4-235 four-cylinder rhombic-drive Stirling engine.

connected in a closed feed back loop so that the valve *S* closed when the cylinder pressure attained a value proportional to the displacement of the accelerator pedal. Release of the accelerator pedal opened valve *D* and working fluid passed from the engine back to the storage reservoir.

Depression of the brake pedal caused the dump valve *D* to be closed and the supply valve *S* and the short-circuit valve *SC* to be opened. The degree of opening of the *SC* valve was determined by the position of the accelerator pedal.

The 'loss regulation' feature of this system was so effective that the engine could be used for braking purposes. Neelen *et al.* (1971) gave the work diagrams reproduced in Fig. 10.8. These show the output work diagram of the engine under full load conditions and the corresponding diagrams with the 'loss-regulation' short-circuit valve open. When partly open the engine produces some power but at a reduced level. When the short-circuit valve was fully open the engine absorbed power and could therefore be used in braking. Neelen *et al.* (1971) forecast that a maximum braking torque of 60 per cent of the rated engine torque would be possible. In one test an engine output of 244 N m (176 ft lb) changed to a braking torque of 140 N m (100 ft lb).

Other test results achieved a response time from no load to full load within 0.3 seconds. With the short-circuit valve operating a similar response time (0.3 seconds) was obtained for sudden release of the load. Without the short-circuit valve operating it took the compressor 30 seconds to reduce the pressure from the full load to the zero load value.

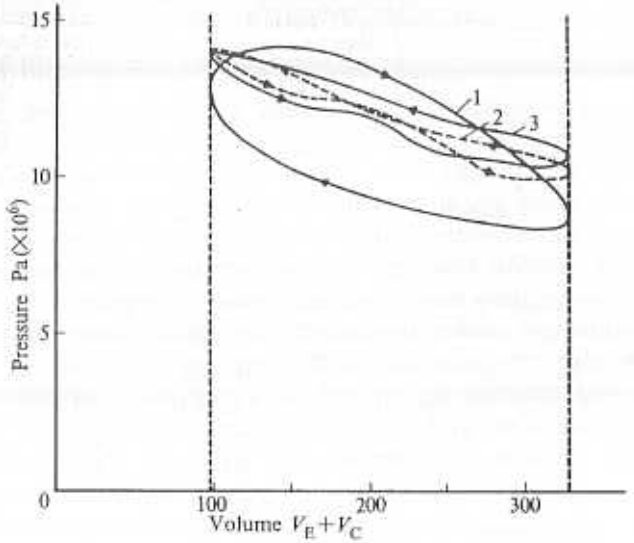


FIG. 10.8. Work diagrams for Philips Type 4-235 four-cylinder rhombic-drive Stirling engine illustrating the control achieved by the loss-regulation short-circuit feature.

The above system, developed for multiple-cylinder rhombic-drive engines, has been carried over to the double-acting Siemens engines now universally in favour, and van Beukering and Fokker (1973) have briefly discussed relevant control systems. Postma *et al.* (1973) have indicated that the power control system for the Philips/Ford type D.A. 4-215 swashplate engine is essentially identical to the system described above. A similar system was incorporated in the conceptual study for a small engine carried out by Philips/Ford for the Department of Energy (Kitzner 1977a).

Variable speed system (United Stirling)

A schematic diagram of the power control system used on United Stirling engines is reproduced in Fig. 10.9. It can be recognized as virtually identical to the above system. To increase power, the control valve is moved to the right so that gas flows directly from the reservoir to the engine. Hallare and Rosenqvist (1977), in a discussion of the control system, have revealed that a timed supply system is used which admits additional hydrogen to the engine cylinders only when the cycle pressure is near the maximum value. Gas flow into the cylinders without a timed system resulted in an undesirable torque drop during increase in the pressure.

Decrease in power is accomplished by moving the control slide to the left, thus releasing fluid from the cylinder back to the gas reservoir and

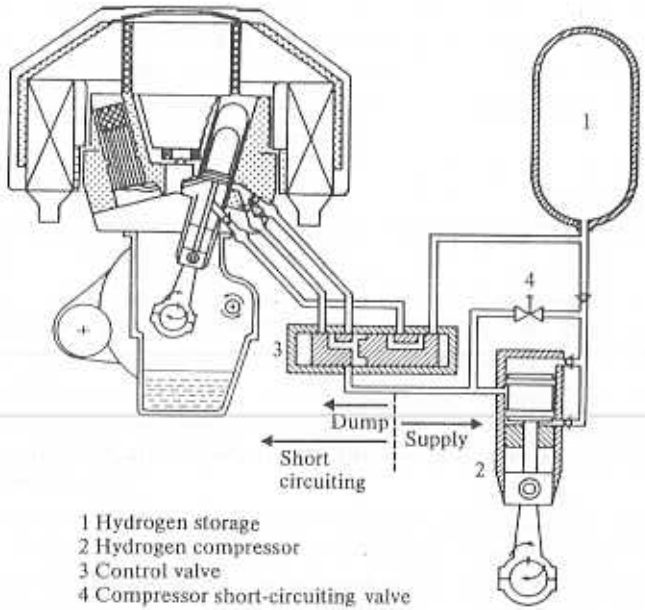


FIG. 10.9. Schematic diagram of power control system on United Stirling engines.

also short-circuiting the various cylinders for rapid response. The hydrogen auxiliary compressor of the United Stirling system is described (Hallare and Rosenqvist 1977) as an oil-free single-stage double-acting compressor with piston rings acting as suction valves. The displacement is 10 cm^3 (0.61 in^3) and the pressure ratio is 10:1. To unload the compressor during the increase of power and steady state conditions the suction and pressure sides of the compressor are connected to each other by a compressor short-circuit valve.

General Motors system

General Motors invested considerable effort in the development of control systems during their decade of intensive work on Stirling engines. Percival (1974) has indicated that on General Motors engines, temperature and torque were always independently controlled. Pressure modulation was principally used for torque or power control.

The small generator sets made by General Motors for the U.S. Army, GM Type GPUI, 2 or 3, required control of the speed to close limits. Stability was to be maintained at 3600 revolutions per minute to within ±10 revolutions per minute. Speed droop was not to exceed 90 revolutions per minute and the surge limit for sudden changes in load was set at

216 revolutions per minute. The recovery time for 100 per cent load change was limited to 6 seconds.

By 1967 the GPU-3 system was capable of holding the stability at ±5 revolutions per minute, the droop did not exceed 10 revolutions per minute, the surge limit was met and the recovery time was reduced to two seconds. Despite this achievement Percival writes feelingly that:

> 'From the standpoint of reliability, however, the entire speed governing system was a constant source of trouble until nearly the end of the programme. The hydrogen compressor was perhaps the major problem in the beginning. It was incorporated into the base of the crankcase of the GPU-3 as an extension of the displacer piston rod in the form of a hydraulic plunger. Hydraulic pressure activated a diaphragm compressor which eliminated the need of a sliding or rotating seal. This made servicing more difficult. The hydraulic plunger required precision machining and was subject to binding. In retrospect, it would have been better to mount an experimental compressor outside the engine and drive it from a gear, with a break-away coupling, or from a belt. On the other hand, an outside compressor requires a good seal to prevent hydrogen leakage.
>
> Another item which often stopped endurance tests was failure of the small ($\frac{1}{2}$ in (1.27 cm) dia.) hydrogen check valves and main control valve—usually the seats were pounded out or distorted sufficiently to leak. The hydrogen control valve was actuated by hydraulic pressure delivered by the speed sensing governor which was mounted on the crankcase and gear driven'.

In 1964, the governor system had 5 separate valving units and 10 adjusting screws; by the end of 1965, it had 2 valving units and one adjusting screw.

Results of endurance testing of the GPU-3 at GMR in 1967 showed that the hydrogen compressor had failed twice in the 1537 hour run and the governor hydrogen valve had failed four times.

In 1969, the GPU-3 at GMR was operated on a more rigorous 500-hour test, equalling a military qualification test. In order to meet military requirements, a 'certified parts list' for the package was established so that all parts were like the engineering drawings. This defined exactly what was being tested and prevented casual substitution of components which would have caused the test to lose significance. At the conclusion, the maximum overhaul life was extended slightly but under more rigorous conditions to 560 hours from the previous 553 hours.

The longest run with no service was extended to 525 hours from the previous high of 196 hours. The longest run without stopping was extended from 159 hours in 1967 to 235 hours. In all, the engine was stopped four times, all caused by building safety interlocks and in no way connected with the GPU operation. The limit of 560 hours was caused by the hydrogen compressor—a small valve assembly failed to function properly. However, the hydrogen check valves and governor control valve were in excellent condition.

An alternative system for compressing hydrogen was investigated briefly in 1961. It was based on electrolytic generation of hydrogen and diffusion through palladium tubes. Pressures to 7.9 MN/m^2 (1150 lbs per sq in) were maintained inside the tubes; but the concept was abandoned when piston rod seals were found to seal hydrogen better than expected.

Pressure-amplitude variation

An alternative power control system for Stirling engines was described by Alm *et al.* (1973) in an account of developments at United Stirling. Power control was achieved by means of variation in the amplitude of the engine cylinder pressure excursion. The system is illustrated in Fig. 10.10. A number of different gas bottles cast into the engine crankcase could be put into direct communication with the working space of the engine by a series of valves operated by the cycle pressure amplitude.

To reduce engine power, one or more of the valves was opened so that the volume of the gas bottle associated with it became part of the dead volume of the working space. Increase in the dead volume decreased the volume compression ratio V_{max}/V_{min} and also reduced the amplitudes of the pressure variation as shown in the pressure/time diagram given in Fig. 10.11.

The gas bottles were of different sizes and the valves could be sequenced so that the power level could be progressively reduced in small

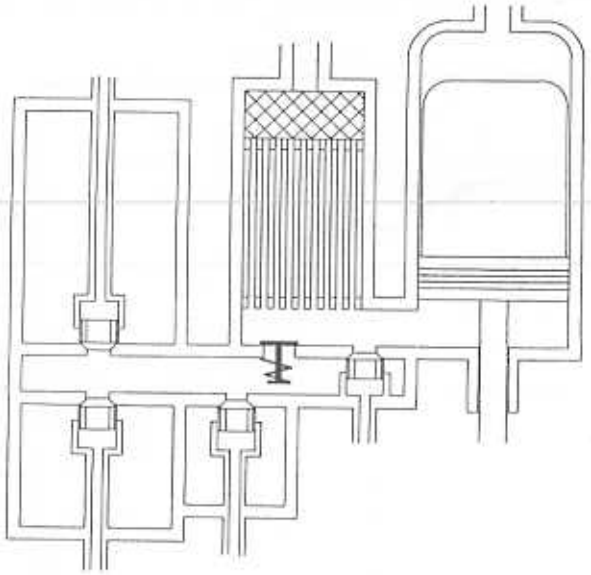


FIG. 10.10. United Stirling system for power control by variation in the amplitude of the cyclic pressure curve.

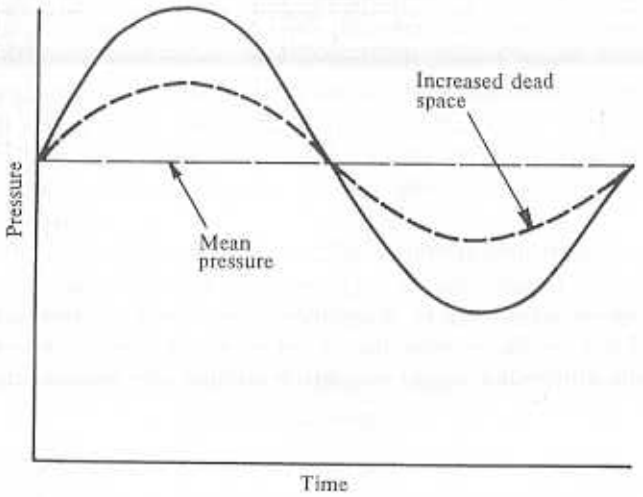


FIG. 10.11. Cyclic pressure characteristic of a Stirling engine with power control by variation in the engine dead space volume (pressure amplitude variation).

incremental steps approximating to the smoothed curve shown in Fig. 10.12.

The system of power control by pressure-amplitude variation was developed by United Stirling as an alternative to the system preferred by Philips of control by pressure-level adjustment. Alm *et al.* (1973) pointed out that the Philips system required a rather complicated and expensive compressor to pump the hydrogen working fluid back to the pressure vessel.

Furthermore, because the compression process required a considerable time, it was necessary to resort to the complementary short-circuiting system and thereby sustain a decrease in engine thermal efficiency. With repeated unloading of the engine, as for example, in a city bus, a substantial efficiency loss would accrue.

To support this contention the figure reproduced in Fig. 10.13 was given by Alm *et al.* (1973). This shows the thermal efficiency for Stirling engine as a function of speed at full load and half load with three different control systems. At half-load the engine with short-circuit power control has only half the thermal efficiency it has at full load. With mean-pressure or pressure-amplitude control the decrease in efficiency is much less pronounced. No details of the engine or any numerical data were given in support of this figure.

It was interesting subsequently to observe that United Stirling appeared to abandon the pressure-amplitude control system in favour of a return to the Philips pressure-level control (Hallare and Rosenqvist 1977). No

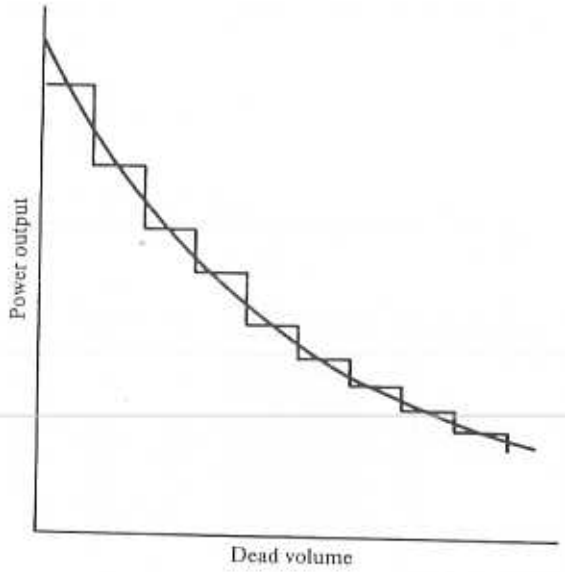


FIG. 10.12. Power output as a function of dead space in a Stirling engine with power control by pressure amplitude variation.

reasons for this change were given and the shortcomings of the pressure-amplitude system are unknown.

A possible explanation may be that, in practice, the decline in engine efficiency with the pressure-amplitude system was greater than anticipated. Some support for this may be found in data presented in the important paper by Neelen *et al.* (1971) following experiments by MAN/MWM on a 7 kW (10 hp) Stirling engine to establish the effect of

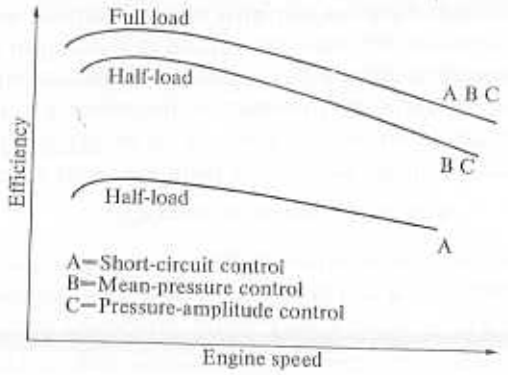


FIG. 10.13. Comparison of Stirling engine efficiency as a function of speed at full load and half load conditions with three different control systems.

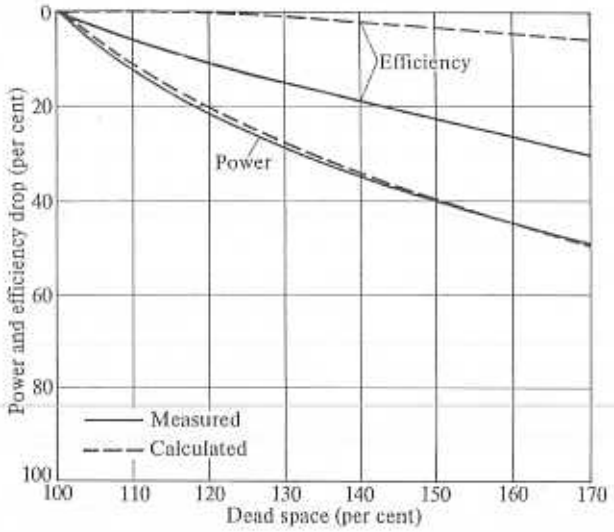


FIG. 10.14. Effect of dead space on the power and thermal efficiency of a Stirling engine thermal. The calculated and measured values are compared as obtained in tests on a 10 hp Stirling engine at MAN/MWM (after Neelen *et al.* 1971).

dead space volume. The dead space of the engine was enlarged artificially and the result, shown in Fig. 10.14, was given as the percentage decrease in power output and thermal efficiency of the engine as a function of the dead space. According to Neelen *et al.* (1971) the power curve declined as predicted by calculation but the efficiency declined at a much greater rate than predicted for reasons that had 'not yet been clarified'.

Phase-angle variation

In a Stirling engine the volume variations of the expansion space lead those of the compression space by the phase angle α, one of the principal design parameters of the engine. Variation in the phase angle is one possibility for engine power control. The power output of a Stirling engine as a function of phase angle is approximately sinusoidal in form, as shown in Fig. 10.15.

At zero phase angle (point A in the figure) the volumes of the compression and expansion spaces vary exactly in phase. The change in total system volume is a maximum and the range of the cyclic pressure excursion is a maximum. However, if we assume isothermal, or adiabatic, processes in the compression and expansion spaces there is no work output from the engine because the pressure—volume diagram is the single line a–b shown on the work diagrams at A and B in Fig. 10.15. The pressure in the engine cylinder is the same on the downstroke of the

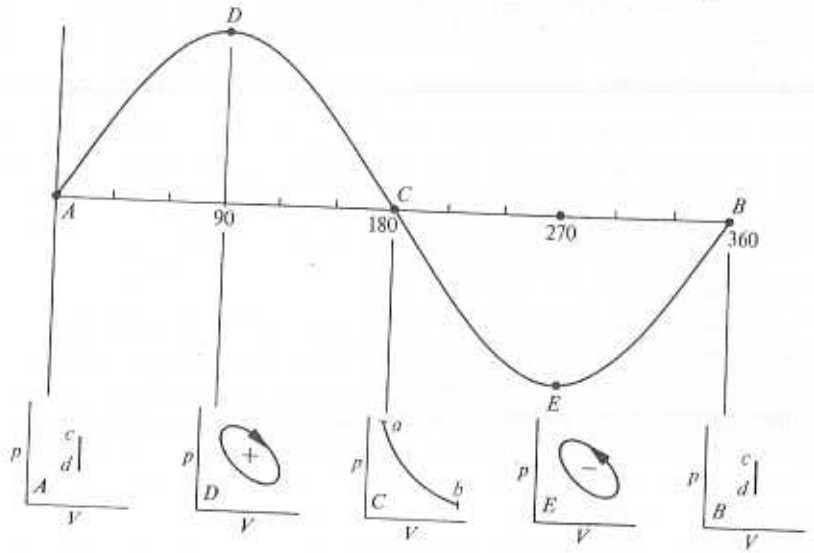


FIG. 10.15. Power output of a Stirling engine as a function of the phase angle between volume variations in the expansion and compression spaces.

piston as it is on the upstroke of the piston. There is no cyclic flow of working fluid through the system.

At point *C*, with a phase angle of 3.14 radians (180°) the converse is true. Here the volumes of the compression and expansion spaces are exactly out of phase so that the variation in the total system volume is at its minimum possible value. If the swept volume in the two spaces is the same, the variation in total system volume is then zero. The flow of working fluid between one space and the other is maximum. The range of the cyclic pressure variation is small because it is due solely to the change in the mean temperature as the fluid moves between the hot and cold spaces at constant volume. Again the work output is zero because the pressure volume diagram is the single line '*c*–*d*' as shown at point *C* on Fig. 10.15.

At any phase angle between 0 and 3.14 radians (0 and 180°) the volume variations lead those in the compression space and the cyclic pressure and volume changes will result in a work diagram as shown at point *D* in Fig. 10.15. If the expansion space is hotter than the compression space the work diagram will be positive and a surplus of work will be available at the shaft to drive the engine and an external load. The net work output will reach a maximum value at a phase angle of about 1.57 radians (90°).

If the phase angle is between 3.14 radians and 6.28 radians (180° and

360°), points *C* and *B* in Fig. 10.15, a similar situation will prevail but now the work diagram will be negative as shown at *E*. An input of work will be required to drive the engine and the maximum value occurs at a phase angle of about 4.71 radians (270°). The direction of heat flow will be reversed. Heat will be transferred from the cooler *to* the working fluid and from the working fluid *to* the heater. In this situation the engine is operating as a heat pump, taking in heat at a low temperature and rejecting it at a higher temperature.

Of course if there is no power input available at the shaft to drive the engine as a heat pump it will simply stop *and then run in the reverse direction* as a prime mover, taking in heat at high temperatures and rejecting it at a low temperature with a positive power output.

Power control by phase-angle variation is characterized by instant response and represents an extremely convenient way to provide a rapid reversing engine. The system was adopted by the Electromotive Division of General Motors for an 590 kW (800 hp) Vee 8 Stirling engine intended as the propulsion motor for coastal surface vessels where good manoeuvering capability was required. The engine had a piston and displacer in each cylinder but they were connected to separate drive shafts. The separate shafts were coupled by sun-and-planet gears so that movement of the sun wheel caused a change in the phase angle between the piston and displacer motion and, hence, a change in the phase angle between the expansion and compression spaces.

The calculated power and efficiency curves for this engine were given by Percival (1974) and are reproduced in Fig. 13.9. Only one bank of four cylinders of this engine was in fact constructed for development work before General Motors abandoned piston-displacer engines in favour of Siemens double-acting engines. In 1967 the author witnessed the operation of this four-cylinder half-engine at the Electromotive Division works at La Grange, Illinois. The response of the engine was remarkable to change in the phase angle by simple adjustment of the sun wheel. It was said at the time that the engine could be reversed in less than a revolution (but only when the stress office engineers were not present!).

Power control by phase-angle variation is obviously not applicable to double-acting engines where the phase angle is limited to the value $6.28/N$ radians ($360°/N$) where N is the number of cylinders. Double-acting engines can be conveniently reversed by switching the interconnected cylinders so that in effect a phase change of 3.14 radians (180°) is introduced. Reversal of the cylinder connections can be done by a simple slide valve fitted to the cold side of the engine. It is not known if the system has, in fact, been used in double-acting engines but the potential for reversing engines was early recognized by van Weenan (1947).

Stroke variation

Power output from a Stirling engine may be controlled by variation in the stroke of the reciprocating elements. This may be either or both the piston and displacer of single-acting engines or the piston-displacer of double-acting engines. This method of power control is more applicable to free-piston or free-displacer machines than to engines with conventional drive mechanisms. In the single-cylinder Beale free-piston Stirling engine, an adjustment of piston stroke to the load condition does occur naturally in the engine. When the load resistance to movement is light the piston stroke extends to the maximum value permitted by the stroke limiting controls of the engine. As the load resistance to motion increases the piston stroke decreases but the force exerted by the piston increases and attains it maximum value when the piston motion is completely restrained. It is an eerie sensation to grasp the pump rod attached to the piston of a Beale engine and to feel the engine respond instantly by an aggressive increase in the driving force as one attempts to restrain the piston movement.

A hybrid Stirling engine has a free displacer and a crank-controlled piston for driving a rotating shaft. The concept was first reduced to practice at the University of Calgary and is being further developed at the University of Bath. Control of engine output by adjustment of the displacer stroke was investigated briefly. By limiting the displacer stroke, the mass flow of gas in and out of the hot and cold spaces was reduced, and so the range of the cyclic pressure amplitude was reduced. The effect was somewhat analogous to the system for pressure-amplitude control described above where the system dead space was increased by opening valves communicating with dead volume gas bottles in the engine crankcase. Limitation of the displacer stroke increases the effective clearance space in either or both the expansion or compression volumes. Clearance space in the cylinders may be accounted as dead volume to correspond with the above system. However, a change in the displacer stroke also changes the ratio of swept volumes in the expansion and compression spaces. This swept volume ratio η is another principal design parameter that may be used for power control independent of the dead space effect.

John Malone (1930) in his liquid engines (see Chapter 8) used a system of displacer stroke limitation as his principal mode of power control. In his paper he shows a rack-and-pinion device on the displacer rod for varying the displacer stroke but the details of the actual mechanism were not disclosed.

In the Thermoelectron Tidal regenerator engine for artificial hearts (see Chapter 17), power control was achieved by variation in the stroke of the piston. A flexible metal bellows acting as the piston was caused to

reciprocate by the action of a ball nut on a screw shaft driven by an electric motor. Rotation in one direction caused the nut and hence the piston to ascend. Rotation in the other direction caused it to descend. An electronic unit controlled the direction and duration of rotation in one direction or the other and so regulated the rise and fall of the piston and hence the output of the engine. The output in this case was, in fact, hydraulic power to operate a blood pump.

11 FREE-PISTON STIRLING ENGINES

THE following chapter was contributed by William Beale, President, Sunpower Inc., Athens, Ohio. Beale invented the free-piston Stirling engine in the late 1950s while a professor of mechanical engineering at the University of Ohio. Over the next decade, with remarkably single-minded dedication, he developed the concept to an operational stage. In the university environment he was unable to secure adequate funds to develop the free-piston engine commercially and so he founded the company he now leads. Sunpower is unique as the only company in the world in commercial production of Stirling engines. Their best known machine is the small free-piston engine demonstrator available as a water pump, electric power generator, or a refrigeration pump. Sunpower's principal business is research and development work on advanced free-piston Stirling engine developments, some of which are discussed below.

G. W.

INTRODUCTION

A free-piston Stirling engine is a machine in which the motion of the reciprocating elements to accomplish the thermodynamic cycle are effected by fluid forces and by the dynamical, fluidic interaction of the components. There are no mechanical linkages coupling the pistons or displacers.

Such machines offer advantages of simplicity, freedom from leakage (since they can be hermetically sealed), low cost, self-starting, and very long life. They are being developed for use as thermally activated heat pumps, solar-electric converters, remote-area power generators, total energy systems, and water pumps.

This chapter is intended to give an elementary description of free-piston Stirling-engine dynamics, to give examples of some of the many possible configurations, and to discuss the present state of development and areas of research. The viewpoints given here are those of the author developed with the assistance of many helpful discussions within his own design group at Sunpower Inc., with the MTI Stirling engine team, and from the publications cited in the text. Since the field is a very active one, the interested reader should view this chapter as no more than an introduction to the subject and should consult the literature frequently for new developments.

DESCRIPTION OF FREE-PISTON ENGINE DYNAMICS

In order to describe the action of free-piston engines, it is useful to assume harmonic motion of the components, which is often nearly true, and to recall that such motions may be described not only as sine waves, but also by the projections on a horizontal or vertical axis of rotating

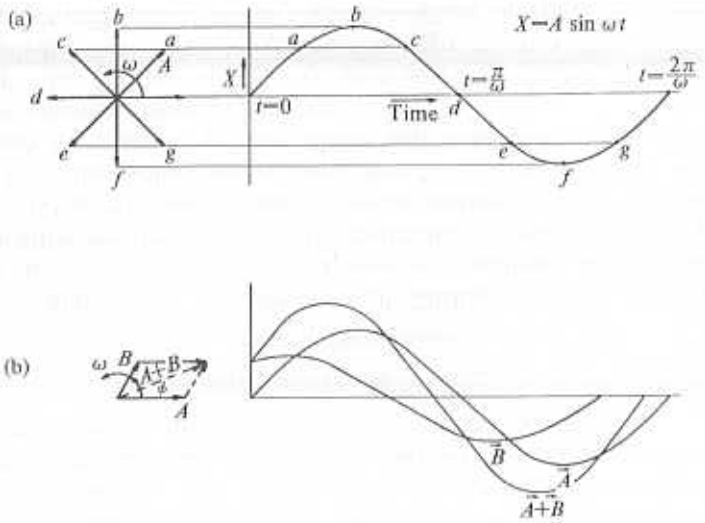


FIG. 11.1. (a) Vector representation of a sine wave. (b) Sum of two sine waves as a vector sum. (c) Vector work.

vectors. Readers who wish to refresh their memory on the rotating vector method of representation are referred to any standard textbook on mechanical vibrations† in which all of what is given below is clearly developed in a simple and convincing manner.

To summarize the vector representation of harmonic motions:

(a) A sine wave $A\sin\omega t$ may be represented as the vertical component of a vector of magnitude A rotating at angular velocity ω radians/second (Fig. 11.1).

(b) The sum of two sine waves $A\sin\omega t$ and $B\sin(\omega t+\phi)$ may be represented by a rotating vector equal to the vector sum of A and B leading A by the angle ϕ.

(c) The work of a sinusoidal force $A\sin(\omega t+\phi)$ upon a displacement $B\sin\omega t$ is represented by the product of the component of the

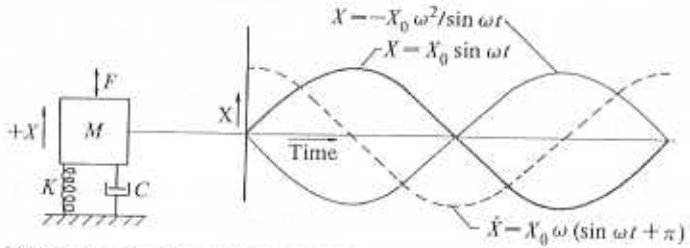
† For example, Den Hartog—*Mechanical Vibrations*, McGraw Hill.

force vector normal to the displacement vector with that displacement vector. If the force vector *leads* the displacement, work is done *on* the displacement. If the force vector *lags* the displacement, work is done *by* the displacement. The magnitude of the work done is $\pi AB \sin\phi$ per cycle or the *power* is:

$$\frac{\text{work}}{\text{cycle}} \times \frac{\text{cycle}}{\text{second}} = (\omega AB \sin\phi)\left(\frac{\omega}{2\pi}\right) = \frac{\omega}{2} AB \sin\phi$$

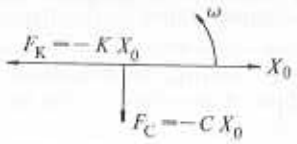
This can also be interpreted to mean that only the component of the force *normal* to the displacement does work, or that only the component of the force *parallel* to the velocity does work. (Note that the velocity is always 90° in advance of the displacement so that a force normal to velocity is parallel to displacement.)

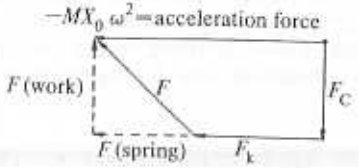
The mass–spring–damper system of Fig. 11.2 illustrates the statements made above. Note that the damping force points down, representing a



Newton's second law: summation of forces = mass × acceleration

$$F + F_{\text{spring}} + F_{\text{damping}} = M\ddot{X}$$
$$F - KX - C\dot{X} = M\ddot{X}$$
$$X = X_0 \sin\omega t$$
$$\dot{X} = X_0\,\omega\cos\omega t = X_0\,\omega\,(\sin\omega t + \pi)$$
$$\ddot{X} = -X_0\,\omega^2 \sin\omega t$$





Work/cycle on damper $= \pi X_0 F_C \sin\pi$
$= \pi X_0(-CX_0)$
$= -\pi X_0^2 C$

FIG. 11.2. Mass spring damper vectors.

work done by the mass, and that as a result, the applied force F must have a component normal to displacement (parallel to velocity) pointing *up*. If there were no damping force, then the applied force would have no normal (work) component and would be in the direction of the acceleration (spring direction). In the discussion to follow, it is useful to consider each force as a combination of *work* components *normal* to the displacement and spring components *parallel* to displacement of the component under consideration.

The necessary requirement for free-piston operation is the dynamic equilibrium of forces as required by Newton's Second Law, i.e. the sum of all forces acting on each moving component must equal the acceleration force of the component. If this situation exists, then the system may in fact oscillate at that condition. The forces to be considered here are damping forces, spring forces and pressure forces. These must all sum to the acceleration vector which has magnitude of $MA\omega^2$, where M is the component mass, A is its amplitude of displacement and ω is its angular frequency in radians per second. The direction of the acceleration force is opposite to the direction of the displacement vector.

Once a thermodynamic analysis has been carried out as described in the previous chapters, and from it the desired motions to the components have been decided upon, the dynamic analysis to permit this desired motion may follow. The procedure for the dynamic analysis is to compute the acceleration force from the desired operating frequency, amplitude, and the mass; the damping forces from the pressure-drop characteristics of the heat exchangers and the gas-flow velocities; the pressure forces from the previous thermodynamic analysis or from an isothermal-cycle analysis as described next; and the spring forces from the mechanical or gas spring characteristics, choosing the spring forces to permit the vector sum of all forces to equal the acceleration force required.

WORKING SPACE FORCES

Gas pressure forces

The working space pressure in real Stirling engines is a complex function of thermodynamic and dynamic influences, and its accurate determination requires a high order computer analysis. However, for the purpose of dynamic analysis, much simpler isothermal estimates are quite useful. Even the isothermal pressure relation is not harmonic, however, but it can be used to get the phase and amplitude of an equivalent harmonic pressure wave as follows:

(a) For the design under study, with known volumes, temperatures, and displacements, compute an isothermal (or higher order) pressure relation and plot a pressure–volume diagram. Find the cycle work as the integral of the P–V diagram.

(b) Equate the cycle work to the vector product of volume and a harmonic pressure wave of amplitude equal to the pressure wave previously computed and phase angle such as to give the same cycle work. $W = -\pi V \cdot P \sin \phi'$. V is the amplitude of the volume vector which may itself be the sum of two or more vectors representing volume changes due to piston motions, P is the amplitude of the harmonic pressure wave, and ϕ' is the pressure phase lag behind the negative volume vector (Fig. 11.3).

This equivalent pressure vector may then be used in the vector representation of the engine dynamics. Its accuracy is good for engines with larger dead volume, but not so good for tight, low dead-volume engines

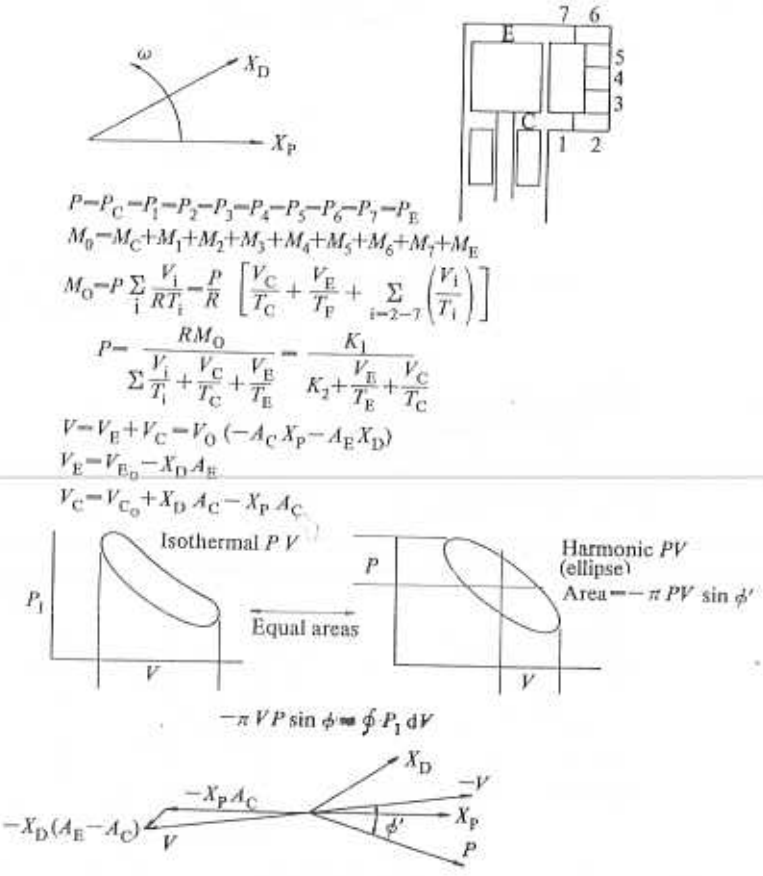


FIG. 11.3. Pressure forces.

such as small annular gap designs, in which the true pressure wave is quite far from simple harmonic form.

Gas spring forces

Gas springs have spring constants $K = \dfrac{\gamma A^2 P}{V}$

where γ = the adiabatic or other appropriate exponent
A = the area of the piston
P = the gas average pressure
V = the volume of the gas spring.

Damping forces

Damping forces in the heat exchanger may be estimated from the steady-flow pressure-drop characteristics of the heat transfer elements and the volume-flow rate to the associated active space.

True damping forces are in general highly anharmonic, and an equivalent harmonic force giving the same energy dissipation should be computed in a manner similar to that described for the pressure forces. This method is definitely a severe approximation to the truth, but can, nevertheless, give reasonably good results because damping forces due to flow pressure drops are usually rather small in comparison with other forces.

TYPES OF ENGINE

The two-piston free-piston engine

Fig. 11.4 shows perhaps the simplest concept with which to exemplify a free-piston dynamic analysis—a two-piston arrangement with power taken from one or both pistons.

In the idealized case, piston 1 is a massless damper and piston 2 is an undamped resonating mass. The pressure vector is collinear with the displacement of piston 2 and at right angles to the displacement of piston 1. Piston 1 sees a pure damping force and piston 2 a pure spring. The resonating frequency is fixed by the mass of piston 2 and the system spring constant, which is the sum of that due to working space pressure P_1 and the bounce space pressure P_2.

In the more realistic example shown, the pressure wave is assumed to lag both pistons, hence its resulting force has work components on both, and both can produce work through a damping load such as a linear alternator. If the pressure wave were collinear with the displacement of piston 2 then that piston would not be able to produce work and would simply resonate. If the pressure wave lay between the two displacements, lagging 1 and leading 2, then the resulting forces would do work on 1 and

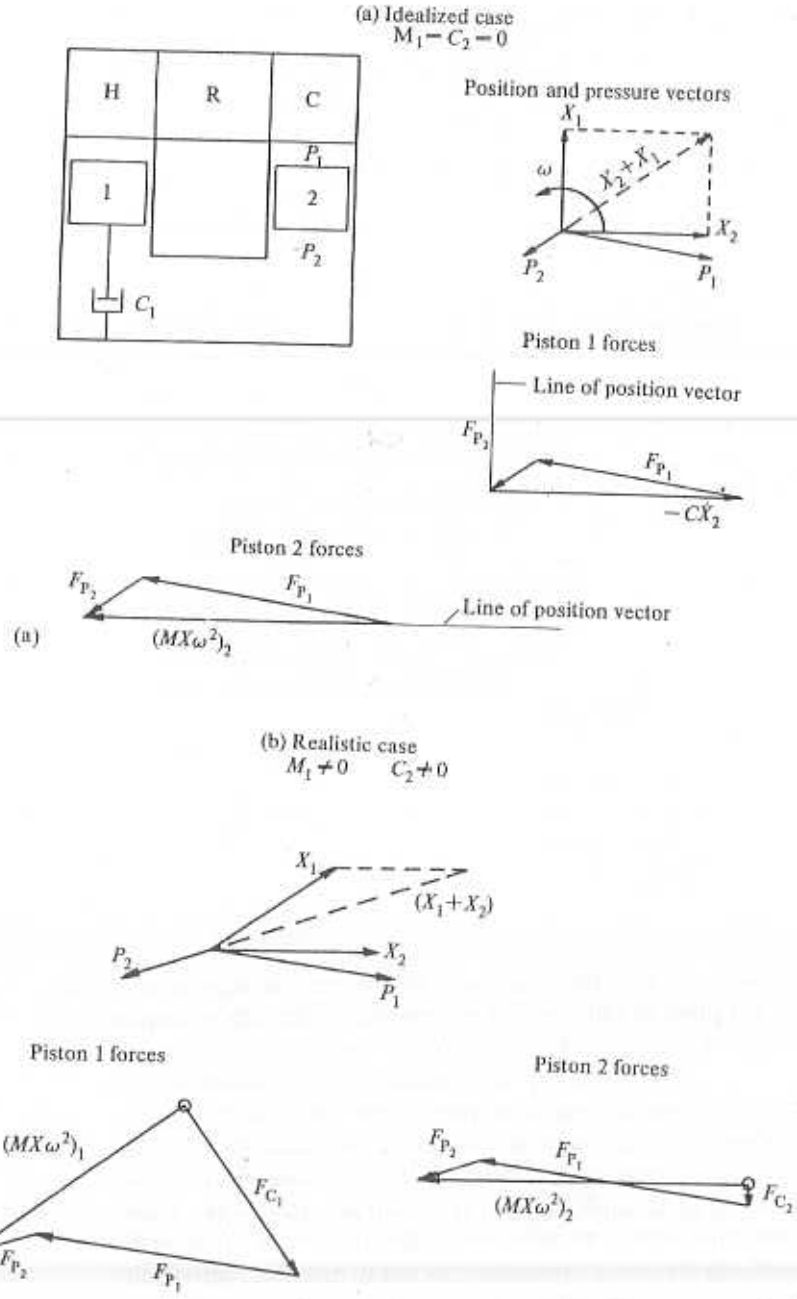


FIG. 11.4. Dynamics of a free-piston engine with two pistons.

absorb work from 2, requiring that it be driven by work from 1 through, for example, its linear alternator acting as a motor.

The piston-displacer free-piston engine

Fig. 11.5 shows a generalized piston-displacer machine with three moving components—piston, displacer and cylinder. This figure defines the nomenclature to be used in the simplified cases described below.

The simplest special case of the general piston displacer machine of Fig. 11.5 is shown in Fig. 11.6, representing the Harwell free-piston electric generator (Cooke-Yarborough 1975). Here the driving energy for the displacer comes from the spring between the displacer and cylinder (K_1) which, as a result of the phase angle between the displacer and cylinder motion, has a work component adequate to overcome displacer damping. The major cycle work flows through the piston damping force (F_{D_1}) represented in the case of the Harwell machine by its alternator.

Fig. 11.7 is another special case of Fig. 11.5—a simple, relatively low-speed engine with the displacer sprung to the bounce space. The displacer rod provides a difference in area between hot and cold ends of the displacer which provides the major drive force for the displacer. This is the design of some of the small Sunpower models. In order for this system to operate, it was necessary to have a very light displacer since the acceleration forces available were weak (Agbi 1973).

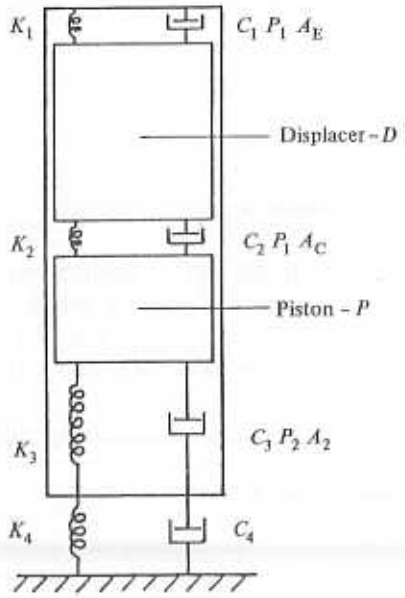


FIG. 11.5. Generalized piston-displacer machine showing nomenclature.

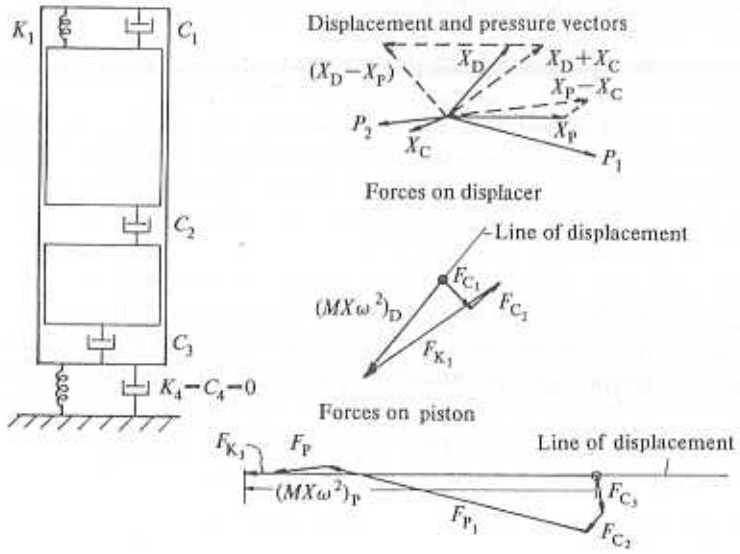


FIG. 11.6. Dynamics of the Harwell free-piston electric generator.

A combination of spring linkage and differential area can provide a strong displacer drive capable of operating the engine at higher frequency with more massive displacers. Fig. 11.8 shows an arrangement used on larger Sunpower designs, in which the displacer is sprung to the cylinder in order to resonate the displacer mass, and a differential area is used to provide a force overcoming displacer damping. The sum of damping and force on the displacer differential area brings the force diagram on the line of displacement, and a spring of sufficient strength to provide the necessary resultant force for resonance is then added. Similarly, a spring K_3 of sufficient stiffness to resonate the piston is added to the spring component of the working space pressure wave in order to permit piston resonance.

For purposes of simplicity, the slight additional effects of the moving cylinder have not been shown. If it is desired to do so, these effects may be added in the manner shown previously in Fig. 11.6.

A drive method intermediate between Figs. 11.7 and 11.8 which is sometimes useful is shown in Fig. 11.9. Here a spring K_2 between piston and displacer assists to resonate the displacer. The fact that neither the pressure nor the spring force are co-linear with displacer displacement results in rather large work flows from the displacer to the piston. In practice this results in hysteresis losses in the gas spring K_2 and reduced system thermal efficiency. Engines made in this way have large rod

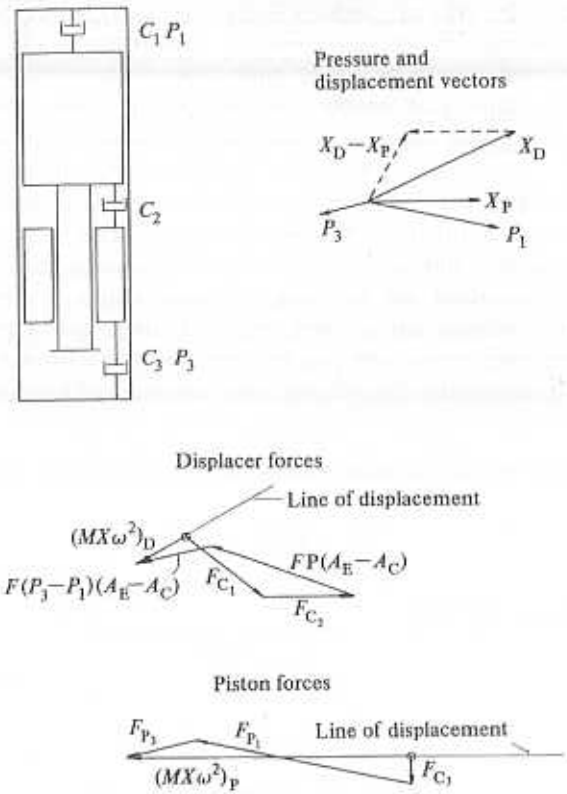


FIG. 11.7. Dynamics of an engine with a common spring in bounce space.

diameters compared to the spring to cylinder models of Fig. 11.8 if operated at the same conditions.

Double-acting free-piston engines

The force diagrams of Fig. 11.10 show the possibility of double-acting free-piston engines. A three piston engine is illustrated but larger numbers of pistons are possible, as well as different connections between cylinders, as a study of the figure will immediately suggest. The three piston double-acting machine was originally suggested by Walker† as a candidate for three-phase power generation.

As the figure shows, there must be a large phase angle between the piston and its associated hot space pressure or the force diagram will not result in positive work. In fact, for the three piston engine, this phase

† private communication

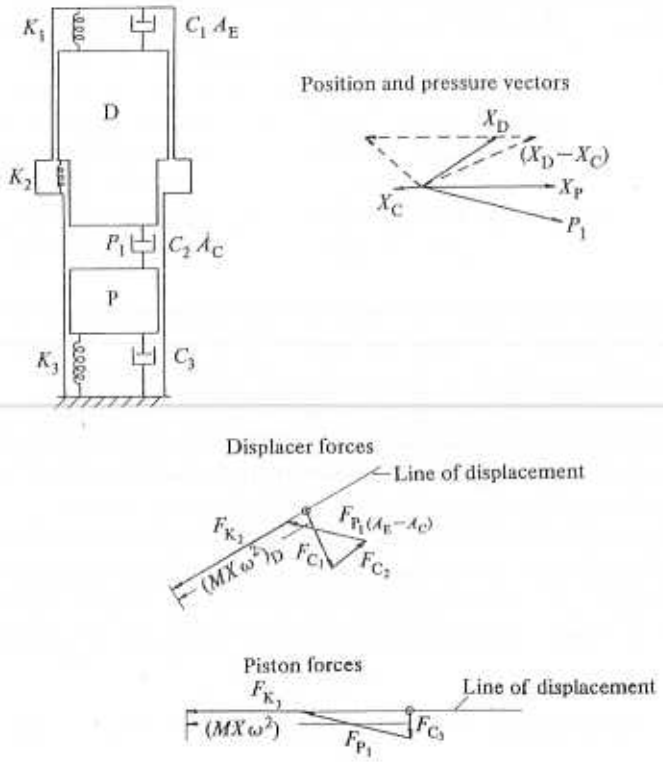


FIG. 11.8. Dynamics of an engine in which the displacer is sprung to the cylinder.

angle must exceed 30° or the resultant of the two pressure forces will lie on or below the line of displacement and hence have a negative work component.

This situation may be avoided by making the cold piston area smaller than the hot piston area, as would naturally be the case with a power rod leaving the cold space. With the rod area subtracted, F_{P_1}, is a smaller force and the force diagram may have net positive work with less pressure phase lag.

Constraints such as the one cited above, resulting from the necessary symmetry of double-acting piston arrangements, make them less flexible and less responsive to optimization procedures than piston-displacer machines. The merit of the double-acting free-piston is the same as its kinematic twin—higher power density and fewer moving components.

Variations of geometrical arrangements

Besides the arrangements already discussed, there seems to exist an almost endless array of variations. For example, in Fig. 11.11 are shown a

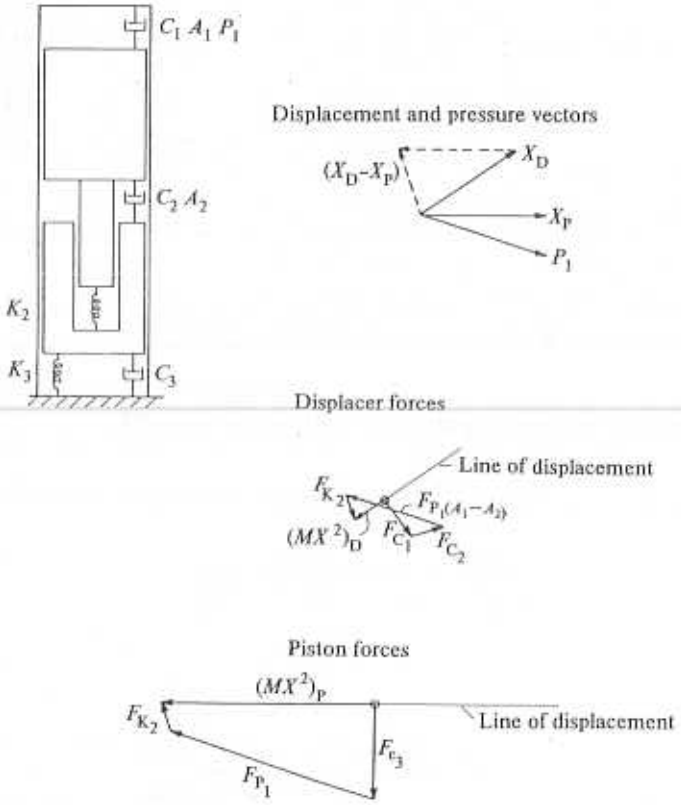
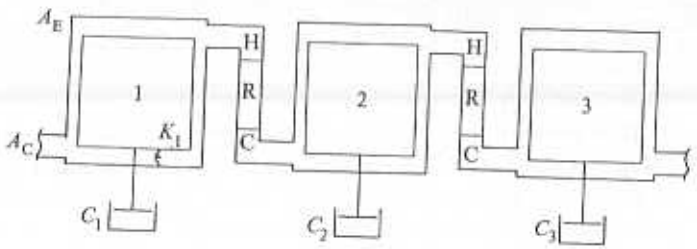


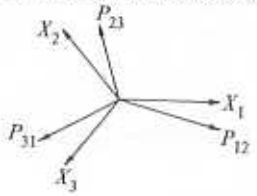
FIG. 11.9. Dynamics of an engine in which the displacer is sprung to the piston.

number of displacer geometries, each completely equivalent to the others. This is by no means an exhaustive list and the reader may doubtless add more after brief reflection. It is the task of the designer to weigh the merits of many possible configurations for his particular applications, taking into account the realities of mechanical design—thermal distortion, leakage, wear, gas spring losses, centering, control, power modulation, fouling with wear particles, cost, ease of manufacture, alignment, operating stability, startup characteristics, and so on.

A host of multi-cylinder arrangements are also possible: heat pumps, cooling engines, heat-driven cooling engines and other combinations. Again the designer must choose with care. The author prefers to stick to the rule cited by Professor Egon Orowon—'Never try something complex until you have failed with something simple'. In this case, if one piston will do, should one be tempted to try two or five?



Position and pressure vectors



Forces on piston 1

Case for zero work, $A_E = A_C$, 30° Phase lag of pressure

Case for 30° phase lag, $A_E > A_C$, work produced

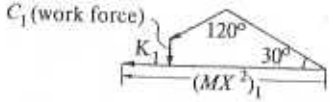


FIG. 11.10. Dynamics of a double-acting machine with three pistons.

Other useful arrangements, and other methods of dynamic analysis are given by Benson (1977b), Martini (1975a), Martini *et al.* (1977), and Rauch (1975).

COMPUTER SIMULATION OF FREE-PISTON DYNAMICS

Figs. 11.12 and 11.13 are a much simplified analogue computer representation of the dynamic model of Fig. 11.5. If the chief aim is a study of dynamics rather than thermodynamics, the analogue computer is a marvellously easy and, thanks to modern electronics, cheap way to study free-piston engine dynamics. It is a quite simple matter to construct such a special purpose analogue from readily-available components and with it to study the free-piston engine. By manipulating potentiometers representing the dynamic components, the student may quickly develop a feel for the machine and he may be able to get good values for the required springs, areas, and loads necessary to achieve desired operation.

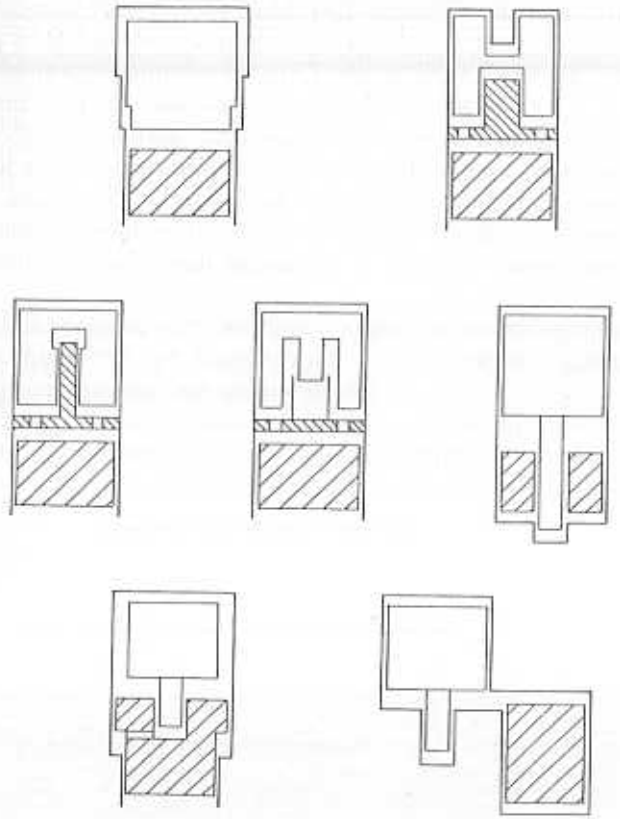


FIG. 11.11. Seven equivalent displacer geometries.

With the addition of non-linear elements and empirically derived correction factors in the analogue, the modelling may be made an excellent representation of a real machine, including accurate pressure drop values and component collision phenomena.

The digital computer may, of course, be used to do precisely the same thing as an analogue with greater versatility, but in most cases, less opportunity for the operator to observe instantly the results of his changes. The IBM Continuous System Modelling Program (CSMP) is a particularly useful tool for digital simulation of analogue circuits, quickly learned by those familiar with analogue computers or problems in vibration (Beale 1969).

THERMODYNAMIC AND DYNAMIC OPTIMIZATION

The previous discussion was aimed only at an illustration of free-piston engine dynamics. The complete design task must start with a

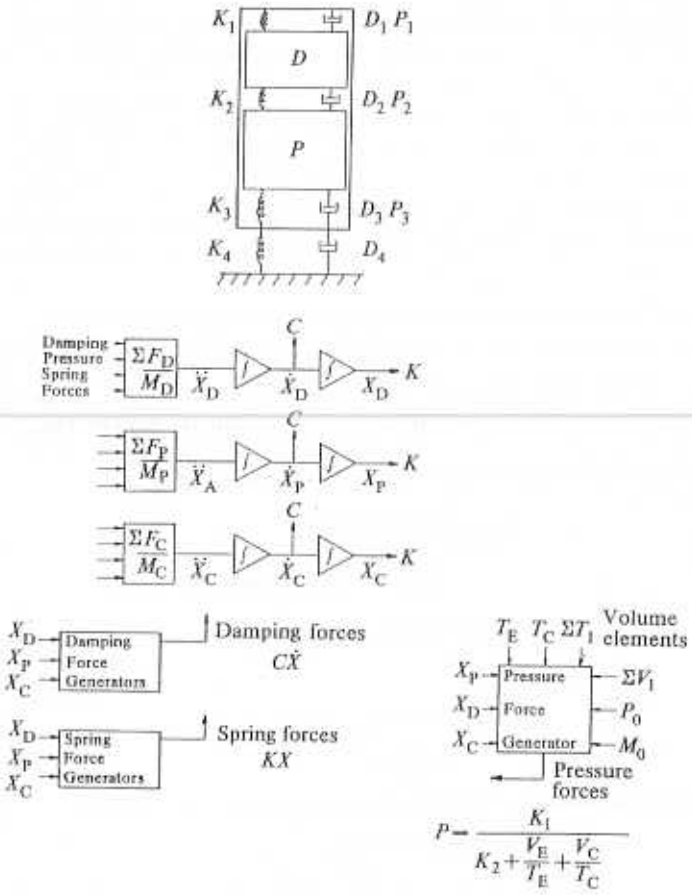


FIG. 11.12. Schematic diagram of analogue computer.

thermodynamic optimization utilizing an appropriate analytical technique which includes within it a dynamic analysis. Gedeon *et al.* (1978) have described the Sunpower procedure, which includes two levels of Stirling engine simulation, a fast running empirically corrected isothermal analysis, and a powerful third order code which is well validated by comparison with engine test results. Associated with these is an automatic optimizing routine which searches in each coordinate direction and arrives at a maximum of a pay-off function previously defined by the designer. Engine dynamics are appropriately considered in the search for the optimum configuration. The Sunpower design procedure is outlined below.

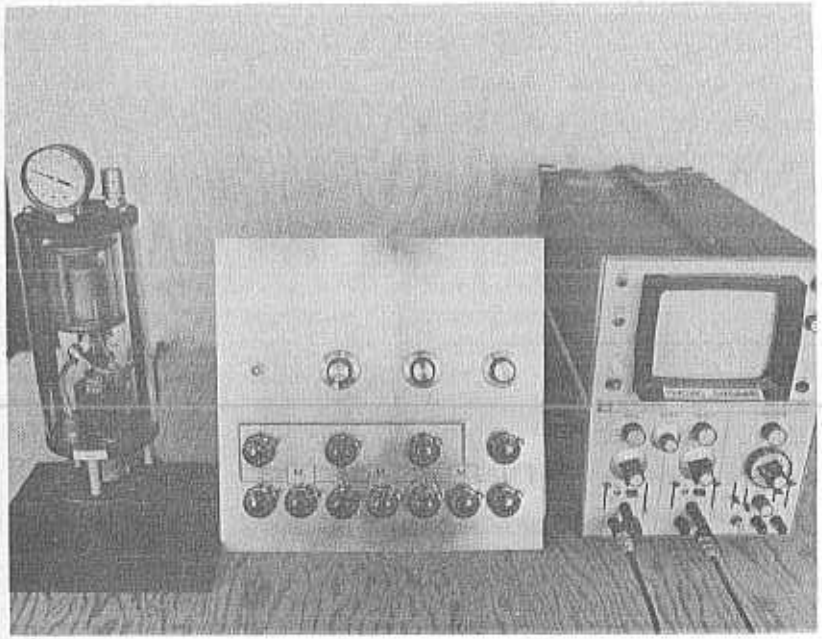


FIG. 11.13. Photograph of analogue computer.

Sunpower design procedure

1. The purpose is defined, which determines the weighting factors to be used in the optimization, i.e. an artificial heart requires maximum efficiency and small size; a water circulator for a wood stove must be cheap and durable but may be inefficient.
2. The other limits on the optimization are set—power, frequency, pressure, temperature, etc., are given values or ranges of allowable values.
3. An automatic optimization algorithm is used to define the thermodynamics, dynamics, and heat-exchanger parameters using appropriate weighting factors on the pay-off functions, and taking due account of loss mechanisms—conduction and shuttle heat transfer, seal leakage, gas-spring hysteresis, gas-bearing pumping power, control and start-ups, and stability requirements.
4. The results of the optimization are studied and adjusted by consideration of other design constraints not presented in the algorithm. It may be decided, for example, that the increase of efficiency from 36 per cent at 20 bar to 37.2 per cent at 40 bar is not worth the cost

and the mechanical design problems caused by the higher working pressure.

5. The final thermodynamic design is simulated in a free-run mode on the high-level computer code to verify performance, stability, start-up, and response to load changes. If it passes these tests, the design is released for mechanical design.
6. Mechanical design is carried out, including structure, bearings, seals, auxiliary pumps, controls, valves, etc.
7. A prototype is constructed, tested, and modified for satisfactory performance. Test results are fed back into the computer design codes to refine them.
8. The design is released for production.

As an example, consider the design of a 1 kW electric generator, propane-gas fired. This design, SPIKE-0, is intended to provide an AC supply of 60 Hz 120 V at 1 kW maximum power for domestic requirements—lights, small appliances, and battery charging. It is intended to have long life, low noise, and moderately high efficiency, with cost not excessive in comparison to competition (e.g. internal combustion engine–generator set). An overall efficiency of 20 per cent fuel-electricity is acceptable for this application. Assuming the burner-recuperator efficiency to be 75 per cent and the alternator 80 per cent, then the required engine thermal efficiency is $(0.2)/(0.8 \times 0.75) = 33$ per cent.

A crude size estimate based on a power parameter guess of 0.10 watts/cc Hz P_0 gives a 25 bar engine with 80 cc swept volume at 2 cm stroke (a convenient stroke for the 60 hz alternator) to deliver about 1200 watts to the alternator. From experience it is known that displacer phase angle should be about 40° leading the piston, and displacer amplitude should be approximately equal to piston amplitude. Using these as the initial starting conditions for optimization, the computer automatically searches among the allowed variables of pressure, phase angle, heater, cooler, and regenerator characteristics for a maximum efficiency point.

The mechanical layout for this design is shown in simplified form in Fig. 11.14. It incorporates a wound field alternator, gas bearings on moving components, and a thin-walled displacer pressurized from cycle pressure peaks. Shuttle heat transfer is kept low by a gap between the displacer and cylinder wall and axial conduction is minimized by the thin displacer cylinder and regenerator walls.

DESIGN PROBLEMS IN FREE-PISTON ENGINES

Piston centering

It usually comes as a surprise to practitioners new to free-piston engine design that the pistons tend to work their way gradually away from the

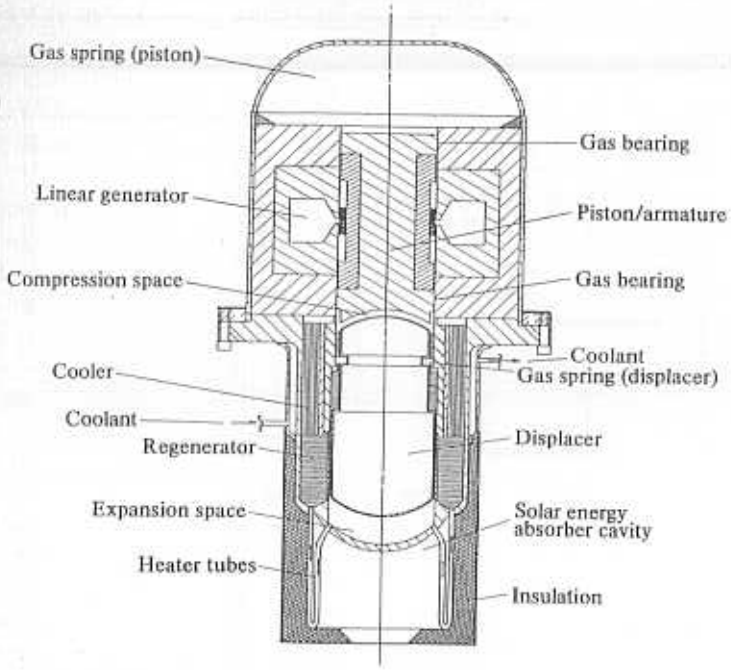


FIG. 11.14. Drawing of SPIKE engine design.

desired operating point, usually toward the working space, ending up rattling madly against the hot end or whatever mechanical impediment first intervenes. The designer's usual first reaction to this perverse behavior is to put a mechanical spring in the system intended to force the piston to behave. But this seldom works since the spring required is usually far too stiff for practical application and/or the piston capriciously decides to wander too far in a direction opposite the one it first sought.

Upon reflection, the designer will discover that there is a non-linear flow in close annular gaps proportional to the differences of the squares of the pressures, and that because the pressure wave is not a pure sinusoid, but is a bit peakier at the high pressure than at the low, a resulting net loss of working fluid from the working space results, and the piston creeps toward the working space.

If other seals than close fits are used—rings, cups, or the like—then the problem is even worse, since these seals tend to be quite unpredictable and will cause piston creep one way or the other in an arbitrary manner deeply frustrating to the test engineer. The reason is that an ordinary piston ring will move somewhat in its groove and will seat best on one

side or the other depending on the chance motion of wear particles or the alignment of minute imperfections.

There are fortunately several effective solutions available, the simplest being merely a centre-point leak deliberately provided to allow a brief but large flow of gas between working space and bounce space at or near the desired centre position of the piston (Fig. 11.15).

This method is to be preferred for centering gas-spring displacers where it is both efficient (low loss) and effective. It is less efficient when applied to power pistons since it allows leaks across a significant pressure difference at both in and out passages by the port and hence causes unrestrained expansion loss. The sudden expansion loss does not occur in the gas spring nearly so much since there is little or no difference in pressure between in and out passages past the port in a properly designed gas spring and the centre port will soon equalize the mean gas spring pressure with that of the bounce space or other space to which it is referenced.

A preferred non-active method of piston centering is shown in Fig. 11.16 in which the piston uncovers a hole allowing a correcting flow only if it reaches a limit of desired travel. Engines designed with close fit pistons (no rings) may be kept from moving too far in by only one such hole and check valve arrangement, but if the piston seals are such as to allow drift in either direction, then two limit circuits are required.

A third method, and one preferable for high performance systems, employs an active control loop comprised of a position sensor, a logic circuit, and a control valve admitting gas to or from the various spaces as desired. In Fig. 11.17 for example, the piston position is sensed by a non-contact sensor—capacitive, magnetic, or the like. The resulting signal is contemplated by a microprocessor and compared with the desired operating condition, and a signal is duly sent to a spool valve which adjusts its position to give the desired change in flow in or out of the working space. This method is capable of allowing long-term centering

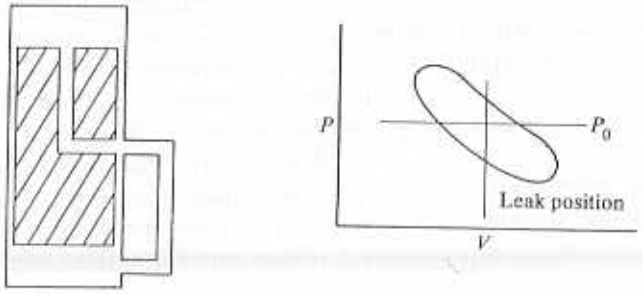


FIG. 11.15. Leak technique for centering the piston.

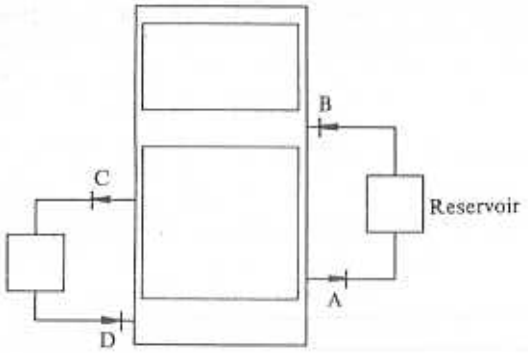


FIG. 11.16. Store-and-dump technique for centering the piston.

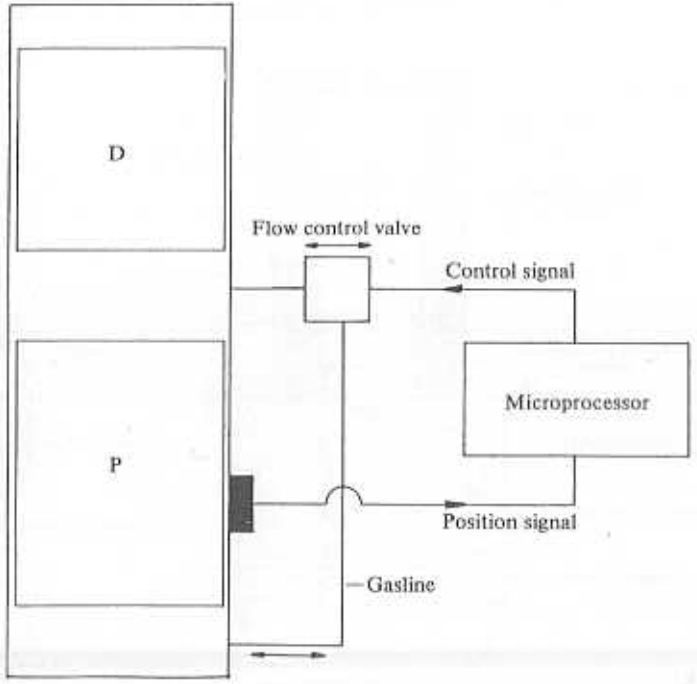


FIG. 11.17. Active method for centering the piston.

without being unduly affected by wear, distortion of the components, or change in fluid properties, all of which can disturb the simpler non-active control methods. The active method is also adaptable to power modulation.

Gas lubrication

Gas lubrication may be arranged by use of the natural pressure swings in the working space or gas springs to pump up a gas bearing reservoir through a check valve. Or an independent pump especially designed for the purpose may also be used, although this method usually adds unnecessarily to complexity. Gas bearing surfaces must be hard and very closely fit; a 50 mm piston, for example, could employ an alumina surface with clearance gap of 8 microns over a 20 mm length.

Mechanical interferences

The mechanical interference problem is best avoided by designing the engine and load so that bumps do not occur over the entire range of loading. If this is not possible, dashpots may be included in the design to prevent destruction of the parts, or a bypass port between working space and bounce space may be opened by an appropriate sensor, and power then dissipated.

Power modulation

Power modulation is often not necessary, as in a water pumping or other constant-load application, but when it is needed it may be done by changing the displacer spring rate, or damping, or by changing piston average position via the piston centering control. Fig. 11.18 shows mechanical arrangements to do this. Displacer spring changes have been found to give greater change in power for less loss of engine efficiency than do displacer damper changes, although the latter is easier to do mechanically.

Power drops quite rapidly as the piston mean position is moved away from the hot end, as is readily predicted from simple isothermal analysis. This method works only with some loads which will permit any position of piston. Alternators usually require a fixed piston average position and are less readily adapted to power modulation by this method.

Starting

The free piston engine will self-start if a sufficient temperature difference exists, and low enough friction forces are present to be overcome by environmental perturbations. Test engines made with close-fit hard surfaces have consistently self-started if mounted in a vertical position and if the displacer was supported from resting on the bottom by magnetic or mechanical springs.

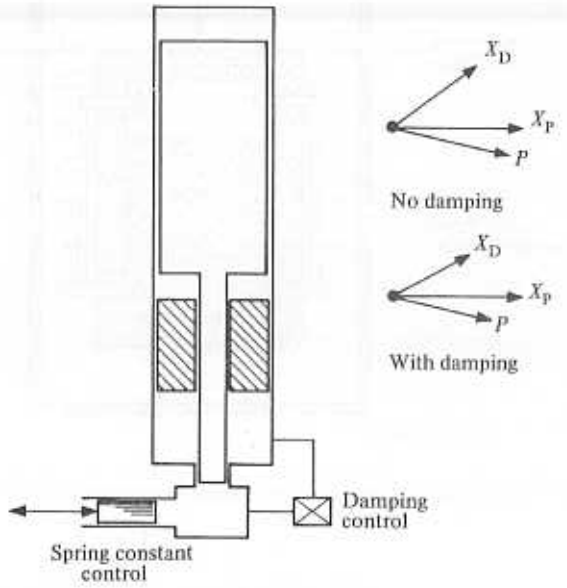


FIG. 11.18. Power modulation by damping or spring.

In any circumstance, properly designed free-piston engines are extremely easy to start by a small applied vibration. Alternator engines may be started by a slight amount of power fed through the electric circuit.

Solar engines mounted on a sun-tracking mount will self-start as a result of the changing inclination to the gravity field, but in order to do so the piston must be suspended above bottom, preferably by a relatively soft and friction-free magnetic spring.

The flame-induced vibration and rapid temperature rise of liquid or gas fired engines gives them a good means for self-starting.

One definite impediment to self-starting is a massive hot-end heat exchanger, especially those containing unheated passages which act to cool the first flow of gas through them and thus reduce the pressure rise during the starting cycle.

ADVANTAGES OF RECIPROCATING POWER

The majority of engineers customarily think of rotating shafts as the embodiment of mechanical power. For this reason the free-piston engine with its pure reciprocating motion may seem strange and relatively useless. Yet there are many virtues to reciprocating motion not possessed by rotating power plants. The reciprocator has no side loads as are

imposed by conventional connecting rods, and as a consequence, its piston lubrication problem is much less severe, and may readily be accomplished by gas bearings using only a very small fraction of developed power to pump the gas.

Reciprocators may also use flexing seals which are hermetically shut with zero leak diaphragms, hollow-spring feeds, torsion tubes or the like. With such seals there need be none of the seal or lubrication problems which plague crank machines.

But one might ask, what use is a pure reciprocator, regardless of its other virtues, since conventional loads require a rotating shaft? This objection is readily dismissed if one notes that often a load with a rotating input contains a crank and piston mechanism to return rotary to reciprocating motion. For example, a conventional internal combustion engine driving a positive displacement compressor first turns reciprocating motion to rotating motion in the engine, then transmits power through the rotating shaft to another crank to a reciprocating piston which finally compresses the gas. A free-piston engine simply transfers work from the power cycle directly through a single reciprocating piston to the compressed gas, thus eliminating lubrication, bearings, windage, wear, and much ironmongery and cost.

RECIPROCATING LOAD DEVICES

There are a few reciprocating load devices which merit particular consideration for use with free-piston Stirling engines. The first is the linear alternator, which can be made in many different ways, some of which are illustrated in Fig. 11.19. In that figure, (a) is a moving-coil type similar to the commonly-used loudspeaker drive. The chief advantage of the moving-coil alternator is that it has no side forces, there being no ferromagnetic components in the coil, but only conductors upon which an axial force is induced by the interaction of the current and the magnetic field at right angles to it. The Sunpower model 10-B uses this alternator, with rolling shim stock for the electric current.

Alternator (b) uses a flux-switching moving plunger which causes a complete reversal of the flux direction through the winding as it moves from one end of its stroke to the other. By this means the maximum power output per unit of flux may be generated. There are side forces on the plunger which must be kept to reasonably small values by accurate centering of the plunger. Gas bearings are required to resist the side force and prevent rubbing contact which could quickly wear out the sliding surfaces.

Another uniquely suited load for the reciprocating free piston engine is the inertia compressor or pump, Fig. 11.20. This uses a flexing hollow spring as the means of feeding the pumped fluid to and from the

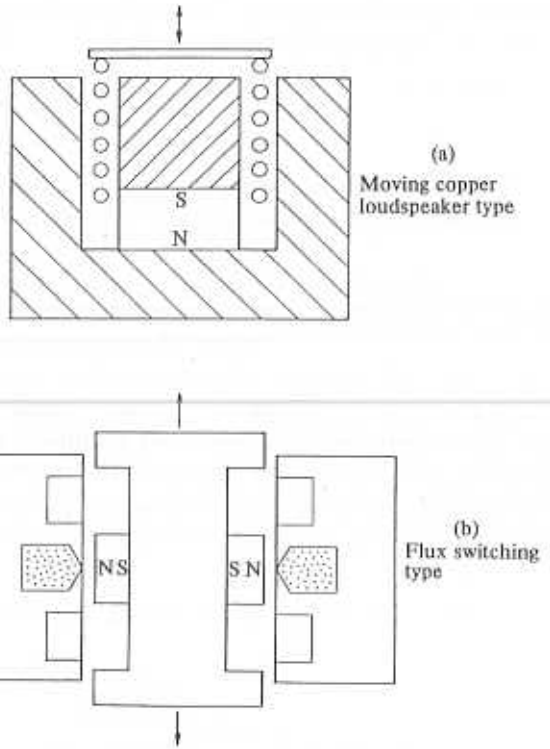


FIG. 11.19. Linear alternators: (a) moving-conductor type, (b) flux-switch type.

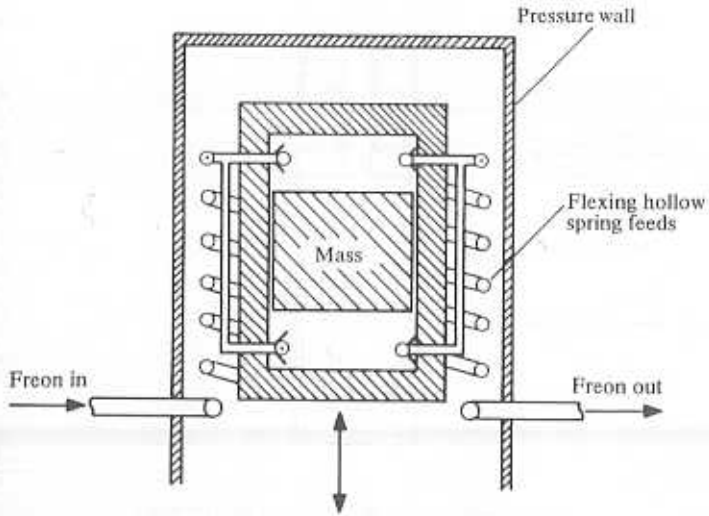


FIG. 11.20. Inertia pump.

compressor through the surrounding high-pressure working gas of the Stirling engine. The compression takes place in the spaces between the relatively stationary inertia mass and the moving enclosure surrounding it, which is attached to or contained within the engine piston. By means of the inertia compressor, a hermetic seal is assured yet a flow of fluid through the compressor carries away power from the engine. This combination may be used as a refrigeration compressor, a water pump, a hydraulic pump, or an air compressor. The flexing hollow spring may easily be designed for very long life and its stiffness is useful to help provide the desired resonant frequency of the engine.

A third important method of removing power from a purely reciprocating engine is the free-cylinder power take-off whereby the engine cylinder is the power transmitting component, rather than the piston, which is made heavy so that its motion is small in comparison with that of the displacer or cylinder. The free cylinder may be directly attached to its load such as an air blower or alternator (Fig. 11.21).

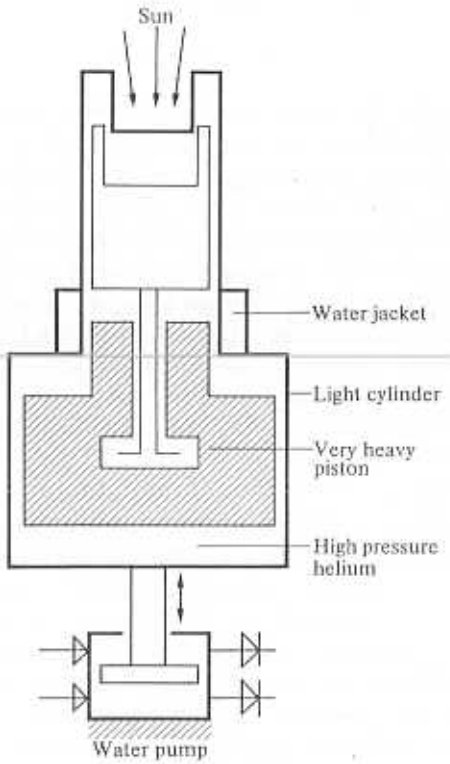


FIG 11.21. Free-cylinder engine.

The free-cylinder engine is probably the simplest hermetically sealed engine, and because of this, is suited to tasks requiring minimum cost and long life such as water-circulating pumps in home heating systems, solar-powered irrigation pumps, or air blowers for hot-air heating systems. Free-cylinder engines, while they appear to be even more unorthodox than free-piston engines and hence even less appealing to many, are not at all intrinsically inferior in performance. In fact, very simple free-cylinder engines have demonstrated thermal efficiency over 10 per cent (Beale *et al.* 1973). They also have the very desirable characteristic that they cannot be stalled out by an overload, since if the cylinder is prevented from moving, the cycle energy is transmitted to the piston, and an increasingly strong reaction force is transmitted to the load. This characteristic makes the free cylinder engine a potential candidate to replace the noisy air hammer as a concrete breaker.

Another hermetically sealed load for the free-piston engine is the Stirling cooling cycle (Fig. 11.22). In this arrangement the power developed by the heat engine is transmitted directly to the cooling cycle, which uses the same working fluid at the same average pressure. Here the

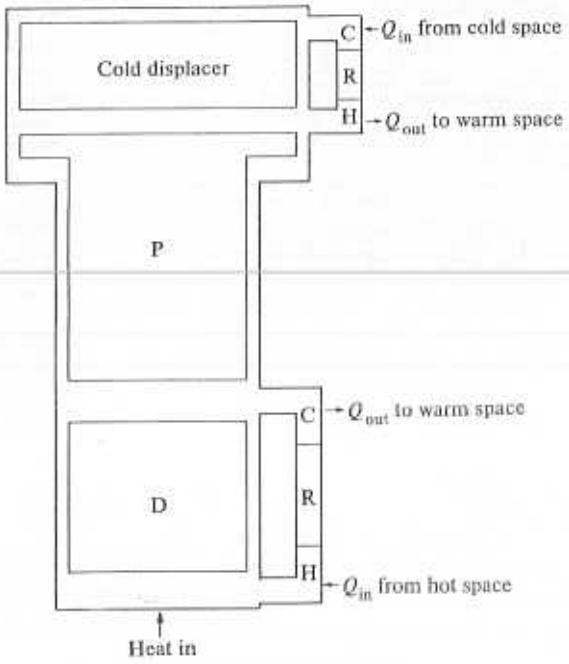


FIG. 11.22. Stirling–Stirling heat-driven heat pump.

entire thermodynamic process goes on in the same pressure enclosure, no mechanical power is transmitted outside, and the leak and lubrication problems are reduced to the relatively trivial ones of piston sealing and gas lubrication. This Stirling-Stirling cooling machine is a very promising one and at this time under-exploited. As far as is known, only a small model, made from two identical Sunpower model 10-B demonstrator engines has been built and operated. It was able to keep the cold end at the freezing point of water despite a poor match of engine and heat pump and the use of only annular spaces around displacers and regenerator in both engines (Fig. 11.23).

Free-piston engines may also drive loads through a reciprocating shaft seal, but if required to do so, they must face the same very difficult problem as does the conventional crank machine—leakage of working fluid past the seal. One such possible application is the hydrostatic vehicle-drive engine, Fig. 11.24. It uses a seal which separates leaked gas and oil and pumps them both back into their respective spaces.

Diaphragm seals are obviously possible, and have been put to use by the McDonnell-Douglas artificial heart design group, by the Harwell group (Cooke-Yarborough *et al.* 1974b) and advocated by Benson (1977b). Close-fit sliding seals lubricated with working gas are believed by the Sunpower design team to be much more versatile.

APPLICATIONS

Artificial hearts

The power required by an artificial heart is about 3 to 5 watts delivered to the blood pump. This power may be delivered in the form of liquid, gas, or electric power, any of which may be provided by the free-piston engine. The liquid power version has been extensively developed by McDonnell-Douglas. The gas power method of energy transfer has been chosen by the artificial heart group at Aerojet General. But the electric output free-piston Stirling engine is probably the simplest and most durable and gives promise of high efficiency. Its mass and reaction forces need not be greater than those of the other candidates. This engine has not yet been tried for artificial heart applications.

Heat pumps

The Stirling-Rankine heat pump is being very actively developed by several large industrial groups and is likely to have an important commercial application in the near future. It combines high overall energy efficiency with long life, low noise, and low pollution, and as a result is a very attractive competitor for the central power-plant and electric heat pumps already established (Auxer 1977) (Fig. 11.25).

Electric power generation

The heat-driven electric power generator is probably the best application for a free-piston engine, since here full advantage can easily be taken of hermetic sealing—nothing penetrates the pressure enclosure but easily sealed electric conductors. Small laboratory models have been commercially available since 1974 and have proven to give adequate performance

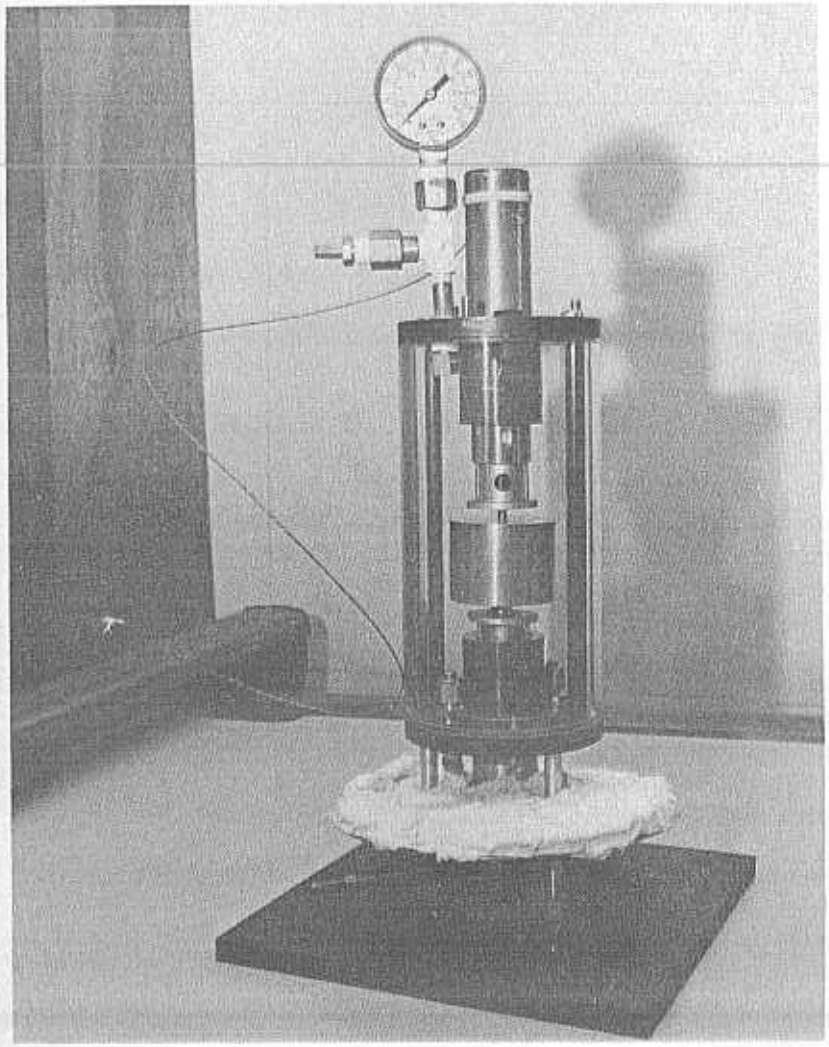


FIG. 11.23. Photograph of Stirling–Stirling heat pump.

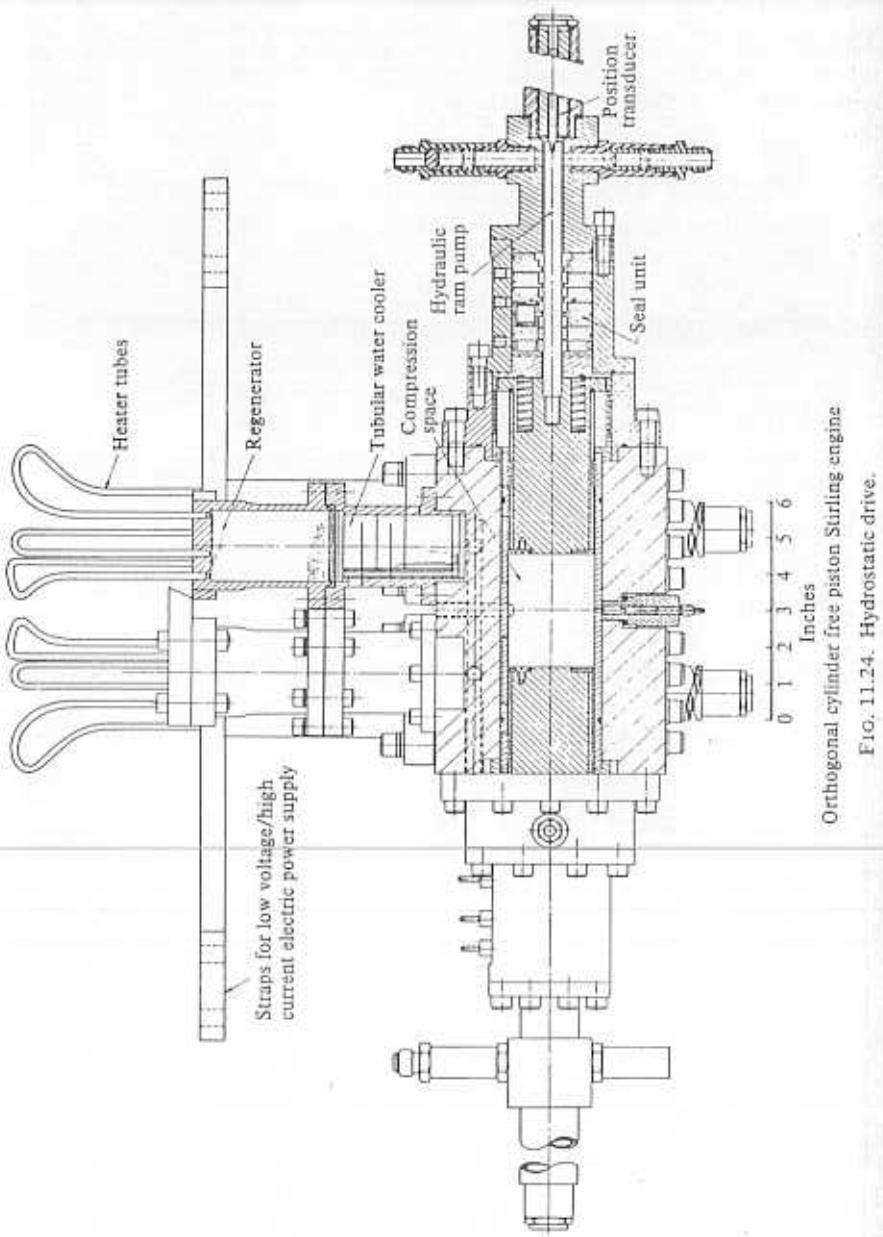


FIG. 11.24. Hydrostatic drive.

and life (Fig. 11.26). Larger models up to 1 kW are coming on the market, and automotive-sized units are likely in the near future. These larger units should have outstanding virtues for domestic applications, since they may be made to be exceptionally durable and almost inaudible. Homes heated with natural gas could derive much of their needed electric power and heat as well from the same amount of fuel which now gives only heat. In fact, with the electric generator combined with a Stirling

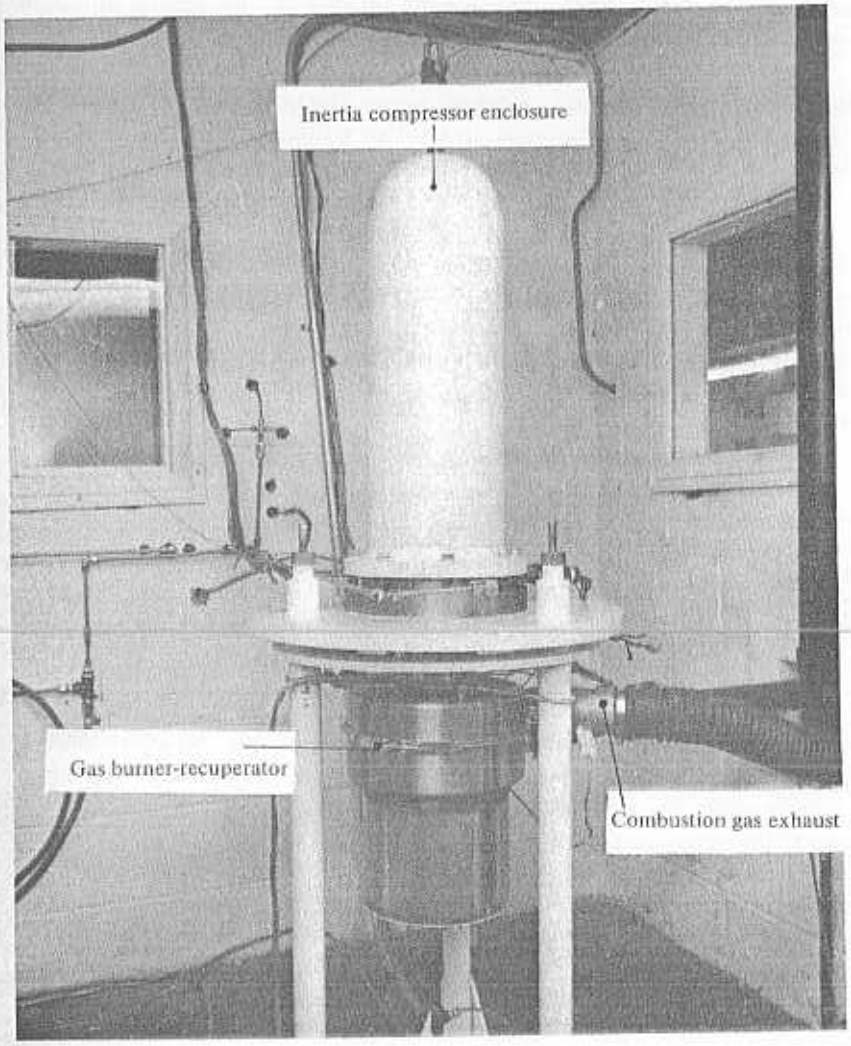


FIG. 11.25. Gas-fired heat pump.

heat pump, heating, cooling, and electric power could be provided for the same or less natural gas or oil which now gives only heat. The promise of this energy conserving mechanism is so great that it will almost certainly be widely used in the near future, now that fuel has become so much more expensive, and the waste of availability represented by combustion heating systems has become widely recognized.

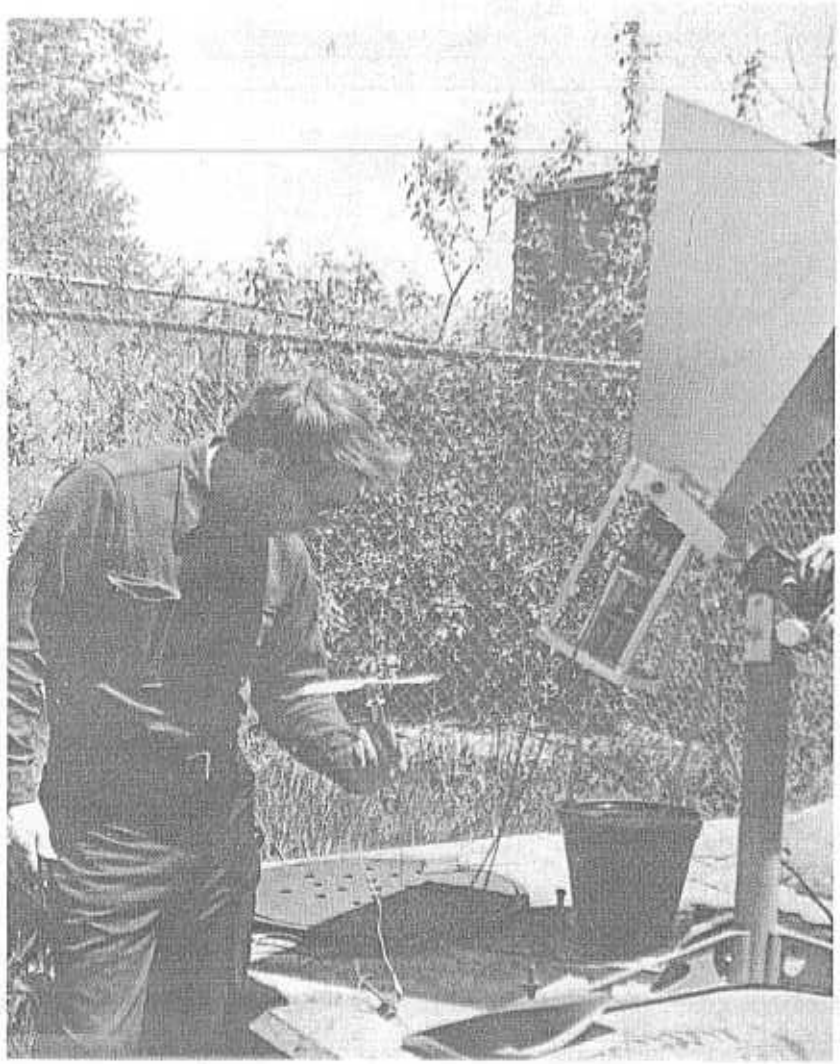


FIG. 11.26. Small free-piston electric generator driven by solar energy.

Solar electric power

The free-piston engine coupled to an alternator is a feasible candidate for large-scale power generation systems using many relatively small 20 kW units mounted at the focal point of a parabolic concentrator. Preliminary studies indicate that system peak efficiencies of nearly 30 per cent are possible with already proven components. High efficiency is vital for solar-electric systems because of the dominant cost of collectors, the engine alternator representing a much smaller fraction of total cost. Made in large numbers, the engines could benefit from economics of mass production (Fig. 11.26).

Third World power

In countries where petroleum products are already prohibitively expensive, and incomes are low, free-piston Stirling engines operating on field waste or solar energy could give useful amounts of power for irrigation, food preparation, cold storage of food and the like (Beale *et al.* 1973).

PERFORMANCE

The free-piston engine is, at this writing (April 1978), undergoing a very rapid development, and new designs benefiting from the Sunpower optimization codes are at present under final assembly but not yet tested. Their performance is expected to be in the 30–40 per cent thermal efficiency range, with hot wall temperatures of 650 to 730 °C, depending on design and application. For example, the MTI-DOE engine is a highly refined machine designed to very close mechanical specifications and intended for use as a space power-plant. It is expected to go on test at MTI in summer, 1978.

The Sunpower SPIKE-0 engine is the first in a series of Sunpower commercial engines designed to provide reliable power below 1 kW for long periods of time, utilizing either propane, kerosene, sunlight, or solid fuels as heat sources.

The model 10-B demonstrator is a very simple model designed as a classroom lecture aid. It uses an annular gap regenerator. Its power and efficiency are low but adequate for its intended purpose. It is also adapted for operation on solar energy using a Fresnel lens concentrator. It has as loading devices an alternator, an inertia water pump or a refrigerant inertia pump.

The Ohio University free-cylinder water pump is an early model constructed in ignorance of the necessity for great care to avoid parasitic losses. Under these circumstances, its performance is remarkable.

AREAS FOR FURTHER WORK

This brief description of free-piston Stirling engines leaves many important questions unexplored. Active areas of research of current interest

to free-piston engine designers include:

(a) Mapping of possible geometries and their ranking according to merit for various applications.
(b) Development of simplified thermodynamic-dynamic analyses for use by designers.
(c) Studies of start-up, dynamics, stability, and load response.
(d) Materials for long-term unlubricated wear.
(e) Efficient position-control devices.
(f) Methods of power modulation.
(g) Load devices capable of efficient matching to free piston engines.
(h) Linear alternator design.
(i) Stirling-Stirling heat-driven heat pumps.
(j) Hybrid-free displacer-crank piston engines.
(k) Analysis of free pistons with working fluids other than ideal gases.
(l) Applications in which free-piston engines would have special competitive advantages.

CONCLUSION

While the practitioners of the crank engine are pursuing the elusive goal of a competitive automotive engine, the free-piston engine rapidly develops toward useful applications unsuited to the dominant internal combustion engines. It is likely that the first significant modern commercial application of Stirling engines will be the Stirling free-piston heat pump and free-piston alternator. No fundamental problems remain, aside from those related to cost and lack of operating experience.

12 PHILIPS STIRLING ENGINES

INTRODUCTION

WORK on Stirling engines has been in progress since 1937 at the Research Laboratories of N. V. Philips Gloeilampenfabrieken, Eindhoven, Netherlands, the large international company well known for electrical and electronic products.

The work on Stirling engines, extending over 40 years, can be broken into distinct phases:

The *initial phase*, from 1937 to 1954, was primarily concerned with the establishment of new ideas and concepts about air engines and the development of small engines.

The *cryogenic phase*, from 1945 to the present (1978) was concerned with the development and production of Stirling engines working as cryogenic cooling engines.

The *rhombic phase*, extending from 1954 to 1970 saw the invention, development, and maturation of single-acting displacer machines with the rhombic-drive mechanism.

The *double-acting phase*, extending from 1969 to the present (1978) saw emphasis of effort on double-acting engines with the swashplate drive.

INITIAL PHASE

Origin

Interest in Stirling engines at Philips sprang from the need to provide a simple lightweight electric-power generator for their radio receivers and transmitters in areas with no electric-power supply. Various thermally-activated systems were considered, including steam engines and thermoelectric generators. Stirling engines were chosen for development because the actual thermal efficiency of the hot-air engines available or known about at the time was so very low compared with the ideal value. Professor Holst, director of the program, thought there was a high potential for improvement in Stirling engines with the application of heat-resistant steels and recent knowledge about heat transfer and fluid flow.

It is said that interest in the hot-air engine was stimulated by a visit to the Museum of Technology in Paris by one of the Philips engineers, where he saw hot-air engines exhibited. The early history of the program has not been extensively recorded, however, and this may be apocryphal. It is known that work continued during the German occupation of Eindhoven in the Second World War, but progress was hampered by

the need for secrecy and the lack of heat resisting steels and other materials.

Early publications

The first public announcement of the program, essentially the *renaissance* of the Stirling engine, was two publications which came out in 1946 (Rinia 1946a), Rinia and du Pre 1946). These were followed a year later by two other papers (de Brey *et al.* 1947, and van Weenan 1947). These four papers were in fact the only source publications on the initial phase of the work, but together they represent a substantial foundation for all that was to follow. They excited attention world-wide at the time of their publication and even today are essential reading for those with a serious interest in the subject.

The theory presented by Rinia and du Pre (1946) followed that developed much earlier by Schmidt (1861). In the initial paper mention was made of engines capable of operating at 3000 revolutions per minute with 'most satisfactory figures for weight and efficiency'! Most of the discussion was about 'air engines', but, in closing, mention was made of the use of Stirling engines as refrigerating machines with alternative working fluids. One experimental engine with hydrogen as the working fluid had attained a temperature of 80 K (144 °R).

De Brey *et al.* (1947) discussed considerations for the development of high-speed, high-output engines in terms of the variable parameters in the engine, including the cycle pressure, speed, and design of heater, cooler, regenerator, and air preheater. Van Weenan (1947) discussed some aspects of construction of Stirling engines including single-acting two-piston and piston-displacer engines and double-acting machines. Photographs were included of an experimental single-cylinder piston-displacer engine, of about 0.7 kW (1 hp) at 2000 revolutions per minute (reproduced in Fig. 12.1) and the four-cylinder double-acting engine, shown in Fig. 12.2 delivering 11 kW (15 hp) at 3000 revolutions per minute. Further papers were promised containing details of the test results on these engines, but unfortunately these were never published. In his paper, van Weenan mentions that double-acting engines with a wobble-plate drive were thought to be limited to 15 to 22 kW (20 to 30 hp) capacity. For higher powers other driving mechanisms had been worked out based on an arrangement of 'the cylinders in V form one behind the other'. An early engine of this configuration is shown in Fig. 12.3.

Double-acting engines

The existence of double-acting engines at this early stage in the Philips work is remarkable. The principle was described by Rinia (1946a) but the first discussion of substance was given by van Weenan (1947) along with the photograph of the high speed 11 kW (15 hp) engine. Even more remarkable, however, is the historical fact that double-acting engines were invented nearly a century before. Babcock (1885) ascribes the original double-acting engine to Charles Louis Franchot, who patented

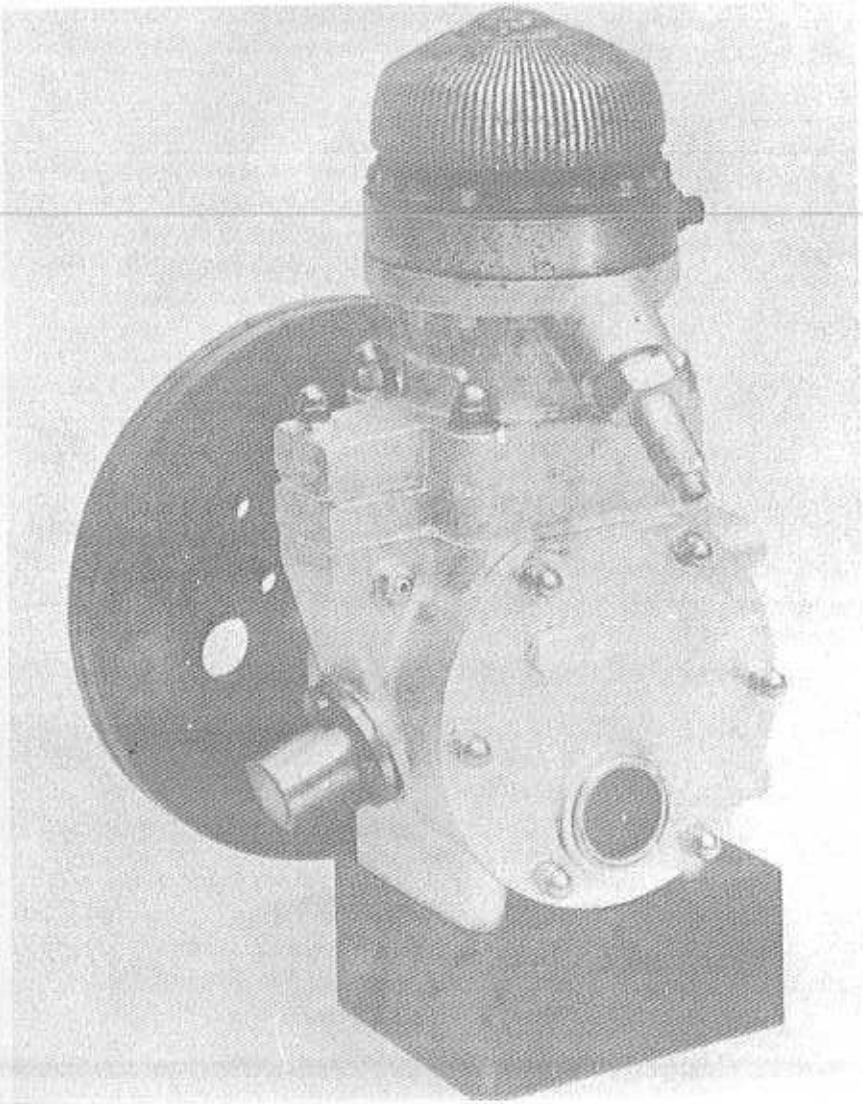


FIG. 12.1. Single-cylinder crank-drive displacer-type air engine, (a) general view, (b) diagrammatic cross-section (after Meijer 1969a).

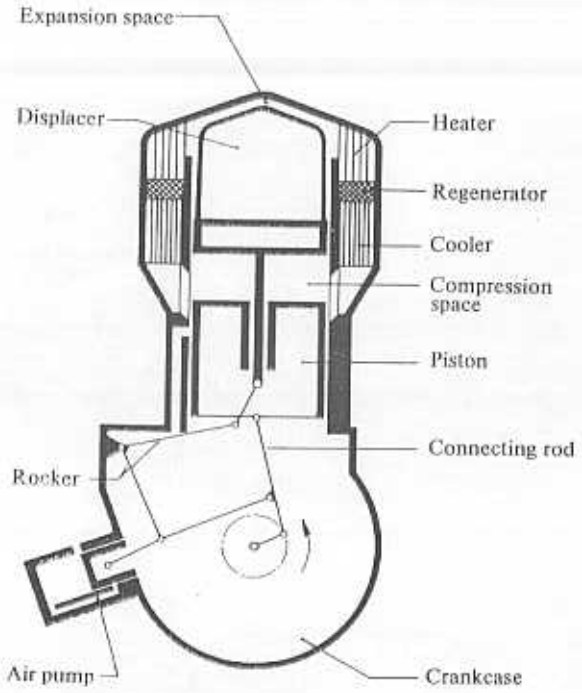


FIG. 12.1(b).

the double-acting two-cylinder configuration in 1853. Babcock (1885) also describes the invention, patented in 1860 by Sir William Siemens, of an engine having 'four cylinders each, hot at one end and cold at the other, all connected to one shaft through a wobbling disc at equal quarters of the revolution and so arranged that the hot end of one communicated through an economizer with the cold end of the other in order'.

This describes exactly the multiple-cylinder double-acting Philips engines of the 1946 era and also the present-day machines developed following a return to the double-acting engine in the 1970s. So far as is known Sir William did not reduce his novel concept to practice. He was prolific with ideas and concepts but his perception was truly remarkable. Babcock in his account provides a prophetic comment on the promise of the Siemens engine for development. It is not known if Siemens' engine was known to Philips or if the engine was reinvented 80 years after the original.

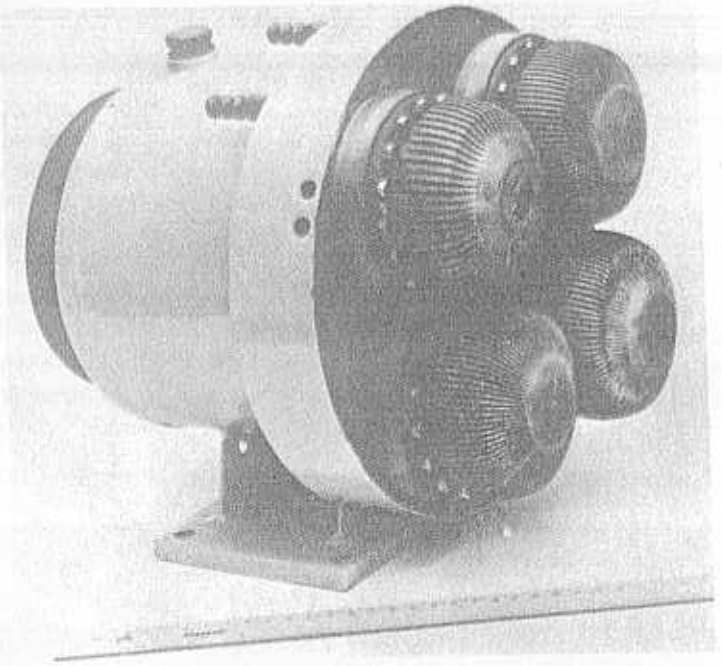


FIG. 12.2. Four-cylinder double-acting air engine with wobble-plate drive mechanism (after Meijer 1969e).

Small power generators

It is difficult to overestimate the accomplishment of the Philips engineers in this initial phase of the program. As Ross (1977) says:

'In the few short years from their first tentative interest in 1937 until these articles in 1947 and with a World War in between Philips had taken the Stirling engine and increased its power per pound by a factor of fifty, reduced its size per unit of power by a factor of 125 and increased its speed by a factor of ten.'

Work on the small fractional power engine generators continued in the late 1940s. Market surveys indicated the need for a self-contained electric-power generating set of 150 to 200 watts (0.2 to 0.27 hp). Eventually, after much detailed engineering work, the Type 102C shown in Fig. 12.4 evolved. This unit was to receive considerable development including a duration test of 2000 hours without maintenance, high-altitude testing on the Jungfrau Joch in the Swiss Alps, in thin air at temperatures of −23 °C (−9.4 °F). Low temperature tests to −40 °C(−40 °F) were also carried out. Ross (1977) describes one test in which an engine with water cooler and other modifications to increase the power to 2 kW (2.7 hp) was fitted to a row boat and motored for 80.5 km (50

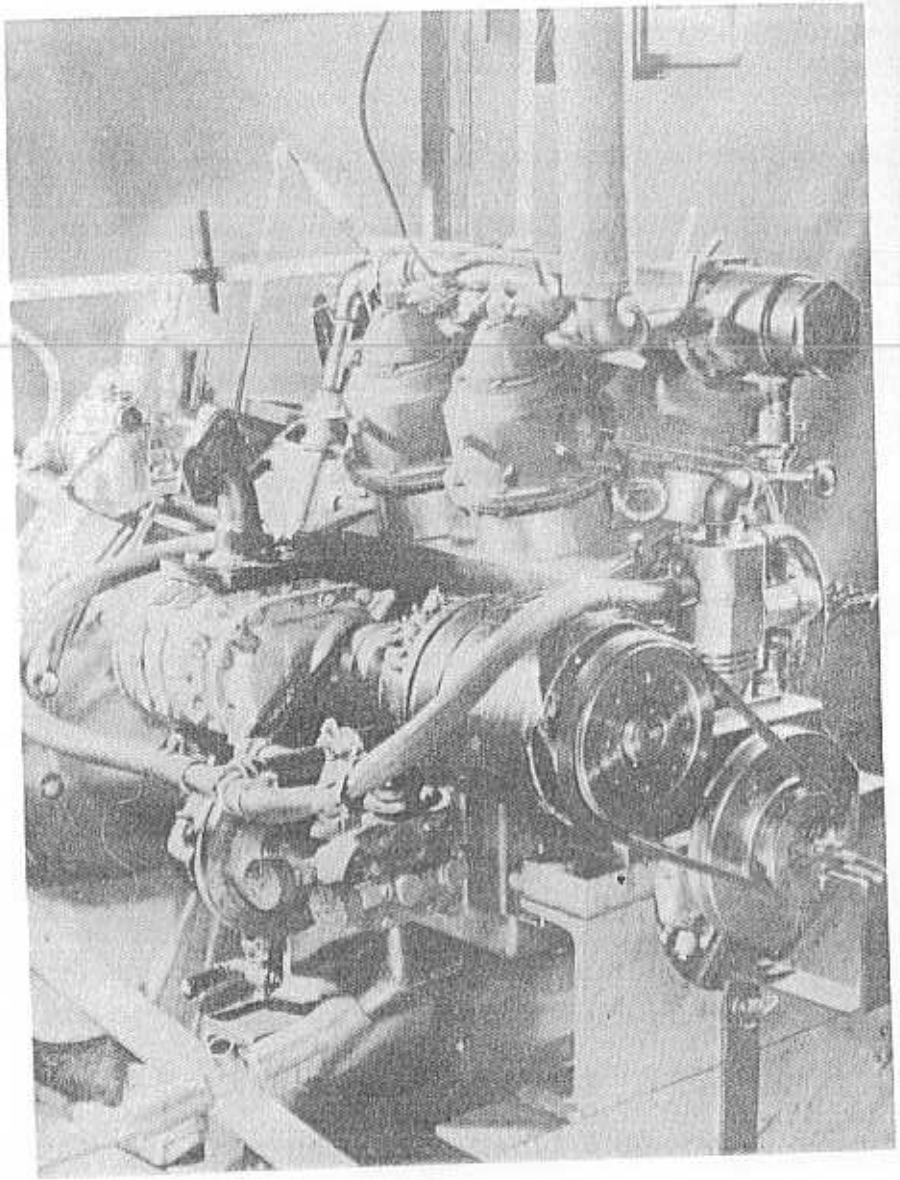


FIG. 12.3. Four-cylinder double-acting air engine with crank-drive mechanism (after Meijer 1969e).

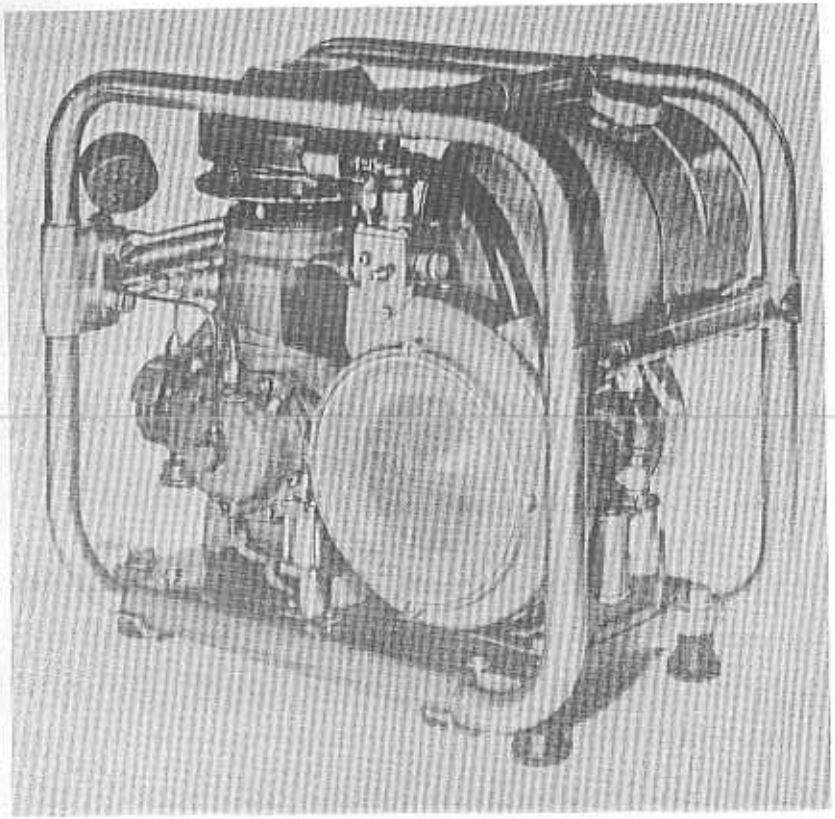


FIG. 12.4. Type 102C Stirling-engine electric-power generating set (after Meijer 1969e).

miles) at a speed of 4.6 m/s (9 knots) on the Dutch canals. Viewers on shore were apparently startled at the sight of the boat with several passengers moving without noise through the water with no visible means of propulsion. The engine was concealed under a cardboard box.

The decision was made in 1952 to start production of the 102C generator sets. Initial production was set at five series of 50 engines. In the end about 100 were made before production was stopped in 1954 and further effort was concentrated on larger engines. The reason was not so much the fault of the engine as the lack of a market.

The original application for radio sets had largely disappeared with the invention of the transistor, the development of electric storage batteries, and improvements in radio valves. No substitute market could be found or foreseen and finally, about 1958, the Type 102C generators were dispersed to universities and technical colleges in Europe.

An unexpected application of the unit was made in February 1953 as part of the relief work at the time of the great sea floods in Holland. A contemporary newspaper report filed from the inundated village of s'Gravendeel describes how seventeen of the generators from the Philips factory at Dordrecht were used for emergency lighting in hospitals, and refugee centres and for amateur radio installations providing communications for the Red Cross. One flat-bottomed boat with two hot-air engines for propulsion carried blankets and foodstuff for a thousand people!

The engine of the Type 102C generator perhaps represented the acme of air engine technology, for subsequently all development was concentrated on larger engines using helium or hydrogen as the working fluid. The performance characteristics of the Type 102C engine were never published by Philips nor, surprisingly, by any of the recipients of the engine after their final dispersal. It was left to Ward (1972) to conduct careful measurements of the engine modified to operate with water cooling and on liquid petroleum gas. The operating characteristics were finally published by Walker *et al.* (1978) and are reproduced here in Chapter 9. The U.S. Navy developed an inquisitive interest in the small engines developed by Philips. Several engines were acquired and tested with results reported by Schrader (1951).

Many other varieties of engines were also considered by Philips. Fig. 12.5 shows a configuration of opposed piston engine that was investigated experimentally.

Little is known of the 11 kW (15 hp) double-acting engine. A contemporary newspaper report contains an account and photograph of a demonstration of the engine to Henry Ford II in 1948. Percival (1974) recounts how the attempt in 1950 by General Motors to form a working agreement with Philips was rebuffed as premature.

Meijer (1959a) identifies the reasons for the failure to persist with the development of the double-acting engine as 'an exceptionally intractable lubrication problem', thought to be the dry-rubbing piston seals in each cylinder separating the hot and cold spaces. The similar problem in displacer engines was eventually overcome with the development of Rulon seals and this permitted the resumption of work on double-acting engines. Another less significant factor was the aerodynamic and thermodyamic penalties of this type of engine. The volume variations in the hot and cold spaces could no longer be chosen freely as regards magnitude and phase.

CRYOGENIC PHASE

Köhler (1965) has recounted how one of the small Philips air engines was found to operate effectively as a refrigerator. With hydrogen as the working fluid the cylinder head (expansion space) cooled to such a low

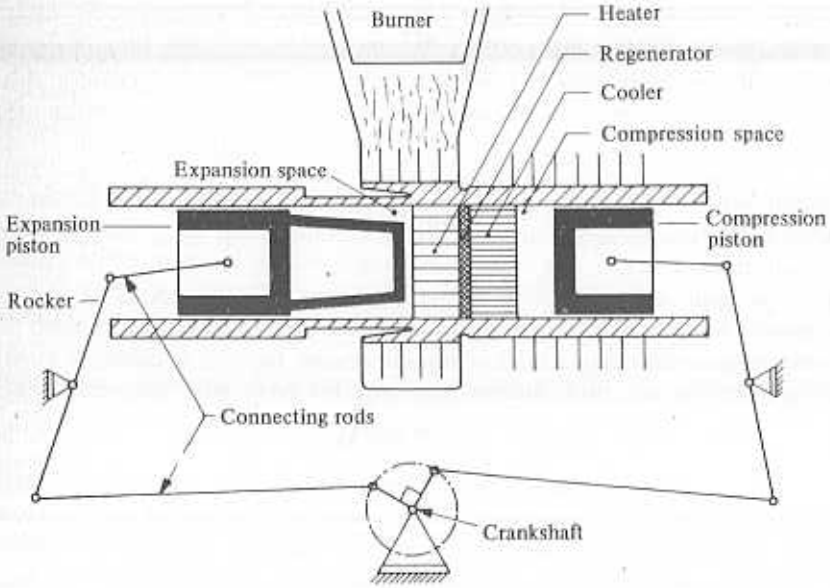


FIG. 12.5. Configuration of opposed-piston engine.

temperature that atmospheric air condensed upon it. This led to the formation about 1948 of a research group, headed by Köhler, to develop the cryogenic cooler. This team was brilliantly successful and the first cryogenic cooling engines were being sold in 1954. These machines produced liquid air at the rate of about 2.22 cm^3/s (8 litres per hour) Köhler and Jonkers (1954a and b) have provided a full account of the fundamentals and construction of the gas refrigerating engine.

This machine and other later developments were manufactured in large numbers and have been sold all over the world. It may be considered the outstanding commercial success for modern Stirling engines.

Further details of the Philips work on refrigerating machines and associated cryogenic equipment are not included here because of limited space, but there are many important contributions from Philips in the cryogenic literature. The material is fully covered in a forthcoming companion work to this volume entitled *Regenerative Cooling Engines*.

RHOMBIC PHASE

A fresh start

The rhombic drive for Stirling engines was invented by Rolf Meijer in 1953. Its adoption in 1954 for all engines represented virtually a fresh start on Stirling engines for Philips.

A first account of the new engine configuration was given by Meijer (1959a)—some twelve years after the final paper by van Weenan about the early air engines. Meijer described the single-cylinder rhombic-drive engine of 30 kW (40 hp: the Philips Type 1-365) shown in Fig. 12.6. This engine had a cylinder bore of 88 mm (3.46 in), a piston stroke of 60 mm (2.36 in), a speed of 1500 revolutions per minute and it used hydrogen as the working fluid at nominal heater and cooler temperatures of 700 °C (1290 °F) and 15 °C (60 °F). The mean pressure was 10.3 MN/m^2, (1500 lb per sq in) the maximum pressure was 13.7 MN/m^2 (2000 lb per sq in) and the compression ratio p_{max}/p_{min} was two. Various operating characteristics of the engine, reproduced in Chapter 9, were given for output and efficiency as a function of the pressure of the working fluid and the engine speed. The effect of heater and cooler temperatures on engine power and thermal efficiency at constant speed and mean pressure were also given. The paper included a comparison of the fuel consumption contour chart, in terms of mean effective pressure and speed for the new Stirling engine and a commercial diesel engine.

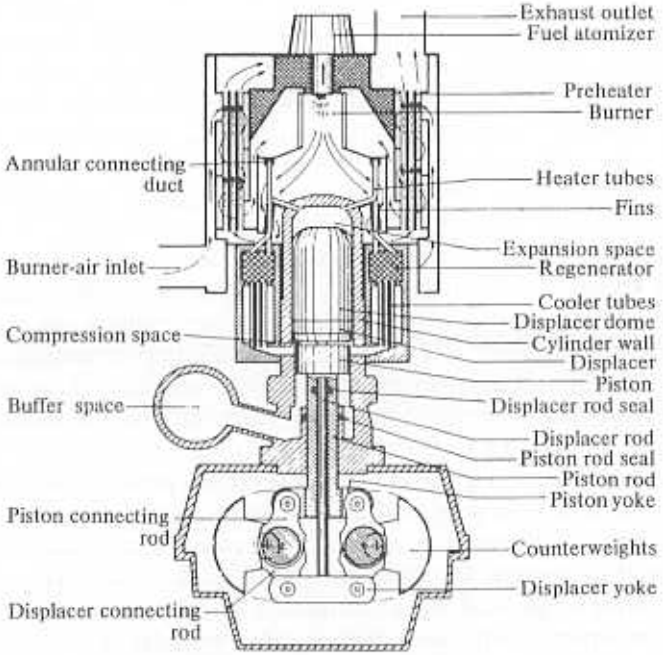


FIG. 12.6. Cross-section of rhombic-drive engine (after Meijer 1969e).

Rhombic drive

The same data, but in considerably more detail, was given by Meijer (1960a) in a reprint of his doctoral thesis. This is a most valuable document, being a comprehensive mathematical analysis of the dynamics of the rhombic drive. Some features are discussed in Chapter 6. A similar kinematic arrangement for reciprocating engines was invented by the renowned British automotive engineer, Lanchester in 1893 (Crabtree 1976). The Lanchester drive for horizontal flat twin internal-combustion engines is so remarkably similar to the rhombic drive, that it must be considered a blood relation even if not truly the grandfather of the rhombic drive. All the essential features are there, just as the Siemens engine predated the double acting engine by 80 years.

Tubular heater

The new rhombic-drive engine was dramatically different to the earlier small air engine. A cross-section of the rhombic-drive engine is shown in Fig. 12.6. The heater, also shown in Fig. 12.7, consisted of a complicated tubular structure with a horizontal annular manifold at the top. Alternate heater tubes were connected to the cylinder head and to the regenerator housings with the tubes to the regenerator housings provided with fins for improved heat transfer.

Split regenerator

The regenerator on the new engine was divided into ten units contained in small cups instead of being a single annulus around the cylinder. This overcame the problem of thermal stress distribution of earlier designs. Three heater tubes were connected to each regenerator cup.

Water cooler

Water cooling was used on the new engine with the working fluid passing through a tubular cooler mounted below the regenerator cups. The design of the cooler was adapted to that of the regenerator so that a number of tubes were bunched together in groups, the ends of each cooler unit protruding into a regenerator cup. The tubes were mounted with a sliding fit in the cooler housing so as to allow unrestrained axial movement of the regenerator cups due to thermal movement of the heater tubes.

Air preheater

The heater was surrounded by the exhaust-gas/air preheater shown in Fig. 12.8. This was a recuperative type heater with spirally curved plates forming narrow passages through which exhaust gas and fresh air passed in alternate channels.

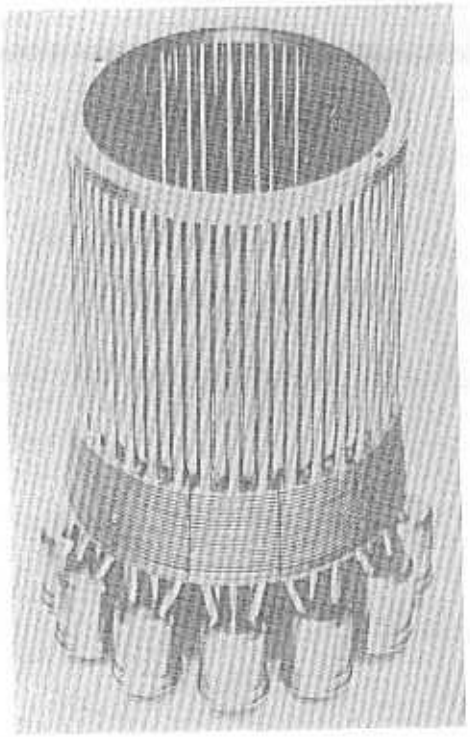


FIG. 12.7. Heater assembly for rhombic-drive engine (after Meijer 1969e).

The burner arrangement is shown in Fig. 12.8. The single burner was centrally located within the ring of heater tubes and was of the 'swirl-chamber' type suitable for use with a variety of gaseous and liquid fuels. An atomizer was used for operation with liquid fuels.

Power control for the engine was obtained by a hydraulic governor. This adjusted the pressure of the working fluid in the engine to maintain a pre-selected speed of operation. A thermostat control was included to regulate the fuel supply to maintain the heater tubes at a constant maximum temperature. These controls are described in more detail in Chapter 10.

Gas seals

To save weight, the crankcase of the rhombic-drive engine was not pressurized and so the question of sealing the reciprocating piston and displacer was critical. Seals were required against the egress of working

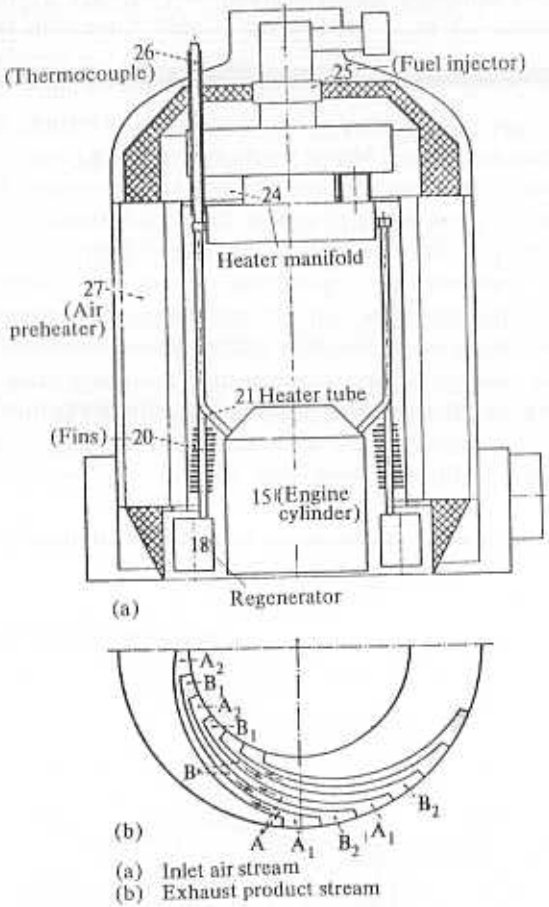


FIG. 12.8. Cross-section of hot-parts assembly of the rhombic-drive engine, (a) vertical section, (b) horizontal section.

fluid (hydrogen) and also the ingress to the working space of crankcase lubricant. Seals are not mentioned in either of the papers by Meijer given in 1959 and 1961 but some details of Philips contemporary seal practice have been given by Percival (1974).

On the early air engines it was found that carbon piston-rings similar to those used in air compressors resulted in unacceptable levels of contamination of the heat exchangers, and probably blockage of the fine interstices of the regenerator. Therefore, on the displacer of the air engines, a close clearance wear band was used. For the power piston a combination

of three to five conventional iron piston rings was found to be adequate for reasonable life but it was difficult to prevent the ingress of oil to the working space with consequent contamination and blockage of the regenerator.

Development of the rhombic drive eliminated the side loading of the piston against the cylinder wall that is characteristic of conventional crank-slider mechanisms. Because of this, piston rings were not required on the piston for sealing and guiding. Instead from about 1955 to 1960 Philips developed close tolerance seals for both the power piston and the displacer. Percival (1974) describes these as a tin-lead alloy band with circumferential grooves and treated with molybdenum disulphide. The piston band was machined slightly oversize then shrunk in a dry ice bath for initial fitting. The piston was then honed into the cylinder by motoring the crank-piston assembly for several hours. Percival remarks that the procedure proved to be more of an art than a science and usually had to be repeated if the piston was removed for any reason. At best the seals were effective against leakage and had a slightly higher mechanical efficiency than piston rings.

Rolling seals

Clearly such a procedure was unsuitable for anything other than special applications. An apparent solution was at hand with the invention of the rolling seal described by Rietdijk *et al.* (1965). Work on rolling seals, described also as 'positive seals' started at Philips in 1960. The result is shown in Fig. 12.9. It consists basically of a thin membrane of flexible material held by clamping rings against the stationary part and against the moving piston or rod. A small pressure difference across the seal was sufficient to hold it snug against the piston or cylinder wall so that it rolls off these surfaces without creasing. Fig. 12.10 is a photograph of a rolling diaphragm.

By this time the working fluid pressures were as high as 14 MN/m^2 (2000 lb per sq in) of hydrogen. It was simply not possible to contain this pressure across a diaphragm that was sufficiently thin and flexible to act as a rolling seal. Therefore the system shown in Fig. 12.11 was adopted. The flexible diaphragm was supported on a cushion of oil and the pressure of the oil was varied in sympathy with the gas pressure variation in the cylinder. A small pressure difference of less than 0.5 MN/m^2 (75 lb per sq in) was maintained across the diaphragm and a high pressure difference was taken by an oil seal of conventional type separating the oil-filled space beneath the diaphragm from the crankcase. This was a much less demanding seal problem. To maintain the oil space at constant volume as the piston moved in the cylinder, it was necessary to 'step' the

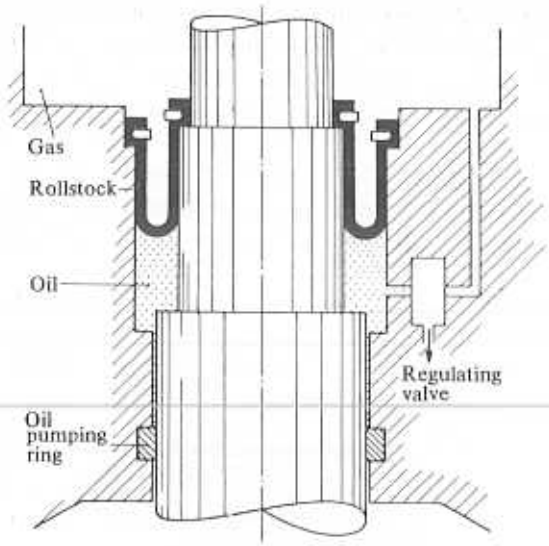


FIG. 12.9. Rolling seal.

piston and cylinder such that:

$$d_m^2 = \tfrac{1}{2}(d_a^2 + d_b^2) \qquad (12.1)$$

where d_m, d_b and d_a are piston or cylinder diameters as shown in Fig. 12.11.

The success of the rolling seal was absolutely dependent on the material used. The requirements were a high fatigue strength, high creep resistance, and resistance to chemical attack by oil or hydrogen. Promising results were obtained with a polyurethane rubber. In rig tests it was found that endurance was largely dependent on three parameters, temperature, pressure difference across the seal, and the ratio of the diaphragm thickness and the piston/cylinder wall clearance. Temperature was found to be a critical parameter. Seals running at 1500 revolutions per minute would endure for over a year (10 000 hours) at a temperature of 25 °C (77 °F) but would fail at 150 hours if the temperature was raised to 100 °C (212 °F). This was attributed to the strong dependence on temperature of the tensile strength of the seal material. At 100 °C (212 °F) it had only 20 per cent of the tensile strength at room temperature.

Basic designs of displacer-type Stirling engines with rhombic-drive mechanisms and using rolling diaphragm seals are shown in Fig. 12.12 (Rietdijk *et al.* 1965). The single-cylinder machine required four seals

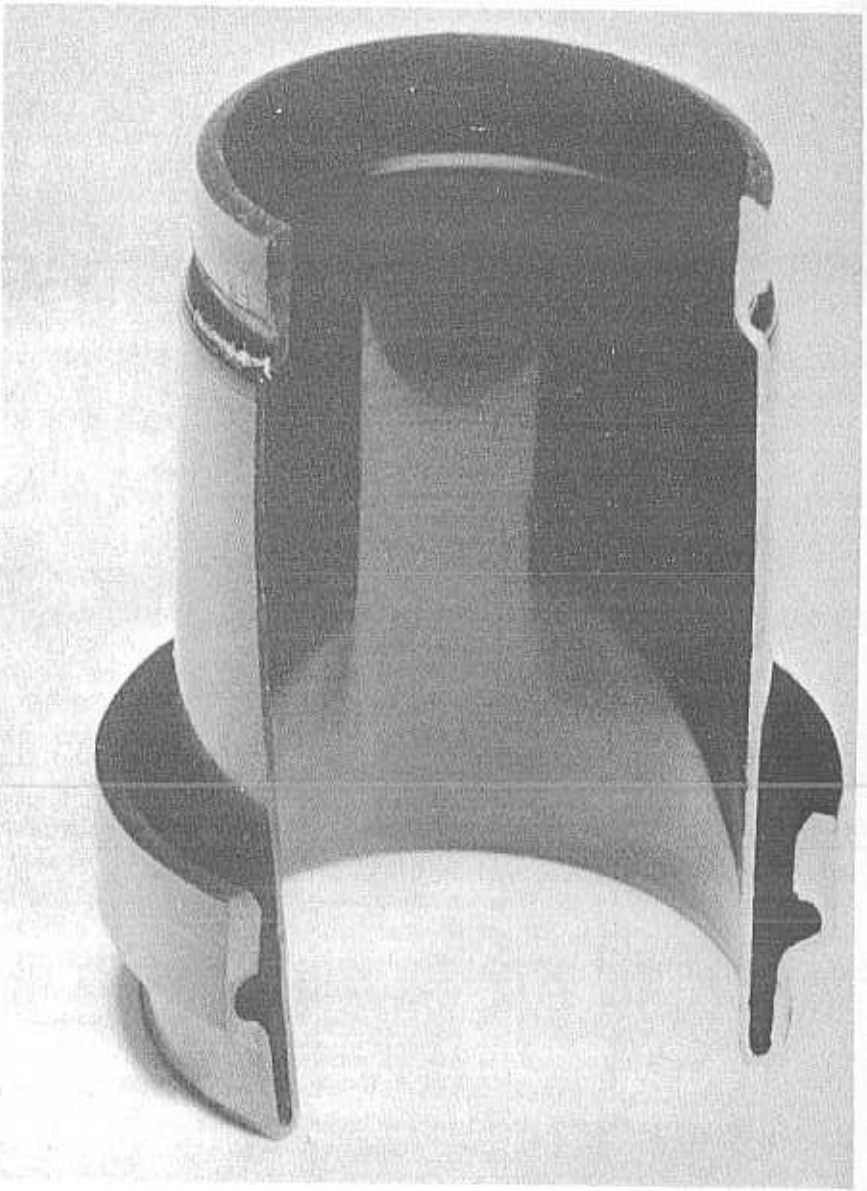


FIG. 12.10. Photograph of rolling seal.

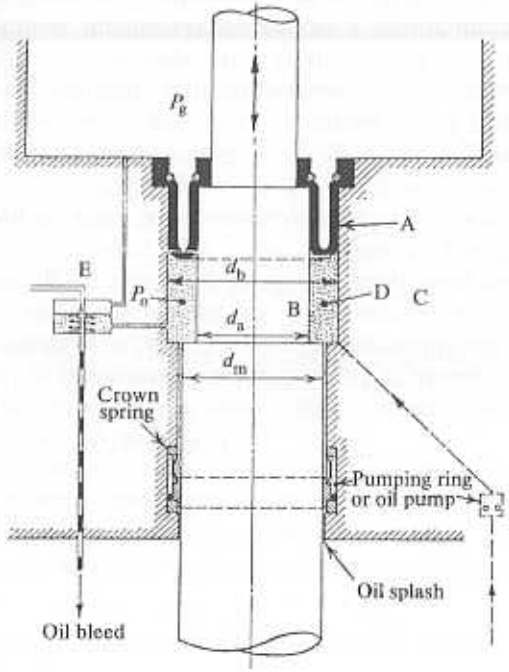


FIG. 12.11. Rolling diaphragm seal with fluid support in a stepped system.

because of the need to provide a high-pressure 'buffer' space below the piston to reduce the mechanical loading on the drive mechanism. In the two-cylinder-opposed design there was no requirement for a buffer space and therefore only two seals per cylinder were required.

General Motors licence

The invention of the rhombic drive and the rolling seal provided an apparently invincible combination for the development of successful Stirling engines. The pace of development at Philips was greatly stimulated and was no doubt enhanced by the association with General Motors in the United States following the licence agreement in 1958. Single cylinder prototype engines of 7.4 kW (10 hp: the Philips Type 1-98 engine) and of 65 kW (90 hp: designated GM Type 1-51050) were developed in addition to the original 30 kW (40 hp) engine. The photograph shown in Fig. 12.13 of this trio of prototype engines was given by Meijer in a review paper (Meijer 1969c). The four-cylinder 265 kW (360 hp) engine (designated GM Type 4-S1210) shown in Fig. 12.14

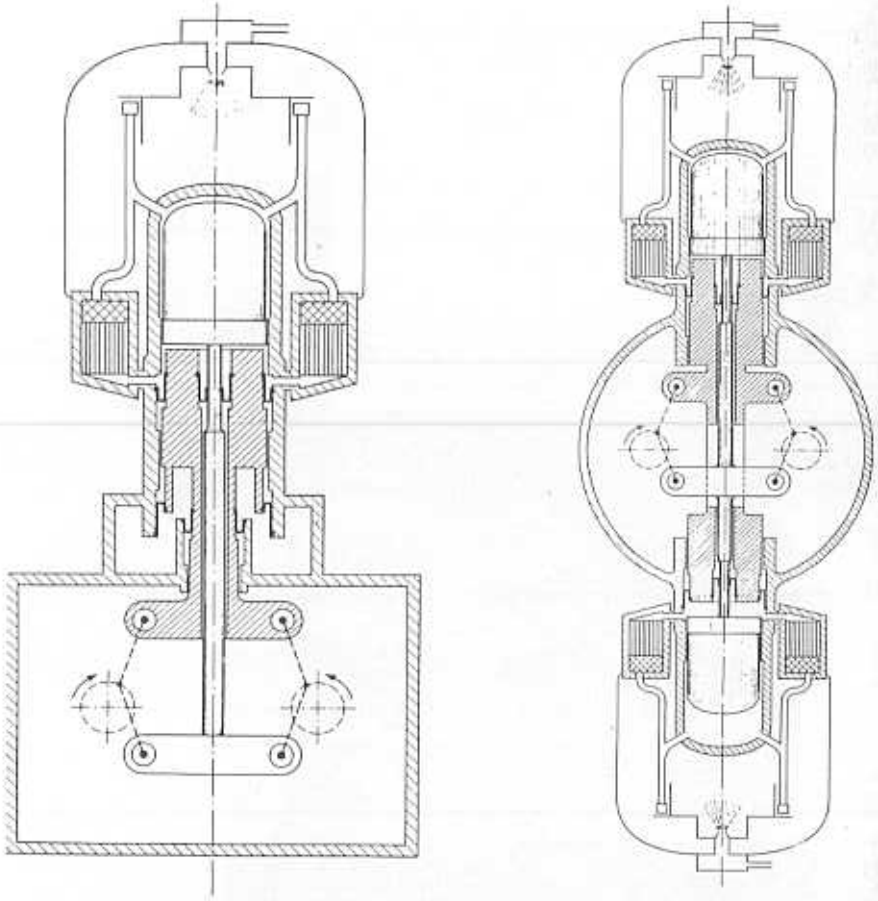


FIG. 12.12. Displacer-type engine with rhombic drive: (a) single-cylinder version, (b) two-cylinder-opposed version.

incorporated four 65 kW (90 hp) single-cylinder engines into a common crankcase. It was designed and built by Philips for the Electromotive Division of General Motors and subsequently was tested by the U.S. Navy in acoustic (Schab 1964) and performance (Loftus 1964) studies.

The 1-98 *engine*

The small single-cylinder Philips Type 1-98 engine became the work-horse for numerous developments. It most likely provided guidance and inspiration for both the General Motors Ground Power Units (GPU) (Heffner 1966) and the Allison solar space power plant (Parker and Malik

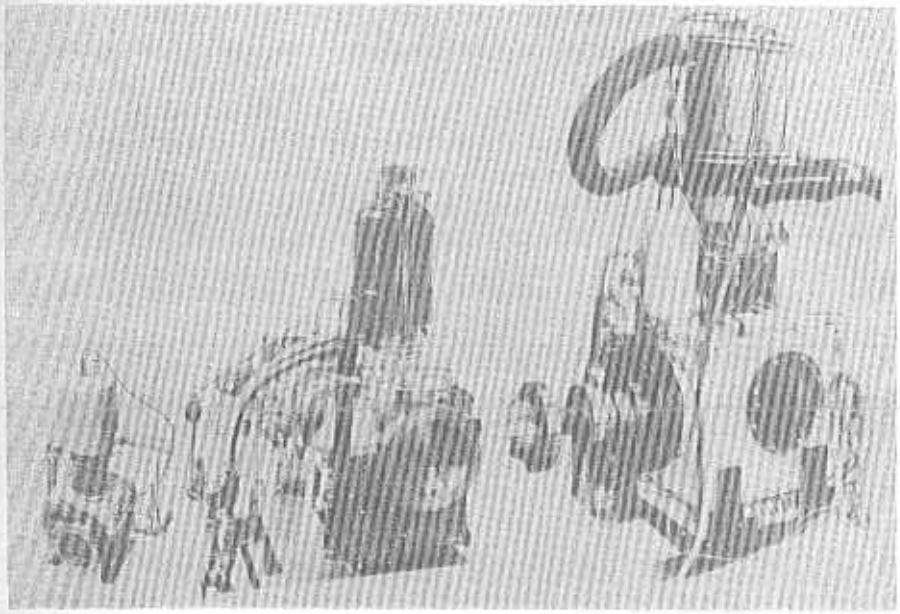


FIG. 12.13. Trio of prototype engines (after Meijer 1969c).

1962). A considerable number, perhaps as many as thirty (Michels 1976), 1-98 engines were built at Philips. They were much used in rig testing for seals and other component developments. In 1969 they were supplied for evaluation to the new licensees, United Stirling of Malmo, Sweden, and the West German group MAN/MWM.

A 1-98 engine was used in the remarkable machine shown in Fig. 12.15. This was built to demonstrate both the perfect dynamic balance of the rhombic-drive mechanism and the omnivorous multifuel capacity of the Stirling engine. Containers of different fuels were provided along the frame of the generating set including crude oil, lubricating oil, olive oil, salad oil, diesel fuel, gasoline, and liquid petroleum gas. The engine would run happily on all these individually or as mixtures. On one visit to Eindhoven in 1966 the author witnessed the machine running, at 3000 revolutions per minute, on a mixture of crude oil containing large bubbles of gasoline and alcohol, with one of the multiple-sided English three-penny pieces standing vertically and motionless on the crankcase.

A type 1-98 engine was incorporated as the power unit in a garden tractor, for presentation on his retirement to a senior member of the research laboratory. Another, incorporated in a 2.5 kW (3.4 hp) generating set has been on trial with the Swedish navy for a decade.

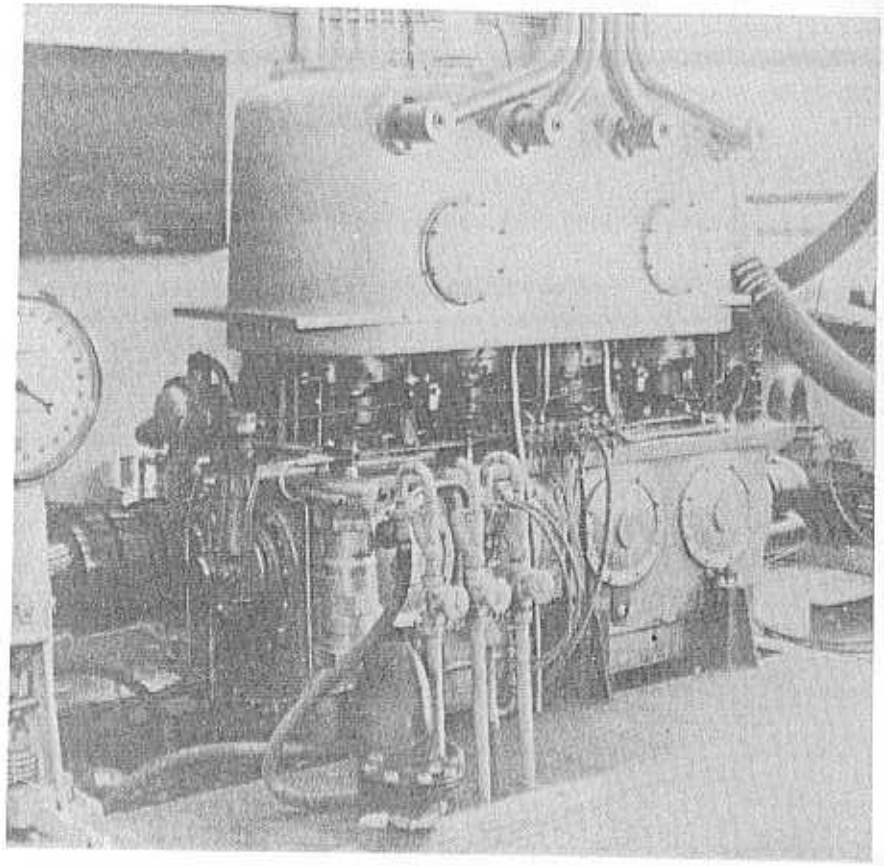


FIG. 12.14. Experimental four-cylinder 360 hp engine with rhombic drive.

Marine engines

A 30 kW (40 hp) single-cylinder engine (the Philips Type 1-365) was installed in the motor yacht *Johan de Witt*, to gain experience on pleasure-boat installations and for the development of engine auxiliary equipment. Following this, the four-cylinder opposed-piston (Philips Type 4-235 Boxer) engine shown in Figs. 12.16 and 12.17 was designed and built (Meijer 1965). This engine was intended as an underfloor engine for pleasure boats or other marine installations. It was designed to produce 85 kW (115 hp) at 300 revolutions per minute and to have a maximum thermal efficiency of 41 per cent. Research on this engine was suspended when the need appeared for a four-cylinder in-line engine (van Beukering *et al.* 1973), for bus and truck propulsion.

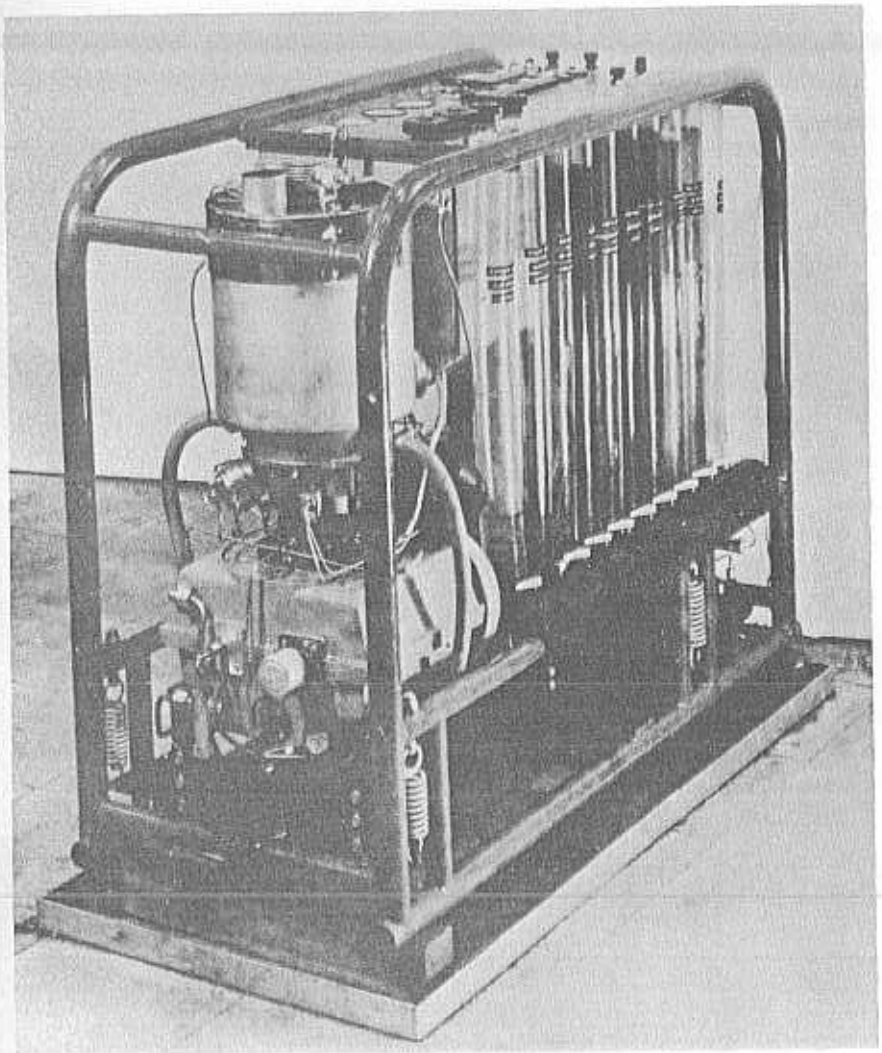


FIG. 12.15. Philips Type 1-98 Stirling engine with electric-power generator, built to demonstrate the perfect dynamic balance of the rhombic-drive mechanism and the omnivorous multifuel capacity of the Stirling engine (after Meijer 1969e).

Vehicle engines: the new licensees

The cylinder module of the opposed-piston engine was incorporated into the new four-cylinder (Type 4-235 In-line) engine. This development was undertaken at the request of the new licence holder, United Stirling, to assess the suitability of the Stirling engine for vehicular applications. It

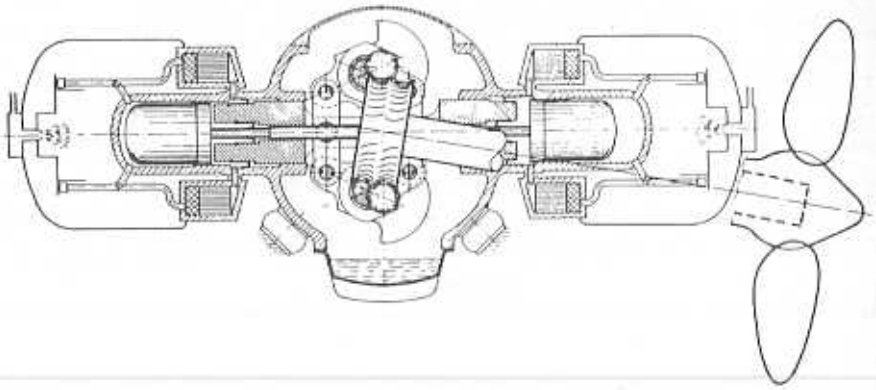


FIG. 12.16. Cross-section of four-cylinder opposed piston-displacer engine (Boxer) for underfloor marine installation (after Meijer 1963).

was about this time the German consortium MAN/MWM also negotiated a licence agreement with Philips.

Up to that time, Philips appear to have given little serious attention to the use of Stirling engines in vehicles. In a wide ranging comprehensive review paper directed principally to the potential for applications with a radioisotope power source van Witteveen (1966) listed a variety of civil and military uses for Stirling engines. He did not emphasize vehicular applications to any degree but mentioned the possibilities for heavy traction.

Once started, work on the in-line 4-235 engine proceeded apace and progress was reported by de Wilde de Ligne (1971) and by Neelen *et al.* (1971). The engine shown in Fig. 12.18 had four cylinders, a bore of 8.28 cm (3.26 in), stroke of 5.00 cm (1.97 in) and operated at 300 revolutions per minute with a mean pressure of 21.6 MN/m^2 (3140 lb per sq in) to produce 147 kW (200 hp). The engine was installed in a rear-mounted flat underfloor arrangement in a DAF bus chassis model S3200 as shown in Fig. 12.19.

The original radiator (front surface area 0.42 m^2) (651 in^2) of the bus was retained along with the existing electrically driven fans. It was supplemented with additional radiator capacity (0.67 m^2/1038.5 in^2) fitted at the rear above the engine, but fans were not fitted to the supplementary radiator. The mean water temperature was calculated to be 62 °C (143.6 °F) at an air temperature of 25 °C (77 °F) with the engine at the full power of 120 kW (163 hp) and maximum speed 25.5 m/s (57 mph). Neelen *et al.* (1971) provide many other interesting details about the installation including the power and fuel control system and the bus

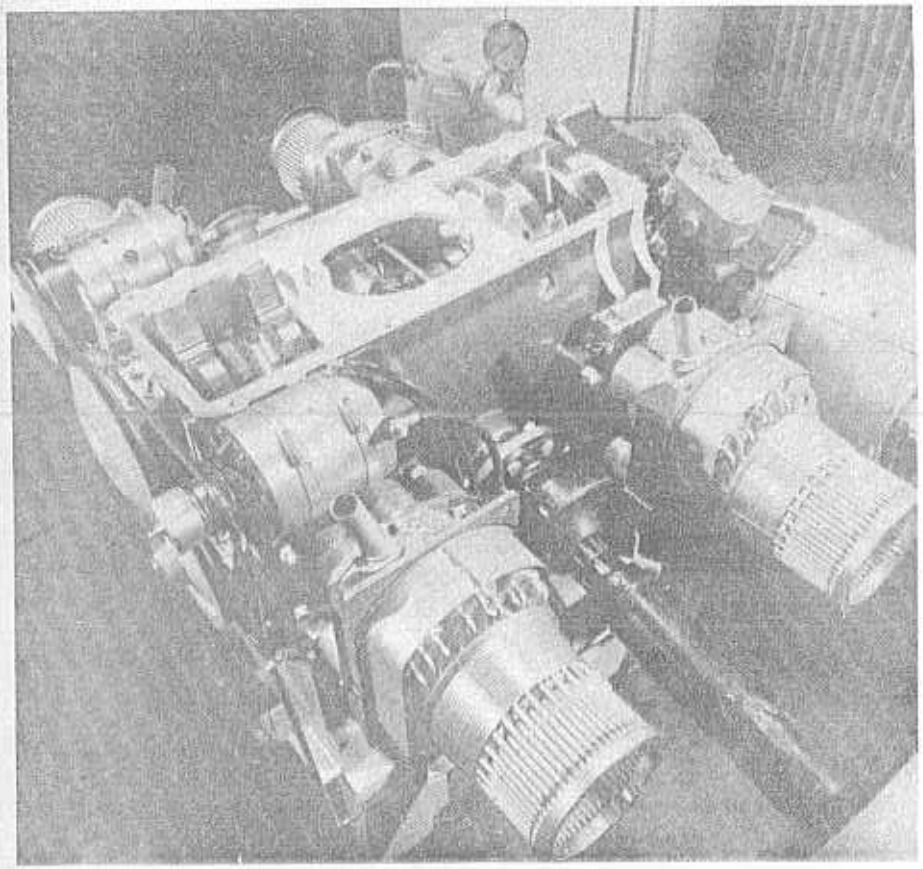


FIG. 12.17. Four-cylinder opposed piston-displacer engine (Boxer) for underfloor marine installations (courtesy Philips Research Laboratory).

gearbox and transmission particulars. Van Beukering *et al.* (1973) indicated that commencing in 1971 the bus had undergone extensive testing in respect to cooling, power control, and vehicle 'driveability' characteristics.

Similar engines of this type were supplied to United Stirling for installation in a MAN bus and a motor yacht. According to Hallare and Rosenqvist (1977) the Swedish engines were rated at only half power because the hot parts were constructed of stainless steel rather than the superior heat-resisting steels necessary for the full pressure and high temperature to achieve the rated 147 kW (200 hp). It is likely that the

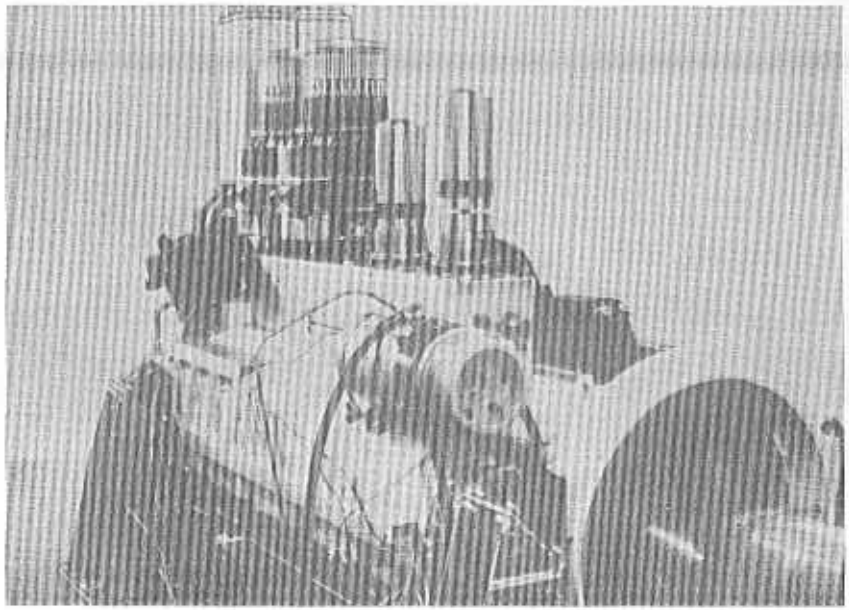


FIG. 12.18. Four-cylinder in-line Philips Type 4.235 vehicle engine (courtesy Philips Research Laboratory).

engine of the Philips bus installation was similarly restricted. No substantial report of the bus performance and operation has been given, but hearsay has it that the rolling seals have been a persistent source of difficulty.

The Philips Type 4-235 in-line engine was the last major multiple-cylinder development of the rhombic-drive displacer-type engine, but Meijer (1970a) described extensive studies of advanced displacer engines for city buses with hydrogen and fossil fuel and with thermal storage systems.

However, detailed work by United Stirling, MAN/MWM, by General Motors, and perhaps by Philips themselves, all combined to show that double-acting engines would have a size, weight, and cost of one half or less than multiple-cylinder rhombic-drive displacer-type machines. The principal saving was, of course, in the simplified drive and the need for only one reciprocating element per cylinder. Thus as the sixties drew to a close there was a concerted movement by Philips and the three licensees away from rhombic-drive engines, and a resumption of work on double-acting engines.

The fifteen years of development effort on rhombic-drive engines had advanced the technology of Stirling engines immeasurably beyond the

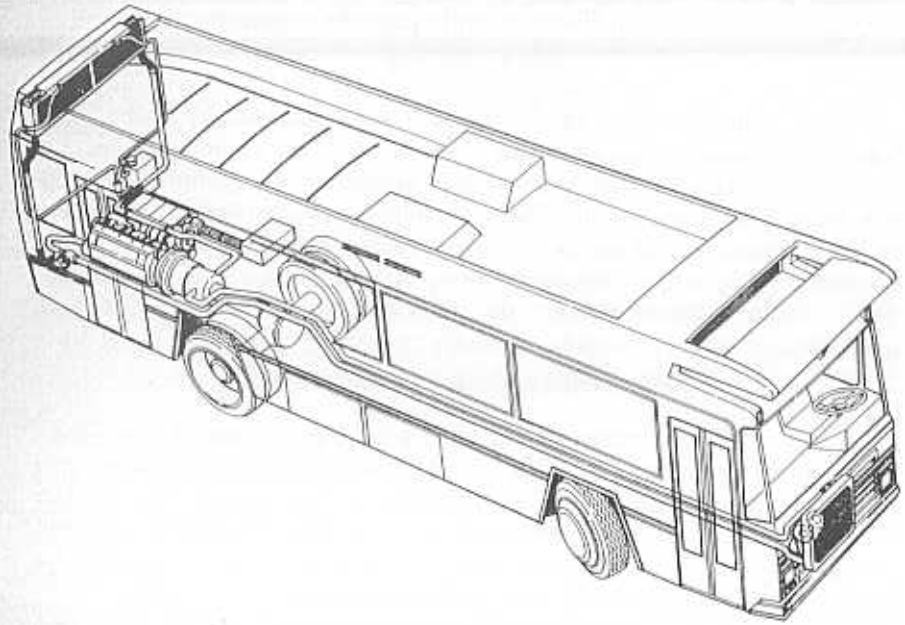


FIG. 12.19. Philips Type 4.235 Stirling-engine installation in a DAF bus (after Neelen *et al.* 1971).

small air engines of the early Philips programme. The rhombic-drive engines were nearly ready for work. They were comparable in size and weight to diesel engines, were low in noise and emission products, had an omnivorous fuel capacity, good torque characteristics and excellent part-load performance. But the cost was high, perhaps three times the cost of a diesel, and there was simply no means in prospect to halve the cost.

DOUBLE-ACTING PHASE

The Swash-plate drive

Van Beukering *et al.* (1973) have disclosed that a return to the double-acting engine was made at Philips in 1968 with the design of the Type 4-65 D.A. engine shown in Fig. 12.20 and conceptually in Fig. 12.21.

Elsewhere, Percival (1974) identifies 1965 as the date of the revival of interest in double-acting engines at General Motors, specifically as a propulsion unit for torpedo motors, and probably with liquid metal combustion. It is clear from Fig. 12.21 that the swash-plate engine has a configuration well suited to torpedoes. At General Motors a six-cylinder

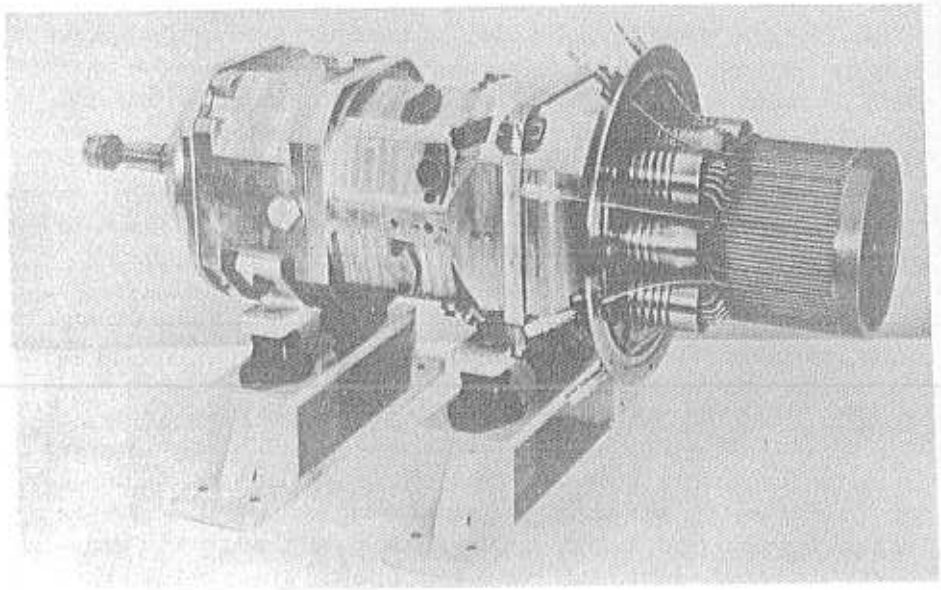


FIG. 12.20. Philips Type 4-65 double-acting engine. (Courtesy Philips Research Laboratory).

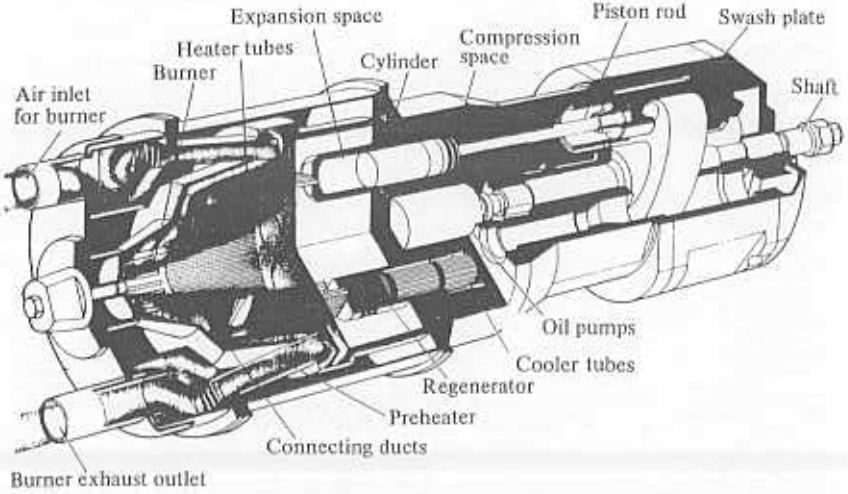


FIG. 12.21. Conceptual view of Philips Type 4-65 double-acting engine. (Courtesy Philips Research Laboratory).

370 kW (500 hp) design was completed in 1966, and by 1967 experimental work was under way on swash-plate drive components. This suggests that Philips were somewhat delayed in their return to double-acting engines and, as with thermal energy systems, followed the lead of their licensee, General Motors. A measure of support for this view comes, indirectly, from their use of nomenclature. Whereas early (1940s) double-acting air engines were said (van Beukering *et al.* 1973) to have wobble-plate drive mechanisms the later (1960s) double-acting engines were described as having a swash-plate drive mechanism. A distinction between a swash-plate and a wobble-plate is sometimes hard to make. Many dictionaries make no distinction and treat the two terms as interchangeable. Maki *et al.* (1971) define a true swash-plate drive mechanism as one that features an inclined disc rigidly attached to the rotating shaft whereas the wobble-plate does not rotate with the shaft but merely rotates. In any event, Philips have now adopted 'swash-plate' to describe the drive on recent engines and, interestingly, retain the term 'wobble-plate' for the older engines (van Beukering *et al.* 1973). Further support for the view that General Motors initiated and led a return to double-acting engines comes in a discussion on the Philips Type 4-65 DA engine by van Beukering *et al.* (1973) from the references to work on swash-plate drives and bearings by Maki *et al.* (1977) and Hays *et al.* (1971), both these papers emanating from General Motors.

The Philips Type 4-65 DA engine is a four-cylinder double-acting engine designed for 44 kW brake power (60 brake hp) and in 1973 was said (van Beukering *et al.* 1973) to have been running over 2000 hours on test. No other details of performance have been disclosed in the literature and the engine probably served as a workhorse for the development of subsequent engines of the same form but of larger capacity.

Termination of General Motors licence

In 1970 the three licensees, General Motors, United Stirling, and MAN/MWM were all working on double-acting engines with crank/connecting drives of one form or another while Philips concentrated on double-acting engines with the swash-plate drives. Then, in early 1970, General Motors did not renew their licence agreement and their programme was suddenly and unexpectedly terminated. It is not difficult to imagine the consternation this must have aroused at Philips, for General Motors had been a partner in development since 1958.

The Ford licence

Vigorous negotiation resulted in the announcement in August 1972, of a licence and development programme with the Ford Motor Company, Dearborn, Michigan (Ford, 1972). Under the terms of the agreement

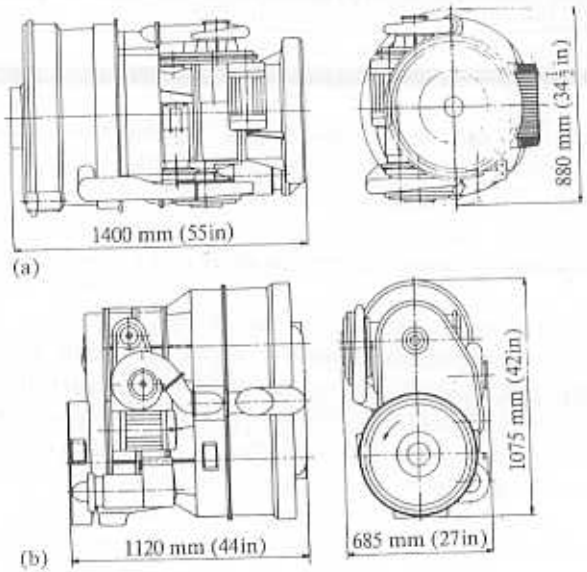


FIG. 12.22. Schematic view of two heavy-duty Philips Stirling engines with swash-plate drive mechanisms (A) Type 4-1400 D.A. engine (B) Type 8-500 D.A. engine (after van Beukering *et al.* 1973).

Ford obtained

'exclusive worldwide licence rights for Philips know-how and patents for car, truck, tractor, bus, military vehicle, industrial and surface vessel Stirling engines,† and a non-exclusive licence for all other Stirling engines. Both licences are subject to rights reserved for certain European countries. An initial three year joint development program of a seven year plan was initiated with Philips to design and build experimental engines for Ford.'

Hard on the heels of this agreement came the important review paper of van Beukering *et al.* (1973) emphasizing the suitability of the Stirling engine for automotive use. Three double-acting Stirling engines with swash-plate drive were discussed in addition to the 44 kW (60 hp) Philips Type 4-65 DA engine mentioned above.

The three engines were a car engine and two heavy-duty truck or coach engines:

(a) Philips Type 4-215 DA Passenger car engine of 125 kW (170 hp).
(b) Philips Type 4-1400 DA engine of 295 kW (400 hp).
(c) Philips Type 8-500 DA engine of 295 kW (400 hp).

Schematic views of the two heavy-duty engines are shown in Fig. 12.22. One was a double-acting four-cylinder swash-plate unit and the other was

† Note omission of underwater power systems (author).

an eight-cylinder twin swash-plate unit. Both engines were designed for long life and high efficiency at full load.

Some projected characteristics for the 4-1400 engine are torque at maximum power of 2.1 kN.m (1590 ft-lb_f) at 1300 revolutions per minute and a maximum torque of 2.6 kN.m (1900 ft-lb_f) at 400 revolutions per minute with the maximum efficiency near 40 per cent. For the 8-500 engine a torque at maximum power of 1.5 kN.m (1090 ft-lb_f) at 1900 revolutions per minute and maximum torque of 1.8 kN.m (1302 ft-lb_f) at 400 revolutions per minute with the maximum efficiency near 40 per cent. These data were calculated for a coolant temperature of 70 °C (158 °F), a working fluid pressure of 22 MN/m^2 (3200 lb per sq in) of hydrogen, and heater-tube wall temperatures of 700 °C (1292 °F) with no allowance made for the auxiliary power consumption for fan, alternator, and power steering. A 10–15 per cent allowance for these would therefore reduce the peak thermal efficiency to about 35 per cent. No other details of the heavy-duty engines have been published.

The Philips Type 4-215 *double-acting engine*

Most work has been done on the Philips Type 4-215 DA engines for passenger cars shown in Fig. 12.23. Some aspects of the design of this

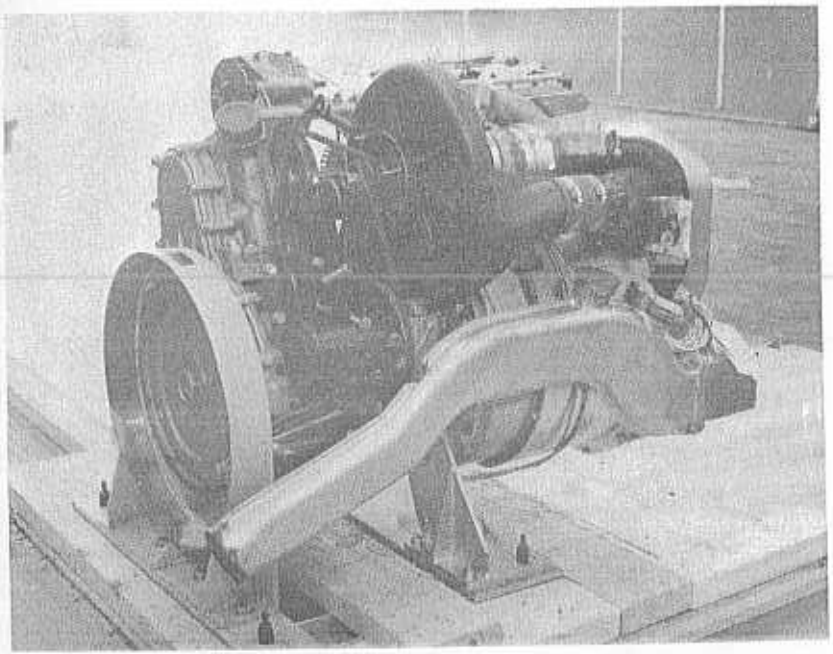


FIG. 12.23. Philips Type 4-215 D.A. engine with swash-plate drive for passenger car application. (Courtesy Ford Motor Company).

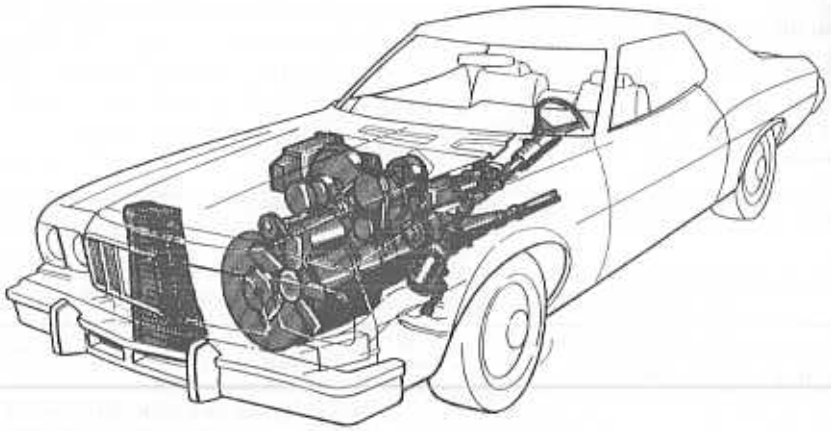


FIG. 12.24. Installation sketch for Philips 4-215 engine in a 1973 Ford Torino (after Postma *et al.* 1973).

engine have been reviewed by van Giessel and Reinink (1977). Fig. 12.24 is an installation sketch of the engine in a 1975 Ford Torino car and Fig. 12.25 is a photograph of the engine compartment of the actual Torino test vehicle. Fig. 12.26 shows the predicted performance characteristics for the Type 4-215 DA engine reproduced from Postma *et al.* (1973).

The review paper by Postma *et al.* (1973) disclosed that the first meeting of Ford and Philips was held in late 1970 (it will be recalled the General Motors program was terminated in early 1970). Following that meeting, a joint technical program was undertaken to investigate the applicability of the Stirling engine to cars. More specifically, the aim was to replace the Ford 5752 cm^3 (351 in^3) displacement V8 gasoline engine in the Ford Torino passenger car of intermediate size. The result of that initial joint program was a decision announced in August 1972 to continue with a second phase to design, build, and develop Stirling engines for cars.

The initial program was most likely a joint exploratory venture with Ford and probably involved no cash royalty or fee payments. The objectives of the program were to demonstrate Stirling engine emission capability, to investigate installation in cars, to predict vehicle performance and fuel economy, and to identify the major unknowns requiring further effort.

According to Postma *et al.* (1973) each company assumed specific tasks:

'Ford provided specifications for engine design, conducted package studies, projected vehicle performance and fuel economy, designed the accessory systems and provided customer acceptance criteria.

FIG. 12.25. Photograph of the engine compartment of 1973 Ford Torino car with Philips Type 4-215 double-acting Stirling engine installed. (Courtesy Ford Motor Company).

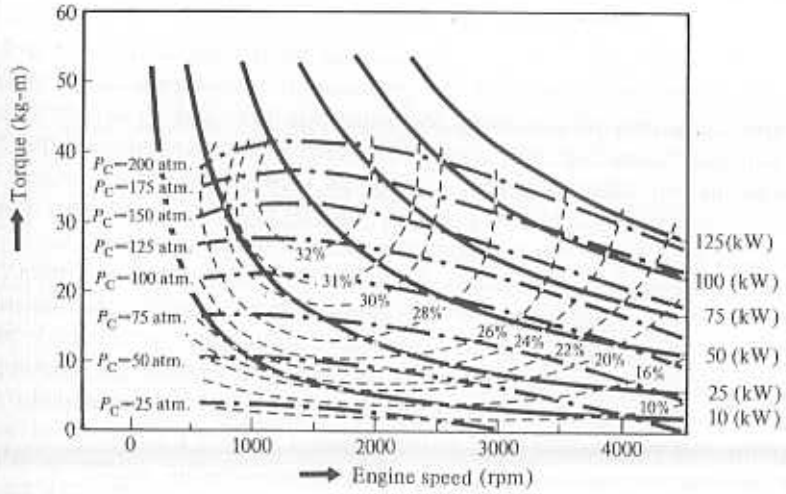


FIG. 12.26. Performance characteristics of the Philips Type 4-215 D.A. engine (after Postma *et al.* 1973).

Philips designed the engine, provided engine drawings, conducted simulated California Vehicle Standards (CVS) emission tests, provided basic engine performance and specific fuel consumption and furnished information on general engine operating characteristics.'

The results achieved in this preliminary phase were sufficiently encouraging for Philips and Ford to undertake the development of a prototype installation in a 1975 Ford Torino car. Philips' task was to design, build, test and develop the basic engine while Ford designed and built the accessory and cooling systems, supplied the vehicle, installed the engine, and performed vehicle tests and evaluations.

At the start of the second phase of the program there was a note of cautious optimism in the paper by Postma *et al.* (1973) but some concern was evident about the size of the cooling system, the use of hydrogen as a working fluid, the feasibility of maintaining low emission levels over the projected life of 80 467 km (50 000 miles) and the cost of manufacturing.

Progress in the development of this program has been well documented in the semi-annual contractors coordination meetings for advanced automotive power systems organized by the U.S. Department of Energy (formerly the Energy Research and Development Administration). At the meeting in May 1975, Ford were able to announce that the Type 4-215 engine had been under extensive test and development at Philips for the past year, had been installed in the Torino test car and would be in the United States before the end of 1975. In the subsequent report (November 1975) it was confirmed that the Stirling powered car was received on schedule from Philips. In addition, a feasibility study was sponsored by ERDA for a Stirling engine of 60 to 73 kW (80 to 100 hp) for use in a passenger car of 1134–1361 kg (2500–3000 pound) weight. Preliminary data was included for this smaller engine obtained by scaling the 4-215 engine. Ambitious plans were projected for the extensive development of Stirling engines for automotive and other applications under joint Ford-ERDA-other sponsorship.

The subsequent report (May 1976) described the progress of design studies for the small engine. An important development was that:

'the assembly, difficulty, cost and possible reliability problems with the roll sock seal used to contain the high pressure hydrogen at the piston rod/block interface has prompted a look at an alternative system'.

About this time Ford had been working also with United Stirling of Sweden with the installation of one of their engines in a Ford Taunus estate car. The Ford report of May 1976 briefly discussed the United Stirling sliding seal and also disclosed that the material of the seal was Rulon, a proprietory filled PTFE (teflon) plastic with exceptional wear properties. By May 1976, 'some 2–300 miles (322–483 km) of driving on the cars (sic) had been accumulated but "they have not yet been in a satisfactory enough condition to obtain meaningful test data"... Problems to be resolved before serious testing can begin were general engine durability, power control stability and air/fuel control stability.' A major mechanical problem had been 'the eccentric crosshead rotation with consequent noise and friction'. The same report also noted that the DAF bus with the Type 4-265 in-line rhombic drive engine had been brought to the United States for demonstration purposes and press review.

The Philips Type 4-98 double-acting engine

In the October 1976 report on passenger car trials, the small engine referred to six months earlier had become identified as the Philips Type 4-98 DA engine of 60 kW (84 hp) shown in Fig. 12.27. The projected performance characteristics of the engine are shown in Fig. 12.28. The working fluid was hydrogen at 20.2 MN/m^2 (200 atm) mean pressure with a heater inside wall temperature of 750 °C (1382 °F) a cooler temperature of 80 °C (176 °F) and a maximum engine speed of 5400 revolutions per minute. The engine had four cylinders, a swash-plate angle of $\pi/10$ radians (18°), swept volume per cylinder of 98 cm^3 (5.98 in^3) a volumetric ratio of expansion/compression of 1.10 and a regenerator filling factor of 38 per cent. Further work was described on the development of a sliding seal as an alternative to the rolling seal. Another interesting aspect was the comparison made between two engines having identical Type 4-98 parts, one with a swash-plate drive and the other with a dual crank drive (similar to contemporary United Stirling engines). The comparison was made for installation in a 1976 Ford Pinto car. The dual crank engine with front-wheel drive was contained in the existing engine compartment

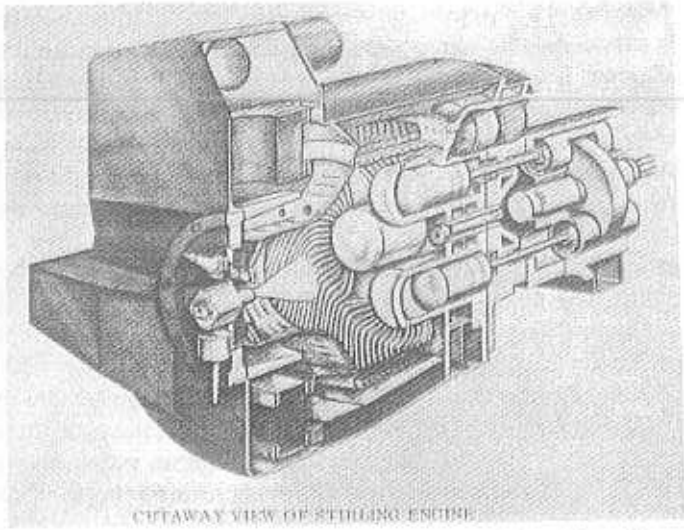


FIG. 12.27. Philips Type 4-98 D.A. engine for use in a Ford Pinto (after Kitzner 1977a).

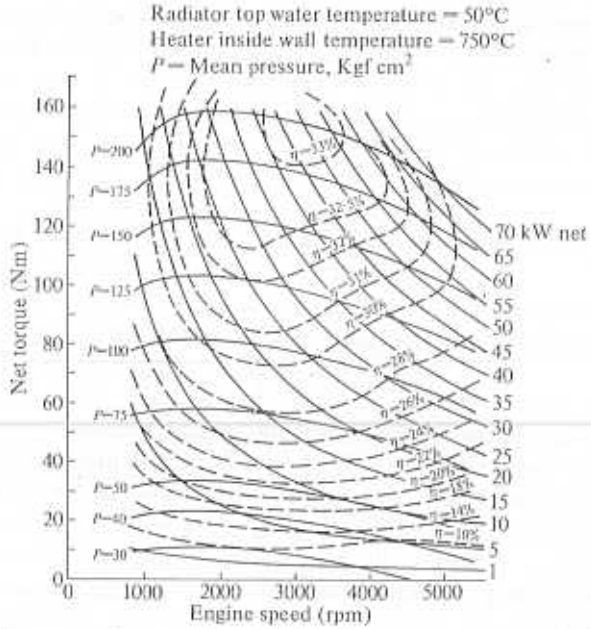


FIG. 12.28. Performance characteristics of the Philips Type 4-98 D.A. engine (after Kitzner 1977b).

without modification. To accommodate the swash-plate drive engine it was necessary to increase the length of the vehicle by 8.1 cm (3.2 in).

The Torino tests

In the October 1976 report no details of the Torino tests were given but it was disclosed that the program 'had not proceeded as quickly as originally hoped, because a variety of . . . failures have prevented running a sequence of engine calibration tests.' The design of the swash-plate drive mechanism had been modified and, more importantly, to permit early resumption of the engine development test program the rolling seals were replaced by the piston rod seals used for the smaller engine although rolling seals were 'still believed to be a possibly proper approach for production'.

A year later the October 1977 (Kitzner 1977a) report contained an extensive review of the work on the Philips Type 4-215 engine. The major innovations of the engine were identified as:

1. 20.2 MN/m^2 (200 atm) instead of 15.2 MN/m^2 (150 atm) for working gas pressure.
2. first engine with a rotary regenerative ceramic preheater system.
3. new air/fuel control system to satisfy dynamic requirements.
4. new power control system for automobile demands.
5. three times larger than previous swash-plate engines.
6. half the specific weight of previous Stirling engines.
7. packageable within existing engine compartments.
8. 4000 rev/min capability instead of the 2000–3000 rev/min with rhombic-drive.
9. first engine with exhaust gas recirculation.
10. unique coolant flow through cooling units.
11. new lubrication system.
12. first engine designed to drive full range of automotive type accessories.

The major technical problems encountered in the engine both in dynamometer and in the Torino vehicle tests were divided between problems resolved and problems unresolved.

The problems resolved were identified as:

1. swash-plate surface galling.
2. drive system noise due to non-concentric crossheads.
3. regenerator end-plate bending.
4. crankcase failure.
5. engine out of balance.
6. piston attachment failure.
7. insufficient exhaust gas recirculation.
8. unstable air-fuel control system.
9. corrosion of check valves.
10. unstable combustion and power control contamination.

The problems unresolved were identified as:

1. roll sock seal system failure.
2. preheater leakage.
3. preheater binding.
4. fuel burning on preheater core.
5. heater-head temperature distribution.
6. excessive warm-up time.
7. insufficient burner air supply.
8. power control instability.
9. heater-head cracking.

DOE-sponsored work

Kitzner (1977b) have reported comprehensively on a feasibility study for a 60 to 73 kW (80 to 100 hp) automotive Stirling engine, conducted jointly by Ford and Philips, and commissioned by the U.S. Department of Energy. This report puts into the public domain a good deal of technical data and information not previously available. It is recommended for close study by anyone interested in advanced Stirling engine systems. Earlier, Kitzner (1977a) gave some details of a comprehensive cost-shared Ford/ERDA/NASA Stirling-engine development program extending over

eight years with fundings at the level of $160 million. A program of similar magnitude is being undertaken by United Stirling/MTI/American Motors with ERDA or, as it is now, Department of Energy (DOE) backing. Thus, it would appear, forty years after the initial effort by Philips, Stirling engines are finally to be given the benefit of adequate funding to permit comprehensive development to be undertaken.

RELATED WORK

Since the late 1960s the main thrust of Philips's efforts seems to have been directed to automobile applications. Meijer (1970a) discussed applications of heavy-duty Stirling engines to buses. At the same time he introduced the concept for vehicle applications of the combination of Stirling engines and thermal storage systems (heat buffer) for completely pollution-free operations. He also introduced the possibility of using hydrogen as the fuel for engines following the discovery at Philips of the means to 'store' large quantities of hydrogen in hexagonal intermetallic compounds of rare-earth metals and nickel or cobalt (van Vucht *et al.* 1970). It had been found, for example, that in the material $LaNi_5$ the density of hydrogen absorbed at 0.25 MN/m^2 (2.5 atm) pressure and at room temperature was nearly twice as high as the density of liquid hydrogen.

In a paper published about the same time (Meijer 1970b) the prospects for application of the Stirling engine to vehicles were further discussed with first mention of the new double-acting swash-plate engines. The discussion included a review of the advantages of indirect heating using heat pipes and a heat storage system using lithium fluoride. Further details of the hydrogen storage system for vehicle fuels were also given. The results of calculations were presented for a variety of vehicles: cars, vans, taxis, and buses operating with Stirling engines in combination with thermal storage systems or hydrogen fuel. Conclusions were that both systems were well suited for vehicular application except in the large American cars requiring a wide radius of operation.

More recently, Asselman *et al.* (1977) have contributed further work on the Stirling automotive engine with thermal energy storage. Boser (1977) has discussed some aspects of safety with high-temperature thermal storage systems. The vehicle Stirling engine with thermal energy storage is believed to have an important part to play in the future when fossil fuels are no longer so readily available. Clearly Philips believe this and are at the forefront of research in this important field. Various other papers, important to the literature of Stirling engines, have been contributed by Philips workers.

Asselman *et al.* (1972) reported on the development of a high-performance radiator. A design called the 'folded front' radiator was

described that was found to be lighter in weight, smaller, and with better heat-transfer properties with less fouling than conventional radiators. This work was stimulated by the handicap of the very large cooling-system load that is characteristic of vehicles equipped with Stirling engines. Use of an improved radiator or other cooling system is of course not confined to Stirling engines. It could be just as easily adopted with great advantage for diesel engine use, and so the penalty of double-sized cooling systems would remain with the Stirling engine.

Michels (1972) presented the results of studies of the emission characteristics of the combustion systems used on Stirling engines in support of the development of the Type 4-215 DA engine. Earlier, Michels (1971) had reported on the theoretical and experimentally measured effects of exhaust gas recirculation on the NO_x content of exhaust gases from a Stirling engine.

Later, Michels (1976) presented interesting data in a study of the effect of temperature and different working fluids on the efficiency of a Stirling engine with particular reference to the Philips 1-98 displacer engine with rhombic drive. Some of these results are reproduced in Chapter 8.

Hermans *et al.* (1972) and Uhlemann *et al.* (1974) reported on the possibilities for the combination of a nuclear isotope heat source and Stirling engine for remote unattended power generation.

In the field of thermal energy storage systems in combination with Stirling engines, Asselman and Green (1973a and b) discussed the technology of heat pipes and their application at Philips to Stirling engines. Gawron and Schroeder (1972) presented information on the thermal storage aspects of eutectic fluoride mixtures, and van der Sluys (1975) and Bierman (1975) presented experimental data about lithium sulphurhexafluoride heat sources in combination with Stirling engines for underwater propulsion systems. Design studies for underwater power systems and for total-energy systems were described by Jaspers and du Pre (1973).

Under ERDA contract the Philips Laboratories in North America investigated the application of Stirling engine prime movers to total-energy power-generation systems in hospitals, office buildings, and a residential complex. The results of this study were reported by Lehrfeld (1977). Asselman (1976) described experimental work with fluidized bed coal combustion to provide heat to a Stirling engine with a heat pipe.

In addition to all the above work, major research and development programs on Stirling engines as cryogenic cooling machines have been carried out continuously since 1948 at the Philips Research Laboratories at Eindhoven and, later, at the North American Philips Laboratories in New York. None of the cryogenic work has been reviewed here but is

included in the companion volume†. It should be understood there was much interchange between the different groups working on prime movers and refrigerating machines with developments in one field finding their application in another.

Another important program not covered here is the joint North American Philips/Westinghouse program to investigate the feasibility of the radioisotope artificial heart under contract to the U.S. Atomic Energy Commission (AEC), and later the Energy Research and Development Administration. This program is reviewed in Chapter 17.

† *Regenerative Cryogenic Cooling Engines*—in preparation.

13 GENERAL MOTORS STIRLING ENGINES

INTRODUCTION

BETWEEN 1958 and 1970, much valuable work on Stirling engines was carried out in the United States by the General Motors Corporation (Percival 1974). Interest at General Motors was stimulated in 1948 by the publication of the three classic papers about the early Philips work (Rinia and du Pre 1946, de Brey, Rinia and van Weenan 1947, and van Weenan 1947). General Motors proposed a working agreement with Philips but the idea was rejected as premature. Later, in 1957, Philips considered their work sufficiently advanced and negotiated a licence agreement with General Motors. The agreement provided for a ten year information exchange with provision for mutual licensing of patents relating to Stirling engines.

At General Motors, the interest in Stirling engines was focused on applications in marine propulsion, locomotive power, generating sets, and various military and space applications. Percival (1974) notes that in 1958:

'There was no interest by anyone 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. However, G.M. made no investigation of Stirling vehicle propulsion until 1962 . . . ,

The work at General Motors can be broadly classified into three principal elements identified by the division responsible for execution of the program, namely,

(a) G.M. Research Laboratories, Technical Center, Warren, Michigan.
(b) Electromotive Division, La Grange, Illinois (and earlier, Cleveland Diesel Engine Division).
(c) Allison Division, Indianapolis.

By far the biggest effort was invested at G.M. Research working on their own programmes and on work related to the other two division programmes. The size of the team, or the total man-years of effort expended on Stirling engines have never been disclosed. An estimate of 50 engineers for 10 years would perhaps not be unrealistic.

Very early G.M. Research was encouraged by the U.S. Army Engineer Laboratories, Fort Belvoir, Va., to develop Stirling engines for outboard motors and small generating sets. Percival (1974) attributes the 'near certainty of an Army contract' as a major consideration influencing the corporate decision to proceed with the licence arrangement.

Subsequently the Army did commission the development of a small (3 kW (4 hp)) electric-power generator designated the Ground Power Unit (GPU). This contract ran for several years and was in fact the only

included in the companion volume†. It should be understood there was much interchange between the different groups working on prime movers and refrigerating machines with developments in one field finding their application in another.

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significant financial support received from external sources by G.M. Research during their whole program. Most of the other work appears to have been carried on with funding provided by the corporation. A particular difficulty in securing funds from U.S. government sources was the severe restrictions of the licence agreement on the release of technical information.

In 1958 the Allison Division were also anticipating the award of an Air Force contract for a Stirling engine power plant for a solar-heated satellite. An experimental engine was in fact commissioned and evaluated in a program extending through 1959/1960.

Finally, at that same period, the Cleveland Diesel Engine Division believed that the Stirling engine could compete with diesel engines for river and harbour work, including boat propulsion as well as submarines. Several large Stirling engines were made by or for the Cleveland Diesel Engine Division and, later, for the Electromotive Division following the dissolution in 1962 of Cleveland Diesel.

G.M. RESEARCH ENGINES

Component development

Percival (1974) has indicated that in the first five years of research and development work on Stirling engines effort was concentrated on component development, specifically:

(a) seals for the piston and piston rods in rhombic drive engines to prevent both leakage of gas and ingress of oil to the working fluid,
(b) reduction of gear noise,
(c) improvement in combustion and burner nozzle designs,
(d) durability of the preheater,
(e) engine speed governing,
(f) reduction of regenerator cost,
(g) endurance testing,
(h) refinement of the cycle analysis,
(j) numerous studies and demonstrations of thermal energy storage systems in combination with Stirling engines.

Effort in the last five years was concentrated in other areas including:

(a) cooler and heater heat transfer,
(b) rolling seal quality control,
(c) low-cost preheaters,
(d) swash-plate drive bearing studies,
(e) reduction in engine volume,
(f) stress analysis of heater cylinders,
(g) vehicle applications,
(h) exhaust emissions,

(j) governor refinements,
(k) controls reliability,
(l) reduction of friction.

Percival's report contains a wealth of detail about all these matters.

In twelve years between the start of the program in 1958 and its abrupt termination in 1970 over 30 000 hours operating experience on Stirling engines had been gained. The accelerating pace of the project was such that over 50 per cent of the operating experience was gained over the final three years and 75 per cent in the last five. In addition many thousands of hours of operation were accumulated on seal, bearing, combustion, regenerator, and heat transfer rigs.

Ground power unit

The principal visible achievement of the G.M. Research program was the Ground Power Unit Stirling engine generator. These units sustained a decade of development. The final model, a GPU3, is shown in Fig. 13.1. The engine was a single-cylinder machine with rhombic-drive having a bore of 6.98 cm (2.75 in) and a stroke of 3.05 cm (1.2 in). Figs. 13.2 and 13.3 show the operating characteristics of the engine (reproduced from Percival 1974). In the course of development the engine experienced many vicissitudes, principally in the governor control and hydrogen compressor systems (see Chapter 10). However, by 1967 the engine was capable of stable operation to within ±5 revolutions per minute and satisfactory response to sudden increases or decreases in load. Reliability was progressively increased so that in 1969 the system passed a rigorous 500 hour military qualification test.

One of these engines was used in an experimental vehicle installation, the Stir-Lec 1, the hybrid Stirling-engine electric-drive car shown in Fig. 13.4. Details were reported by Agarwal *et al.* (1969). The GPU3 engine

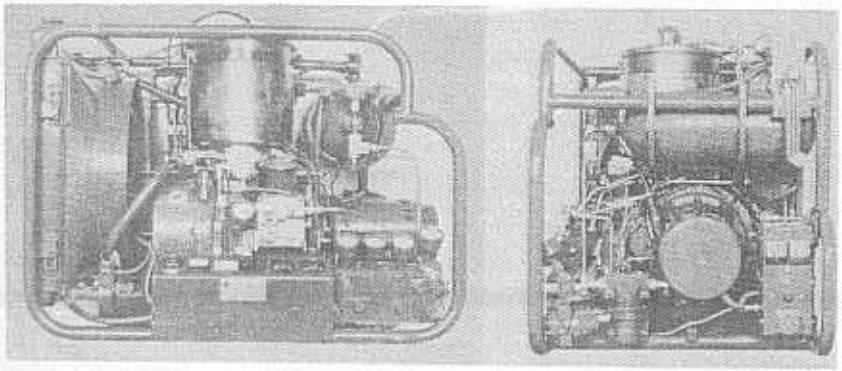


FIG. 13.1. General Motors Stirling engine generator GPU3 (after Heffner 1966).

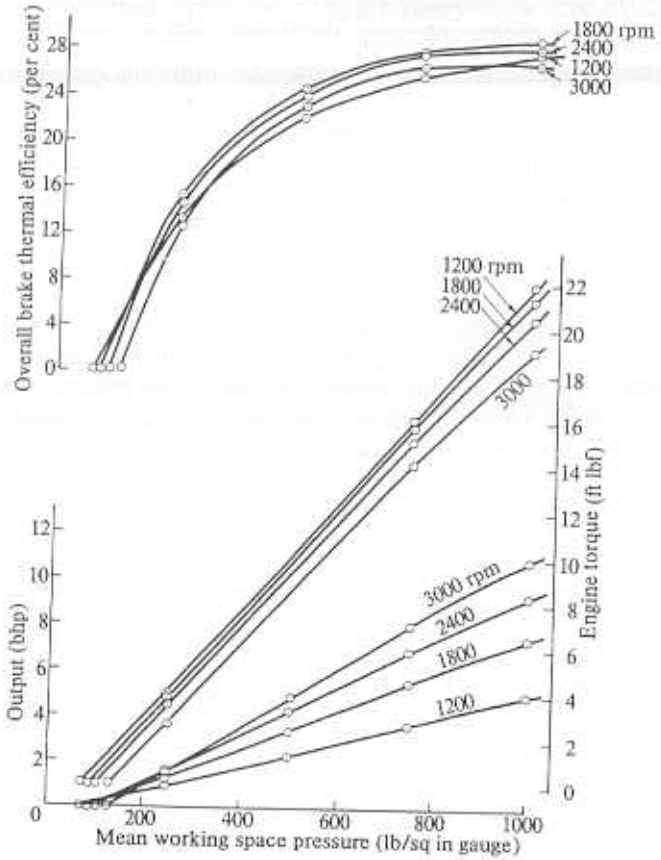


FIG. 13.2. Performance data for GPU3 engine (after Percival 1974).

was installed in the rear compartment of a standard Opel Kadett. The engine was not connected directly to the drive train of the car but drove, at constant speed, a three-phase alternator. Alternator output was rectified to charge electric storage batteries. The battery power was modulated and inverted to a three-phase induction motor driving the car.

This exercise was largely a demonstration of the feasibility of a low-emission engine/electric-drive hybrid vehicle. Agarwal *et al.* (1969) conclude that '... the extra cost and complexity of the hybrid system would make it difficult to compete with simpler propulsion approaches for private passenger vehicles ...'

Earlier vehicle work was undertaken in 1964 with the installation of a 22 kW (30 hp) Stirling engine in a Corvair. This vehicle, christened the

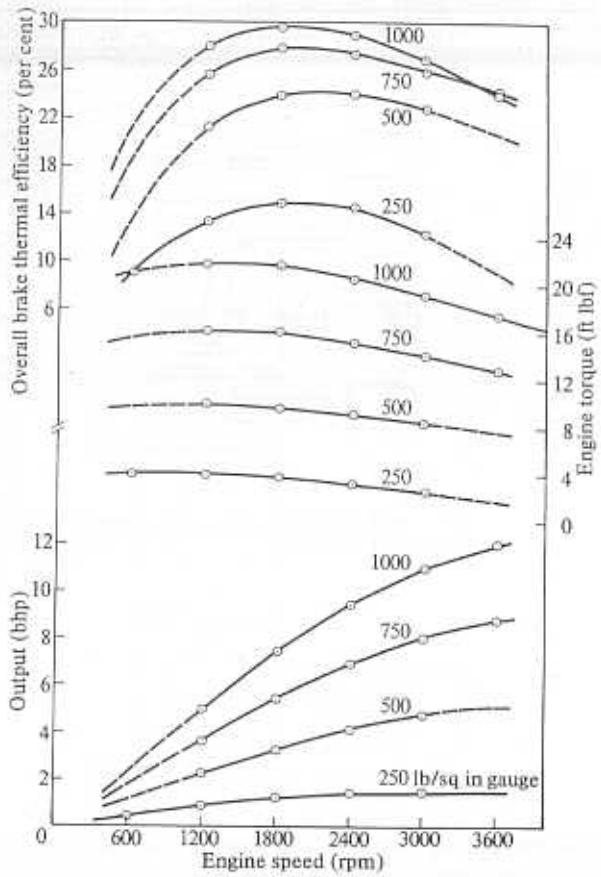


FIG. 13.3. Performance data for GPU3 engine (after Percival 1974).

'Calvair', was energized from a tank of heated alumina and formed part of the studies in progress for thermal energy storage systems.

Underwater power systems

The double-acting Stirling engine was invented at Philips in the late 1940s in both swash-plate and crank configurations, but was abandoned because of problems with the piston seal. In 1965, the concept was revived at G.M. Research specifically for advanced torpedo motors. An important paper about this work was published (Mattavi *et al.* 1969) which reviewed various studies of compact double-acting engines and contained performance data projected for a number of underwater powerplant engines up to 110 kW (150 hp). The paper also contained some discussion of energy-storage materials and metal combustion systems.

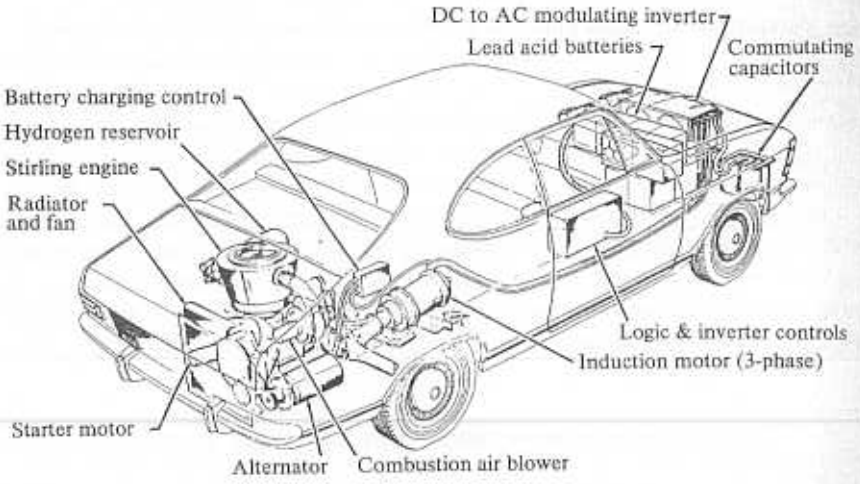


FIG. 13.4. Stir-Lec hybrid Stirling engine electric-drive car (after Agarwal *et al.* 1969).

Metal combustion involves the rapid oxidation of a liquid metal using the resultant heat of reaction as the primary energy source. For underwater power systems it is attractive primarily because the reaction products can be retained 'on board' thereby avoiding detection and, in deep submergence systems, avoiding the need for compression equipment to discharge the effluent.

Vehicle engines

These studies for compact underwater engines coincided with the development of public interest in the emission characteristics of vehicle engines. The promising efficiencies and specific outputs achieved with Stirling engines coupled with their low noise and low exhaust emission characteristics generated intense interest at General Motors in potential vehicular applications. In 1967 the Pontiac and Oldsmobile Divisions cooperated with G.M. Research in a study of a 185 kW (250 hp) swash-plate Stirling engine for a Torino car. At the same time the Truck and Coach Division was interested in a Stirling engine of 110 kW (150 hp) for city buses as a replacement for the diesel engine. It was said that the advantages of reduced noise, smoke, odour, hydrocarbon, and NO_x emission with essentially no oil consumption would be worth up to $15 000 more per bus.

Another possibility carefully considered in the late 1960s was the production of small cars for shopping and short-distance city travel. Part of the work involved study of a 18 kW (25 hp) swash-plate Stirling engine

complete with accessories, transmission, and power train. The compact engine studies carried out embraced by 1968 the five fundamental arrangements shown in Fig. 13.5. Comparative sizes of these five arrangements showed the double-acting engines to be about half the size of rhombic in-line or rhombic opposed-piston configurations. Thereafter, detail designs were undertaken for compact double-acting engines of 90 kW (120 hp) output from four cylinders. The results, shown in Fig. 13.6, indicate that swash-plate design is not necessarily the most compact form of double-acting engine. Subsequently (in 1968) the design of a four-cylinder engine of 110 kW (150 hp) was undertaken for vehicular applications as part of the Truck and Coach Division demonstration program. The engine, designated Type 4L23 was a double-acting, four-cylinder, in-line, single-crank configuration with crossheads and piston displacement of 377 cm^3 (23 in^3). The engine was designed for operation at 2000 revolutions per minute with a mean hydrogen pressure

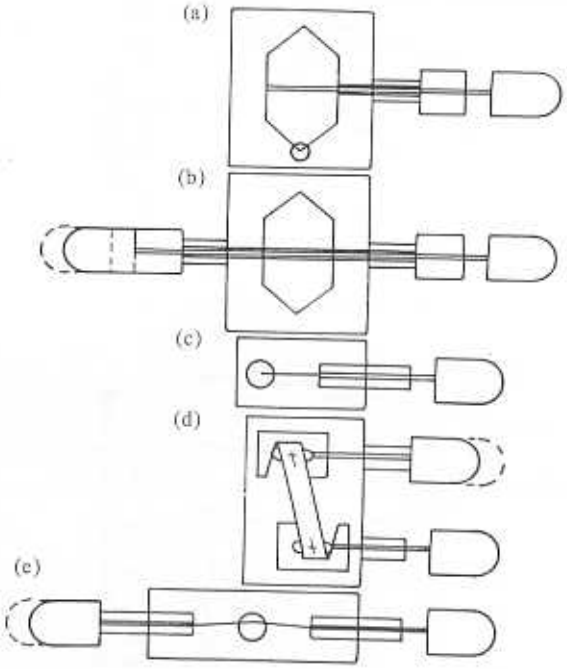


FIG. 13.5. Compact engine arrangements (after Percival 1974): (a) Single-acting, single-cylinder rhombic-drive or in-line; (b) Single-acting, opposed-cylinder rhombic-drive; (c) Double-acting, multiple-cylinder in-line; (d) Double-acting, multiple-cylinder, swash-plate drive; (e) Double-acting multiple-cylinder, opposed-piston, in-line.

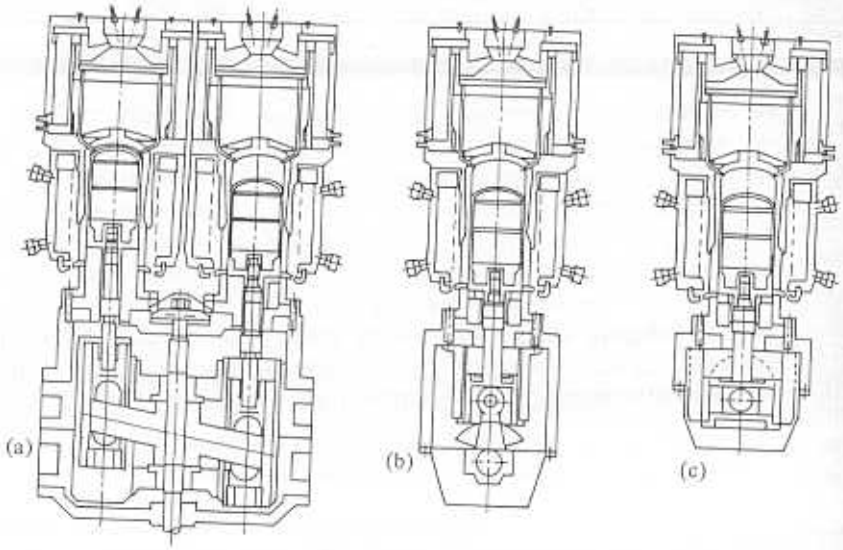


FIG. 13.6. Relative volumes of double-acting Stirling engines of 120 hp with different drive mechanisms (after Percival 1974): (a) Swash-plate; (b) Crank and connecting rod; (c) Scotch yoke.

of 10.3 MN/m^2 (1500 lb per sq in). To provide a margin for future development of the hot parts the drive mechanism was designed for a mean pressure of 20.6 MN/m^2 (3000 lb per sq in).

Calculated performance data for the model 4L23 engine are shown in Fig. 13.7 and a comparison of the engine with equivalent General Motors diesel engines is given in Table 13.1. Percival (1974) notes laconically:

'Approximately 95 per cent of the basic engine parts were on hand early in 1970 and the engine was being motored on the dynamometer as a part of the balancing procedure when the programme was halted on February 27, 1970. Target date for start of the coach installation had been May 1, 1970.'

Publications

In the course of their duodecade of effort on the Stirling engine, G.M. Research contributed several papers of permanent interest and value to the open literature in addition to those referenced above. The first by Flynn *et al.* (1960) was a lengthy paper primarily surveying the Philips rhombic-drive engine but included an interesting historical section. Another paper by Flynn *et al.* (1962) surveyed the possibilities for heat

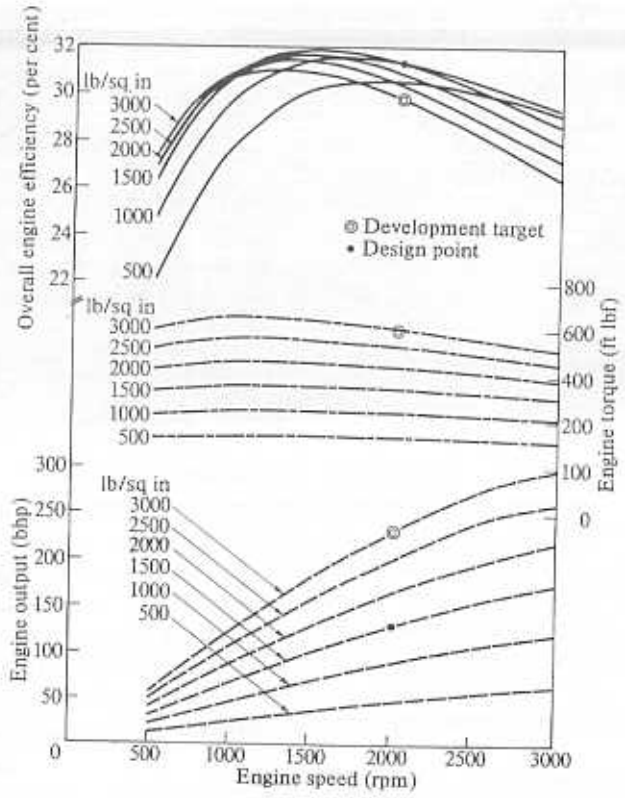


FIG. 13.7. Performance data for the Type 4L23 Stirling engine: a four-cylinder in-line double-acting engine of optimized design for use in buses (after Percival 1974).

engines in combination with thermal energy storage systems with particular reference to the Stirling engine. Subsequently Heffner (1966) reviewed the progress of the G.M. Research program with emphasis on the Army GPU. The paper was also of interest because of the photographs and passing references to the other larger engines, to some aspects of the analytical procedures and to work on the reciprocating seals.

Percival (1967) discussed naval applications of Stirling engines and covered some of the same ground, but less substantially than the subsequent paper by Mattavi *et al.* (1969).

A report on the favourable environmental characteristics of Stirling engines in terms of smoke, odour, noise, and exhaust emissions was presented by Lienesh and Wade (1968) and contained measured data on the GPU 7.3 kW (10 hp) engines. Other measurements made by Philips

Table 13.1. Calculated performance data for the General Motors Stirling engine Type 4L23 double-acting vehicle engine and a comparison with General Motors diesel engines of comparable power and size.

Parameter	Units	GMR Stirling	DDAD Diesel	DDAD Diesel
model		4L23	4–53N	6–71N
bore	in	4.00	3.875	4.25
stroke	in	1.83	4.50	5.0
cylinders		4	4	6
displacement	cu in	92	212	426
heater temperature	°F	1400		
water temperature	°F	135		
working gas		H_2		
length	in	52.75	40	56
width	in	23.75	29	33
height	in	39.25	34	44
volume	cu ft	28.50	22.8	47
weight	lb	1600	1190	1960
working gas pressure		1500		
rated bhp/rpm		130/2100†	130/2800	
min. bsfc/rpm		.425/1400†	.40/2300	
working gas pressure		3000		
rated bhp/rpm		225/2100†		218/2100
min. bsfc/rpm		.415/1500†		.37/1800

† Net values after combustion blower and water pump power have been deducted.

were also included. Further data in this area was presented at the G.M. Symposium on Emissions (1972). Papers about swash-plate mechanisms highly relevant to compact Stirling engines were presented by Maki and DeHart (1971), and by Hays and Maki (1971).

ELECTROMOTIVE DIVISION ENGINES

Cleveland Diesel Engine Division

Work on Stirling engines at the Electromotive Division was not well documented but useful information about this activity can be compiled from the various references listed above, principally Heffner (1966), Percival (1967), Mattavi *et al.* (1969) and Percival (1974).

Work on heavy, high-power Stirling engines apparently started at the Cleveland Diesel Engine Division of General Motors about 1958. Cleveland Diesel had supplied the majority of submarine engines during the Second World War and had strong connections with the U.S. Navy. In addition, they were builders of heavy diesel engines for locomotives and surface vessels. Their interest in Stirling engines was for use as an alternative to diesel engines for submarines, river and harbour work boats, and for locomotive propulsion. The Cleveland Diesel Engine Division was dissolved in 1962 and the Stirling engine project was transferred to the Electromotive Division at LaGrange, Illinois.

Large marine engine

The first substantial project was the design and construction of a four cylinder 265 kW (360 hp) engine designated the Model 4-S1210 and shown in Fig. 13.8. This was basically four 65 kW (90 hp) Philips rhombic-drive engines compiled in an in-line arrangement on a common crankcase and with a common air preheater hood. The engine was constructed at Philips and, at the Electromotive Division, was fitted with a modified locomotive generator. Both units were mounted on a monstrous bed plate said to be equal to 15 per cent of the engine generator weight (Schab 1964). The engine weight was 3175 kg (7000 lb), the generator weight 2721 kg (6000 lb) and the total system was 6895 kg (15 200 lb). With a nominal power rating of 295 kW (400 hp) the specific weight was therefore 23.4 g/W (38 lbs per horsepower)! The unit was put together in a crash program to demonstrate the low noise and vibration characteristics. It was delivered to the Navy and the noise characteristics were evaluated (Schab 1964) with favourable results but with no great enthusiasm.

In addition to the noise evaluation studies, the performance of the engine was separately evaluated by Loftus (1964) principally with hydrogen, but also with helium as the working fluid. The engine achieved a maximum power of 265 kW (360 hp) at 1500 revolutions per minute with a hydrogen pressure of 13.8 MN/m^2 (2000 lb per sq in). Maximum efficiencies in the range of 28 per cent were observed at 1500 revolutions per minute and a power output of 205 kW (275 hp). On test, the engine experienced numerous operational difficulties and ran for only about 50 hours. Loftus concluded the engine was feasible for operation where

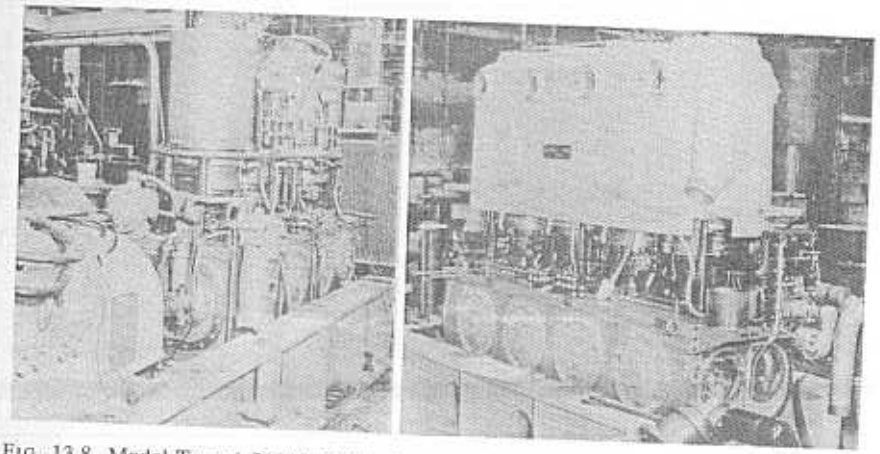


Fig. 13.8. Model Type 4-S1210 380 hp four-cylinder rhombic-drive engine (after Heffner 1966).

'extreme quietness and presumed adaptability' for low-grade fuels were of importance. However, 'the fuel economy . . . is 10–15 per cent poorer than would be expected in a diesel engine of similar size and speed'. In retrospect it is difficult to avoid the feeling that this unit did more harm than good to Navy interest in Stirling engines. What a pity that a proper level of time and effort was not expended to produce a fully developed machine for Navy evaluation. It is understood that the unit was finally consigned to the Smithsonian Institution in Washington, D.C.

Heffner (1966) in his paper includes photographs of a single-cylinder Type 1-S1050 65 kW (90 hp) engine and a two-cylinder Type 2-S1210 130 kW (180 hp) engine, both attributed to Electromotive but little is known about these engines. They were most likely both rhombic-drive engines used basically for component development in dynamometer work.

Vee engine with variable phase angle

In 1965 the Electromotive Division built and tested a four-cylinder 295 kW (400 hp) engine using four Type S1050 cylinder assemblies (Percival 1974). It was intended as an eight-cylinder vee engine but for test work only one bank of four cylinders was constructed. A photograph of the unit was given by Percival (1967). This engine was remarkable not only for its size but, principally, because the method of power control was by variation of the phase angle between volume variations in the compression and expansion spaces. Each of the four cylinders contained a conventional piston and displacer but the four displacers and the four pistons were connected to different crankshafts. Phasing between the two shafts was maintained by a sun-and-planet gear system at the rear of the engine. The phase angle between compression and expansion spaces could be readily varied by simple rotation of the sun gear thereby increasing or decreasing power as required. Sufficient rotation of the sun wheel resulted in zero and then negative power so that, in fact, the engine reversed in rotation. Response was virtually instantaneous and reversal could be obtained in little over one revolution. This facile reversal of the direction of rotation was the principal attraction of the engine for use in tugs and other marine coastal vessels requiring high manoeuverability. Fig. 13.9 contains the calculated forward and reverse characteristics of this engine, reproduced from Percival (1974). The figure appears to refer to output per cylinder.

In 1967 Electromotive Division began construction of a double-acting version of a displacer engine, called the 'W' model, having one double-acting pistion and two displacers operating in separate cylinders. The engine had a power output of 105 kW (140 hp) but no other details of this engine were given by Percival (1974).

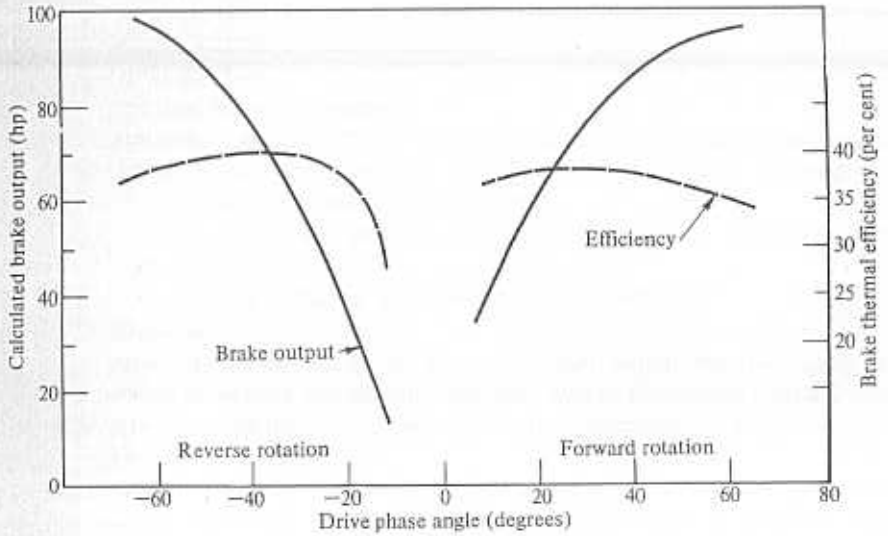


FIG. 13.9. Forward and reverse characteristics of the Electromotive four-cylinder variable-phase-angle Stirling engine (calculated; after Percival 1974).

ALLISON DIVISION AND THE SOLAR ENGINE PROJECT

So far as is known, the Allison Division was actively engaged in only one Stirling-engine development: a 3 kW (4 hp) solar space power plant. The work was done for the U.S. Air Force and has been excellently documented in a series of ten technical reports (Parker and Malik 1962) of which Volume 1 on engine design, and Volume 10 on engine experimental evaluation, are most relevant. The remaining reports are concerned with other aspects of the system. The engine was also reported by Parker and Smith (1960), by Welsh *et al.* (1959), and by Welsh and Monson (1962). The engine designated the Type PD46, shown in Fig. 13.10, was a single-cylinder rhombic-drive machine having a cylinder bore of 6.03 cm (2.375 in), a stroke of 2.84 cm (1.118 in) with helium as the working fluid at a mean cycle pressure of 10.3 MN/m^2 (1500 lb per sq in). The rated electrical output of the generator driven by the engine was 3 kW at a speed of 3000 revolutions per minute. The design overall thermal efficiency was 30 per cent.

The engine was noteworthy principally because the cylinder head and heater tubes were heated by liquid metal (NaK) at a temperature of 677 °C (1250 °F). This represents an historic first in view of the subsequent developments that have been made and are yet to come with thermal energy storage, solar power, and nuclear energy systems.

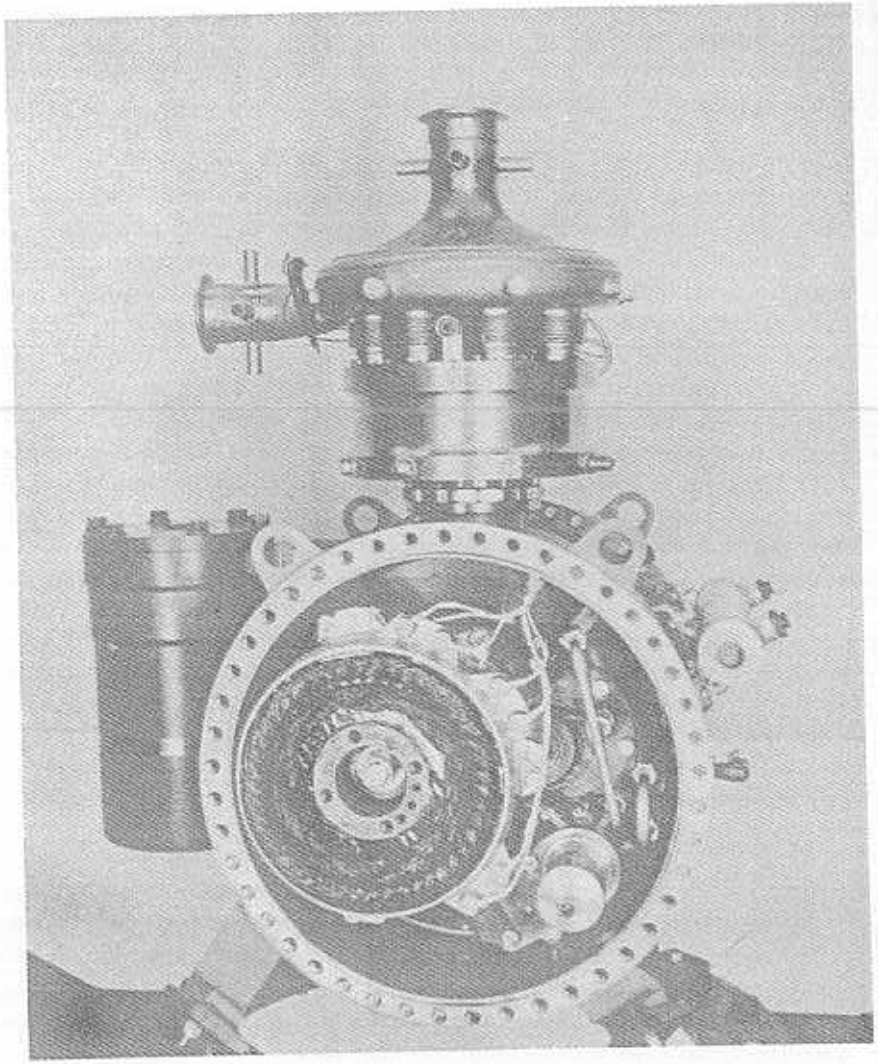


FIG. 13.10. Allison Type PD46 Stirling engine for solar-heated space power-plant.

The engine was also noteworthy as the only significant expenditure of U.S. government funds on Stirling engines for space power plants during the era of high space expenditures in the 1960s. For reasons that have never been explained the National Aeronautics and Space Administration (NASA) resolutely neglected to consider the Stirling engine for space applications and the Air Force invested only relatively minor expenditures in the Allison program.

The PD46 engine generator operated a total of 76 hours on test. The design objective of 3 kW (4 hp) electrical output at 30.5 per cent was not achieved. The actual maximum output was 2565 watts (3.5 hp) at an efficiency of 23 per cent. Such a near-miss of objective is typical of initial efforts and there is little doubt that subsequent development would have seen significant improvement in performance. Regrettably, funding for an advanced flight-qualified engine was not forthcoming and the development was abandoned. Volume 10 of the report referred to contains detailed test results of the engine operation with descriptions of the various difficulties encountered. It is a prime repository of practical operational experience and is recommended for close study.

Following the termination of the PD46 engine work, no further Stirling engine development is known to have occurred at the Allison Division. Percival (1974) has disclosed that various conceptual design studies were made for solar, chemically fuelled, and isotope-heated space power plants, for torpedo engines and for residential air conditioners.

SUMMARY

The contributions to Stirling engine technology made by General Motors in the period 1958–1970 were summarized by Percival (1974) as follows:

1. Their program developed smokeless, turbine-type burners and the direct ignition of diesel fuel.
2. Their program developed complete, self-contained Stirling Ground Power generator sets and introduced these to the U.S. Army Research Laboratories for evaluation. These were the only modern low-noise engine packages to pass formally-specified military performance tests.
3. Their program developed the first precision-control constant-speed governor for Stirling engines, which embodied entirely new concepts to the field of governing and which accomplished the following:
   (a) Speed variation at constant load $\pm\frac{1}{3}$ per cent
   (b) No load to full load droop $\frac{1}{4}$ per cent
   (c) 100 per cent sudden load change, off speed surge of 4 per cent and recovery in 6 seconds.
4. Their program developed the first automatic fuel controls for starting a Stirling engine over a wide ambient temperature range, without smoke emissions.
5. It developed the first engine to be manually started and operated entirely from hydraulic controls.
6. It developed two new hydrogen compressor systems having zero leakage capability.

7. It developed the first mechanical temperature control of proven reliability for many thousands of hours.
8. The program also initiated and completed analyses and tests of low-cost regenerator materials, particularly the proprietary MetNet material, which have shown a reduction in cost per unit power output over 50 to 1.
9. It made detailed cost and design studies with several General Motors' Divisions of a 35 kW (50 hp) single-cylinder engine, which included cost reduction techniques in cylinder construction, preheater sheet metal construction, and simplified crankcase construction.
10. The program made the following contributions to the analysis of the Stirling cycle:
    (a) It corrected errors in cooler calculations so that thermodynamic cycle heat rejection and cooler heat transfer rate were equal.
    (b) It introduced routine flow measurements of heater, cooler, regenerators, and assemblies, to provide correct data for calculation of flow losses.
    (c) It developed a theory for heat flux distribution on the Stirling heater tubes and ran experiments to show the effect of radically different heat flow patterns.
    (d) It made the first analytical and experimental determination of heat transfer film coefficients on the heater tubes and concluded they were 2 to 3 times greater than could be deduced from the literature.
    (e) It developed an original analysis of combined thermal and pressure stresses in non-uniform wall cylinders.
    (f) It added and refined calculations related to drive mechanisms.
11. The program developed piston-sealing technology which included the following items:
    (a) Systematic piston-seal studies to eliminate white-metal clearance seals which were Philips' standard at the beginning of program.
    (b) The first screening of Teflon and other self-lubricated materials (in the Allison Division studies).
    (c) A method of measuring piston leakage in operating engine.
    (d) Experimental verification of accuracy of piston-seal leakage power loss.
    (e) Data indicating an optimum combination of piston-seal leakage and friction.
    (f) Concepts for controlling the relationship between working-space and buffer-space pressures by slots in cylinder walls.
    (g) Basic ideas for the present Rulon piston rings (the development of the rings used in all Philips and GMR engines is a closely integrated result of contributions from both sources).
12. The program developed new engine-drive mechanisms and explored different types of cylinder/piston arrangements, including the following, which were all designed and tested:
    (a) The first modern, four-cylinder, 275 kW (350 hp) phase-angle control engine, which could also be directly reversed (Electromotive Division).
    (b) The first 'W' configuration, double-acting engine of 100 kW (140 hp) (Electromotive Division).
    (c) The idea of swash-plate-type axial Stirling engines (the program provided analytical and experimental verification of the low-friction characteristics of properly designed swash-plate mechanisms with hydrodynamic type bearings).
    (d) The first hermetically-sealed 7.3 kW (10 hp) engine with pressurized crankcase (Allison Division).
    (e) A single crank-displacer engine of 7.3 kW (10 hp).
    (f) The first minimum weight, 1.5 kW (2 hp) 'V' type displacer engine, with separated cylinders, which could be adapted to isotope heat sources.
    (g) The first 90 kW (120 hp), four-cylinder, in-line double acting Stirling engine for a proposed bus installation. The General Motors program was cancelled as testing just started.
13. The program developed special fuels and heat sources as follows:
    (a) The only potassium-sodium (NaK) heated engine, under a Government contract (Allison Division.)
    (b) The concept of thermal energy (heat) storage with the Stirling engine. (Initial work on heat storage began at G.M. Research in the early 1950s and led on to the first heat-storage power-plant using specially shaped aluminum oxide pellets; the first Stirling-engine automobile (the Calvair) with heat supplied from an aluminum oxide heat-storage container; and tests on lithium fluoride and lithium hydroxide as energy storage materials, including a joint program with Oak Ridge National Laboratory which determined suitability of various alloys for containment of lithium fluoride.)
    (c) Tests to demonstrate controlled combustion of lithium fuel with Freon oxidizers, including operation of a Stirling engine from lithium combustion heat.
    (d) The first Stirling engine with natural gas fuel.

Table 13.2. Summary of design or test data for General Motors Stirling engines.

	Test data							Design data								
Designation	10–36 engine	GUP 2 engine	GPU 3 engine	3015 (Calvair)	4S1210 Navy 4 Cyl.	EMD 1-S1050	EMD 2W17A	GPU 2 package	GPU 3 package	Philips Boxer	Philips in-line	GMR in-line	GMR in-line	GMR Vee 1	Swash-plate	Torpedo***
max. hp	7.47	7.3	11.2	40	380	75	138	3 kW	3 kW	120	200	148†	129	2	22	690 (gross)
@ rpm	3600	3600	3600	2500	1500	1500	1800	3600	3600	3000	3000	2000	2000	5000	2400	3000
max. efficiency	26.3	28.03	26.5	39	35	28	28.4HHV			(36.5)		33	30	28.2	28.5	52 (no comb. loss)
@ rpm	1800	2400	1900	1400	750	1200	900			3000		1500	1000	3200	2400	1200
heater wall (°F)	1400	1400	1400	1270	1202	1270	1100			1292	1292	1400	1400	1300	1264	1500
water in (°F)	75	126	100	70	90	100	100			104	104	125	125	170	150	60
working gas	H_2	H_2	H_2	H_2	H_2	H_2	H_2	H_2	H_2	He	He	H_2	He	H_2	He	H_2
F.L. Pressure (psi)	1000	1000	1000	1560	1500	1436	1100	1000	1000	1720	3140	1500	1500	1500	1500	3500
cylinders (no.)	1	1	1	1	4	1	2			4	4	4	4	1	4	5
bore (in)	2.362	2.375	2.75	3.47	5.70	5.70	6.50			3.26	3.26	4.00	4.00	1.18	1.57	3.40
stroke (in)	1.238	1.238	1.238	2.37	2.90	2.90	3.20			1.97	1.97	1.83	1.83	1.26	1.57	2.30
total displ. (in^3)	5.44	5.47	7.32	22.3	296	74.0	212.0			66.0	66.0	92	92	1.38	12.1	104.0
hp/in^3	1.37	1.33	1.53	1.79	1.28	1.01	0.65			1.82	3.04	1.61	1.40	1.45	1.82	6.64
piston speed (max. rpm)										—	984 —	—	610 —	1050	629	1150
weight (lbm)	127.4*	90*	127*	550*	5000	2300‡	3800‡	382	350.0	850	880	1000	1000	24.6‡	200	600
L (in)	14	12	14	17.5	74	36	36.25	35	38.25	59	44.5	41	41	10	30	21 (Inc. gear)
W (in)	14	16.5	15.5	17	40	27.5	62.25	24	24.75	39.7	17.3	18	18	10.5 dia	12 dia	19 dia
H (in)	28	26.5	28	37.25	76	65	85.63	32	28.6	17.3	37.9	34	34			
Volume (Ft^3)	3.17	3.04	3.51	6.42	130.0	37.2	111.0	15.6	15.7	23.5§	16.9§	14.5§	14.5§	.50	1.96	3.44
hp/ft^3	2.36	2.40	3.19	6.25	2.92	2.02	1.24			5.1	11.82	10.2	8.9	4.00	11.21	201.00
lbs/hp	17.1	12.3	11.30	13.75	13.20	30.60‡	27.50‡			7.10	4.40	6.75	7.75	12.30	9.10	.87
lbs/ft^2	40.2	29.6	36.2	86.0	38.5	61.90‡	34.20‡	24.5	22.3	36.2	52.1	69.0	69.0	49.2	102	174
preheater weight (lbm)		21	21	64		97	194			150						
BMEP (psi)	151	147	168	284	339	268	143			240	400	318	277	115	299	815

* bare engine with preheater.
‡ without flywheel.
§ volumes include accessories.
** with ring for isotope heat but less isotope capsules.
*** all data without comb. system.
† 200 bhp, 3000 rpm, 1600 psi.

14. The program designed, built, and operated the 7.3 kW (10 hp) engine generator sets for the first Stirling-electric hybrid passenger cars, known as the Stir-Lec I and II. It also made the first exhaust-emission tests on a Stirling powered vehicle.
15. The program performed the first endurance tests approaching 'military standard' severity on a Stirling generator set.
16. The program made the following application and design studies:
    (a) It designed complete Stirling engine submarine powerplant with LiF heat storage and submitted proposal to the U.S. Navy in 1959.
    (b) It made a design study of a high-performance swash-plate drive torpedo engine of 440 kW (600 hp).
    (c) It included design studies of 7.3 and 22 kW (10 and 30 hp) Stirling-LiF thermal energy storage plants for small research submarines, and 735–3680 kW (1000–5000 hp) plants for larger military submarines incorporating aluminum oxide heat storage.
    (d) It designed rhombic engines for installation in buses.
    (e) It included conceptual designs of a swash-plate automobile engine for Oldsmobile.
    (f) At the Allison Division, it included conceptual design studies of a solar-heated power-plant, a chemically fuelled space power-plant, an isotope-heated space power-plant, a rhombic-drive torpedo engine, an ASW power-plant to operate from hydrogen peroxide, a residential air-conditioner engine, and a back-pack power-plant to operate from indigenous fuels.
17. The program published eight engineering papers on Stirling engines, including their heat sources and application studies.
18. It also produced internally 330 research reports and technical

memoranda at the Research Laboratories.

Table 13.2 is a summary given by Percival (1974) of test or design data for many of the General Motors Stirling engines.

CLOSURE

Work on Stirling engines at General Motors started in 1958 and formally ceased early in 1970. It stands as the most sustained and concentrated effort ever made outside the Philips Research Laboratories and the achievements realized are impressive. The big question, of course, is why did they stop? Why did they stop, suddenly, in February 1970, with the Model 4L23 bus engine on dynamometer test and projected for vehicle installation a month or two later?

It is a question that has never been addressed in the literature but is important, for it has profoundly affected the development of Stirling engines in North America and the world. It is clear that the people working on the program were taken by surprise when the decision was made to terminate it. They were by no means at the end of the road. In fact, the program was rapidly accelerating with the culmination of many independent development studies in the design of optimized engines.

Hearsay has it that, at the time, the top management of General Motors was preoccupied by incipient problems concerning the brakes on G.M. school buses. Thus, when the time for renewal of the Philips licence came, it was easier to stop the work than to evaluate the situation properly. So it was that one day, out of the blue, the Stirling engine team was told to 'stop work on engines; tomorrow start on emission controls'. If hearsay is correct, then, in light of developments in the seventies, the decision must qualify as one of the most crass and stupid that industrial management has ever made.

Percival (1974) attributes the stimulation of interest in Stirling engines at General Motors to Mr. Arthur Underwood, not only for the initial interest in 1948 but also for successful negotiation with Philips of a licence agreement in 1958. In the light of the abrupt termination of work in 1970, it is not without interest that Mr. Underwood retired in 1969 as Manager of the General Motors Research Laboratories; he remains a public champion of Stirling engines (Underwood 1976).

The fate of the Stirling engine hardware developed in the course of the General Motors program is unknown. At least one of the GPU3 engines came into the possession of Wayne State University, Detroit, Michigan, and was used for the combustion and emission studies reported by Davis *et al.* (1971 and 1972). The other GPU3s developed for the U.S. Army were acquired in 1976 by the NASA Lewis Research Centre as a part of the Stirling engine program for highway vehicles. Cairelli and Thieme (1977) have reported on tests carried out with these engines.

In May 1978 a collection of about 100 internal reports and memoranda prepared by General Motors during the tenure of their licence period was transmitted to NASA Lewis Research Center as a G.M. Special Publication (Heffner 1978). The reports are expected to be reproduced and circulated by the Stirling Engine Project Office of NASA Lewis or by the National Technical Information Service.

Superficial examination of a copy of the reports indicate that they contain much useful and interesting information but that they have been heavily expurgated of much material that would have added to their value. Further, it is well to recall that although a hundred reports have been made available this is in fact less than one-third of the 330 items referred to above. Many of the reports included appear to contain material referred to by Percival (1974), and if Percival's report provided the basis of selection for the material now released by General Motors, it therefore becomes the 'executive summary' of the new Special Publication. It is to be hoped that these reports will be widely distributed and further, that the remaining two-thirds of the G.M. technical reports and internal memoranda will eventually be released to the public domain.

14 MAN/MWM STIRLING ENGINES

INTRODUCTION

In 1967, two West German diesel engine manufacturers, Maschinenfabrik Augsburg-Nürnberg (MAN) and Motorenwerke Mannheim (MWM) formed the 'Entwicklungsgruppe Stirlingmotor MAN-MWM', and a licence and cooperation agreement was negotiated with Philips. The new company at Augsburg, West Germany, undertook scientific and development work directed to the development of heavy vehicle engines and underwater power systems.

Single-acting engines

Initial work included the acquisition of a 7.3 kW (10 hp) Philips 1-98 engine. Some test results obtained with this engine were included in the important review paper by Neelen *et al.* (1971) containing reviews of the Dutch, German, and Swedish developments on Stirling engines for vehicular use. The German contribution to the paper included the results for torque and efficiency characteristics, a noise spectrum distribution and an assessment of the significance of the dead space on the efficiency and power output of the 7.3 kW (10 hp) engine.

Following this early experience MAN/MWM designed a single-cylinder rhombic-drive engine, designated the Type 1-400, which developed 22 kW (30 hp) at 1500 rpm. The engine was equipped with all the auxiliaries for independent engine operation and was intended principally as a test unit for the development of a four-cylinder 90 kW (120 hp) engine designated the Type 4-400.

Alm *et al.* (1973) gave further information about the Type 1-400 and Type 4-400 engines. This paper was presented jointly with the Swedish licensees, United Stirling, but did not include a contribution from Philips. It was principally a recitation of the favourable environmental characteristics of Stirling engines (low noise and exhaust emissions). For vehicular use, however, it is noteworthy because it contains recognition by both MAN/MWM and United Stirling of their inability to reduce the production costs of rhombic-drive engines to an acceptable level. Both were therefore forced to seek cheaper alternatives in the form of multiple-cylinder double-acting engines of the Siemens type.

Double-acting engines

A cross section of the MAN/MWM four cylinder double acting engine is shown in Fig. 14.1. Two important features incorporated in the design are the simplified heater head and the air preheater assembly. Fig. 14.2 is a section of the heater-head configuration and Fig. 14.3 is a further detail

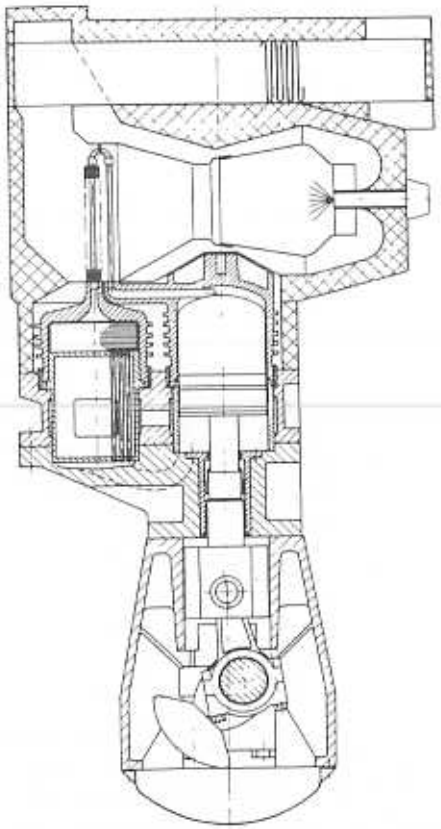


FIG. 14.1. Cross-section of MAN/MWM four-cylinder double acting Stirling engine (Alm *et al.* 1973).

of the heater tubes. The heater consists of straight tubes of heat-resistant steels, one set of which is finned, joined in pairs at the upper ends by brazing a cast U-shaped tube element. The lower ends of the tubes are brazed to the regenerator casing or the expansion space cylinder. The tubes are arranged to form a plane heater screen.

The accordion-type preheater developed by MAN/MWM is shown in Fig. 14.4. This can be made at relatively low cost by simply folding a sheet metal strip to produce the plate-type recuperative heat exchanger shown. It will be recalled that low oxides of nitrogen in the engine exhaust have been achieved by recirculation of a significant proportion of the combustion products. One way to achieve this might be to use a 'leaky' recuperative exchanger. The accordion-type preheater is clearly

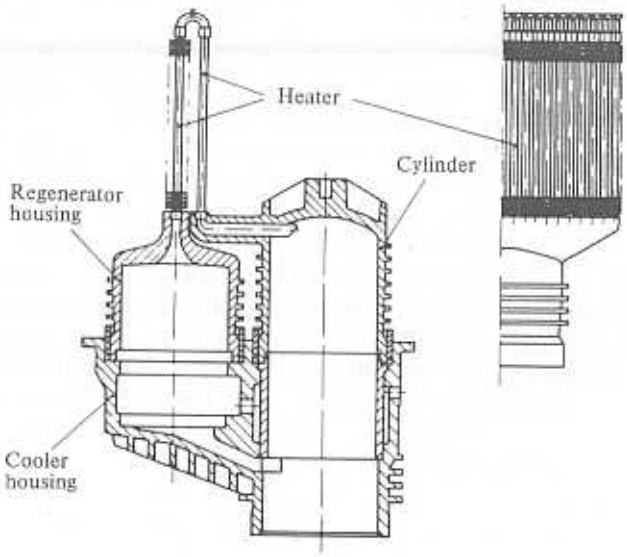


FIG. 14.2. MAN/MWM heater-head configuration (Alm *et al.* 1973).

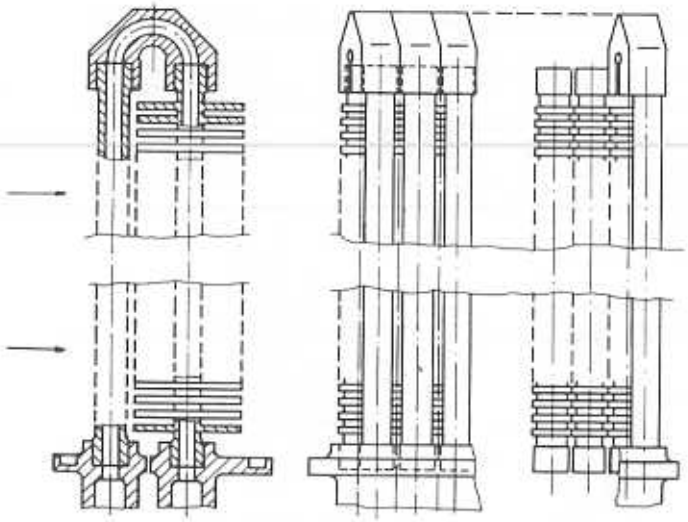


FIG. 14.3. Detail of heater tubes (Alm *et al.* 1973).

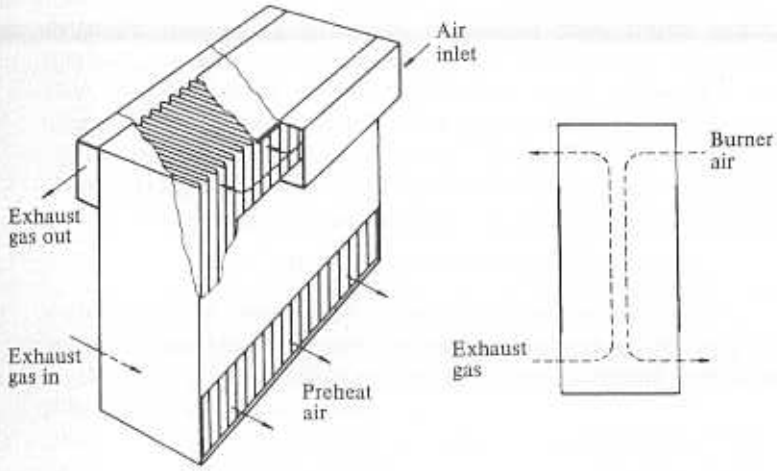


FIG. 14.4. Accordion-type air preheater (Alm *et al.* 1973).

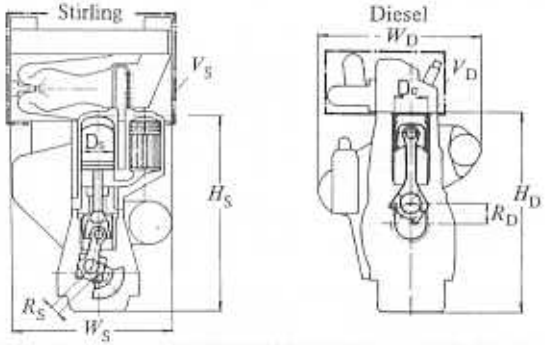
well suited for this application. If rigorous separation of the fluid stream is not necessary, the unit could be made at very low cost indeed. It provides an economically attractive alternative to the more complicated brazed plate-type recuperative exhaust heat exchangers featured elsewhere.

The new heater head and air preheater could be used on multiple or single cylinder rhombic-drive engines but it is clearly evident from their publications that MAN/MWM plan their future developments to be principally double-acting Siemens engines.

COMPARISON OF STIRLING AND DIESEL ENGINES

Zacharias (1974) drew a comparison of diesel and Stirling engines in terms of size and volume. Some of his data is summarized in Fig. 14.5. In the crank-driven, double-acting Stirling engine it was necessary to use a crosshead. Furthermore, it was customary to use a long piston/displacer so the displacer seal operated always in the cold space of the engine. Good sealing of very high pressure hydrogen or helium was necessary where the piston rod leaves the cylinder. All this conspired to increase the height of the engine measured from the oil pan to the crown of the piston to be about 25 times the crank throw. In contrast, conventional diesel practice called for a height of about 10 times the crank radius. Parenthetically Zacharias noted the corresponding height for rhombic-drive and swash-plate engines was about 32 and 25 times the crank radius, respectively.

Some savings in height for the Stirling engine could be achieved by the



$H_S \sim 25 \times R_S$	Height of crank gear	$H_D \sim 10 \times R_D$
$L_S \sim 1{\cdot}6 \times D_S \times Z$	Length of crank gear	$L_D \sim 1{\cdot}4 \times D_D \times Z$
$W_S \sim W_D$	Width of crank gear	W_D
$P_S \sim 18$ bar (at 110 bar m.p.)	BMEP	$P_D \sim 8$ bar
$V_S \sim 0{,}002$ m^3/kW	Volume combust. syst.	$V_D \sim 0{,}001$ m^3/kW

FIG. 14.5. Size proportions of Stirling and Diesel engines (Zacharias 1974).

use of a small oil sump. The lubrication requirements were far less demanding than for the diesel engine and a reduced quantity of oil was required.

The cylinder centre-line distance of Stirling engines was larger (1.6 × cylinder diameter) than that of diesel engines (1.3 × diameter) because the very high pressures used required thick walls. Also, the cylinders were divisible into hot and cold regions with heat-resistant steels used for the hot parts and a cheaper metal for the lower temperatures.

The width of both Stirling and diesel engines was found to be about the same with the inclusion of all the auxiliaries.

The work done per unit piston displacement or the brake mean effective pressure (bmep) was quoted by Zacharias to be about 0.8 MN/m^2 (8 bar) for non-turbocharged diesel engines and 1.8 MN/m^2 (18 bar) for Stirling engines having a mean cycle pressure of 11 MN/m^2 (110 bar). This significant difference (in the ratio of 18/8) was almost exactly the inverse ratio of the difference in height in terms of crank radius (from above $H_s/H_d = 25R/10R$). Thus for engines of the same height the power output was about the same despite the smaller crank radius (and hence piston displacement) of the Stirling engine.

Zacharias estimated the combustion chamber and inlet-air preheater of a Stirling engine to be about 2 cm^3/W. (90 in^3/hp) For the non-turbocharged diesel vehicle engine he estimated the cylinder heads, gas manifolds, mufflers and related parts to be about half this volume, i.e. 1 cm^3/W (45 in^3/hp). Zacharias also pointed out that the Stirling engine does allow some flexibility in arrangement so that, in fact, the increased

size of the equipment related to combustion could be accommodated such that the space requirement of the engine would be no greater than that required for a diesel engine.

In vehicular use a further handicap for the Stirling engine compared with a diesel was that the cooling system had to handle about twice the load. Furthermore, the efficiency of the Stirling engine was directly related to the cooling water temperature and increased as the temperature was reduced. This was in contrast to the diesel engine where the efficiency increased with *increase* in the cooling temperature. In the Stirling engine therefore the quantity of heat to be transferred was double and the temperature limitations were more stringent.

Zacharias was less precise in regard to a weight comparison but made the point that the specific weights of both diesel and Stirling engines were comparable. With regard to cost he was able to do no more than reiterate the attention MAN/MWM have paid to cost reduction by careful design for series production of a range of engines using common components, connecting rods, crossheads, piston rods, piston seals, heater head assemblies, regenerator/cooler units, air preheaters, etc. Aluminum was used for the cold parts of the engine, heat-resistant steels for the hot parts and, for the heater tubes, cast super alloys joined by vacuum brazing.

CYLINDER ARRANGEMENTS

Elsewhere Zacharias (1973) discussed a variety of arrangements for the MAN/MWM engine, some of which are reproduced herein. Fig. 14.6 shows a cross-section and plan for a four-cylinder double-acting in-line engine. It will be noted that this machine was designed with three combustors and had a common exhaust-gas/inlet-air preheater. Fig. 14.7 shows the cylinder connection arrangements for four- and six-cylinder double-acting in-line engines.

Fig. 14.8 shows a cross-section of a double-acting Vee-6 engine. The cylinder connections for this machine are shown in Fig. 14.9. A schematic arrangement of this engine is shown in Fig. 14.10.

HEATER-TUBE DEVELOPMENT

Some insight into the very careful and detailed development carried on by MAN/MWM was given by Zacharias (1973) in a scholarly discussion of the interrelation of computation and design in the development process. Part of this presentation included a discussion of stress and temperature distribution in heater tubes. Fig. 14.11 shows the elastic stress distribution in a heater-tube wall due to thermal strain and internal pressure. Fig. 14.12 shows the computed temperature distribution in a heater tube and Fig. 14.13 shows the computed creep-limited life expectation for heater tubes of different nickel-base steels.

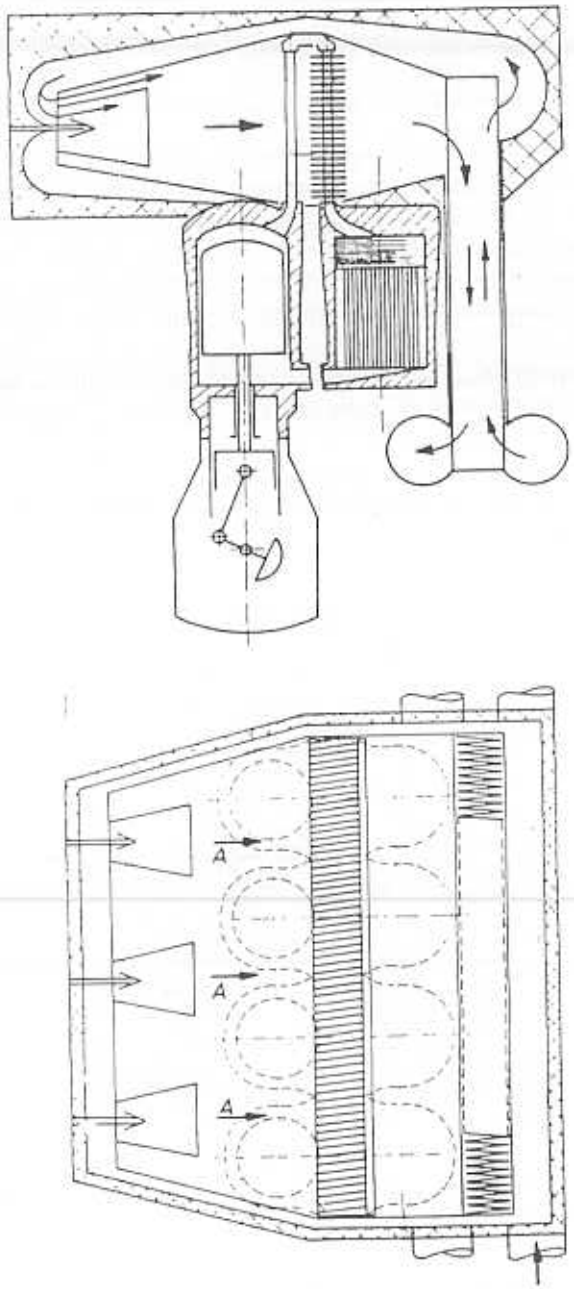


FIG. 14.6. Schematic diagram of four-cylinder in-line double-acting Stirling engine (Zacharias 1973).

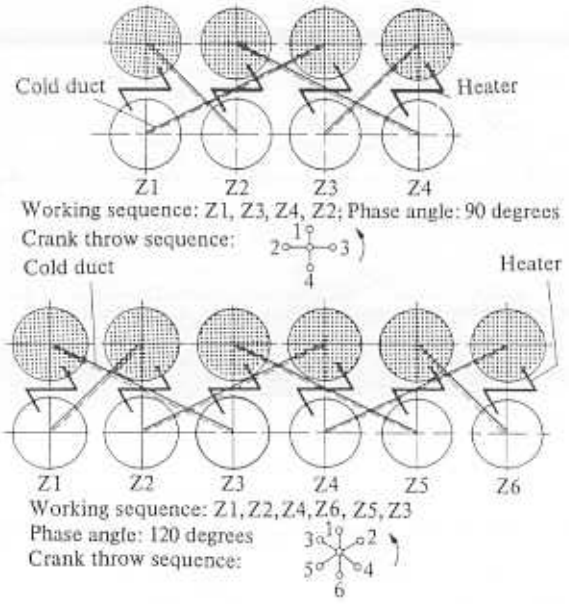


FIG. 14.7. Cylinder connections for four- and six-cylinder in-line double-acting Stirling engines (Zacharias 1973).

The arrangements for transferring high-temperature heat available from the source (combustion gas) to the working fluid through the cylinder walls or heater tube walls are always of particular interest to the designer of Stirling engines. The MAN/MWM heater was an elegant and functional system but it can be seen from Fig. 14.12 that the temperature distribution was by no means uniform in the tube. The highest temperatures occurred on the upstream or leading face of the tube. Thus the thermal expansion of the tube along its length, at the front, was greater than at the rear and so to minimize thermal stress the tube must be allowed to bend. The second row of tubes in the heater was finned to increase the area for heat transfer and so compensate for the lower downstream combustion gas temperatures. Because of the fins, a similar temperature distribution to that for the first tube row was likely to be found in the second tubes. These would therefore bend sympathetically with the first tubes and so could be joined at the top end by the rigid U connection as shown in Fig. 14.3. To minimize dead volume in the engine the heaters were short and of small diameter. For high efficiency they operated at the maximum temperature tolerated by the heater-tube material. Satisfactory design could only be achieved by careful matching of the internal engine thermodynamic cycle with the temperature dis-

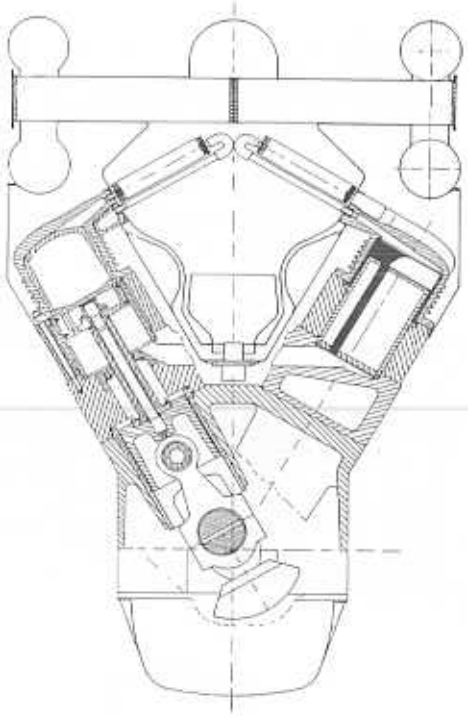


FIG. 14.8. Cross-section of MAN/MWM Vee 6-cylinder double-acting Stirling engine (Zacharias 1973).

tribution resulting from the heat-transfer process, and the strength characteristics resulting from internal pressure, thermal strain, and anchor restraints.

Some insight into the computational procedures utilized at MAN/MWM was provided by Feurer (1973) in a discussion of some aspects of the 'degrees of freedom in the layout of Stirling engines'. The material presented was intensely interesting and so far as is known has not been given elsewhere. It is reproduced in extended form in Chapter 4, on the theoretical analysis of Stirling engines.

COMPOUND ENGINES AND MULTI-STAGE COMBUSTION

Earlier, Zacharias (1971b) examined the external system of combustion in a Stirling engine. The study was a scholarly work (in German) beyond the scope of our discussion here but provided an interesting catalogue of losses and the effects on combustion efficiency. He considered the

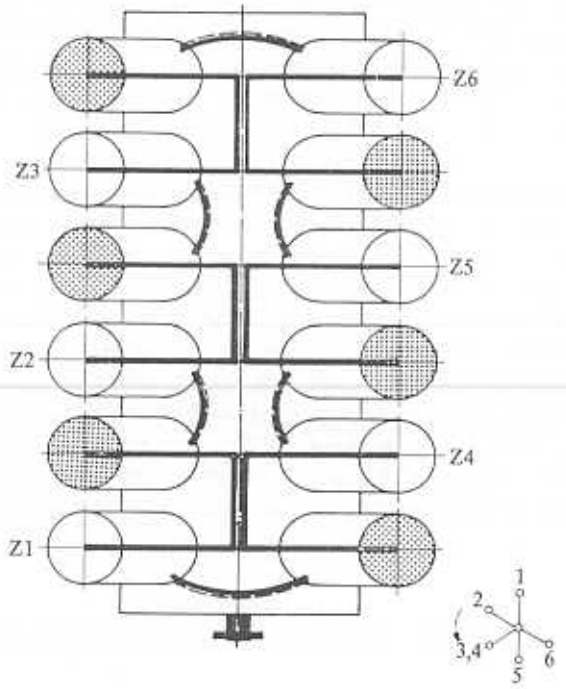


FIG. 14.9. Cylinder connections for Vee 6-cylinder double-acting Stirling engine (Zacharias 1973).

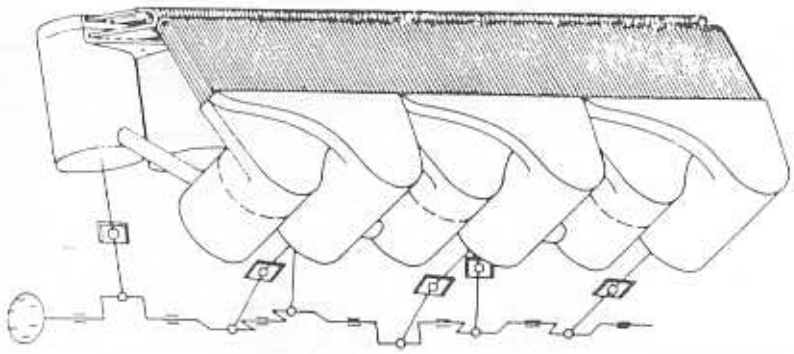


FIG. 14.10. Schematic diagram of Vee 6-cylinder double-acting Stirling engine (Zacharias 1973).

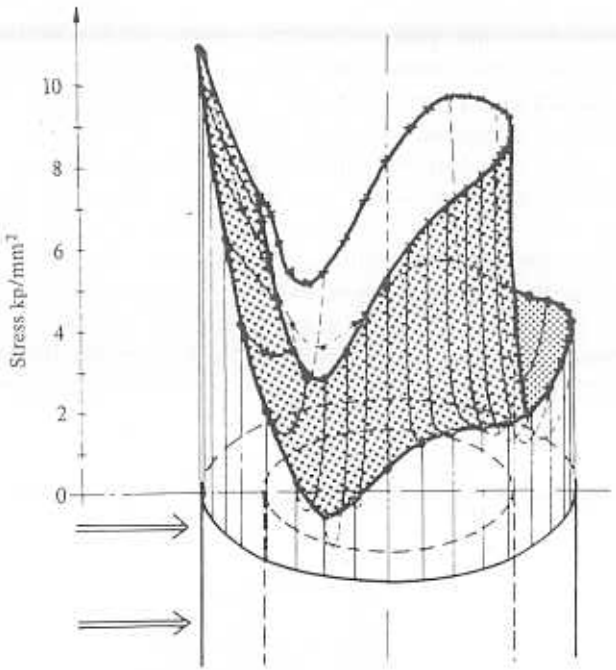


FIG. 14.11. Elastic stress distribution in heater-tube walls due to thermal strain and internal pressure (Zacharias 1973).

requirements that must be met by the combusion system, and examined a number of alternative designs for the combustion system including:

(a) conventional atmospheric-pressure single-stage combustion with air movement by mechanically-driven fan.
(b) multi-stage combustion at low pressure with mechanically-driven fan.
(c) single-stage combustion at elevated pressures in combination with a turbine and compressor (a compound engine).

Zacharias concluded that there was no thermodynamic advantage to be gained from the use of a turbine/compressor unit provided an effective exhaust-gas/inlet-air preheater can be incorporated into the low pressure system. He further concluded that multi-stage combustion systems should be reserved for high power units where the complex control system might be justified. For engines of low to moderate output (e.g. vechicle engines), the conventional single-stage low-pressure combustion system had advantageous economic and thermodynamic characteristics.

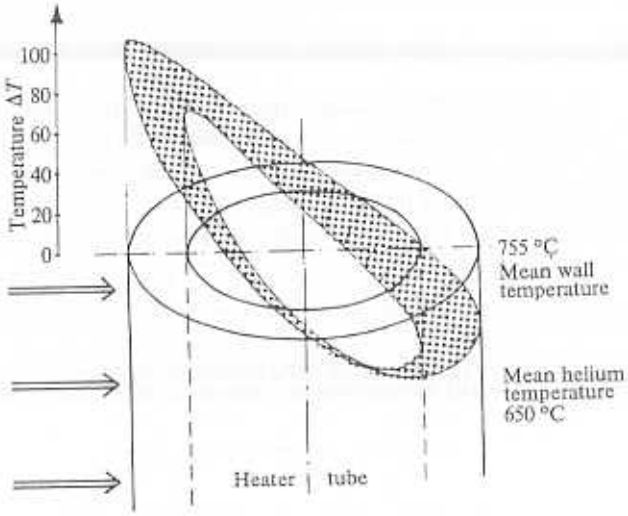


FIG. 14.12. Temperature distribution in heater-tube walls (Zacharias 1973).

CURRENT ACTIVITIES

Little is known about current activities and recent developments at MAN/MWM on Stirling engines. Following the important paper by Zacharias in 1974 little further information was published until the general review paper by Zacharias (1977a). This contained a broad survey of the characteristics of double-acting engines and a brief account

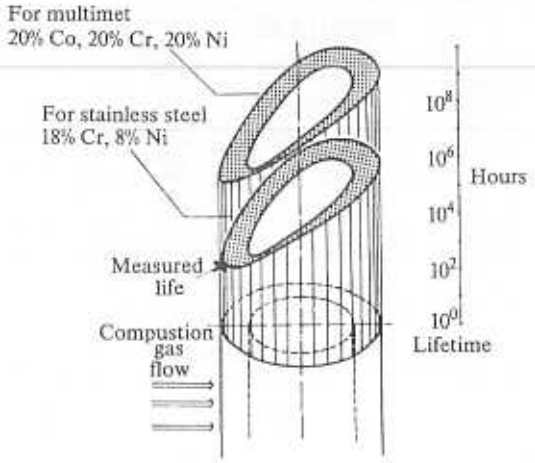


FIG. 14.13. Computed creep life expectation for heater tubes of nickel-base steels (Neelen *et al.* 1971).

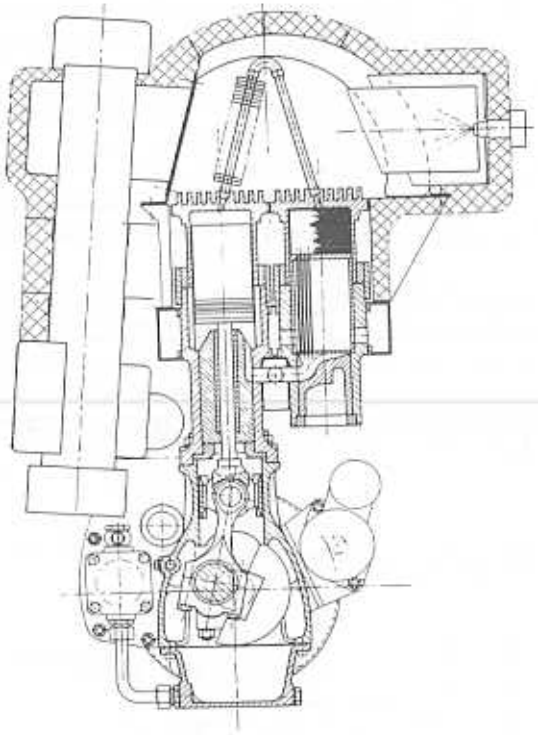


FIG. 14.14. Cross-section of MAN/MWM four-cylinder Type 4-100 double-acting Stirling engine (after Zacharias 1977).

of various Stirling-engine programmes in progress throughout the world. The principal activity of MAN/MWM was the development, in four-cylinder versions, of the MAN/MWM Type 4-100 engine shown in Fig. 14.14. Six- and eight-cylinder versions were said to be in prospect. Another interesting machine, briefly discussed by Zacharias in the 1977 paper, was the Stirling engine hydroelectric generator shown in Fig. 14.15. This unit was the result of a cooperative development program between MAN/MWM and the Battelle Institut, Frankfurt. The unit was rated at 15 kVA and operated at 3000 revolutions per minute with helium as the working fluid at 12 MN/m^2 (120 bar) pressure. The overall system efficiency was 25 per cent, the system envelope measured 970 × 600 × 600 mm (38.18 × 23.62 × 23.62 in) and the total system weighed 300 kg (661.5 lb_m).

It is understood that marine applications for Stirling engines, both surface and underwater, have always been part of the MAN/MWM

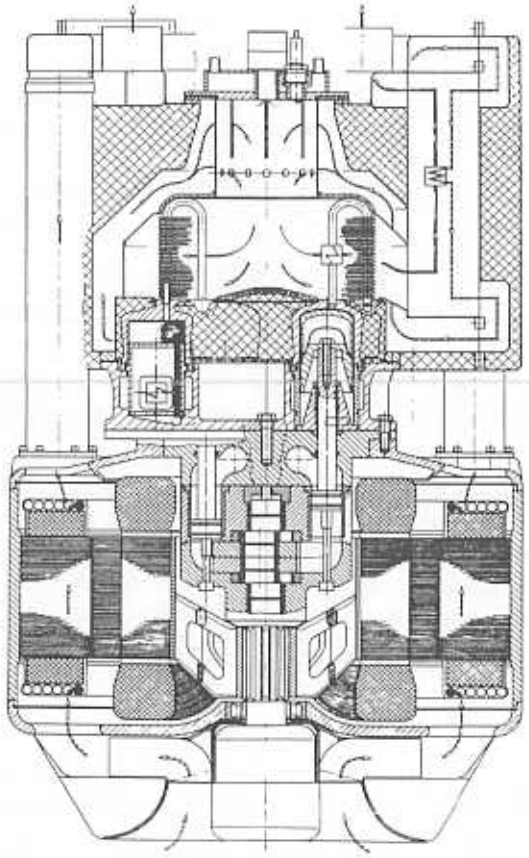


FIG. 14.15. Cross-section of MAN/MWM—Battelle Institut 15 kVA Stirling-engine hydroelectric generator (after Zacharias 1977).

development program. Yet none of their publications have acknowledged this. It is further understood that this work is classified and sponsored by the German Ministry of Defence. It is said that the principal emphasis of the work is the development of underwater power systems of 515 to 735 kW (700 to 1000 hp) capacity. This, of course, is simply hearsay and should be treated speculatively until independently confirmed.

15 UNITED STIRLING ENGINES

INTRODUCTION

THE Swedish company, United Stirling A.B., was formed in 1968 by Forenade Fabriksverken (FFV) and Kockums Mekaniska Verkstads A. B. FFV is a defence-related industrial group owned by the Swedish government. Kockums is a large publicly-owned Swedish company having its main business in shipbuilding and the lumber industry.

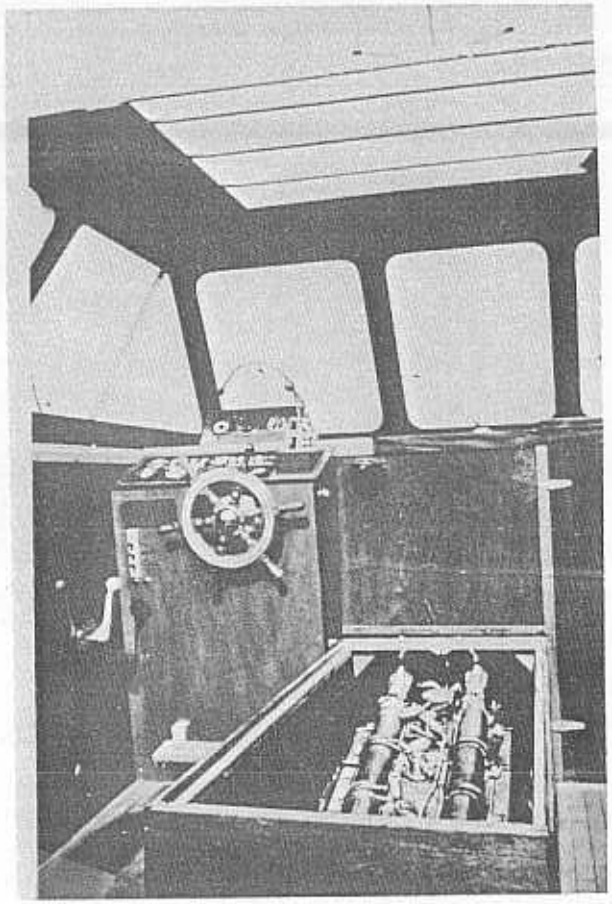
The declared aim of United Stirling is to commercialize the Stirling engine. Following a decade of development they appear to be well on the way to achieving this aim with both the technology and the will to do it. United Stirling will not manufacture engines on a production basis, but rather, will serve as design and development consultants for established engine manufacturers. An important development announced in 1977 by the U.S. Department of Energy was the formation of a second major Stirling engine development group for automotive engines including United Stirling, of Malmo, Sweden, Mechanical Technology Inc., of Latham, New York, and American Motors Inc., of Detroit, Michigan.

SINGLE-ACTING VEHICLE ENGINES

Since inception of United Stirling a decade ago, the vehicular application of Stirling engines has dominated their papers in the open literature. Principal emphasis has been given to the engines' advantageous characteristics from the aspect of atmospheric pollution, noise, and more recently, omnivorous fuel capacity and high thermal efficiency. Marine applications, in particular underwater power systems, have also occupied substantial effort and attention, but little in this field has been reported.

Soon after it was founded, the company became a Philips licence holder. In the important paper by Neelen *et al.* (1971), outlining work by United Stirling, MAN/MWM, and Philips, it was mentioned that work on the four-cylinder in-line rhombic-drive engine Type 4-235 was started at Philips in mid-1968 at the request of United Stirling. This engine has been described by deWilde de Ligne (1971). Swept volume was 235 cm^3 per cylinder, the nominal speed and power were 3000 rpm and 145 kW (200 hp) at a mean pressure of 22.3 MN/m^2 (220 atm) helium with heater tube and cooler temperatures of 700 °C and 60 °C (1292 °F and 140 °F), respectively. The early engines of this type were limited to a mean pressure of 11.1 MN/m^2 (110 atm) because the heater heads were made from conventional stainless steels rather than the high nickel alloys. Because of this low mean pressure the engines delivered only half power.

At United Stirling, engines in this form were installed for evaluation in a pleasure boat and a medium-sized bus (Alm *et al.* 1973). In the boat,



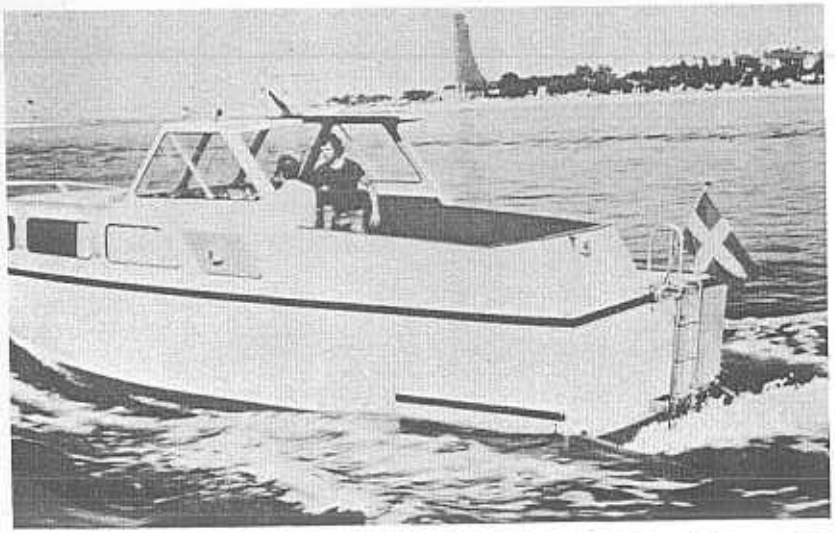


FIG. 15.1. Stirling engine Type 4-235 installation in pleasure boat. (Courtesy United Stirling)

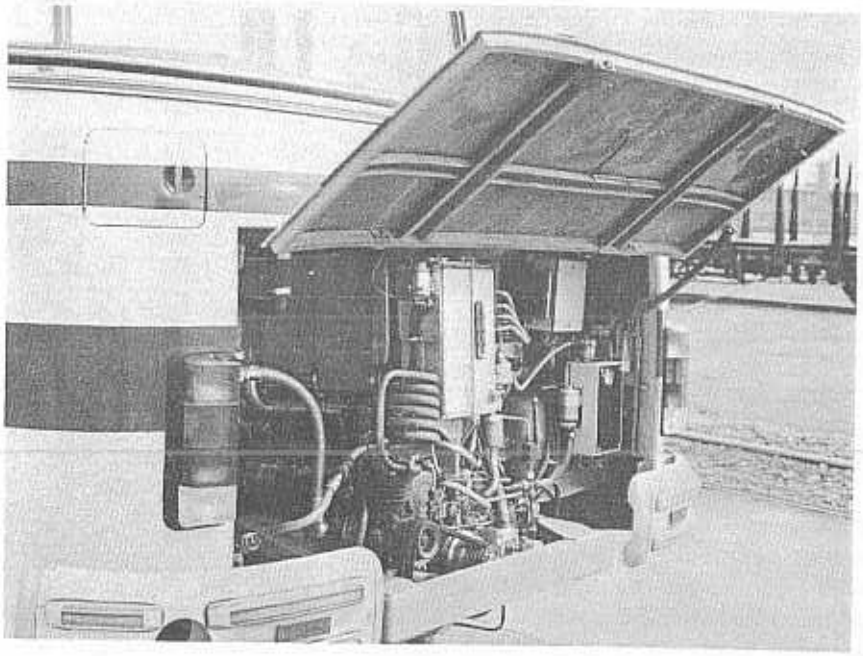


FIG. 15.2. Stirling engine Type 4-235 installation in bus. Note the cooling system installation in luggage compartment. (Courtesy United Stirling).

shown in Fig. 15.1, the engine was rated at 75 kW (100 hp), and at a speed of 2200 rpm drove the boat at 5.66 m/s (11 knots).

The bus installation is shown in Fig. 15.2. The engine was installed at the rear, in the same place as the conventional diesel engine, and was matched to an automatic split-torque type gear box comprising a hydrokinetic torque converter and two mechanical gears. Cooling was obtained by a hydrostatically-driven radiator fan. The cooling system could not be accommodated in the engine compartment but was installed in the underfloor baggage compartments. The grilles for this large cooling system can be seen in Fig. 15.2. Even with this it was necessary to increase the cooling water temperature and hence reduce power of the engine in the bus from 75 kW (100 hp) to 65 kW (80 hp) (Hallare and Rosenqvist 1977). No performance details of the bus or boat have been published but, no doubt, both provided invaluable installation and operational experience.

United Stirling also had available for laboratory use a number of the Philips single-cylinder rhombic-drive Type 1-98 engines. One of these was installed in a pleasure boat for evaluation. Another was engineered into the 2 kW (2.72 hp) generator set shown in Fig. 15.3 under a contract

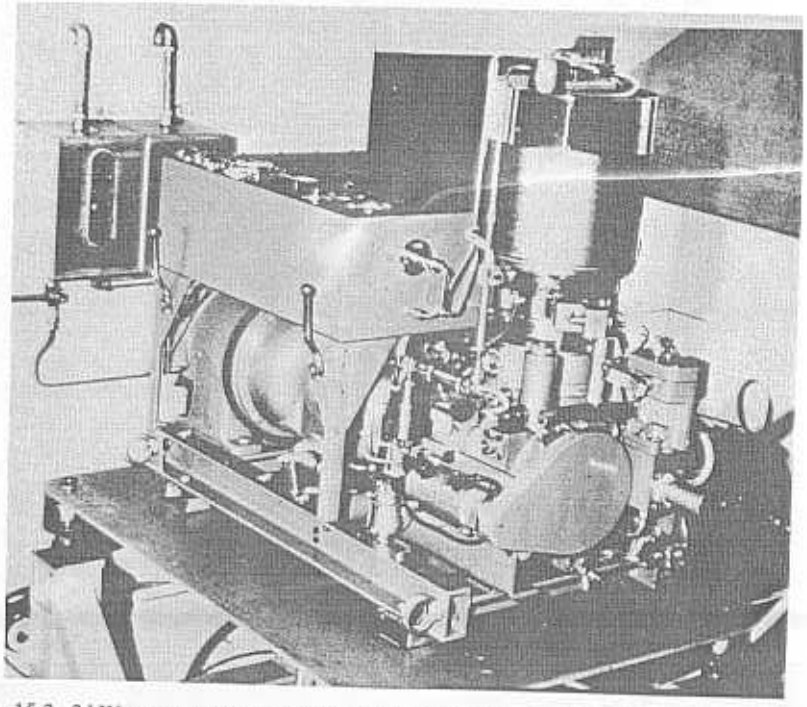


FIG. 15.3. 2 kW generator set incorporating a Philips Type 1–98, single-cylinder rhombic-drive Stirling engine. This unit has been in service with the Swedish Navy for 10 years. (Courtesy United Stirling).

with the Royal Swedish Navy. Much later (Hallare and Rosenqvist 1977), this unit was said to be still in use on a Navy patrol boat. This machine must therefore qualify for the record 'time in service' for modern Stirling engines. A report on the maintenance and operating experience with the engine would be illuminating.

Neelen *et al.* (1971) identified the first engine to be produced by United Stirling as the Type 4-615 engine shown in Fig. 15.4. This engine was similar to the Philips Type 4-235 engine in that they were both four-cylinder displacer-type rhombic-drive engines of 145 kW (200 hp) rating, but the United Stirling engine was much larger.

In the Type 4-615 engine the efficiency was accorded a high priority and, to achieve this, a large slow-running engine was designed. Whereas the speed of Philips Type 4-235 engine was 3000 rpm, the speed of the United Stirling Type 4-615 engine was only 1550 rpm with internal gears to increase the shaft output speed to 2400 rpm. This slow speed and conservative values for the heater temperature and mean cycle pressure made necessary the large swept volume of 615 cm^3 (37.5 in^3) compared to 235 cm^3 (14.34 in^3) of the smaller high speed engine. Despite these obvious handicaps Neelen *et al.* (1971) were sufficiently optimistic to

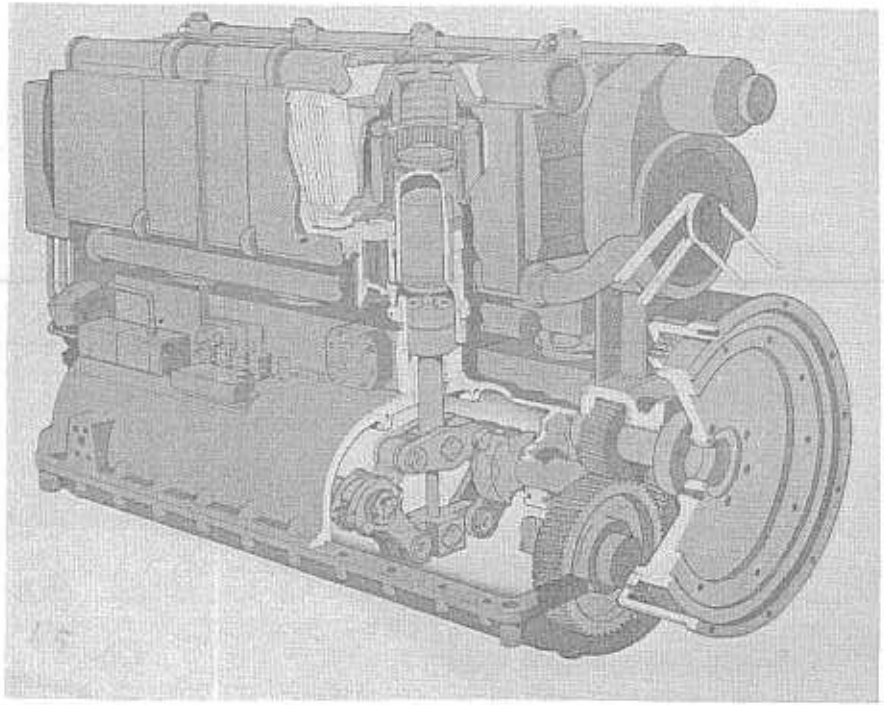


FIG. 15.4. United Stirling engine Type 4-615.

project prototype operation in 1971 and for development to series production of the engine in 1976. Installation in a city bus was projected for 1973. As in so many other things, however, economic realities prevailed and these projections were never realized.

The optimism of the 1971 paper was tempered two years later with the publication of the milestone paper by Alm *et al.* (1973). There it was wryly noted:

'The evaluation of the 4-615 engine with regard to performance, emission and costs has led to the following conclusions:

(a) although performance is satisfactory and emission characteristics are excellent the production cost of the engine makes it suitable for special applications only;

(b) major simplifications regarding basic engine design as well as components and control systems are necessary to make the engine generally competitive.'

A conversation in the early '70s is recalled at which a target figure for the cost of a production series Stirling engine was said to be twice the cost of a bus diesel engine of corresponding power. Typically, in a bus, the cost of the engine is about ten per cent of the gross vehicle cost. It was

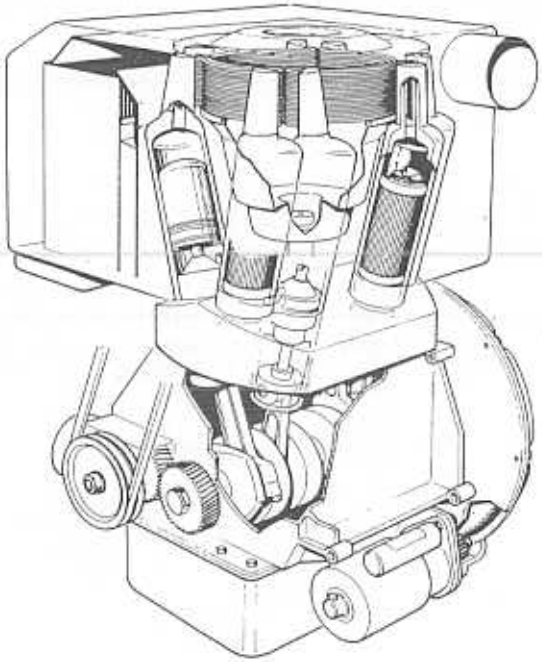


FIG. 15.5. United Stirling double-acting engine Type V4X. (Courtesy United Stirling).

said that for a ten per cent increase in the bus price (allowing the engine cost to double) operators would be glad to have the quiet, low-pollution characteristics of the Stirling engine. The target figure of twice the cost of a diesel engine could not be achieved. It is understood the most realistic estimates for production costs for multi-cylinder displacer-type rhombic-drive engines were calculated to be nearer *three* times the cost of diesel of the same power.

DOUBLE-ACTING ENGINES

Subsequent development effort was switched from single-acting to double-acting engines. In the paper by Alm *et al.* (1973) both United Stirling and MAN/MWM announced their abandonment of the rhombic-drive engine in favour of double-acting engines with crank and connecting-rod drive mechanisms. It is likely that both companies were influenced in their decisions by the experience of the other licensee, General Motors, who had been actively developing double-acting engines since 1965.

The new engine developed by United Stirling, designated the Type V4X, is shown in cut-away form in Fig. 15.5, and in cross-section in Fig. 15.6. A schematic view of the connections for the four-cylinder double-acting engine is shown in Fig. 15.7. The new arrangement allowed the use

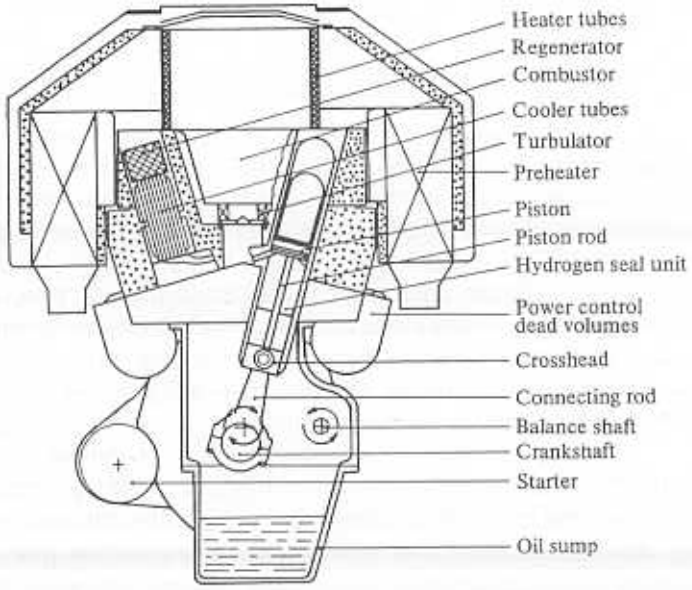


FIG. 15.6. Cross-section of four-cylinder double-acting Stirling engine Type V4X (cylinder bore 50 mm, stroke 46 mm; after Carlqvist *et al.* 1975).

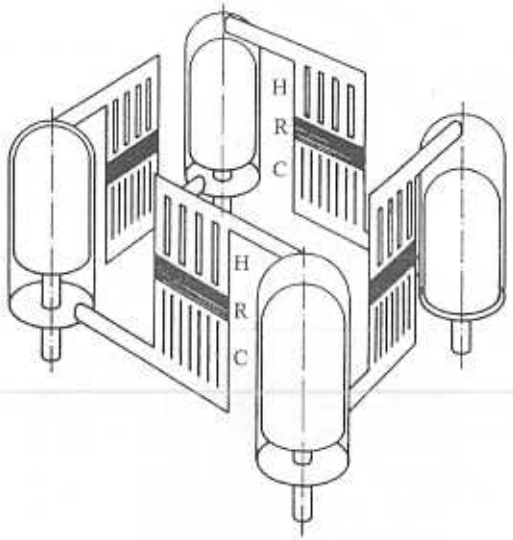


FIG. 15.7. Schematic connections for the four-cylinder double-acting Stirling engine.

of a single burner supplying heat to all four cylinders and a single recuperator for the preheated air. The cylinders, arranged in a narrow vee, allowed for a favourable cold connecting-duct arrangement and for the use of a conventional single crank/connecting-rod/crosshead drive mechanism.

Prototype engines of the new form were operating by 1973 and it was possible to project specific weights of 4.3–5.0 kg/kW (7–8 lb per hp), half the values attained with rhombic-drive machines and accompanied by concomitant savings in cost. Development of the V4 engine has been vigorously pursued, with progress reported from time to time in the literature.

Carlqvist *et al.* (1975) disclosed that the experimental engines had four cylinders arranged at a centre-line angle of $\pi/6$ radians (30°) with cylinder bores of 50 mm (1.97 in) and a stroke of 46 mm (1.81 in). Engine power was in the range of 30–40 kW with efficiencies up to 30 per cent and speeds up to 4500 rpm using hydrogen as the working fluid. The engine was designed to use established engine technology where possible, particularly in the area of bearings, crankshafts, and other components.

The shape and size of the engine were such that it could be accommodated in the engine compartment of vehicles with little modification. Rosenqvist *et al.* (1977) described the first vehicle installation undertaken in 1972/74 in a joint U.S. Ford/United Stirling program. The engine, designated V4X31, was installed in a 1972 Ford Pinto car which ran for

the first time in March 1974. Tests were terminated in December 1974 with total test distance of about 643.7 km (400 miles). This vehicle was equipped with a power train including a torque convertor and automatic transmission. The power control system of the engine was a United Stirling development utilizing a variable dead volume to control the amplitude of pressure variations and hence power output.

A second vehicle installation was undertaken with engine V4X35 in a 1974 Ford Taunus estate car. The engine and transmission are shown in Fig. 15.8 and the vehicle engine compartment in Fig. 15.9. This machine utilized a single dry-plate clutch and an all-synchronized four-speed manual gearbox. The control system for this engine was of the conventional type where power variation was obtained by adjustment of the mean pressure level in the engine.

PRODUCTION ENGINES

Despite these interesting and practical demonstrations the V4X engine was basically intended as a workhorse for component and system development. Much was accomplished and now three different production series engines have been defined (Hallare and Rosenqvist 1977) to

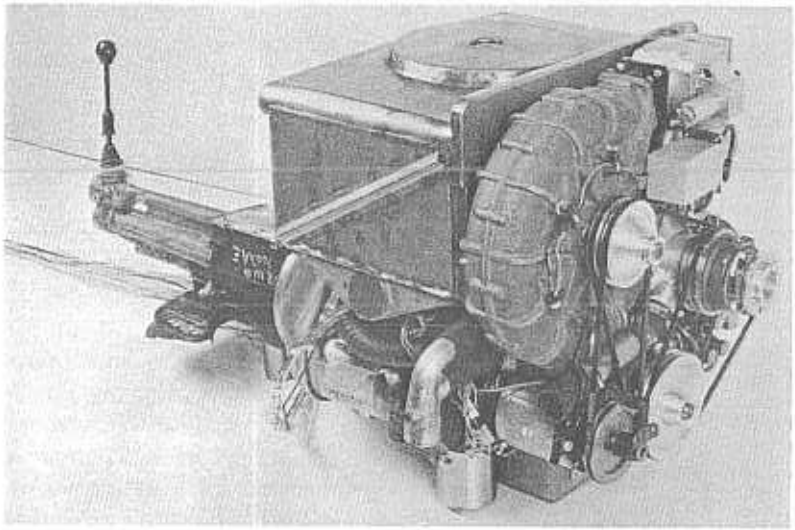


FIG. 15.8. United Stirling double-acting engine Type V4X35 with clutch and four-speed manual transmission (after Rosenqvist *et al.* 1977).

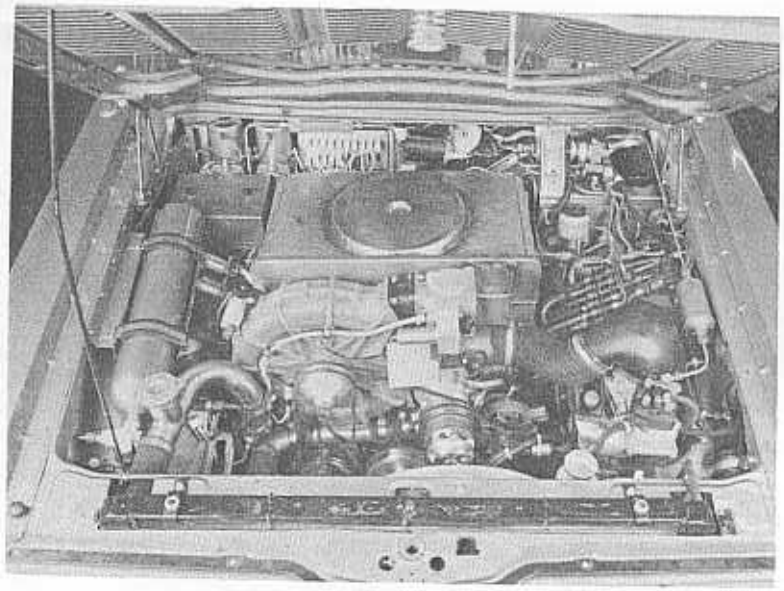


FIG. 15.9. Engine compartment of Ford Taunus car with United Stirling V4X engine installed (after Rosenqvist *et al.* 1977).

encompass a range of vehicular and other applications. The main performance criteria for these production engines are:

Type Designation	P40	P75	P150
Power (kW)	40	75	150
Speed (rpm)	4000	2400	2400
No. of cylinders	4	4	8
Maximum efficiency (installed in vehicles)	35	37	37
Weight with auxiliaries (kg)	180	350	650

Details of the bore, stroke, mean pressure and operating temperatures are unknown. The P40 engine is most likely a production series version of the V4X engine with a bore of 50 mm (1.97 in) and stroke of 46 mm (1.81 in) (Carlqvist *et al.* 1975).

A cross-section of the P150 engine is shown in Fig. 15.10. (Carlqvist *et al.* 1977). This engine is basically twin modules of the P75 engine mounted on a common crankcase. Fig. 15.11 (Carlqvist *et al.* 1977) shows a P75, V4 engine module with auxiliaries. Fig. 15.12 (Carlqvist *et al.* 1975) shows the installation envelope of a P150, V8 engine having an

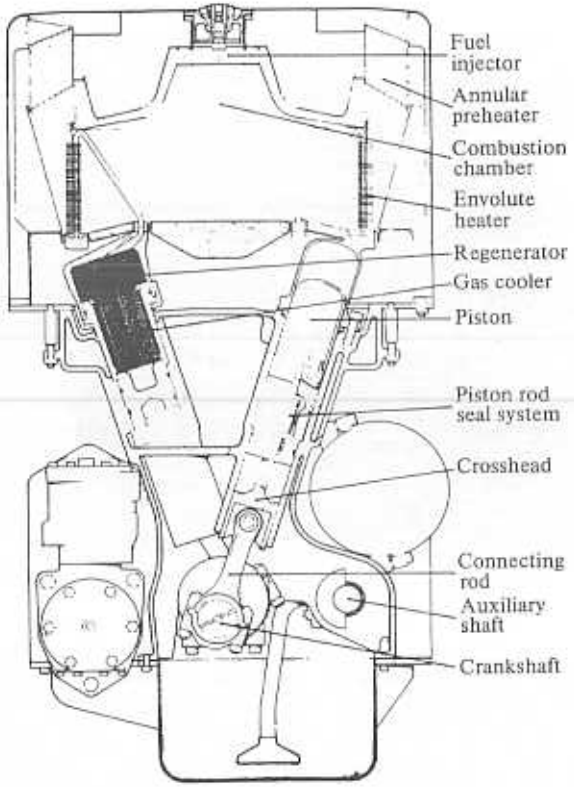


FIG. 15.10. Cross-section of United Stirling P150 double-acting Stirling engine (after Rosenqvist *et al.* 1977).

overall length of 1200 mm (47.24 in), width 680 mm (26.78 in) and height 980 mm (38.6 in). Fig. 15.13 (Rosenqvist *et al.* 1977) shows the calculated performance characteristics of a P150 engine. In 1977 four P75 V4 engines were said to be in laboratory testing, with field testing scheduled for 1979 in mine vehicles and city buses (Hallare and Rosenqvist 1977).

COMPONENT AND SYSTEM DEVELOPMENT

Much effort over the past few years has been invested in component and system development, principally the heater system, the control system, and the sealing system.

Heater system

Few specific details of the heater system have been disclosed but most of the recent review papers include a general reference to heater design.

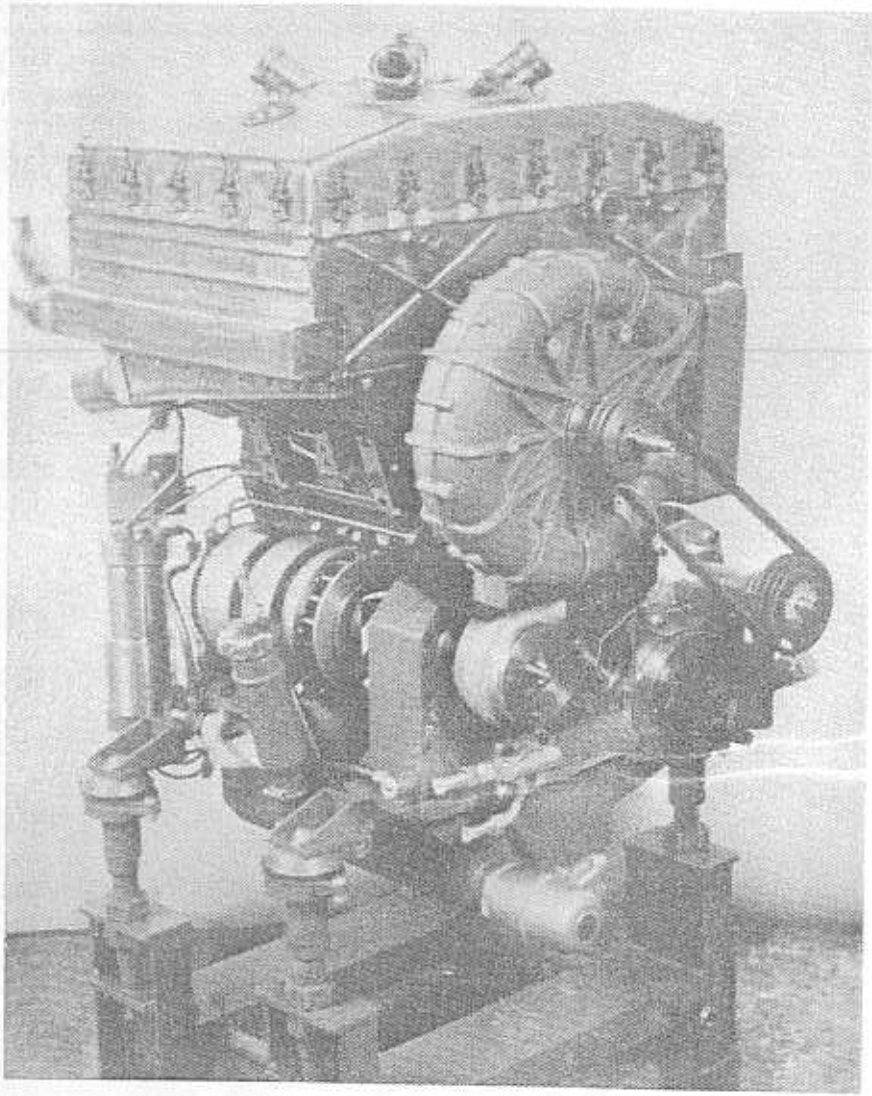


FIG. 15.11. United Stirling 75 kW double-acting engine Type P.75.V4. (Engine module with auxiliaries after Hallare and Rosenqvist 1977.)

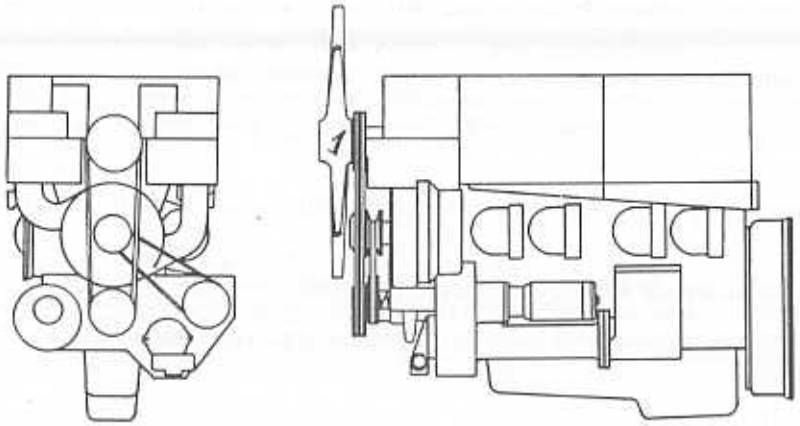


FIG. 15.12. Installation envelope of United Stirling double-acting 150 kW P150, V8 engine. (Length 1200 mm, width 680 mm, and height 980 mm; after Carlqvist *et al.* 1975.)

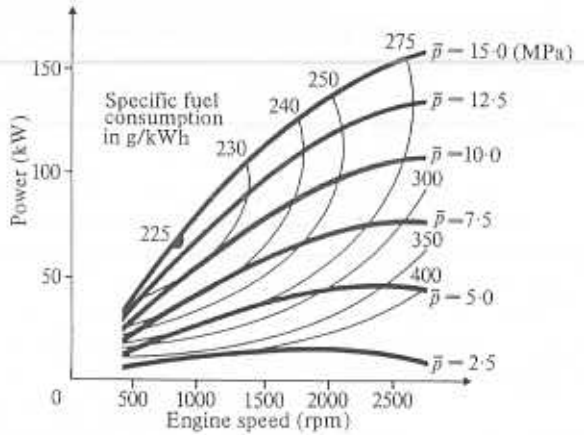


FIG. 15.13. Calculated performance characteristics of the United Stirling, double-acting 150 kW P150, V8 engine (after Hallare and Rosenqvist 1977).

Carlqvist *et al.* (1977) enumerates some of the important parameters in heater design as:

'Combustion gas flow distribution.
Heater tube temperature distribution.
Radiation contribution to heat transfer.
Heater tube stresses, both creep stresses and thermal fatigue.
Heater tube outside heat transfer.
Heater tube inside heat transfer.
Heater tube inside flow losses.
Exhaust gas leakage (exhaust gas bypassing the tubes).
Manufacturing methods and costs'.

In recent papers United Stirling show the three types of heater-head tube arrangements that have been evaluated. These are shown in Fig. 15.14. The 'tower' type heater consisted of a double row of curved

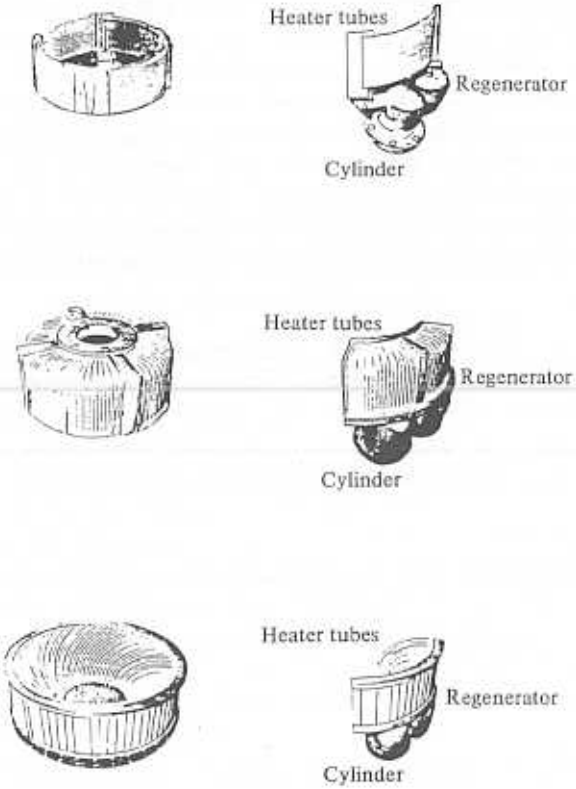


FIG. 15.14. Types of heater head evaluated in United Stirling program (after Hallare and Rosenqvist 1977).

horizontal tubes connected to vertical manifolds, or towers, coupled to the cylinder head or regenerator case.

The 'temple' type had vertical tubes connected with horizontal manifold rings at the top and bottom of the tube assembly. Temple heaters with both single and double rows of tubes have been made.

The 'envolute' type heater comprised a double row of tubes between horizontal upper and lower manifolds. The inner tubes were inclined at varying angles so that the clearance between the tubes was constant. The outer ring of tubes was vertical with extended area fins to improve the heat transfer.

In the recent papers it was emphasized these three types of head represent valuable background for future improved designs. They report that endurance testing has started, that production methods are being developed and, while corrosion has not yet been a problem, it is under study for different fuels and contaminants.

Hallare and Rosenqvist (1977) mention that, 'Sufficient knowledge has been available for quite some time how to design for high performance, avoiding creep temperature shocks, local overheating etc...' They also comment that, 'Several attempts to develop a simplified heater head have resulted in lowered engine performance...'

The materials of the heater assemblies have not been discussed in the recent papers. However, Carlqvist *et al.* (1977) classify efficiency levels of 38–40 per cent as E1, defined to be:

'Engines optimized for good efficiency using expansion gas temperatures in the range 650–700 °C (1202–1292 °F). This level represents targets for the present development and involves a well established materials technology.'

Level E2 (thermal efficiencies in the range 40–42 per cent) were defined as:

'Engines optimized for good efficiency using expansion gas temperatures in the range of 700–750 °C (1292-1382 °F). This level represents an increase in operating temperature that is considered to be compatible with the use of improved but still metallic materials in the heater.'

Level H (for engines having thermal efficiencies in excess of 43 per cent) was defined as:

'Engines optimized for good efficiency using expansion gas temperatures in the range of 1000–1100 °C (1832–2012 °F) require materials of improved heat resistant properties, for example, ceramics. Such high temperature engines represent a major development step and are only mentioned here for reference.'

This latter level provides a tantalizing glimpse of the future. Elsewhere Carlqvist *et al.* (1975) have referred to the use of 'high strength silicon carbides and silicon nitrides... for future Stirling engines. Maximum wall temperatures of the order of 1300–1400 °C (2372 to 2552 °F) seem attainable with these materials...'

Another important element of the heater system is the air preheater utilizing heat in the exhaust gas but little has been disclosed of United Stirling's effort in this area. Carlqvist *et al.* (1975) mentioned that the V4X engine was equipped with a recuperative metal plate type mainly because the technique for manufacturing was well established. The preheaters were of the counterflow type and it was found convenient to have one on each side of the engine, outside the vee.

It was also mentioned that testing of ceramic rotary regenerators for preheaters was in progress. The preheater in Fig. 15.10, a cross-section of the double-acting P150 V8 engine, looks to be of the recuperative plate type but the annular design and the general shape clearly lends itself to adaptation to the rotary ceramic type.

Piston rod seal

The change from displacer-type rhombic-drive engines to double-acting engines effected a simplification to the seal problem. In the rhombic-drive machine there are two pressure seals (working space to ambient) per cylinder, one for the pistons and one for the displacer rod (passing through the piston). In the double-acting machine there are still two seals, one for the piston and one for the piston rod. However, the seal for the piston is an 'internal' seal separating two working spaces at different pressures. Therefore, only one seal per cylinder (that for the piston rod) is required to separate the pressurized working space from ambient pressure.

Philips developed the rolling seal for the rhombic-drive engine and have carried this over to the swash-plate double-acting engines now being developed with the U.S. Ford Company.

United Stirling have sought alternatives to the rolling seal. One system of sliding seals, shown in Fig. 15.15, was described by Carlqvist *et al.* (1977) as having 'been tested for many thousand hours both in component and engine test rigs and has gradually been developed to such a degree that a viable piston rod seal can now be considered to exist...'

The United Stirling sliding seal had three elements. The upper seal element acted principally as a throttling device between the engine working space in the cylinder and the seal space inside the seal element. The pressure in the working space above the top seal varied harmonically over the full pressure range of the cycle ($p_{max} - p_{min}$). Inside the seal space the pressure was nearly uniform at a value close to the minimum cycle pressure. The seal space was connected to the working space through a gas-oil separator and filter element and one-way check valve. When the pressure in the seal space increased (because of leakage past the top seal

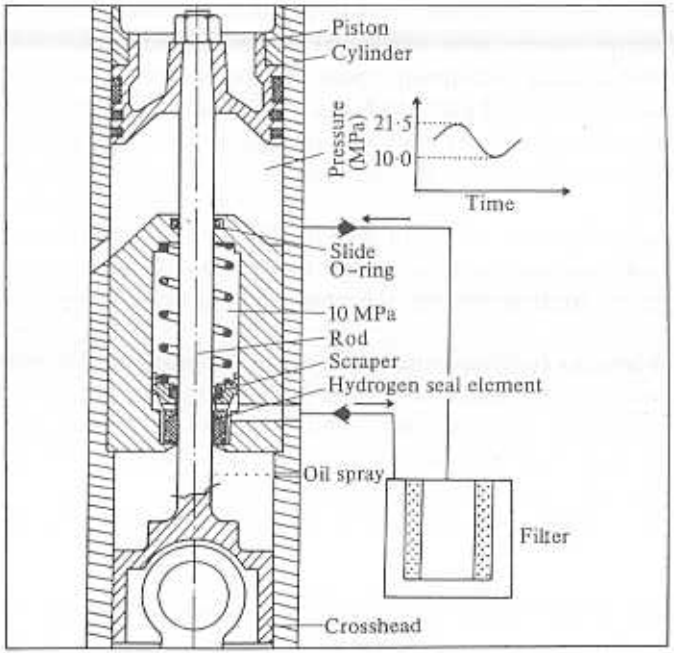


FIG. 15.15. United Stirling multiple-stage sliding seal for the displacer rods in double-acting engines (after Carlqvist *et al.* 1975).

element) the check valve opened as the pressure in the working space approached the minimum cycle pressure and fluid was returned to the working space.

The second principal element of the seal was a scraper ring which cleaned lubricating oil from the piston rod. The oil that was collected drained into the gas-oil separator and hence to the engine sump.

The third element was the main seal which maintained the pressure difference between minimum cycle pressure (of hydrogen or helium) in the seal space and the ambient pressure (of air and lubricating oil) in the crankcase space.

The exact form of the material of the reciprocating pressure seal has not been disclosed by United Stirling, but was described by Kitzner (1977b) as Rulon. Clearly, it is not a hermetic seal. Carlqvist *et al.* (1977) have advised that with this seal:

'Leakage rates of working gas, normally hydrogen in modern engines, has continuously been reduced by intensive development work on component test rigs and on engine test rigs. A characteristic hydrogen leakage rate for a P150 engine would at present development status be 1.0 nl/h at 15 MN/m^2 mean working pressure.'

To cope with seal leakage the provision of a storage container of the working fluid was foreseen. A P150 V8 engine operating 8 hours per day with a 3-month refill interval would require a 6.5×10^3 cm^3 (397 in^3) gas bottle decreasing in charge pressure from 25 to 20 MN/m^2 (3625 to 2900 lb per sq in). These values were obtained in a 600-hour test and endurance runs up to 3000 hours have been made. The total seal friction losses were stated (Carlqvist *et al.* 1977) to be 2.5 kW (3.4 hp) for a P150 V8 engine at full speed on full load.

Serious questions concerning the seal unit remain unanswered or unaddressed in recent papers by United Stirling. An obvious difficulty is the seal life. Reference to Fig. 15.10, showing a cross-section of the P150 V4 engine, suggest that the seal replacement will require major maintenance work on the engine. For vehicle diesel engines the customary expectation for operation between major maintenance is between 400 000 and 800 000 km (250 and 500 thousand miles). This corresponds (at 48 km per hour) to a range of 8000 to 16 000 hours. To be competitive the Stirling engine must also provide that level of reliability. Routine maintenance-free operation for 16 000 hours probably requires reproducible rig testing of twice that value, say 30 000 hours. This is an order of magnitude greater than the endurance tests discussed above.

Disposal of hydrogen leaking into the crankcase is another matter for concern. It will mix with air and oil vapour and the wide flammability limits of hydrogen in air must arouse interest in the measures taken to combat the chance of an exposion. No doubt the leakage rates of seals in good operating condition are sufficiently low to avoid problems. However, the prospect exists that any one of the 8 seals in a P150 V8 engine may become damaged, worn, or defective with consequent high levels of leakage into the crankcase. One possibility is to vent the crankcase to the combustion space but this may lead to contamination of the preheater with oil vapour.

Power control system

United Stirling have, in experimental testing, evaluated three types of system for power control:

(a) mean pressure
(b) pressure amplitude variation
(c) bypass control.

A simplified diagram of the mean pressure variation power control system is shown in Fig. 15.16. The main constituents of the system were a storage vessel, compressor, and control valves. To increase engine power the control valve moved to allow high pressure fluid from storage to enter the engine. Admission of working fluid to the cylinders was restricted to that part of the cycle when the pressure was close to maximum. Unselec-

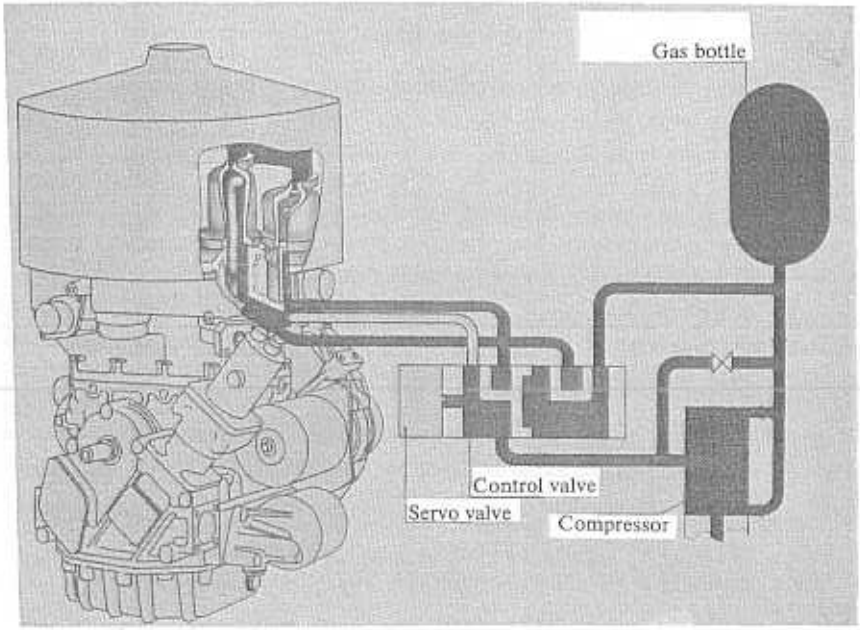


FIG. 15.16. Simplified diagram for the power-control system used on United Stirling engines. (Courtesy United Stirling).

tive supply of working fluid to the engine was said (Hallare and Rosenqvist 1977) to cause a drop in engine torque during the charging process.

To decrease power, working fluid was withdrawn from the engine cylinder and returned to the storage vessel via the compressor. Rapid decrease in power output could be gained by putting the various cylinders into mutual communication (called short-circuiting) by further movement of the control valve.

In a vehicle engine the link between the accelerator pedal and the control valve was a servo-system which for different accelerator levels moves the control valve to gain and maintain the appropriate mean pressure levels in the engine.

With this control system, a remarkably effective response characteristic can be gained as shown in Fig. 15.17. This is reproduced from Hallare and Rosenqvist (1977) and shows the speed and pressure response for the V4X, 35 engine in a free-running mode disengaged from a dynamometer but equipped with all auxiliaries. The engine was initially idling when the accelerator pedal was suddenly depressed causing a rapid increase in the engine speed and pressure level. It will be recalled that engine Type V4X, 35 was the unit used in the Ford Taunus vehicle installation.

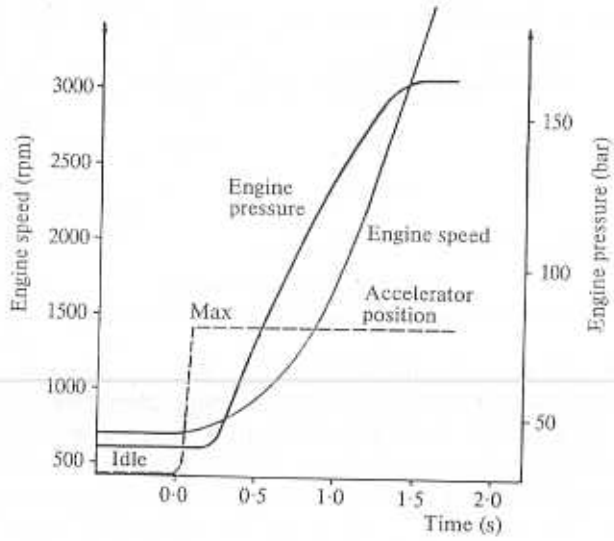


FIG. 15.17. Speed and pressure response for a United Stirling double-acting engine Type V4X, 35, in the free-running mode, i.e. driving auxiliaries but disengaged from dynamometer load (after Hallare and Rosenqvist 1977).

Air/fuel control

A secondary element of the power control system for a Stirling engine is the air/fuel control. Increase in engine output requires increased heat transfer in the heater and thus an increased fuel supply to the combustion chamber so as to maintain constant heater-head temperature. Similarly, a reduction in power calls for a reduction in the fuel supply. Regulation of the fuel supply could be the principal mechanism of power control but engine response is too slow in anything other than special constant-speed, constant-load applications. Therefore, in most cases an alternative power control system is used with supplementary air/fuel regulation.

United Stirling have sought to develop an air/fuel control system having favourable emission and fuel economy characteristics. The air/fuel system shown diagrammatically in Fig. 15.18 is reproduced from Hallare and Rosenqvist (1977) and includes a Bosch K-Jetronic unit for precise air/fuel ratio control. The temperature of the heater tubes was measured by thermocouples and the signal obtained used to regulate air flow through a throttle to the combustion air fan. The Bosch unit sensed the rate of air flow and adjusted the fuel flow to an appropriate level.

Remarkably low exhaust emission levels have been achieved. Hallare and Rosenqvist (1977) record emission levels from a 75 kW (102 hp)

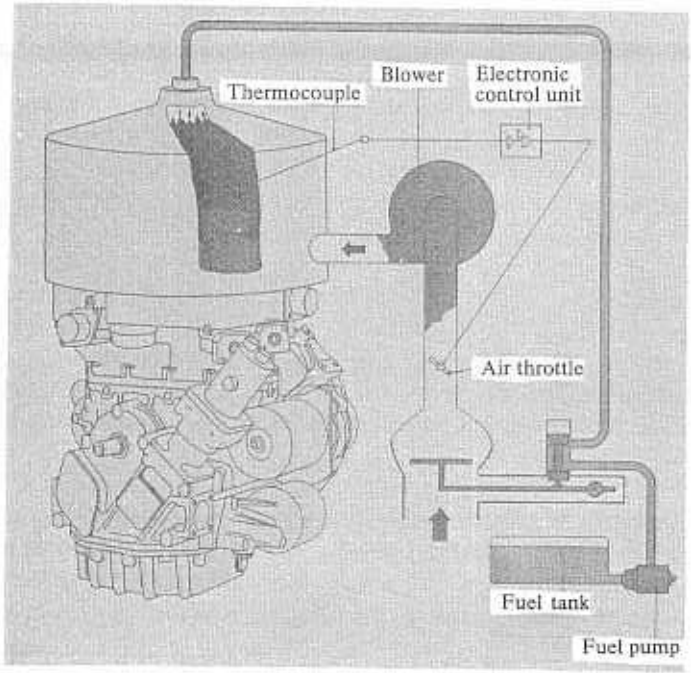


FIG. 15.18. Simplified diagram of the fuel/air control system used on United Stirling engines. (Courtesy United Stirling).

engine (presumably a Type P75), with no exhaust gas recirculation, as:

Hydrocarbons and nitrous oxides:	2.85 grams per kW hour (2.1 grams per hp hour)
Carbon monoxide:	2 grams per kW hour (1.5 grams per hp hour)

This compares with the predicted emission requirements (in California) of 4.76 and 13.6 grams per kW hour (3.5 and 10 grams per hp hour) respectively.

COSTS AND APPLICATIONS

No specific data on the costs of United Stirling engines have been published. Comparative estimates for total operating costs have been given by Rosenqvist *et al.* (1977), and by Carlqvist *et al.* (1977) for 150 kW (204 hp) Stirling and diesel engines for the typical 13 000 kg (28 665 lb_m) city-suburban delivery truck shown in Fig. 15.19. Initial cost for the Stirling engine was assumed to be 50 per cent higher than the diesel engine. The radiator cost was assumed to be twice that for the cost of the diesel engine radiator. A variety of other assumptions, some credible, some questionable, were made with the result that the Stirling engine

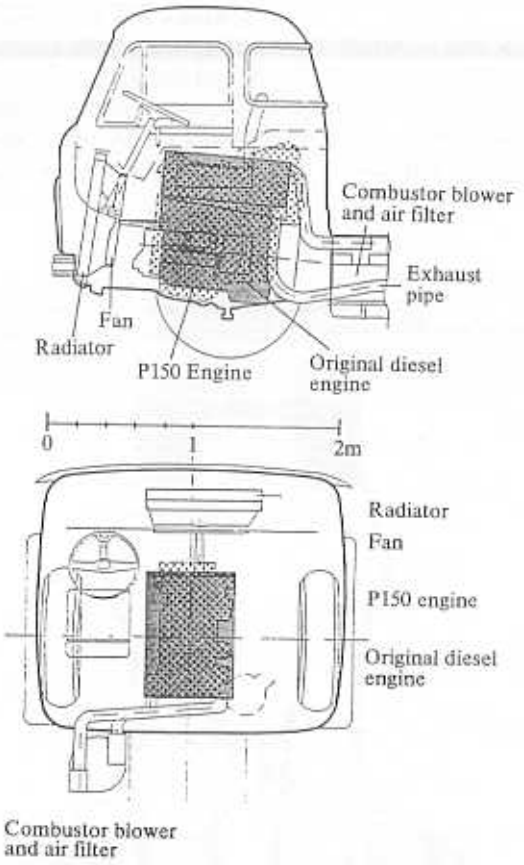


FIG. 15.19. United Stirling P150, V8 Stirling engine installation in 13-ton medium-duty truck (after Hallare and Rosenqvist 1977).

installation had a weight advantage of 40 kg (88.2 lb) compared with the diesel installation. The higher initial cost of the Stirling engine was offset by the lower operating cost so that a pay-back period of 0.5 to 5 years will be achieved depending on utilization in the range 16 000 to 100 000 km/year (9942 to 62 137 miles/year). The economic analysis depends on predicted price levels for fuels and lubricants in the 1980s and on performance penalties imposed on the diesel by anticipated regulatory emission requirements. In both these areas prognostication is so speculative that the credibility of such comparisons must be low. It would be better if specific information for the Stirling engine in terms of original and operating costs could be published for other, disinterested, parties to undertake cost comparisons.

The main thrust of United Stirling effort is clearly directed to vehicle

application but other uses are foreseen. Marine applications, both surface and underwater, have always been of interest. Some aspects of the marine and stationary applications were discussed by Carlqvist *et al.* (1977). In such applications it is possible to use lower water cooling temperatures (10–20 °C (50–68 °F)) compared with automotive use (60–70 °C (140–158 °F)), but the gain in performance is largely offset however by the preference for helium as the working fluid rather than hydrogen. It is to be expected that marine versions of the P40, P75, and P150 engines will be available.

Stationary applications will no doubt include electric power generation. Hallare and Rosenqvist (1977) touched briefly on the attractiveness of the Stirling engine for heat pumps. They outlined a district heating scheme for space heating (and cooling) and hot water production for large industrial or commercial residential buildings and estates of single or multiple family dwellings.

Lia and Lagerqvist (1973) have discussed experimental work on Stirling engines heated by heat pipes rather than direct combustion heating. Such a programme is indicative of interest in high-temperature, high-performance engines and of engines used in association with thermal energy storage systems for emission-free vehicles or underwater power systems.

Recently it was announced (NASA, 1977) that United Stirling was to be a partner with Mechanical Technology Inc. and American Motors in one of the two teams to be funded by the Department of Energy in the United States for development of an automotive Stirling engine.

Recent developments

Recent developments at United Stirling include the engine configuration shown in Fig. 15.20. This is a double-acting engine having four cylinders connected as shown in Fig. 15.7. It is therefore similar to the four-cylinder Vee engines described earlier. The big difference between the new engine and the previous Vee engines is that the cylinders are parallel and two crankshafts are used with two pistons coupled to each crankshaft. Twin gears are used to couple the crankshafts.

The relative merits of parallel versus Vee cylinders were discussed by Kitzner (1977a) as part of the study carried out by Ford on behalf of the U.S. Department of Energy, for an 60 to 74 kW (80 to 100 hp) automotive Stirling engine. The parallel cylinder design leads to a more complicated drive mechanism but allows a very compact heater and air preheater arrangement. Machining costs for the parallel cylinder arrangement are likely to be much reduced.

It is not known which of the two engine forms is projected for production series engines.

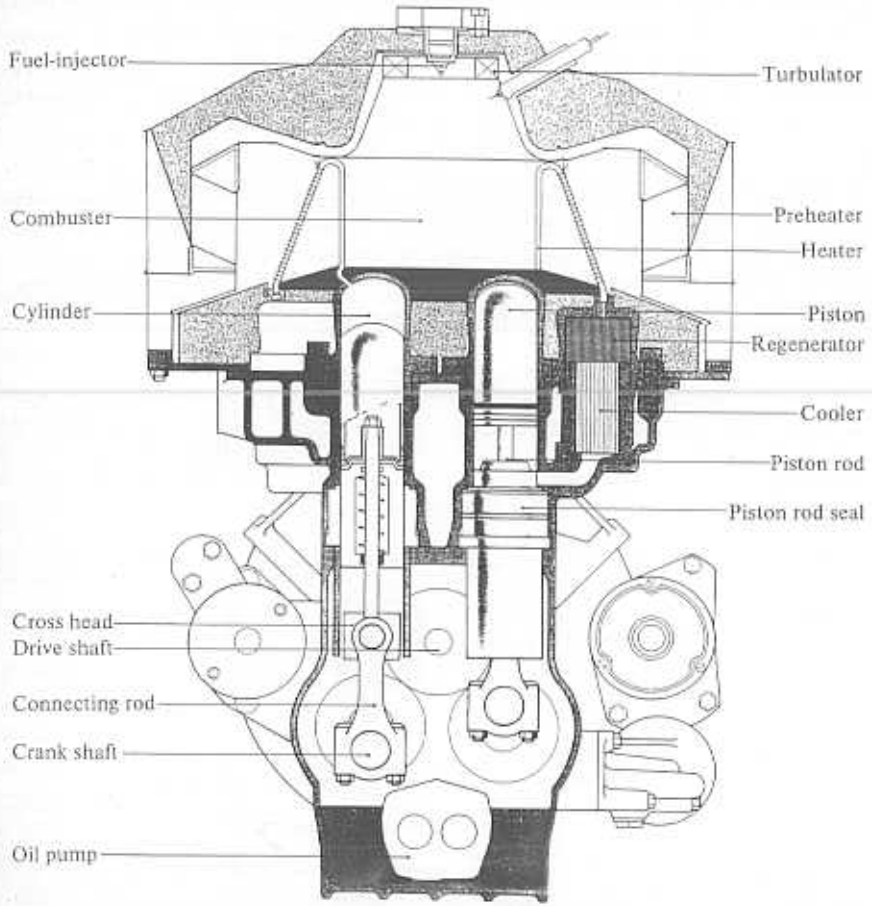


FIG. 15.20. Four-cylinder double-acting Stirling engine with parallel cylinder axes. (Courtesy United Stirling).

Conclusion

United Stirling now has a decade of experience on Stirling engine design, development and operation and they appear an aggressive, practical, commerically oriented team. This accumulated experience coupled with the substantial funding for development available from U.S. sources provides the greatest possibility that commercially viable Stirling engines will become available in the 1980s.

16 STIRLING ENGINES FOR AUTOMOTIVE APPLICATIONS

INTRODUCTION

STIRLING engines are attractive for automobile applications. They operate quietly, have low exhaust emissions, and can operate with any fuel or source of heat with high efficiency and rapid response. They are comparable in size and weight with internal combustion engines. Their part-load performance and torque speed characteristic is favourable to vehicular applications They have the promise for long operating life, extended periods between maintenance, and low lubricating oil consumption.

Cost is the principal disadvantage of Stirling engines compared with internal combustion engines. This arises out of the need to use heat-resistant steels or ceramic components in the hot parts of the engine. Moreover, the heat exchange and control systems are relatively complicated and the cooling system has to handle double the load of an internal combustion engine of comparable power since the exhaust heat from the Stirling engine must be kept to a minimum.

Serious interest in Stirling engines for automotive applications developed at General Motors in the mid-1960s. This followed their revival of Siemens double-acting Stirling engines and the emergence of compact engine designs for underwater power systems. Contemporaneously, public interest in alternative engines was stimulated by increased concern about vehicle exhaust emissions and atmospheric pollution.

General Motors' interest was focused on the Stirling engine for bus applications. A four-cylinder bus engine was designed and built before their Stirling engine program was abruptly terminated in 1970 (see Chapter 13).

In 1968 Philips undertook the development of a bus engine at the request of a new licensee, United Stirling of Sweden. These engines were installed in medium-size DAF and MAN buses. Later, United Stirling developed Siemens double-acting engines and evaluated these in a Ford Pinto car, a Ford Taunus station wagon, and in a suburban delivery truck, (see Chapter 15).

On termination of the licence agreement with General Motors Philips became associated with Ford and continued the development of Siemens double-acting Stirling engines with swash-plate drive for passenger cars. A 125 kW (170 hp) four-cylinder engine was developed and installed in a Ford Torino car and a smaller engine for compact passenger cars was investigated (see Chapter 12).

GROWTH OF U.S. INTEREST

The oil embargo and energy crisis of the early 1970s focused public attention on the economic vulnerability of the United States to extra-territorial pressures. It resulted in a massive surge of research and development activity related to energy. The Energy Research and Development Administration (ERDA) was created to coordinate all energy-related research of the U.S. Government. This was later reorganized as the Department of Energy.

In the early 1970s the Ford Company provided one half million dollars for the Jet Propulsion Laboratory (JPL) of the California Institute of Technology to conduct an independent evaluation of automobile power systems for the 1980s (Stephenson 1975). This comprehensive study arrived at conclusions favourable to development of the Stirling engine as a potential automotive engine.

Simultaneously the joint Ford–Philips program for development of a 125 kW (170 hp) swash-plate Stirling engine for the Ford Torino car was proceeding. Progress was sufficiently encouraging for the U.S. Department of Energy to commission the feasibility study of a 60 to 74 kW (80 to 100 hp) Stirling engine (Kitzner 1977a).

The optimistic projections of this study, coupled with the recommendations of the JPL work, and the urgent need to explore viable alternatives to existing engines, led ERDA (now the DOE) to enter a cooperative Stirling engine development program with Ford and Philips. This cost-shared program was outlined by Kitzner (1977b). It was anticipated the program would require 8 years for completion and require total funding of some U.S. $160 million. The proposed program was divided into twelve separate tasks extending to 1985.

A second Stirling engine development effort of corresponding magnitude was proposed by a consortium of Mechanical Technology Incorporated and American Motors of the United States, and United Stirling of Sweden, with funding provided by the U.S. Department of Energy. A preliminary announcement of this program was made (NASA 1977) but the final details and scope are presently unknown.

The Department of Energy has vested the responsibility for the management of these two main development programmes in the Stirling Engine Project Office at the Lewis Research Centre, Cleveland, Ohio, of the U.S. National Aeronautics and Space Administration. A review of the Stirling engine project status was given by Ragsdale (1977).

The objectives of the program were stated to be:

(a) the development of an 'improved' engine to permit an engineering production decision (by 1984–85).
(b) define the 'advanced' Stirling engine (by 1983).

(c) develop the technology for advanced engine production in the 1990s.

An improved Stirling engine was defined as one using existing or near-term technology and exhibiting a 30 per cent fuel economy advantage over a comparable 1976 spark-ignition engine while meeting the statutory emission requirements of the Clean Air Act. The 'advanced' engine was defined as having a 60 per cent fuel economy advantage over a comparable spark-ignition engine. Both engines are to be economically attractive compared with alternative engines.

If the efficiency of the 1976 spark-ignition engine is accepted as 30 per cent, the target efficiency of the advanced Stirling engine is therefore 48 per cent—a challenging, but not impossible, target. To achieve that and yet remain economically attractive with other engines does not seem to be realistic. However, time will tell.

In addition to the two principal Stirling engine development programmes a number of supporting research and development activities were described by Ragsdale (1977). These included (a) materials technology assessment for Stirling engines (Stephens *et al.* 1977), (b) studies of hydrogen permeability in metals and ceramics, (c) studies of seals for Stirling engines, and (d) a survey of available design analysis techniques for Stirling engines (Martini 1978). Future activities were anticipated to expand or initiate work on materials, seals, heat exchangers, advanced engine definition, and fabrication of a general purpose test engine.

In-house research on Stirling engines at the Lewis Research Centre was initiated to develop a capability in the technology and to provide basic experimental data for the validation of computer simulation programs. This program involved the restoration of two General Motors GPU-3 single-cylinder rhombic-drive engines obtained by NASA-Lewis from the U.S. Army Mobility Equipment Research and Development Laboratory, Fort Belvoir, Virginia. Test results obtained with the restored engines were given by Cairelli and Thieme (1977).

Computer simulation programs for Stirling engines are being developed at NASA-Lewis and were briefly described by Tew (1977).

The availability of a general purpose laboratory engine for component evaluation and general test work would undoubtedly enhance the development of Stirling engines in all applications. The design and construction of a Stirling laboratory research engine (SLRE) was undertaken by the Jet Propulsion Laboratory and has been described by Hoehn (1978). The engine is shown in the pictorial section diagram, Fig. 16.1. It consists of a single-cylinder opposed-piston unit with twin crankcases and an in-line heater, cooler, and regenerator interposed between the pistons. Phase control of the two pistons is maintained by toothed flexible-drive belts coupled to a motor-driven countershaft. The engine is designed to

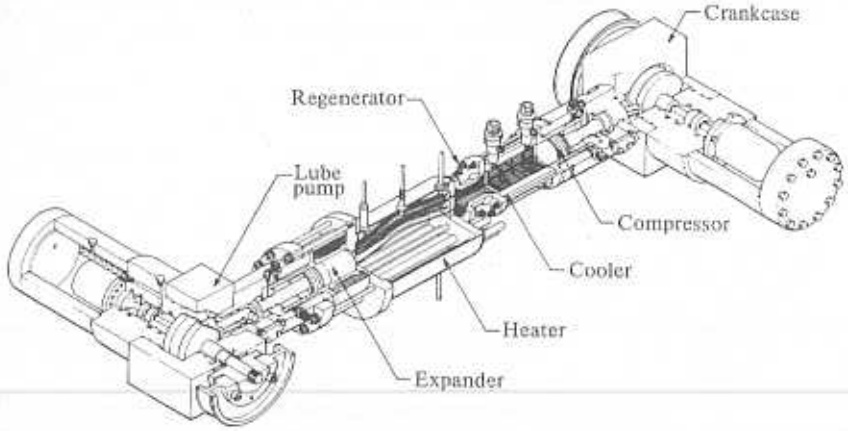


FIG. 16.1. Pictorial cross-section of the JPL Stirling laboratory research engine (after Hoehn 1978).

develop 9 kW (12 hp) at 3000 rpm with a cylinder bore of 73.0 mm (2.87 in), a stroke of 54 mm (2.12 in), and a design mean pressure of 6.9 MN/m^2 (68 atm) of helium working fluid.

The Stirling engine development programmes discussed above are only one aspect of various alternative engine concepts and systems being explored in the United States. It is not the intent of the U.S. Government to enter the vehicle engine production business. Rather, by the provision of financial assistance they are seeking to develop the basic technology to allow industrial evaluation of all the potentially viable alternative automobile engines. It has become customary to review the progress of these various programmes at semi-annual Highway Vehicle Systems Contractors Coordination Meetings held at various locations in the United States and organized by the Department of Energy. The proceedings of these meetings are published and distributed through the National Technical Information Service (NTIS) Springfield, Virginia. These contractor reports will provide the prime vehicle for information exchange on the Stirling engine development programmes although occasional papers will no doubt continue to be included in the technical society meetings.

AUTOMOTIVE APPLICATIONS WITH THERMAL STORAGE SYSTEMS

Stirling engines may be driven from any heat source. One possibility for automobile applications of Stirling engines is to drive the engine from heat drawn from a thermal-energy storage system, a thermal battery. The battery may be charged at intervals (overnight) by natural-gas combustion or, more likely, by electric-resistance heat.

The combination of Stirling engine and thermal storage system was first

explored systematically by General Motors in the 1960s for underwater power systems. This work is discussed in Chapters 13 and 18. Although primarily concerned with marine applications, the program, in 1964, embraced the installation in an automobile of a 22 kW (30 hp) engine driven from a heated alumina thermal-energy source. This was probably the first Stirling engine in a vehicle as well as the first vehicle to be driven from a thermal energy source but was never intended as a serious alternative propulsion system for automobiles. It did serve to demonstrate that a Stirling engine driven from a thermal energy source can be a fraction of the weight of an electric motor/lead-acid battery combination.

Later Meijer (1970a) presented a milestone paper containing the results of extensive studies carried out at Philips concerned with vehicular applications of Stirling engines. Part of the paper was concerned with the combination of a Stirling engine and thermal-storage systems in vehicles.

Fig. 16.2 is a schematic representation of the combination of Stirling engine and thermal storage system discussed by Meijer (1970a). The thermal-storage medium was lithium fluoride stored in thin-walled sealed bottles charged with argon to maintain a slight internal positive pressure to prevent collapse of the bottles. Lithium fluoride melts at 848 °C (1558 °F) and has a high latent heat of fusion with reasonable volume, a combination which makes it the preferred thermal storage medium for use with Stirling engines.

The thermal-energy content (measured from a datum of 550 °C (1022 °F)) of lithium fluoride is shown in Fig. 16.3 as a function of temperature. The thermal capacity, Q of solid lithium fluoride increases steadily with temperature until, at 848 °C (1558 °F), the solid begins to

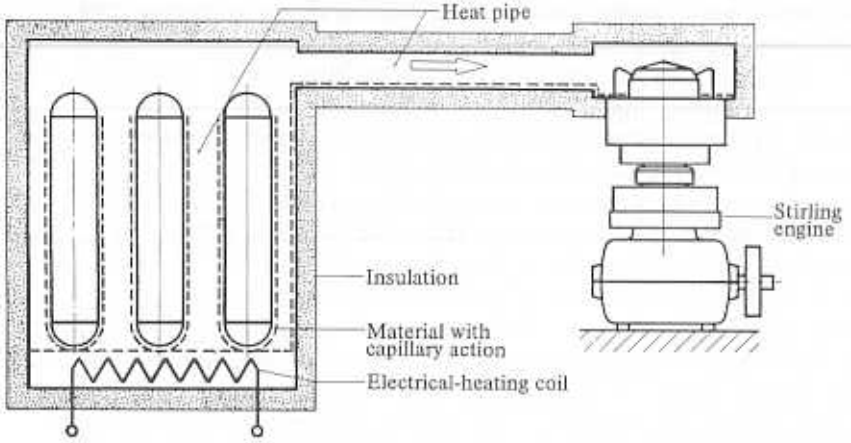


FIG. 16.2. Schematic representation of a Stirling engine with lithium fluoride thermal storage system coupled by sodium vapour heat pipe (after Meijer 1970a).

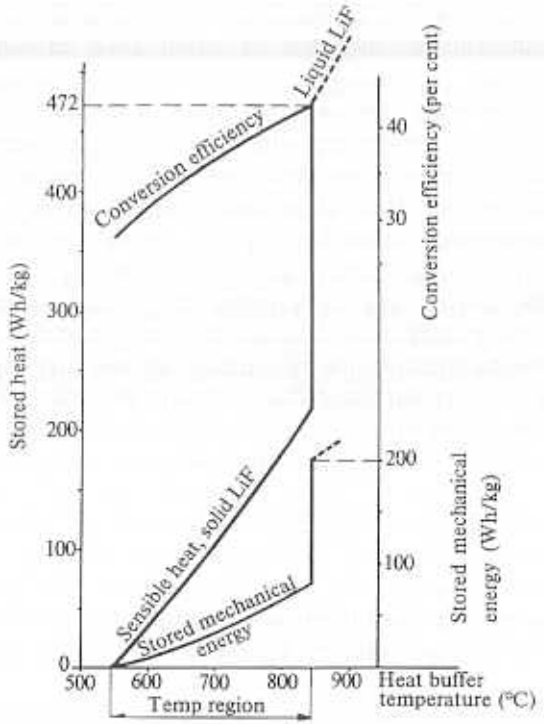


FIG. 16.3. Energy content, Q, per kilogram of lithium fluoride as a function of temperature. The efficiency of Stirling engines η_c is also shown. The product $Q \times \eta_c$ is the mechanical energy potentially available from a Stirling engine driven by thermal storage over the temperature range 850 to 550 °C (after Meijer 1970a).

melt. The latent heat of fusion (heat to melt) is 250 Wh/kg (0.15 hph/lb_m) and so the heat capacity increases by this amount at constant temperature until all the solid has melted.

It was reasonable to anticipate that the heat stored in this way could be used to drive a Stirling engine with a reducing temperature of the storage media down to a temperature as low as, say, 550 °C (1022 °F). The efficiency of the engine was of course a function of the temperature and therefore decreased as the storage temperature decreased. The possible variations in efficiency with temperature is shown in Fig. 16.3. The lower curve shown in the same figure is the potential engine work available. This is the product of the energy stored, Q, times the thermal efficiency, η. It can be seen from Fig. 16.3 that as many as 200 Wh (0.27 hph) of work could be obtained per kg of lithium fluoride contained in the thermal store ranging in temperature from 550 °C to 850 °C (1022 °F to 1562 °F).

The thermal storage system was coupled to the engine by a sodium

vapour heat pipe shown in Fig. 16.2. Liquid sodium boiled on the hot bottles of lithium fluoride salts and condensed on the heater tubes and heater head of the engine so that heat was effectively transferred from the thermal store to the engine. The storage unit and connecting ducts were all thermally insulated so that condensation of sodium vapour occurred only at the engine and thermal loss by conduction was minimized.

Heat could be transferred very effectively by heat pipe systems involving boiling and condensing of the transfer fluid. There was virtually no difference between the supply and delivery temperature and the rates of heat transfer could be orders of magnitude higher than by conventional conduction. Another advantage when used with Stirling engines was that the vapour condensing on the heater tubes and cylinder head ensured a uniform temperature. There could be no 'hot spots' which were virtually unavoidable in direct heating combustion systems. As a consequence, the mean heater temperature could be elevated to the metallurgical limit of the heater-tube material. This gain was often as much as 75 °C (167 °F) (Meijer 1970a) with consequent improvement to the power and efficiency of the engine.

Moreover, the heat transfer rates of condensing sodium vapour were very high, so the size of the heater tube could be reduced to the limit dictated by thermal conduction capacity through the walls of the heater tubes and by internal heat transfer in the tubes to the working fluid. This resulted in short heater tubes giving appreciable savings in the dead volume with further beneficial consequences to the engine power and efficiency.

This system of indirect heating by heat pipes was not limited to Stirling engines with thermal storage systems. It was equally beneficial when used on Stirling engines with fossil-fuel combustion systems. Meijer (1970a) gave the data reproduced in Fig. 16.4 as a comparison of engine performance with direct and indirect heating for engines with helium and hydrogen working fluids. It can be seen that the power and efficiency of indirectly heated engines were markedly superior at higher speeds for both hydrogen and helium. Lia and Lagerqvist (1973) have described similar work with indirectly heated Stirling engines at United Stirling.

Heat pipes have been extensively studied by Philips not only for Stirling engines but also for other applications. Asselman and Green (1973a and b) gave excellent reviews of the basic technology and applications.

Returning again to the Stirling engine with thermal storage system and sodium heat pipe as illustrated in Fig. 16.2, the system could be recharged with heat using the sodium boiler shown at point 3 in Fig. 16.2. An electric heater is shown but a combustion heater could also be used. The thermal battery is charged periodically by heat supply to the boiler. The

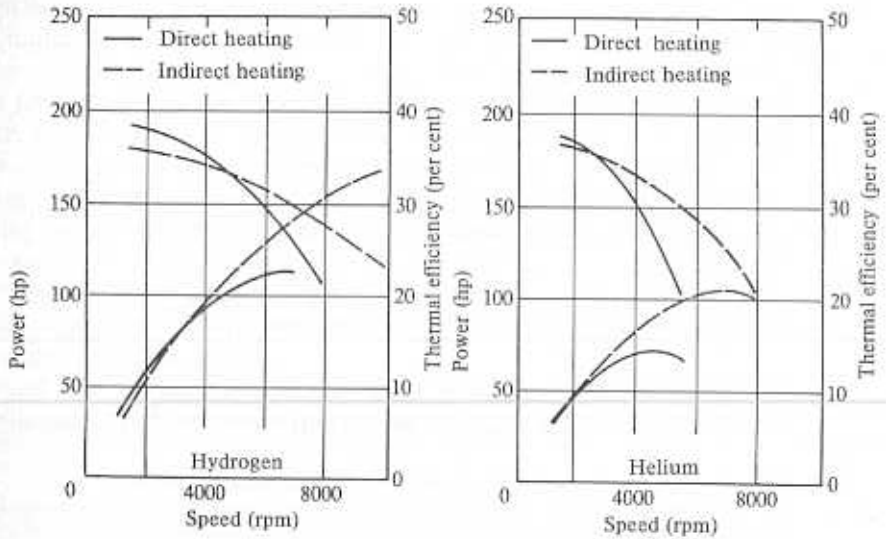


FIG. 16.4. A comparison of the power output and thermal efficiency of Stirling engines with direct and indirect heating and with hydrogen and helium working fluid (after Meijer 1970a).

sodium boils and condenses on the lithium fluoride bottles thereby reheating them. If electric heating is used, the thermal battery may simply be recharged overnight or when the vehicle is not in use. If on the other hand combustion heating is used, there are two possibilities. One is that a portable external combustion system be employed when the vehicle is not in use. The other possibility is that the combustion system be self-contained, on board the vehicle, and is used intermittently when convenient or as required. A unit similar to this latter type was described by Agarwal *et al.* (1969) in reporting the General Motors 'Stirlec' hybrid vehicle. This was an electric car with electric rather than thermal-storage battery capacity and an electric generator driven by a Stirling engine (General Motors Type GPU-3, 7.3 kW (10 hp) engine).

Meijer (1970a) presented the results of calculations made at Philips for six types of vehicles with Stirling-engine/thermal-battery propulsion systems. The calculations were made with the assumption that the battery was charged once only every day and that the vehicle radius of action was the same as that of a gasoline driven car, *circa* 1968. The basic vehicle data was similar to that used elsewhere for a study of electric propulsion of vehicles and is reproduced in Table 16.1. With this basic information, calculations were made to establish the principal parameters for the vehicles equipped with a Stirling engine/thermal battery.

The results of the calculations are summarized in Table 16.2, reproduced from Meijer (1970a). Comparison of the results given in the two

Table 16.1. Some data concerning the principal types of cars at present in use.

		American family car	Small European car (commuter car)	Utility car	Delivery van	City taxi	City bus
1. range of operation	km	322	161	80	97	241	193
2. maximum speed	km/h	161	129	105	90	124	88
3. acceleration to	km/h	97	97	48	64	64	48
in	s	15	30	10	20	15	15
4. maximum power output	kW	70	22	12	49	36	135
5. loaded weight	kg	1815	1135	770	3175	1815	13610
6. total weight assignable to new propulsion system							
a. conventional construction	kg	565	340	225	635	565	2270
b. lightweight construction	kg	795	475	320	910	795	3175
7. energy delivered	kWh	100	20	8	45	75	300

Table 16.2. Data calculated for the cars of Table 16.1 when equipped with a Stirling engine with LiF heat accumulator. Engine and accumulator are connected by a heat pipe. The accumulator is assumed to be cylindrical, with the height equal to the diameter.

		American family car	Small European car	Utility car	Delivery van	City taxi	City bus
1. volume of heat accumulator tank	dm^3	385	77	30	174	289	1154
2. diameter of tank	cm	79	46	34	61	72	114
3. thickness of superinsulation material	cm	0.55	0.95	1.43	0.74	0.63	0.38
4. weight of engine + radiator	kg	216	82	49	162	124	379
5. weight of heat-pipe system	kg	32	12	7	24	19	57
6. weight of heat-accumulator material	kg	530	106	42	239	398	1590
7. weight of container + insulation	kg	62	21	12	37	52	130
8. total weight of propulsion system	kg	840	221	110	462	593	2156
9. weight assignable to propulsion system (Table 16.1)							
a. conventional construction	kg	565	340	225	635	565	2270
b. lightweight construction	kg	795	475	320	910	795	3175
10. difference in weight (item 8 minus item 9)							
a. conventional construction	kg	+275	−119	−115	−173	+28	−114
b. lightweight construction	kg	+45	−254	−210	−448	−202	−1019
11. range of operation required	km	322	161	80	97	241	193
12. actual range of operation							
a. conventional construction	km	172	311	248	157	226	206
b. lightweight construction	km	298	480	387	252	350	308

tables indicates that the Stirling engine with thermal storage has very advantageous characteristics for application to vehicles. It is possible to drive a city bus, taxi, delivery truck, or passenger car around all day, without any noise, without using liquid fossil fuel and without exhaust emissions from the vehicle. Moreover, as Meijer (1970a) pointed out, the system permits the interior of the vehicle to be heated by engine waste heat. In vehicles with electric propulsion, interior heating has always been a particularly difficult problem, especially in cold climates.

A more recent study by Asselman *et al.* (1977) of automotive applications of Stirling engines with thermal energy storage was directed particularly to the compact commuter passenger car.

Independently, Folsom *et al.* (1977) considered a number of automotive propulsion schemes using thermal storage concepts and concluded that the Stirling engine had substantial advantages over Brayton and Rankine cycle engines.

The safety of high-temperature thermal-storage systems was considered by Boser (1977) with an account of the accumulated experimental experience at Philips.

Intense effort is presently being invested in the United States on the development of Stirling engines for automotive purposes, using gasoline or diesel fuel with direct heating. The wisdom of this is questionable. Internal combustion engines are perfectly adequate for use where their noise, exhaust emission, and need for distillate fuels can be tolerated. Stirling engines can probably never be made as cheap and reliable as internal combustion engines have become in the century of effort devoted to them.

A better use of the time and money now being spent might be to develop Stirling engines with thermal-energy storage as automotive propulsion systems for the electric economy that lies ahead. In the electric economy, liquid distillate fuels will be in relatively short supply and available at premium prices. Electric energy will be low cost by comparison. The electric energy will be derived from coal combustion, from nuclear fission reactors and, in the twenty-first century, from large scale photelectric conversion of solar power and by nuclear fusion.

It is conceivable that in the electric economy Stirling engines with thermal-energy storage will become the primary vehicle propulsion system.

AUTOMOTIVE APPLICATIONS IN MINING

Hallare and Rosenqvist (1977) indicated that the first commercial applications for United Stirling engines will be for underground mine vehicles. The use of diesel engines in underground minework is a subject of increasing concern to those involved with industrial health and safety.

There is concern at the long term effects on health of inhaling diesel exhaust emissions. There are dangers of fire and explosion from the high temperature exhausts, particularly after the safety devices have been tampered with by well-intentioned but ignorant mechanics or operators.

Stirling engines for underground mine vehicles have advantages in terms of reduced vehicle emissions, quiet operation, and low-temperature exhausts. Moreover, it is possible to foresee the next step for combination of the engine with thermal storage so as completely to eliminate exhaust emission when operating underground. Electric power is always available for recharging during shift changes or when loading and unloading. For mine locomotives used above and below ground, a combustion-heating system may be turned on when operating on the surface. It would provide the energy for both surface propulsion and to replenish the thermal battery for underground operation.

REGENERATIVE BRAKING AND PROPULSION

In surface mines large trucks are used to move both the overburden and product. Payloads of 100×10^3 and 200×10^3 kg (220 500 and 441 000 lb_m) are common. A prototype unit of 35×10^4 kg (771 750 lb_m) payload, the Terex Titan, has been constructed in Canada and larger units are contemplated. These giant machines are powered by 735 kW (1000 hp) locomotive diesel engines driving an electric generator which in turn supplies power to the propulsion motors in the rear wheels. Smaller trucks, from 35×10^3 to 85×10^3 kg (77 175 to 187 425 lb_m) capacity, have a conventional mechanical drive from the engine through a transmission system to the rear wheels.

An application for Stirling engines exists with these large mining trucks to provide regenerative braking and propulsion. The principle of the combined regenerative braking and propulsion unit is simple. Consider, for example, the cylinder arrangement for the Siemens double-acting engine shown in Fig. 16.5. When operating as an engine producing power the expansion spaces are located in the heated spaces at the upper end of the cylinder. The compression spaces are at the cooled lower ends of the cylinder. The direction of heat flow is *into* the engine at high temperature and *out* of the engine at low temperature with the production of some work at the shaft. The expansion space of one cylinder is connected to the lower end of the succeeding cylinder through the heater, regenerator, and cooler.

Now let the connections between cylinders be switched at the cold end so that the upper end of one cylinder is connected to the lower end of the *preceding* rather than succeeding cylinder. This causes roles of the hot and cold spaces to be reversed and the engine will attempt to run in the reverse rotation direction. However, if sufficient power is applied to the

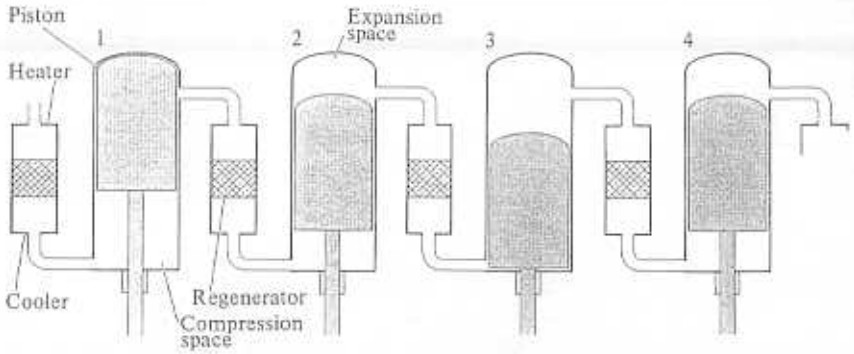


FIG. 16.5. Cylinder arrangement for Siemens double-acting Stirling engine for regenerative braking and propulsion concept.

engine shaft to drive the engine in the original forward direction it will operate, not as a prime mover, but as a heat pump. With the switching of the cylinder connections the lower ends become the expansion space abstracting heat *from* the surroundings to the working fluid. Similarly the upper ends of the cylinder become the compression space and heat is rejected *from* the working fluid to the heater. Thus, by simply switching connections between adjacent cylinders, the engine can be converted from a prime mover using high temperature heat and producing power, to a heat pump which absorbs power and generates high temperature heat.

When this system is applied to a Stirling engine equipped with thermal-energy storage in a mine truck, it can absorb power during down-haul operation to retard or decelerate the loaded truck and, at the same time, store the thermal energy to drive the empty truck back from the load tipple to the mine site.

The regenerative braking propulsion system was conceived by the author in the course of consulting work in coal mines in Alberta and British Columbia. These are located in mountainous regions and long down-haul operations of loaded trucks are commonplace. The same system can of course be applied in railway locomotive engines and in highway vehicles and is particularly attractive in mountain or foothills areas where considerable downhill braking is necessary.

Preliminary studies for mining trucks have shown that for loaded down-haul operations the fuel savings exceed 50 per cent. Economies of this order with 735 kW (1000 hp) engines in large fleets operating continuously are very significant indeed. Further savings are attained in reduced wear and tear on mechanical braking systems and in improved safety in operation.

17 STIRLING ENGINES FOR ARTIFICIAL HEARTS

INTRODUCTION

THE Artificial Heart Program in the United States was established in 1964 by the National Heart Institute (Department of Health, Education and Welfare). The objectives of the program were to develop devices that could assist or totally replace the heart. Some devices were intended for temporary circulatory assistance for patients in hospitals and confined to bed. Other devices were intended for permanent implantation in the body to assist or totally replace the heart in pumping blood and to allow the recipient complete freedom of movement.

Harmison and Hastings (1969) justified the program on the grounds that in the United States, heart disease caused about a million deaths per year and was by far the leading cause of death, 54 per cent, compared with 16 per cent for cancer, the second most common cause. Moreover, data compiled in the early 1960s indicated that about a quarter of the adult population had definite or suspected heart disease. About two million of these were seriously handicapped by the disease. An annual expenditure by the U.S. Federal Government of over 300 million dollars was estimated for welfare benefits to individuals under the age of 65 permanently and totally disabled by heart disease. By way of comparison the total annual budget for the artificial heart program was eight million dollars in 1969.

Wiggers (1957) has provided an excellent summary about the heart and its functions. A more extensive discussion has been given by Longmore (1971). Both these sources are written for the lay person and provide a good foundation for understanding the magnitude of the task facing those seeking to replace the natural heart.

The U.S. artificial heart program was extraordinarily broad in scope. At the First Artificial Heart Conference, held in 1969, ninety-two technical papers were presented by the sixty-three separate contractors on a very wide range of topics. Of this total, sixteen papers were concerned with implantable energy sources and, within this group, only two were devoted to Stirling engines. A third paper discussed a fluidic control device for coupling Stirling engine gas compressors to the blood pump. It is clear, therefore, that work on Stirling engines was but a small part of a large program. However, because of the specific nature of this book the discussion here will be confined to matters related to Stirling engines.

GENERAL CONSIDERATIONS

Two research and development programs on Stirling engines have been sponsored by the Artificial Heart Program of the National Institute of

Health from its beginning in the mid-1960s to the present time. The two research contractors have been the Aerojet Liquid Rocket Company at Sacramento, California and the McDonnel-Douglas Corporation at Richland, Washington. A third program, begun somewhat later at the Thermoelectron Corporation, started with a Rankine-cycle steam engine. Resulting from progressive development, the engine became a 'tidal regenerator engine' and changed its character to a closed Stirling-cycle regenerative engine with a condensing/evaporating working fluid. Finally a fourth program for circulatory assist devices, commencing in the early '70s was separately funded by the U.S. Atomic Energy Commission with the Westinghouse Electric Co., Pittsburgh, Pennsylvania. This started off as a rotary steam engine project but changed to a Stirling engine project with North American Philips Inc., New York, as the principal subcontractor.

All these programs for engine and system development have been regularly reported at the Intersociety Energy Conversion Engineering Conferences held annually in the United States and, more fully, in the quarterly progress and annual or semi-annual contractor reports. Progress with blood pumps and surgical techniques are reported at meetings, and in the *Journal of the American Society of Artificial Internal Organs.*

All the four machines are superficially similar in that they seek to provide the means to convert thermal energy to some form of mechanical work to drive a blood pump. A typical schematic system is shown by the block diagram in Fig. 17.1, (reproduced from Johnston *et al.* 1977). For many years the source of the thermal energy was intended to be radioisotope, with plutonium 238 as the preferred material (Sandqvist *et al.* 1975). Its high specific power, moderate radiation levels, and long half

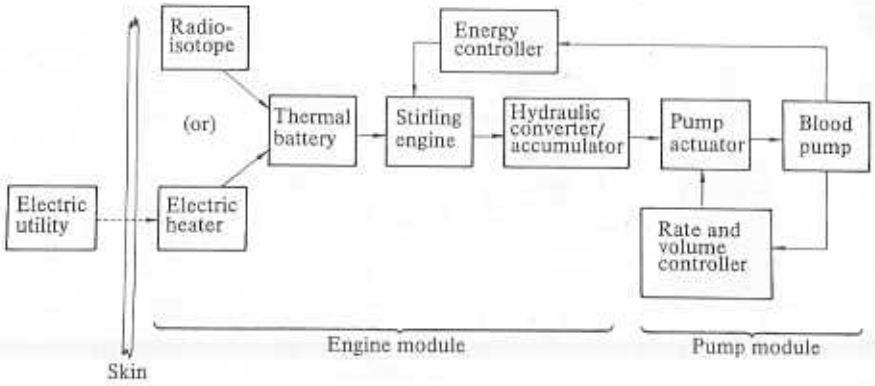


FIG. 17.1. Block diagram for artificial heart (after Johnston *et al.* 1977)

life (89 years) provide the prospect of an essentially permanent heat source (lasting more than 10 years). However, an alternative has been proposed because of the large numbers involved (Harmison and Hastings, 1969, estimate a demand of 200 000 units per year). Other factors are the high cost of the isotope and increasing public concern and government restriction against the use of plutonium in such large amounts. The alternative (Martini 1977b) is an electrically-heated thermal-storage capsule that would give four to eight hours of artificial heart power before recharge would be necessary. This approach demands the development of satisfactory long-term percutaneous (through the skin) power leads (Lee 1969) or the alternative of a transcutaneous (across the skin) power transformer (Newgard and Eilers 1969; Thumin 1969).

For the thermal engine systems the power input levels range from 30 to 50 watts to develop the 3 to 5 watts of actual hydraulic power required for the blood pump. This whole 30 to 50 watt power input must eventually be dissipated from the system to the body and hence to the atmosphere. In all cases the blood is used as the system coolant. For comparison an average man asleep and at rest dissipates about 100 watts of heat. When doing heavy manual work the power level increases to as much as 500 watts. An athlete may be able briefly to exceed a 1 kW power rate.

The dissipation of an extra 30 to 50 watts of heat from the body is not seen as an insuperable problem and this has been confirmed in many animal experiments. It is imperative to avoid hot spots at any external part of the system in contact with the body for blood cells and other living tissues are highly susceptible to temperature damage.

One of the principal attractions of the alternative electrically driven blood pumps is the low power input, for it would reduce the amount of heat that must eventually be dissipated in the body. Electrical systems have paid the Carnot penalty at the power station where the electricity was generated. Therefore, the power input may be as little as 10 watts to develop the 3 to 5 watts power for the blood pump.

The physical structure and arrangement of the circulatory system and the susceptibility of the blood to mechanical damage require that the blood pump, even for total heart replacement, be a pulsing system, operating at a frequency close to the normal rate (about 50 to 100 beats a minute). For a heart assist system, of course, the pump must be exactly synchronized with the operation of the natural heart.

In some of the systems tried the engine operated at a high frequency to make the unit small and compact. Others operated at heart-beat rate but in no case was the engine coupled directly to the blood pump. Instead, the different systems were packaged in two units as shown in Fig. 17.2—an engine module and pump module. A variety of arrangements

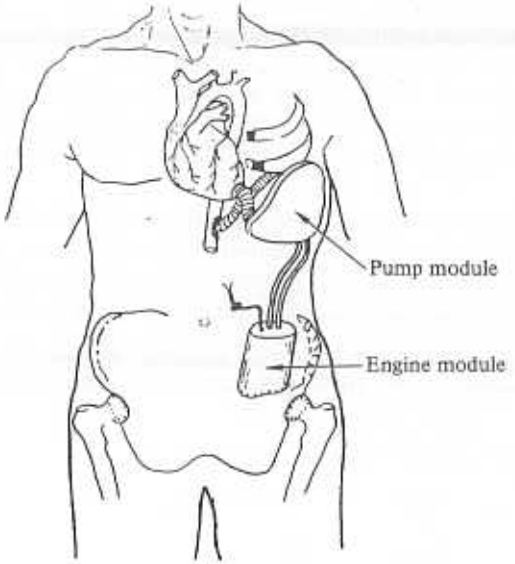


FIG. 17.2. Heart-assist thermal-energy system concept (after Johnston *et al.* 1977).

were used to provide connections between the two modules through which power was transmitted from the engine to drive the blood pump. The McDonnel-Douglas and Thermoelectron systems used hydraulic connections with a hydraulic actuator to drive the blood pump. The Aerojet-General system used a pneumatic connection and pneumatic actuator to drive the pump. The Westinghouse/Philips unit used a flexible rotating shaft with a crank driven blood pump with intermediate speed change gears.

For all systems a control unit was a necessary component and much engineering ingenuity has been exercised in the development of these electronic or fluidic control units.

A brief description of the four systems is given below although space precludes a complete discussion of details for all systems. The general approach followed is briefly to review published accounts of the most recent versions of the four systems, and provide sources for more complete reference. All the systems have experienced successive generations of development; some are now at the fifth and sixth stages. Development has proceeded in parallel with related development of blood pumps, actuators, control systems, operating experience with laboratory systems and with actual installation in animals—usually calves but sometimes pigs have been used. Space precludes a discussion of all this. Further, the ethical, societal, and legal questions concerning artificial hearts have not been addressed.

There is concern about the use of radioisotope heat sources in artificial hearts. A heat source of 30 watts would require 54 grams (0.12 lb_m) of Pu238. Now the critical mass of a fast unmoderated Pu238 system is (Sandqvist *et al.* 1975) 5.2 kg (11.5 lb_m) with a steel reflector. One nightmarish scenario has a convention of 1000 'Arty-Heartys' providing a determined terrorist organization with all the plutonium necessary for a bomb. There is more legitimate concern about the dangers of inadvertent cremation of artificial heart power sources in hotel or residential fires, in automobile accidents or even in routine cremations following death by natural causes, even perhaps mechanical failure of the blood pump. It would be exceedingly difficult to keep track of all the isotope sources if anything other than a few experimental units were involved.

The progress achieved in fifteen years of development of the artificial heart has been remarkable. Complete, operating, heart-assist systems have been installed in calves free to stand and walk about. The longest time of survival to date was about eight months. No attempt has been made here to judge the competing systems for this would require a more intimate knowledge and understanding than can be gleaned from the published reports. Enough progress has been made to demonstrate the technical feasibility of relatively long-term (say five years) mechanical heart-assist systems. There is every reason to believe that total heart replacements are also possible. When and, indeed, if such developments will occur are not certain for there are important issues to resolve.

MCDONNEL-DOUGLAS ARTIFICIAL HEART ENGINE (NOW UNIVERSITY OF WASHINGTON)

Johnston *et al* (1977) have given a good overall summary of progress achieved with McDonnel-Douglas Stirling-engine/hydraulic-pump artificial hearts. A schematic diagram of the system is shown in Fig. 17.3. The engine module contained the thermal energy source, Stirling engine and the hydraulic converter/accumulator. The pump module contained the blood pump and the hydraulic actuator/controller.

Details of the engine module are shown in Fig. 17.4 and some of the performance characteristics are given in Table 17.1 (Johnston *et al.* 1977). The Stirling engine was basically a piston/displacer in separate cylinder units. In one cylinder the displacer oscillated with a short stroke and was caused to move by the balance of pressure forces acting upon it. These were made up of the helium gas pressure acting on the displacer over the whole area at the top and over part of the area at the bottom. This pressure changes during the cycle. The displacer piston moves the displacer by operating between the variable engine pressure and the constant pressure of the buffer space. Variation in engine speed was accomplished by a power control valve V_2 (Fig. 17.3) in the hydraulic

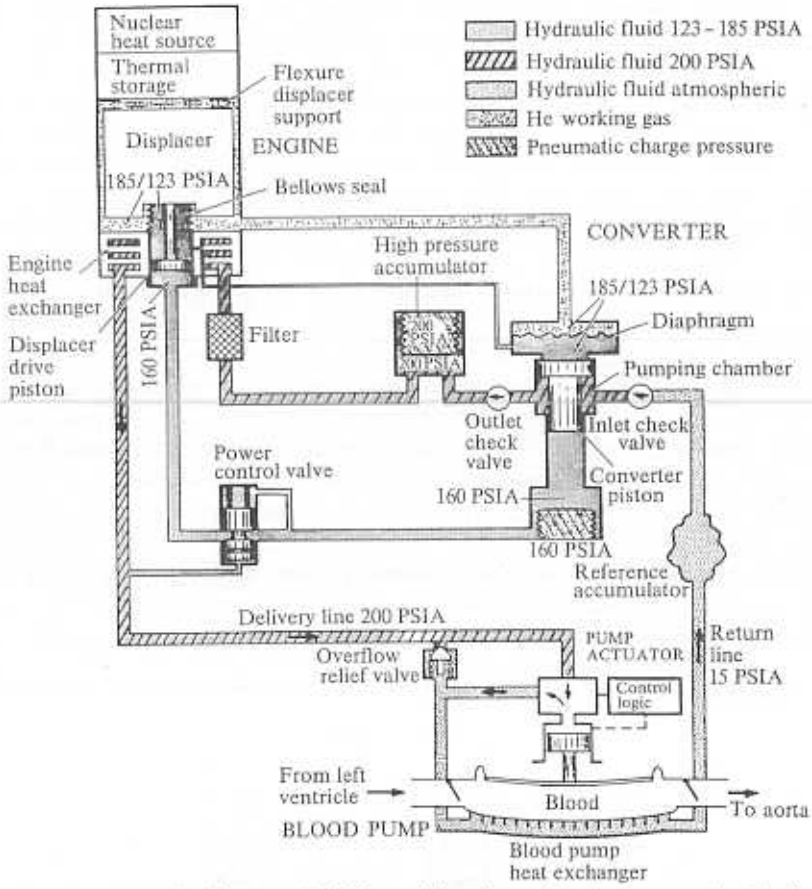


FIG. 17.3. Schematic diagram of McDonnel-Douglas nuclear-isotope or electric-heating Stirling engine and hydraulic blood pump (after Johnston *et al.* 1975).

connection between the displacer cylinder and the converter.

In this engine the 'piston' consisted of a flexing metal diaphragm with helium gas on one side and hydraulic fluid (a petroleum base fluid) on the other. The regenerator was the annular gap between the displacer and the cylinder wall.

Motion of the displacer in the cylinder caused helium to pass through the regenerator from the hot space above to the cold space below the displacer resulting in variation of the pressure level of the helium gas between 0.9 and 1.3 MN/m^2 (135 and 185 lb per sq in). The pressure change acting on the 'piston' diaphragm caused it to flex and pump the hydraulic fluid at the same pressure range. A differential area hydraulic pump created hydraulic pressure (1.4 MN/m^2 or 200 lb per sq in). This was the delivery pressure to the hydraulic actuator of the blood pump.

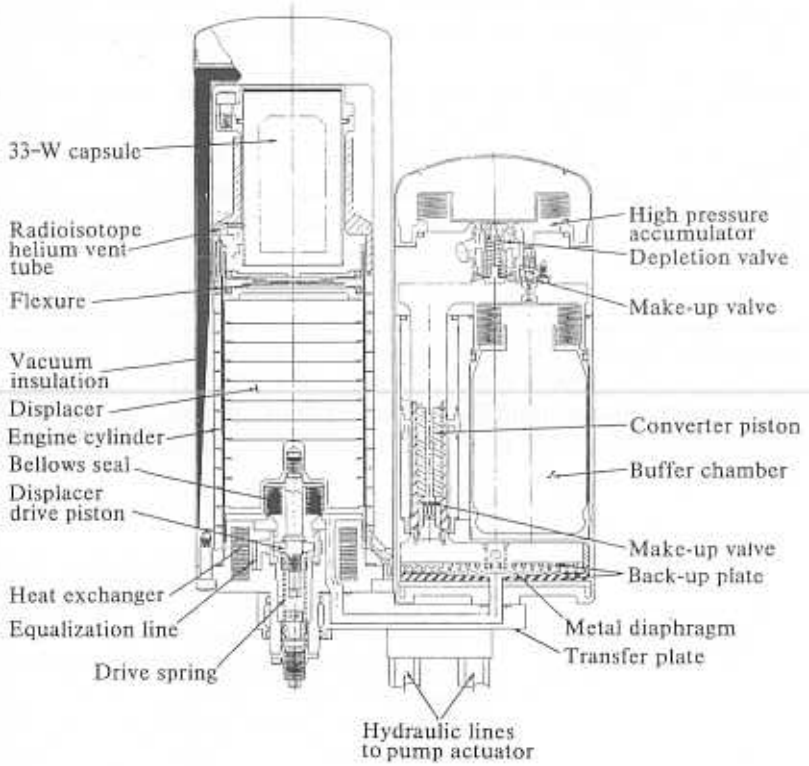


FIG. 17.4. Cross-section of McDonnel-Douglas radioisotope, Stirling engine heated module (after Johnston *et al.* 1975).

The high pressure hydraulic fluid acted as the coolant for the engine and carried the waste heat for final dissipation to the blood in a heat exchanger adjacent to the blood pump.

An interesting feature of the McDonnel-Douglas engine was the flexural element used to locate the upper end of the displacer in the cylinder. This was adopted in preference to a sliding guide ring at the hot end so as to eliminate the generation of wear debris. The original version of the flexing element is shown in Fig. 17.5. The present, improved flexing element has a spiral configuration to provide a long flexure length and so reduce the stress levels as shown in Fig. 17.6, for long life and reduced heat transfer.

Four flexing metal bellows were used in the engine module to separate the hydraulic fluid and helium working gas, the diaphragm, the displacer drive bellows, and the two buffer bellows. All these flexors do not sustain a pressure difference and may therefore have a long life. Nevertheless

Table 17.1. Power source design and performance parameters for McDonnel-Douglas artificial heart (after Johnston *et al.* 1977).

Prototype designation	System 5 (implantable)	System 6 (implantable)	System 7 (implantable)
Overall system characteristics			
Peak thermal input (W)	50	33	20
Peak power delivered to blood pump (W)	4.55	5.10	4.7
Peak system efficiency (%)	9.1	15.5	23.5
Volume (l)*	1.7/2.2	0.93/1.33	0.41/0.61
Fuelled weight (kg)	4.5	2.4	1.1
Peak thermal energy to abdomen (W)	9	3.5	2
Peak thermal energy to chest (W)	4	1.5	1
Peak thermal energy to blood (W)	32	23	12
Daily average thermal input (W)†	—	—	10
Year available	1972	1975	1979
Thermal engine module characteristics			
Hydraulic power to actuator (W)	5.0	5.76	5.0
Working fluid-engine	Helium	Helium	Helium
-hydraulic converter	UCON	NF No. 1	NF No. 1
Peak operating frequency (Hz)	22	25	35
Peak cycle temperature (°C)	470	575	500
Cycle pressure (psia)-engine	255/190	200/160	230/155
-converter	200/14.7	2.15/14.7	250/14.7
Peak engine module efficiency (%)	10	17.5	25
Fuelled weight (kg)	4.0	1.80	0.6
Volume (l)*	1.6/2.1	0.77/1.05	0.25/0.33
Key dimensions (cm)			
Maximum length*	25.4/26.7	16.3/17.3	11.4/12.1
Maximum width*	10.7/11.9	10.2/12.1	7.1/8.5
Maximum thickness*	8.9/10.1	5.1/7.2	3.6/5.0
Hydraulic connecting lines			
Volume (l)*	Not	0.080/0.151	0.080/0.151
Weight (kg)	Considered	0.260	0.15
Pump actuator/controller characteristics			
Working fluid	UCON	NF No. 1	NF No. 1
Peak efficiency (%)	90.7	88.5	93.9
Volume (l)*	0.134/0.144	0.082/0.124	0.082/0.124
Weight (kg)*	0.504	0.319/0.331	0.319/0.331
Height above pump push plate (cm)*	3.73/3.98	3.05/3.56	3.05/3.56

* With no low temperature foam insulation/with insulation.
† For 2W daily average power to blood pump push plate.

their pressure must give rise to concern about the feasibility of achieving the desired 10-year life with no maintainance. The peak operating frequency was 25 Hz and was projected to increase to 35 Hz (Johnston *et al.* 1977). An average frequency of say 25 Hz for 10 years corresponds to 3.28×10^8 cycles. Trouble-free operation of flexing elements for such extended periods poses a formidable challenge indeed.

A good deal of effort has gone into fatigue testing of components and systems. The best achieved so far (Johnston *et al.* 1977) was a laboratory

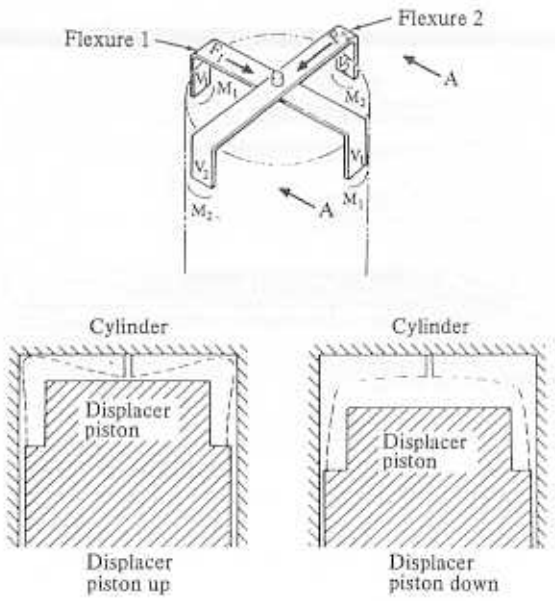


FIG. 17.5. Principles of flexural hot-end displacer guide (after Johnston *et al.* 1975).

bench operation of an engine module for 3 years with continuous maintenance-free operation using a bellows operating with a pressure difference of 30 lb per sq in.

In a decade of development, remarkable reduction in system weight and volume has been achieved with concomitant improvements in efficiency. A heart-assist unit prepared for animal implant experiments was

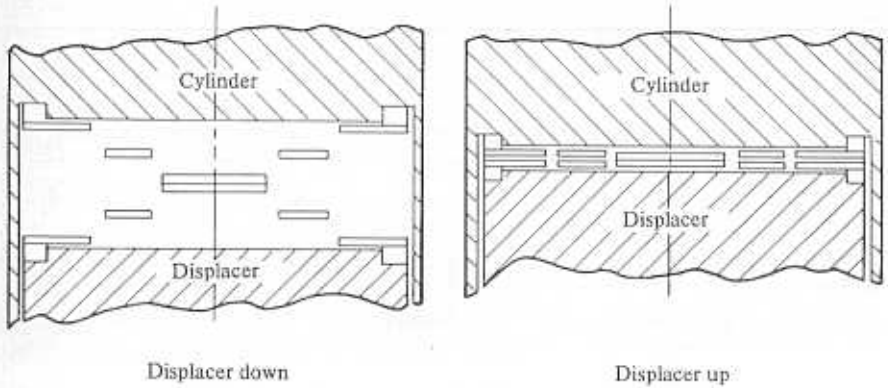


FIG. 17.6(a). Flexural guide of spiral configuration (after Johnston *et al.* 1975).

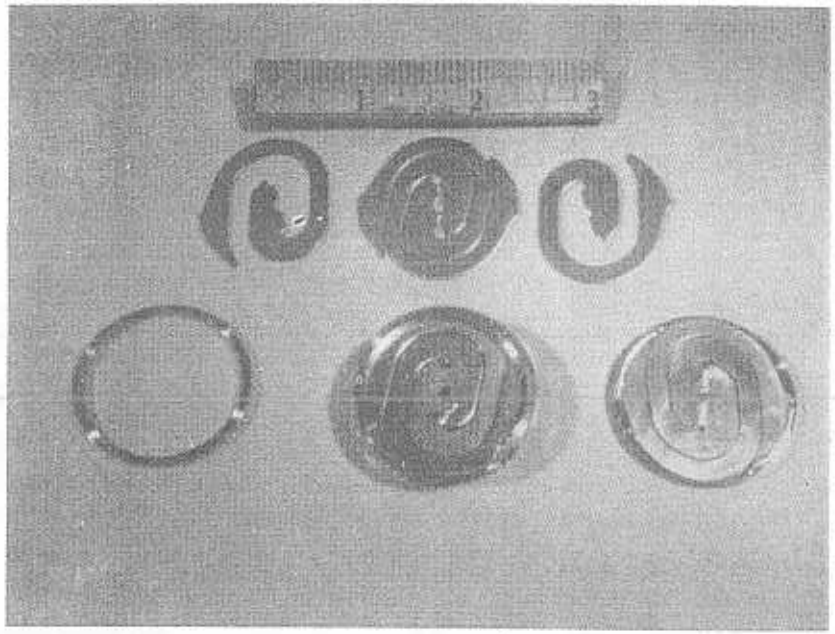


FIG. 17.6(b).

described as having an overall efficiency of 15 per cent with a 33 watt thermal input, a volume of 930 cm^3 (57 in^3) with additional 400 cm^3 (24 in^3) foam insulation and a mass of 2.4 kg (5.3 lb_m).

The McDonnel-Douglas system had been used in a number of animal experiments (principally with calves) carried out by the implant team at the Washington State University, Pullman. Most experiments briefly reported by Johnston *et al.* (1977) had been of the partial system with electric heating of the engine module. Full system tests were scheduled for 1978 in six calves.

Johnston *et al.* (1975) included a history of the McDonnel-Douglas artificial heart program and a complete listing of the eight annual reports and ten technical papers contributed to that time.

AEROJET-GENERAL ARTIFICIAL HEART ENGINE

Moise and Faeser (1977) summarized the status of development of the Aerojet artificial heart project. The Aerojet system utilized a thermal energy source and a regenerative engine to generate pneumatic power (compressed helium gas) supplied to a pneumatic actuator driving the blood pump. A block diagram for the unit is shown in Fig. 17.7. A section view of the engine module is given in Fig. 17.8 and some of the engine operating characteristics are given in Table 17.2.

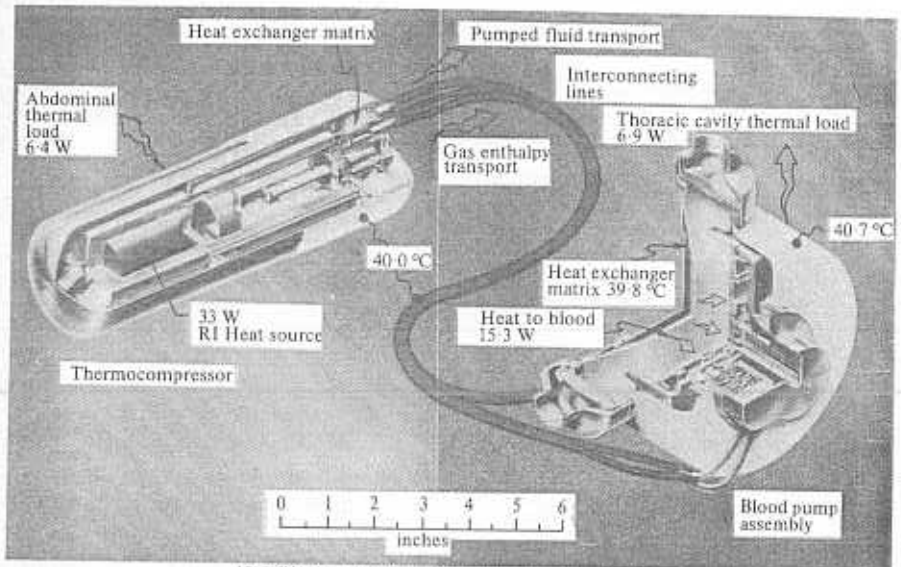


FIG. 17.7. Aerojet heart-assist thermal-energy system (after Moise and Faeser 1977).

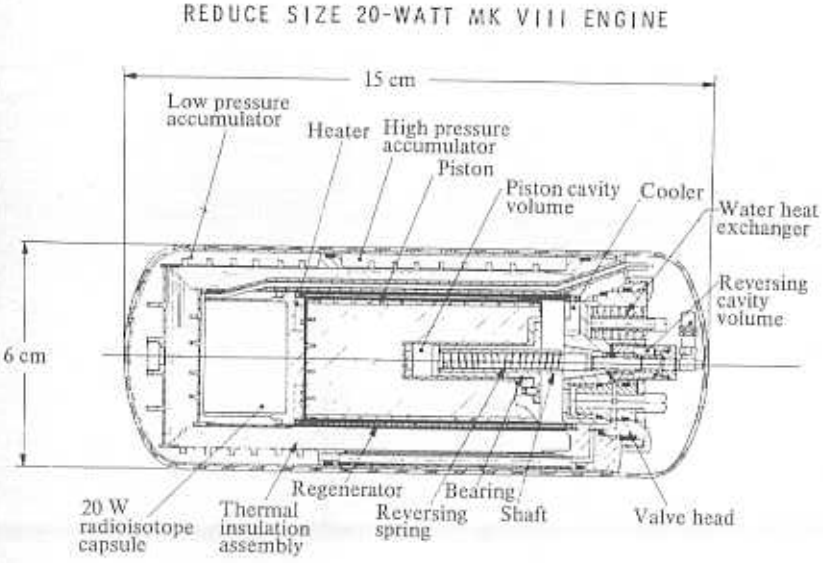


FIG. 17.8. Cross-section of Aerojet heart assist thermal engine mark VIII, 20 watts (after Moise and Faeser 1977).

Table 17.2. Characteristics of two Aerojet-General artificial heart engines and systems (after Moise and Faeser 1977)

Engine characteristics	*MK VII*	*MK VII*
Input thermal power, watts	33	20
Output pneumatic power		
steady state, watts	6.0	3.3
transient peak, watts	N/A	6.0
Peak thermal efficiency, %	18	16.5
Weight (incl. radioisotope), Kg	1.99	0.79
Volume, litres	0.68	0.43
Length, cm	22.12	16.63
Diameter, cm	6.02	5.93
Specific gravity	2.93	1.84

System characteristics	*MK VII/PAC-6*		*MK VIII/PAC-7*	
	Weight (Kg)	*Volume* (l)	Weight(Kg)	*Volume* (l)
Implantable engine (including RI capsule)	1.99	0.68	0.68	0.40
Actuator/controller	0.52	0.07	0.30	0.06
Systems fluids	—	—	—	—
Coolant system lines	0.03	0.07	0.01	0.02
Total thermal system	2.54	0.82	0.99	0.48
PVAD blood pump	0.94	0.52	0.94	0.52
Compliance system	0.21	0.21	0.21	0.21
Total assist system	3.69	1.55	2.14	1.21

The engine of the Aerojet unit operated as a thermocompressor inhaling helium at a low pressure 1.2 MN/m^2 (180 lb per sq in) and exhaling it at a higher pressure (1.5 MN/m^2; 215 lb per sq in) through inlet and outlet valves. The flow was therefore controlled by valves and the engine qualified for classification as an Ericsson rather than a Stirling engine using the definitions given earlier. It was, in fact, an Ericsson engine of the Bush variety.

The Aerojet engine was a single-cylinder machine containing a reciprocating displacer with a heater located at the upper end of the cylinder and a cooler at the lower end. The regenerator was the long narrow annulus around the circumference of the displacer. The displacer was guided in its motion by a linear bearing on a centre-post along the axis of the cylinder. The centre-post also provides the reversing cavity or, in the jargon of other free-piston Stirling engines, the bounce space or gas spring.

Operation of the engine can be understood by reference to Fig. 17.9. The sequence of operations is divided into four phases shown as (a), (b), (c), and (d).

At state (a) both inlet and outlet valves are closed, the pressure in the cylinder and in the reversing cavity is low, and the displacer is at the top

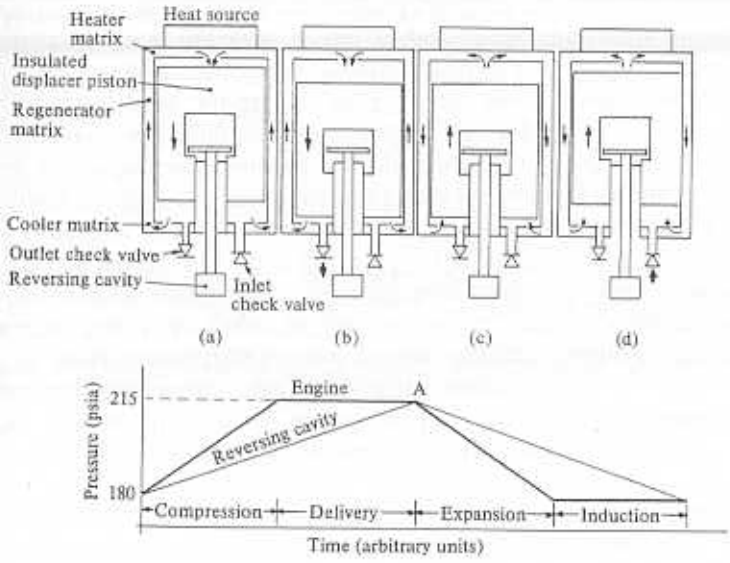


FIG. 17.9. Operating sequence of Aerojet thermocompressor engine.

dead-point and beginning the downstroke. As the displacer moves, fluid flows from the cold end through the regenerator to the hot end. Although the volume enclosed in the engine cylinder remains constant the pressure is raised because the *mean* temperature level increases as more of the gas is heated. As the displacer descends, fluid in the reversing cavity is compressed as the volume of the reversing cavity decreases and consequently the pressure increases. The rate of increase of pressure in the reversing cavity is less than the rate of increase in the engine cylinder. The different rate of increase in pressure causes a pressure differential to be established across the transverse faces of the displacer over the area of the centre-post. The gas force acting on the displacer is, in fact, the pressure difference between the cylinder and reversing cavity pressures times the cross-section area of the reversing cavity centre-post.

The gas force on the displacer then acts to drive the displacer towards the bottom of the cylinder, thereby increasing the gas flow to the hot space which in turn further increases the pressure in the cylinder and consequently the downward force acting on the displacer. This compression process continues until the pressure in the cylinder reaches the pre-set delivery pressure, at which point the outlet valve opens and pressurized helium gas leaves the engine cylinder (Fig. 17.9(b)).

The delivery process continues until the displacer reaches the end of

stroke at the bottom dead-point. The pressure of gas in the reversing cavity is highest at that time and on Fig. 17.9 is shown to be identical to the delivery pressure on the cylinder. This is not a necessary condition but a similarity of pressure levels does ease the sealing requirements between the cylinder and reversing cavity.

The pressure energy stored in the gas spring of the reversing cavity is sufficient to 'jump' the displacer upwards from the bottom position. In practice this is assisted by the provision of a mechanical 'reversing' spring not shown in Fig. 17.9 but which can be seen in Fig. 17.8.

The upward motion of the displacer (Fig. 17.9(c)) causes fluid to flow back from the hot space to the cold space resulting in reduction in the mean temperature of the fluid and hence a reduction in the pressure so the delivery valve closes. The pressure in the cylinder decreases, and as before, at a faster rate than the pressure in the reversing cavity. Consequently, a pressure differential force is established which causes the displacer to continue moving up the cylinder. This in turn increases the flow to the cold space and further decreases the cylinder pressure and consequently increases the accelerating force on the displacer.

The expansion process continues until the cylinder pressure falls to the pre-set value at which the inlet check valve opens and a fresh charge of low pressure (1.2 MN/m^2; 180 lb per sq in) helium is inhaled. The induction process continues until the displacer reaches the top dead-point and the cycle is complete. Fluid pressure in the reversing cavity pressure is lowest at this point.

A schematic diagram of the complete Aerojet heart assist system is shown in Fig. 17.10. The pressurized helium gas flowed from the engine to a pneumatic actuator driving the blood pump. The actual pump work to the blood was accomplished through a novel magnetic coupling allowing the use of a rigid hermetic seal enclosure to contain the helium at the actuator.

Use of the helium working gas for the pump actuation was clearly an attractive feature for it eliminated the pneumatic/hydraulic conversion process of the McDonnel-Douglas machine. However, the advantage was partially offset by the need, in the Aerojet machine, to provide a liquid (physiologic saline) cooling circuit to carry heat from the engine to the vicinity of the blood pump for eventual dissipation in the blood.

Critical features of the Aerojet engine as regards the target 10-year design life were the helium check valves, the centre-post bearing and the displacer reversing spring. Considerable endurance testing has been carried out on these components and on complete systems. Moise and Faeser (1977) claimed a total test time of 67 000 hours and appeared sufficiently confident to project that the 10-year operating life target would be met. The centre-post bearing was perhaps the most critical

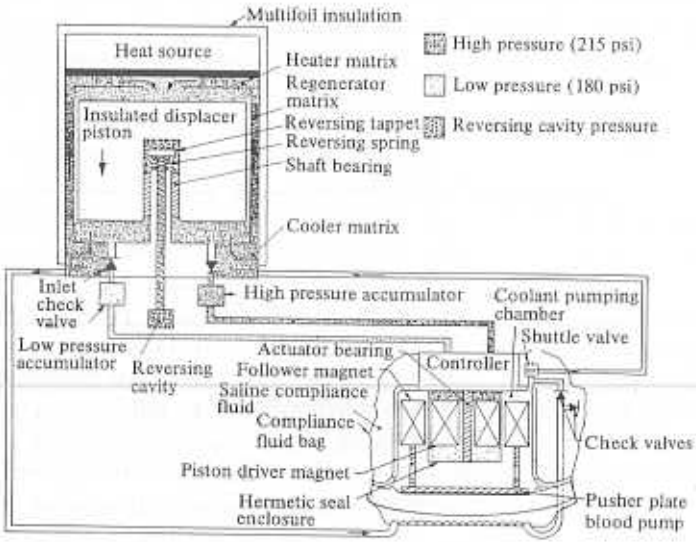
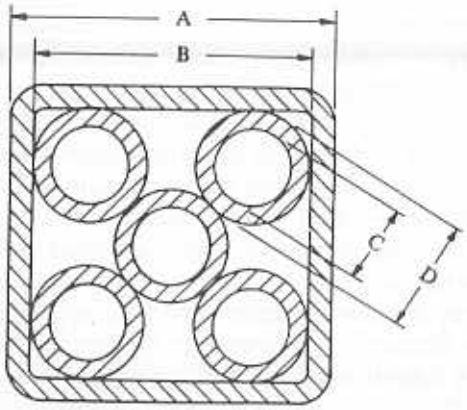


FIG. 17.10. Schematic diagram of Aerojet heart-assist system (mark VII engine).

component. This was a dry lubricated rubbing unit of an alumina sleeve riding on an alumina shaft. It is difficult to share the sanguine view of Moise. With data from only 3943 hours of actual engine running time, to project the wear rate for 10 years (63 000 hours) he allowed that the total quantity of debris generated from all sources in ten years of operation would be 0.05 cm^3 (0.003 in^3) . A speck or two of this in the wrong place on that most critical bearing surface would wreck the chances of achieving the 10-year life. Wear debris on the valve seats or faces would also prevent satisfactory operation of the engine.

A most interesting feature of the Aerojet engine is the use of long hollow fine bore glass 'straws' as the regenerative matrix arranged in annular form around the inside of the cylinder. A cross section of square straw element is shown in Fig. 17.11. The working fluid flowed around and through the straws in passing from the hot space to the cold space. A variety of different cross-section forms for the straws had been evaluated but no technical details of the optimum forms were released. Straws of improved 'arc matrix' form shown in Fig. 17.12 were investigated theoretically by Hoffman (1976) using the Aerojet engine simulation program. He obtained the results shown in Fig. 17.13 for net pneumatic efficiency as a function of the engine cyclic rate with different section straws and radial gap dimensions. Details of the Aerojet simulation work and the degree of correlation between predicted and observed performance have not been published.

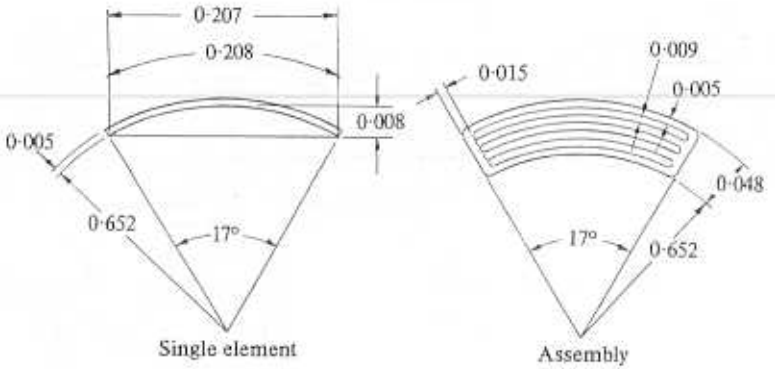


A +0·000 −0·001	B ±0·001	C ±0·001	D ±0·001	L +0·000 +0·000
0·058	0·048	0·012	0·020	3·300
0·052	0·044	0·012	0·018	3·150
0·058	0·048	0·014	0·020	3·300
0·056	0·046	0·015	0·019	3·300
0·059	0·049	0·014	0·020	3·300
0·053	0·045	0·012	0·018	3·300
0·053	0·045	0·015	0·018	3·300
0·059	0·049	0·015	0·020	3·300
0·059	0·049	0·012	0·020	3·300

FIG. 17.11. Cross-section of square straw regenerative element (after Hoffman 1976). Typical dimensions for A and C are, respectively, 0.055 and 0.015 in.

The Aerojet artificial heart development involved considerable animal experiments with components and part systems to establish and evaluate various physiological and anatomical compatability criteria (see Fig. 17.14). Animal test-work for the Aerojet program was carried out in cooperation with medical and veterinary personnel of the University of California, Davis.

A complete discussion of all aspects of the Aerojet program may be found in the series of annual reports, for example see Andrus (1976).



DASH NO.	E .002	G +.000 −.001	H +.000 −.001	θ ±.08°	L +.000 −.010	M +.000 −.001	R +.001 −.000	S	T ±.0005	W +.000 −.001
-1	.015	.009	.008	17.0°	3.300	.049	.652	.208	.0055	.207
-2	.015	.009	.008	17.0°	3.300	.046	.652	.208	.0050	.207

FIG. 17.12. Arc-matrix regenerator straws (after Hoffman 1976). Dimensions are in inches.

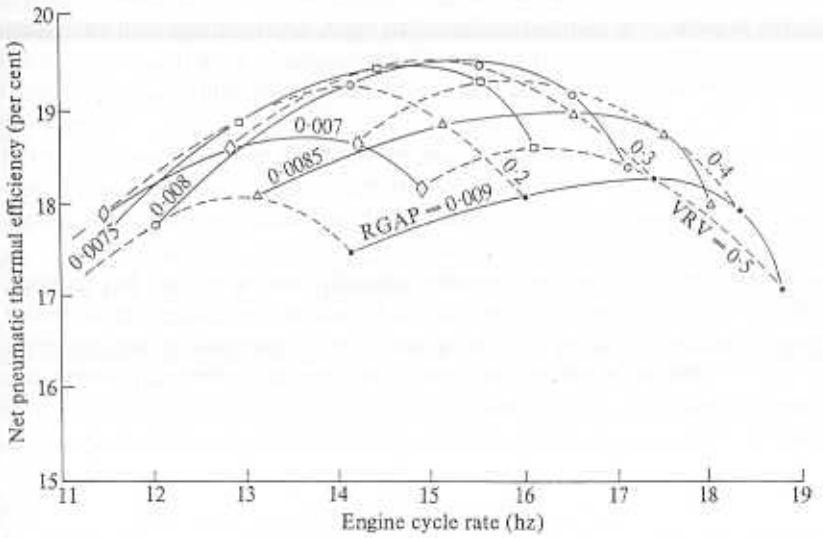


FIG. 17.13. Pneumatic efficiency of Aerojet thermocompressor as a function of engine cyclic rate with different regenerator sections (after Hoffman 1976; RGAP denotes the radial gap in inches; VRV denotes the regenerator void volume).

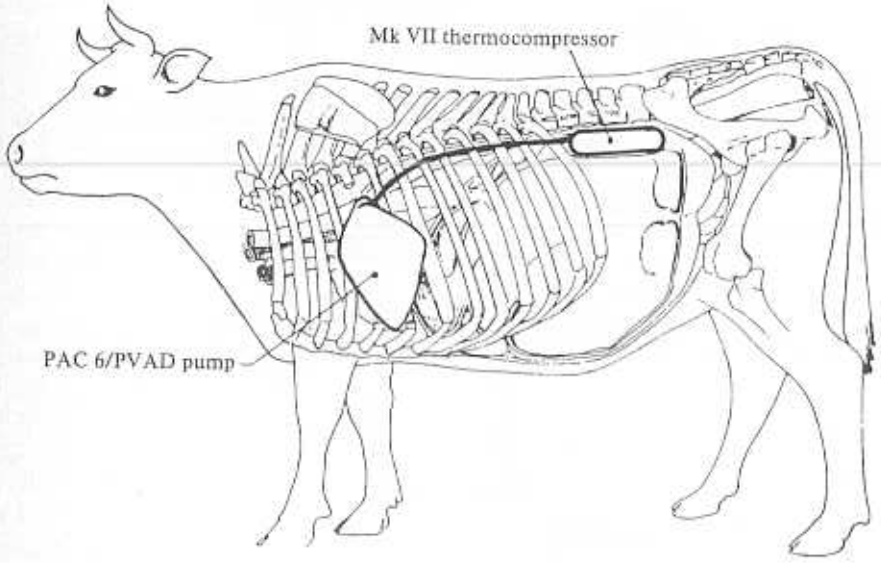


FIG. 17.14. Animal tests of Aerojet heart-assist systems are carried out at the University of California, Davis (after Moise and Faeser 1977).

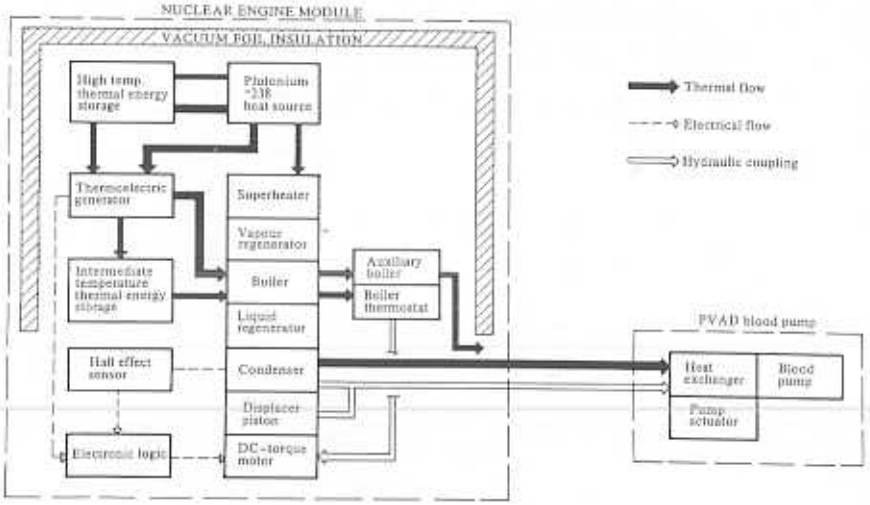


FIG. 17.15. Block diagram of elements in the Thermo-Electron artificial heart system (after Watelet, Ruggles, and Hagen 1976).

THERMO-ELECTRON ARTIFICIAL HEART ENGINE

Work on the Thermo-Electron engine began in 1970 and has been reported regularly in annual reports (e.g. Watelet, Ruggles, and Hagen 1976), the annual Intersociety Energy Conversion Engineering Conference (e.g. Watelet, Ruggles, and Torti 1976), and elsewhere. In the course of development the engine sustained major changes in design. The system described here is the Model 4 unit reported in the above references.

The Thermo-Electron engine was originally called the annular tidal regenerator engine and was not generally reckoned to be a Stirling engine. However, the engine is, in fact, a Stirling engine and furthermore, one of unique character. The working fluid, water, changes phase from liquid in the cold space to vapour in the hot space and so may be described as a one-component, two-phase working fluid†.

A schematic diagram of the Thermo-Electron system is shown in Fig. 17.15. It consists of two principal assemblies; the engine module and a pump module. In the engine module, nuclear or electric heat is converted into mechanical work to operate a hydraulic pump. The hydraulic fluid is then conveyed to the second unit to actuate the blood pump.

A simplified cross-section of the engine is shown in Fig. 17.16. It can be recognized as a Stirling engine of Heinrici form having a piston and displacer in separate cylinders. The expansion space is the heated space

† The reader is referred to Chapter 8, on working fluids in Stirling engines, for further discussion.

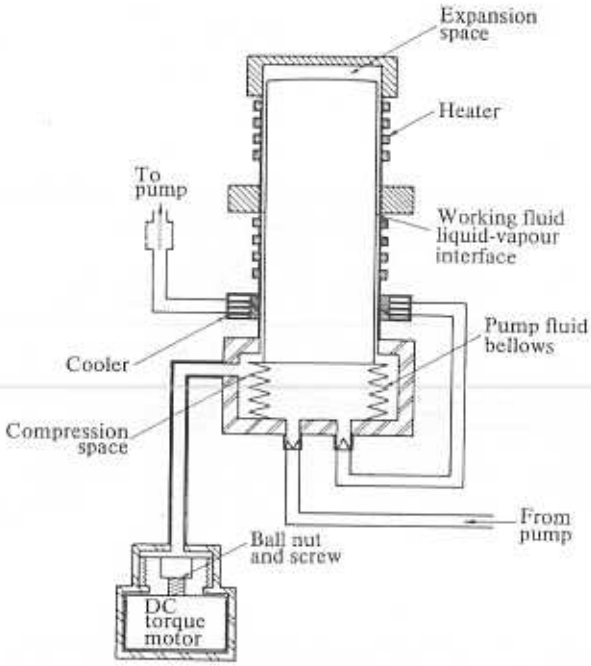


FIG. 17.16. Simplified cross-section of the Thermo-Electron tidal regenerator engine. This may be classified as a Stirling engine having separate cylinders for the piston and displacer and with a one-component condensing working fluid.

above the displacer and the compression space is the cooled spaces below the displacer and above the piston. The hydraulic pump is a metal bellows located below the displacer with inlet and outlet check valves.

Piston motion is induced by an electric motor driving a screwed shaft on which is mounted a nut attached to the underside of the piston. Rotation of the motor in one direction causes the piston to rise; rotation in the other causes it to descend. The motor is controlled by solid-state electronic logic unit in terms of the duration and direction of rotation, and the cyclic frequency. Power for the motor and the electronic control is provided from a silicon-germanium thermoelectric unit heated by the power source.

As the piston ascends, driven by the electric motor, liquid working fluid moves into the displacer cylinder and passes up the annular gap between the displacer and the cylinder wall. In passage through the annular gap it boils to a vapour and is further heated to a superheated vapour.

The phase change to vapour causes an increase in pressure in the working space. When the pressure exceeds that in the hydraulic pump

circuit, in the bellows below the displacer, the displacer begins to descend. This causes more working fluid to be displaced through the regenerator into the hot space, thereby increasing the pressure further, which in turn accelerates the motion of the displacer.

As the displacer descends the bellows below is compressed and hydraulic fluid is expelled through the outlet check valve at the pre-set discharge pressure.

When the piston drive motor reverses rotation the piston descends, the direction of fluid flow reverses and the pressure decreases. When it becomes less than the pressure in the hydraulic circuit the bellows below the displacer expands, driving the displacer upwards. This causes the working fluid to flow back through the annulus where it is cooled and changes phase to liquid resulting in further decrease in pressure and thereby accentuating the upward displacer motion. As the bellows expands fresh hydraulic fluid is drawn through the inlet check valve into the bellows space below the displacer. Waste heat from the engine is carried by the hydraulic fluid to the blood pump which incorporates a heat exchange for dissipation of the heat in the blood.

The Thermo-Electron artificial heart system was different to other Stirling engine systems reviewed here for the engine operated at the frequency of the natural heart. As a heart-assist system it was necessary to have a sensor which synchronized the operation of the engine/pump operation with the natural heart.

No technical details about the engine in terms of pressure range, cylinder diameter, or displacer stroke were given in any of the published material consulted. Engine efficiencies were said to be in the range of 15 to 16 per cent with maximum cycle temperatures of 480 °C (900 °F) to 540 °C (1000 °F). The engine efficiency was defined as the ratio: η = indicated work/(condenser heat + indicated work).

A cross-section of an implantable engine module is shown in Fig. 17.17. This was described by Watelet, Ruggles, and Torti (1976) as 20 cm (7.9 in) long by 6.1 cm (2.4 in) diameter for a volume of 700 cm^3 (42.7 in^3) and a weight of 1.6 kg (3.5 lb_m). In bench tests of the complete system, a power input of 33 watts to the engine module resulted in approximately 3 watts hydraulic output *from the blood pump* at a frequency of 90 to 110 beats per minute.

Endurance testing of a complete inplantable engine module was carried out for 24 hour day operation at 70 beats a minute for 1200 hours. In this time two bellows failures were recorded, both being the main pump bellows beneath the displacer. A second metal bellows was used to seal the piston but apparently this endured the test without incident.

Further improvement, to a value of 20 per cent, in the efficiency of the tidal regenerator engine was anticipated with the development of the

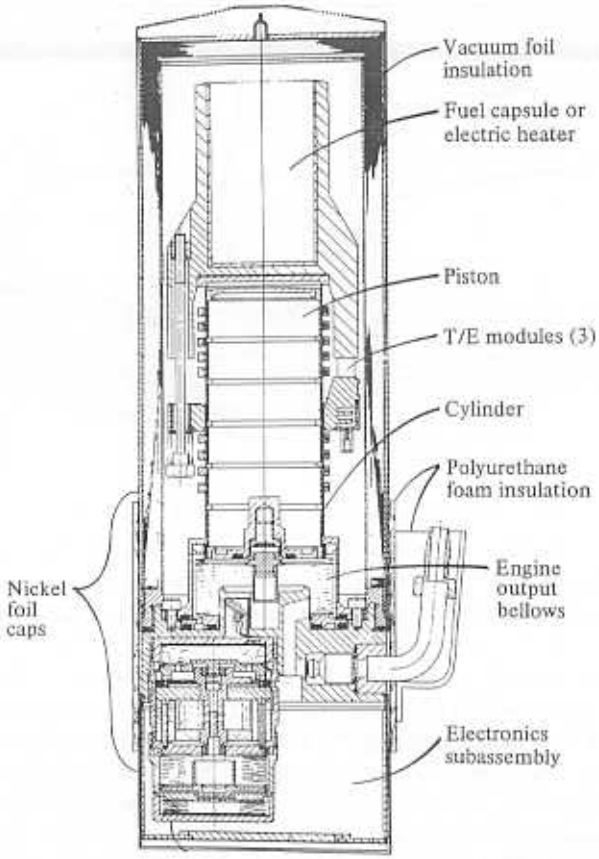


FIG. 17.17. Cross section of implantable engine module of the Thermo-Electron tidal regenerator engine (after Watelet, Ruggles, and Hagen, 1976).

binary tidal regenerator engine. This was a combination in a single machine of two tidal regenerator engines using different working fluids. The two fluids discussed by Watelet, Ruggles, and Hagen (1976) were water and the proprietory fluid Dowtherm A. The two thermodynamic cycles, superimposed on a common temperature/entropy plane, are shown in Fig. 17.18. The engine was essentially the existing tidal regenerator engine with water as the working fluid on which was superimposed another engine using Dowtherm and operating at higher temperature. The Dowtherm displacer was a double-wall unit with the water displacer operating inside it. Heat was supplied to the engine at the maximum cycle temperature in the Dowtherm boiler and was transferred at some intermediate temperature from the Dowtherm condenser to the water boiler. It was rejected from the engine at the low temperature of

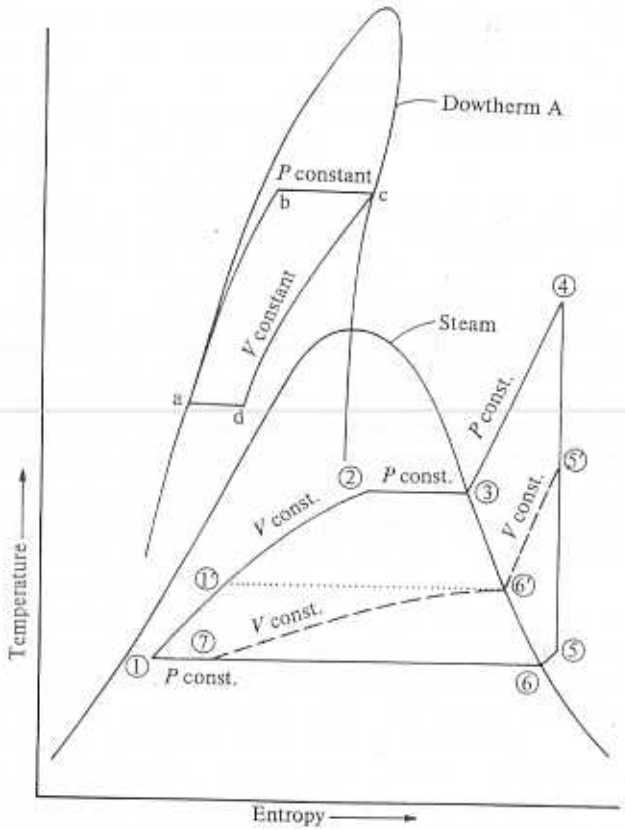


FIG. 17.18. Superimposed thermodynamic cycles of the binary tidal regenerator engine with water and Dowtherm working fluids (after Watelet, Ruggles and Hagen, 1976).

the water condenser and hence dissipated to the blood.

Watelet, Ruggles, and Hagen (1976) described a bench model binary engine at an early stage of development: 'that started easily and generated 3.6 watts power output at a frequency of 100 beats per minute with an efficiency close to 10 per cent. The maximum engine pressure was 0.86 MN/m^2 (125 lb per sq in) corresponding to a Dowtherm saturation vapour temperature of 382 °C (720 °F)'.

Animal experiments associated with the Thermo-Electron system development have been carried out in cooperation with the Cardiovascular Surgical Research Laboratory of the Texas Heart Institute, Houston.

WESTINGHOUSE/PHILIPS ARTIFICIAL HEART ENGINE

The Westinghouse/Philips artificial heart system development was sponsored by the U.S. Department of Energy, formerly the Energy

Research and Development Administration and before that the U.S. Atomic Energy Commission. The AEC program was reviewed by Mott *et al.* (1972).

An early indication of Westinghouse interest in this field was given by Lance and Selz (1968) reporting a conceptual study for a Rankine-cycle rotary steam engine with a Plutonium 238 heat source. When the present program started in the early 1970s the concept had been converted to a radioisotope-fuelled Stirling engine driving a blood pump by means of a flexible shaft. The system was a cooperative effort of the Westinghouse Electric Corporation and the North American Philips Corporation. Philips was responsible for the Stirling engine development.

Pouchot and Daniels (1974) described the concept and early work carried out on a bench model demonstrator. A conceptual view of the complete system is shown in Fig. 17.19 and a schematic diagram in Fig. 17.20. The engine was a low-speed (600 revolutions per minute) rhombic-drive piston-displacer unit. The shaft speed was increased by gearing to 1800 revolutions for the flywheel rotation then down to 900 revolutions for the flexible shaft drive with a further reduction in speed to 120 revolutions per minute at the blood pump. The pump was a double cup unit driven by a crank Scotch yoke mechanism as shown in Fig. 17.21.

Goldowsky and Lehrfeld (1977) have reviewed the progress in the engine development following several years' work. The prototype implantable unit has been defined to the system envelope shown in Fig.

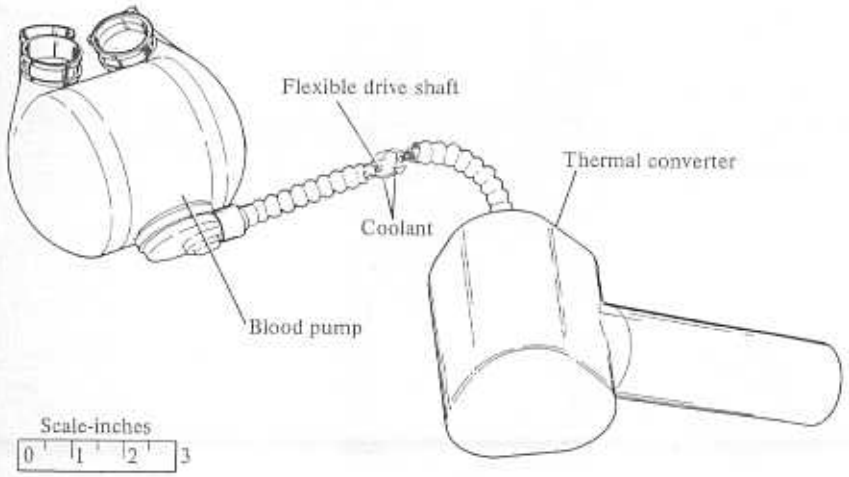


FIG. 17.19. Conceptual diagram of the Westinghouse/Philips artificial heart system (after Pouchot and Daniels 1974).

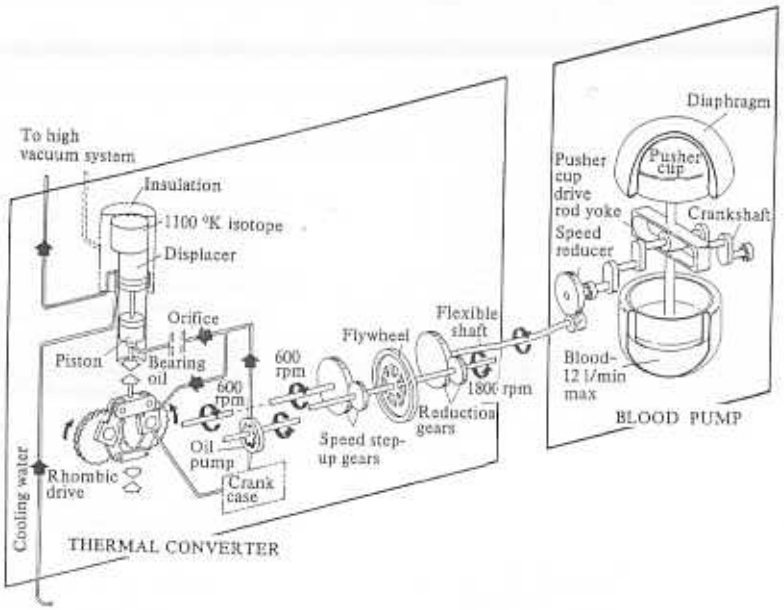


FIG. 17.20. Schematic diagram of the Westinghouse artificial heart system (after Pouchot and Daniels 1974).

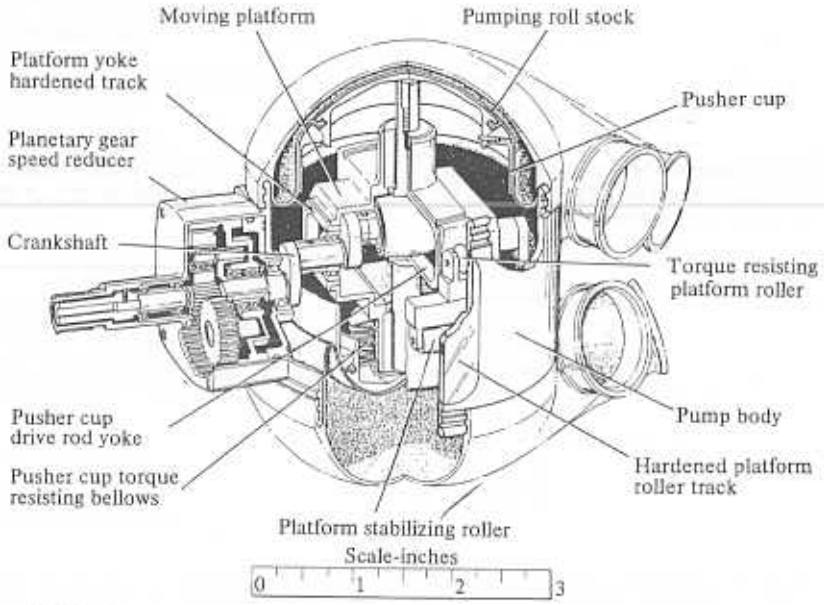


FIG. 17.21. Cross section of blood pump for the Westinghouse/Philips artificial heart (after Pouchot and Daniels 1974).

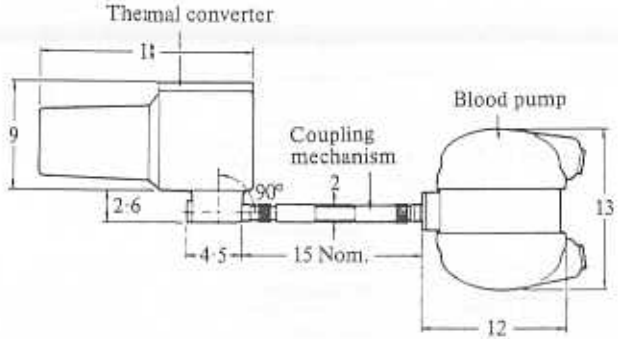


FIG. 17.22. Implantable prototype system envelope for the Westinghouse/Philips artificial heart (after Krasicki and Pierce 1977; dimensions are centimetres).

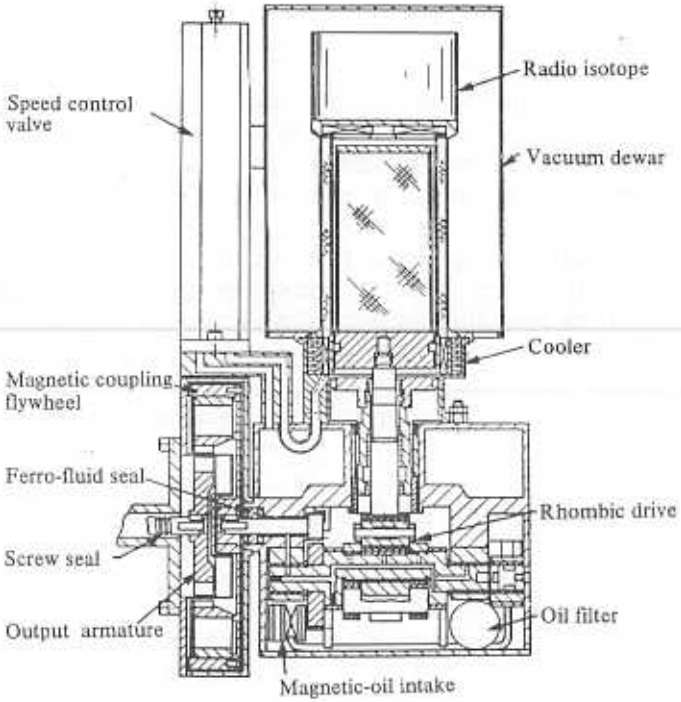


FIG. 17.23. Cross-section of Philips Stirling engine for the implantable prototype artificial heart system (after Goldowsky and Lehrfeld 1977).

Table 17.3. Design details for Westinghouse/Philips artificial heart engine (after Westinghouse 1975).

General specification:	
Input power (BOL)	29.5 watts
Net output power (BOL)	5.4 watts
Gross dimensions	17.9 cm × 7.6 cm × 8.9 cm
Weight	<1.9 kg
Working gas	Argon
Mean working pressure	14.6 atm abs
Crankcase pressure	6.1 atm abs
Expansion space temperature	900 K
Control system	Automatic short-circuit control valve
Hot finger design:	
Max. thin-wall temperature	925 K
Max. stress	44.4 ksi (18.0 atm abs peak pressure)
"Ideal" life at above stress	130 years (LM parameter = 43.4)
Stationary regenerator	5 cm thin-wall, 2.25 cm effective dia.
Insulation heat loss	1.5 watts
Drive mechanism:	
Oil-lubricated rhombic-drive piston stroke	0.8 cm
Displacer stroke	0.8 cm
Piston diameter	2.86 cm
Displacer diameter	2.54 cm
Pressure-fed drive bearings	
Seals:	
Piston and displacer rods	Rolling diaphragm seals
Piston and displacer	Piston rings
Flywheel/magnetic coupling cavities	Magnetic fluid seals
Materials:	
Titanium & steel drive elements	
Titanium crankcase & housings	
Nickel hot-side heat exchanger	
Aluminum/titanium ambient temperature heat exchanger	
External couplings:	
Direct drive starter	600 rpm
Magnetic output shaft coupling	2400 rpm
Mark I prototype weight estimate	
Heat source and vacuum insulation.	300 g
Hot finger and ambient cooler	200 g
Rhombic drive	195 g
Flywheel/magnetic coupling	230 g
Crankcase and flywheel housings	550 g
Oil	65 g
Other (controls, fittings, gears, oil pump, etc.)	250 g
Total	1725 g

17.22. A cross-section of the engine for this prototype unit is shown in Fig. 17.23. Some of the design details for the engine given in a quarterly progress report (Westinghouse 1975) are reproduced in Table 17.3. The projected energy balance for the engine, shown in Fig. 17.24, indicates that a thermal output of 29.5 watts from the isotope source is converted to 5.4 watts delivered to the flexible shaft.

In course of development a number of novel features have been introduced in this most interesting engine, some of which are likely to find application in larger machines. Argon is used as the working fluid. The reasons for this selection were not discussed in any of the references consulted but may have been to facilitate sealing or reduce thermal conduction. At the low speed of operation (600 revolutions per minute) the thermophysical characteristics of the working fluid are not so critical as in high power density machines.

Another innovation is the use of a magnetic lubricating oil or 'ferro-fluid'. This is discussed in detail by Goldowsky and Lehrfeld (1977). Use of the ferro-fluid made possible the development of a low-loss high-reliability engine shaft seal and a magnetic-type oil intake system to the lubrication pump. In future engines, ferro-fluidic-type seals may allow elimination of rolling-diaphragm seals on the reciprocating members.

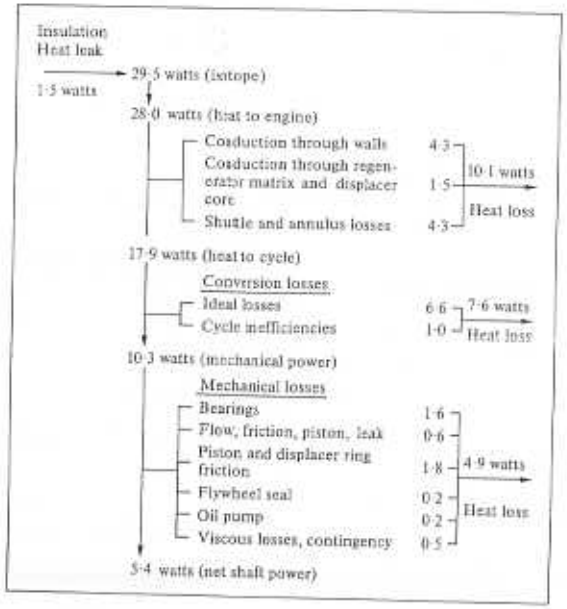


FIG. 17.24. Energy balance for the Philips Stirling engine in the artificial heart system (after Westinghouse 1975).

Another feature of interest is the use of magnetic coupling for transmitting power from the engine shaft to the flywheel ard hence to the output shaft. Construction of the flywheel in two parts coupled magnetically permits the interposition of an impervious titanium membrane and the consequent hermetic sealing of the engine crankcase. Interesting details of these innovative features will be found in the quarterly progress report (Westinghouse 1976).

One particular difficulty of the Westinghouse/Philips system was the transfer of engine reject heat to the blood pump for dissipation in the blood. Early versions used a water cooling circuit energized by a bellows pump incorporated in the blood pump (Pouchot *et al.* 1975). This was supplanted by a flexible heat pipe described by Krasicki and Pierce (1977).

No mention of animal experiments was included in any of the references consulted. It may be these are projected for a later phase of the program when operational prototypes of the system have reached a reasonable level of development.

Complete details of the progress of the Westinghouse/Philips program are reported in the quarterly progress reports (up to No. 29 at mid-1977). These are not readily available and are somewhat tedious to search through, so the general reader may find more useful the annual summary papers published in the Intersociety Energy Conversion Engineering Conference.

CLOSURE

Outside the United States the only known artificial heart research involving Stirling engines is the Messerschmidt-Bulkow-Blohm program in West Germany reported by Frank *et al.* (1974) and by von Reth *et al.* (1975). This is an early stage of development of a Bush-type free-piston thermocompressor engine similar to the Aerojet unit described above. The status of the program is unknown.

The other work on Stirling engines summarized above is a small part of a large program on artificial hearts in progress in the United States.

Frequently, these Stirling engine programs are overlooked because the power levels are so low and the subject so esoteric as to be outside the general interest. This assessment is mistaken. Four teams of exceedingly capable and innovative engineers have laboured for a decade at the absolute frontiers of technology in materials, control systems, heat transfer, fluid mechanics, thermodynamics, instrumentation, and experimental technique. There is much in their work that can be carried over into engines of larger size and power output, and close study of the annual contractor reports and papers in the open literature is recommended to all those professionally interested in Stirling engines.

18 STIRLING ENGINES FOR UNDERWATER AND SPACE POWER SYSTEMS

INTRODUCTION

STIRLING engines are prime candidates in applications where air-breathing engines cannot be used. Such applications are found primarily in underwater and space power systems. Self-contained engines capable of operating independently of the environment are required to convert stored energy to electric power, to mechanical work for propulsion, or to perform other necessary functions. The power level and mission duration requirements vary widely. At one extreme, a few watts for a short time can be adequately provided by batteries. At the other, thousands of horsepower for long periods in large submarines can only be provided by steam-turbine/nuclear-reactor systems. In between there are opportunities for many other combinations of energy source and conversion systems.

ENERGY SOURCES

Energy sources may be nuclear, either reactor or isotope, solar, or chemical. Nuclear energy sources, both reactor and isotope, depend on the radioactive decay or fission energy of certain materials for the production of highly energized particles. The energy of the particles is converted to thermal energy in dense materials enveloping the radioactive source. Reactor and isotope nuclear energy sources are essentially the same thing. In reactor systems, control devices are provided to vary the level of radioactive decay and hence power production, the engines can be shut down, or the output reduced as required. On the other hand isotope sources are essentially uncontrolled. They simply decay at a steady rate producing a virtually constant energy which declines slowly at a predictable rate depending on the isotope used. For example, plutonium 238 has a half life of 89 years. Thus, the power of a plutonium 238 source declines exponentially to half the power at intervals of 89 years. On the other hand, cobalt 60 isotope has a half life of only 5 years. The variations are endless, ranging from less than a second to thousands of years. In general, reactor systems are suitable for high-power, (say above 50 kW (68 hp)), long-endurance (greater than 1 month) applications. Isotope sources on the other hand are suitable for low and moderate power levels (up to 15 kW or 20 hp) and for long duration (greater than six months).

Solar energy is an attractive energy source for space power systems and has been widely used with photovoltaic converters, for the low power

level energies required on small exploratory space craft. The energy is available in virtually unlimited amounts at a rate which is inversely proportional to the square of the distance from the sun. It may be used to produce electricity directly, as in solar cells, or concentrated to produce high temperatures to drive engines.

Chemical energy sources comprise two or more chemical species which, when combined, react to form other compounds together with the production of electricity or heat. Lead-acid batteries, hydrogen-oxygen fuel cells and gasoline-air combustion are all common chemical energy systems. In space power systems chemical energy sources may include electric batteries or hydrogen and oxygen stored at low cryogenic temperatures as liquids for combination in fuel cells or combustion systems. Because of the weight involved chemical sources can only be used in space power for short duration missions at low or medium power levels.

Chemical energy sources are attractive for underwater power systems for short duration missions at all power levels. Weight is much less important than in space power systems but a major consideration is whether or not the reaction products can be stored on board. This is important for deep submergence systems where very high pressures would be necessary to expel the reaction products. It is important too in torpedoes and small tactical vehicles where the absence of a vapour trail of reaction products assists in evading detection. White (1961) has discussed the potential for lithium and sodium reacting with sea water for underwater propulsion.

THERMAL CONVERTERS

Nuclear, solar, and chemical sources can all be used to generate heat. Then a thermal converter may be used to change the thermal energy into electricity or mechanical work. All thermal convertors share a common limitation in the fraction of the heat supplied that can be converted to electric or mechanical power. This fraction, called the thermal efficiency,

$$\eta = C(T_{max} - T_{min})/T_{max} \qquad (18.1)$$

where C is some factor characteristic of the converter system involved and $(T_{max} - T_{min})/T_{max}$ will be recognized as the Carnot efficiency discussed earlier. When C has its maximum possible value of 1, the efficiency equals the Carnot efficiency, the highest possible efficiency between given temperature limits. In practice C ranges from 0.05 to 0.6.

Thermal converters may be split into two classes. Those without moving parts are called static thermal converters. Those with moving parts are called dynamic thermal converters.

Static thermal converters include thermoelectric systems. The absence of moving parts, seals, bearings, pistons, etc. makes it likely they will have

long life but they tend to be large and bulky, and have a low efficiency (less than 10 per cent). They are therefore mainly confined to relatively low-power applications of less than 100 watts (0.14 hp). Radioisotope thermoelectric generators (RTGs) are used in many low-power space and underwater applications.

Dynamic thermal converters tend to have a high power-density and efficiency compared with static thermal converters, although the difference is less marked in small sizes. Dynamic systems are therefore favoured for power levels exceeding 100 watts. There is no known upper limit. Dynamic thermal converters include all types of heat engine, but clearly, air breathing engines such as diesel or gasoline engines cannot be used for space and underwater systems. There are in fact three contending systems, Stirling engines, Rankine-cycle (steam) engines, and Joule/Brayton closed-cycle gas-turbine engines. All these engines operate on closed thermodynamic cycles converting heat to mechanical work to drive electric-power generators, pumps, or drive mechanisms. They can all use any kind of heat source and they must all reject a substantial fraction of the heat from the system. They can exist in a wide variety of forms or mechanical arrangements. Stirling engines are generally thought of as reciprocating machines and Joule/Brayton engines as rotary machines. However, the closed-cycle regenerative gas turbine conforms exactly to the definition of a Stirling engine adopted earlier and may simply be regarded as one type of rotary Stirling engine. Many other forms of rotary Stirling engines are possible. In a similar way the Rankine engine exists both in reciprocating and rotary forms.

Turbine engines operate at high speeds and high power-densities compared with reciprocating piston engines. However, the thermal efficiency of reciprocating engines is usually higher and appreciably so at the lower power levels (less than 50 kW or 68 hp), primarily because of high turbine blade leakage losses, and in turbines without regenerative heating, high stack losses. At high powers the efficiency of turbine systems can approach those of reciprocating systems and the power-density advantage is so great that piston engines are rarely found in large sizes.

At medium and low powers (less than 200 kW or 272 hp) Stirling engines are favoured because in comparison with Rankine-cycle engines they are much less complicated, have a higher efficiency, and a better power-density. Further, they offer more flexibility in the independent selection of pressure and temperature levels than vapour systems where the pressures and temperatures are related.

In most applications, but particularly so in space power-plants, the thermal efficiency is of great concern. A high efficiency requires a reduced isotope or fuel inventory or, alternatively, allows for a higher power or longer mission for the same isotope or fuel weight and cost. Furthermore,

a high efficiency reduces the amount of heat that has to be rejected from the spacecraft.

The process of heat rejection from a spacecraft is a major factor. All the energy used for any purpose on the vehicle eventually ends up in the form of heat and must be rejected from the spacecraft if the temperature is to remain constant. There is no cooling water or cooling air as we have available on the Earth so consequently *convective* heat transfer cannot be used. Radiation heat transfer is the only mechanism for dumping the heat into space.

The controlling equation for radiation heat transfer from a body is:

$$Q_{rad} = \varepsilon A \sigma T^4 \qquad (18.2)$$

where ε = the emissivity, an optical property that is characteristic of the surface, i.e. a dull black surface has a high emissivity whereas a polished gold plate surface has a low emissivity.

A = area of the body for heat transfer (this is the projected area looking out to space—it does not help much to have a convoluted or irregular surface with a large surface area).

σ = a number, called the Stefan–Boltzmann constant.

T = the absolute temperature of the body.

The most important aspect of the above equation is to note that the quantity of heat transferred from a body by radiation depends on the *fourth* power of the temperature. Doubling the temperature increases the heat lost by sixteen times!

On spacecraft, radiators are used to effect the heat transfer to space. If fluids are used to transfer heat from the power-plant to the radiators it is necessary to armour the radiators so as to prevent punctures by particles in space. Radiators therefore tend to be large, heavy units which provide a large area for heat transfer.

However, the rate of heat transfer increases only linearly with the radiator area whereas it increases as the fourth power with temperature. There is, therefore, great attraction in operating the radiator at high temperatures since it can be made smaller and lighter.

Unfortunately this trend is exactly opposite to the requirement for high efficiency. It will be recalled that the efficiency $\eta = C(T_{max} - T_{min})/T_{max}$. The maximum temperature T_{max} is limited by metallurgical considerations in the hot parts of the system and should be as high as possible. The minimum temperature T_{min} is the temperature of heat rejection from the cycle, the radiator temperature and should be as low as possible.

In the design of space power systems there is, therefore, a dichotomy of interest in the minimum cycle temperature between a low value on the one hand and a high value on the other. Considerations of system weight

are dominant. Frequently the radiator is the principal system mass, perhaps as much as 50 to 60 percent of the total system, but can be limited in size by the use of a high minimum cycle temperature which in turn results in a low cycle efficiency. There is great advantage in increasing the maximum cycle temperature and this may lead to the use of relatively exotic materials for the hot parts that are too expensive or difficult to fabricate except for highly specialized applications.

STIRLING ENGINES FOR SPACE POWER

Given the elementary but irrefutable logic above it is difficult to understand the virtual total neglect of Stirling engines by NASA during the 1960s. This was the era of prodigious expenditures on hydrogen-oxygen fuel cells for the Apollo missions, on the Rankine cycle SNAP8 reactor power-plant, and on various Brayton turbines. So are as is known, no NASA funds and only minor Air Force funds were expended on Stirling engines.

One possible explanation for NASA neglect of Stirling engines is that the confidentiality clauses in the Philips licence agreement foredad full-hearted participation by General Motors in Government contracts which required full disclosure. Other companies known to have been interested in the possibilities of the Stirling engine were daunted and discouraged by the impressive thorn-fence of patents that Philips built around the new technology and the exclusive licence with General Motors.

At any event only one program of substance on Stirling engines for space power-systems was carried out in the 1960s, and another small program is presently in progress.

The earlier program, sponsored by the U.S. Air Force, was executed by the Allison Division of General Motors and was directed to the design and development of a 3 kW (4 hp) solar space power-system. The Stirling engine, shown in Fig. 18.1, was a single-cylinder piston-displacer machine with rhombic drive and antecedents that were recognizably Philips. The engine used helium as the working fluid and was heated by solar energy concentrated by a large Fresnel lens. A comprehensive report (Parker and Malik 1962), in ten volumes, was prepared on the project and even at this late stage remains an interesting and worthwhile reference source. Volume 1 deals with the design of the engine and Volume 10 the test and evaluation of the prototype machine. So far as is known the unit never went beyond a first prototype and was never developed to the flight hardware stage.

This engine was particularly notable because it represented the first application to Stirling engines of an intermediate heat-transfer liquid-metal heating system. A sodium/potassium eutectic mixture was used to

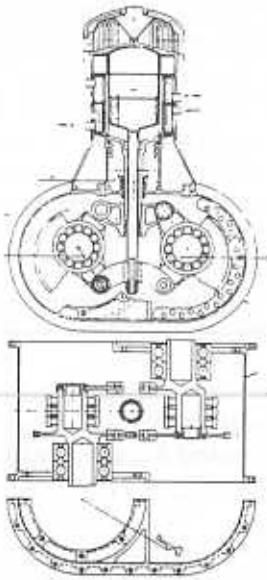


FIG. 18.1. Allison Type PD46 Stirling engine for 3 kW solar-heated space power-system (after Parker *et al.* 1962).

transfer heat from the solar energy absorber to the heater head of the engine. The use of an intermediate heating fluid is now becoming commonplace in advanced Stirling engines for various applications. Further details of the Allison space power-system are given in Chapter 13.

Another application of Stirling engines for space power-plants was initiated in 1975 by the U.S. Energy Research and Development Administration, now the Department of Energy. This is a 1 kW (1.4 hp) isotope power system (KIPS) being executed by Mechanical Technology Incorporated, Latham, New York.

Goldwater and Morrow (1977) have described the concept and the early phases of the development. The unit combines a single-cylinder Beale displacer-type free-piston engine with an MTI linear generator as shown in Fig. 18.2. The engine is heated by a radioisotope energy source. To establish a system in perfect dynamic balance an opposed-piston arrangement is proposed for the ultimate flight engine.

The well-publicized but unscheduled descent in Northern Canada in early 1978 of the Russian satellite with an isotope power-system (thermoelectric generator) aboard casts some clouds on the political acceptability of the KIPS system for the future. It is likely that, in civil applications at least, solar-powered systems will be emphasized. These

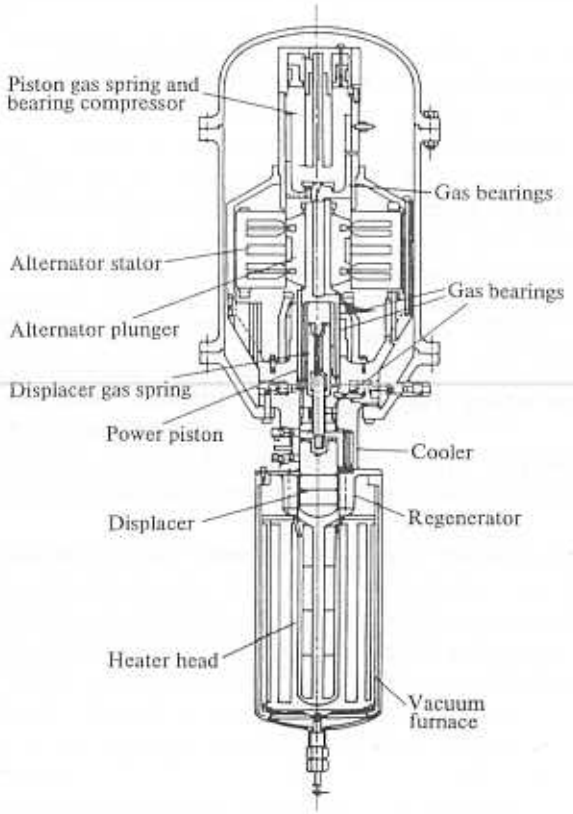


FIG. 18.2. MTI-Beale free-piston Stirling engine and linear alternator for 1 kW isotope space power-system (after Goldwater and Morrow 1977).

can be easily disabled by hostile action so that it is likely that military spacecraft will continue to use isotope sources.

The intensity of solar energy diminishes as the square of the distance from the sun. Therefore spacecraft engaged in exploration of the remote solar system find it increasingly difficult to collect adequate solar energy as their journey proceeds away from the sun. Power levels of a few watts are all that is required to remain in radio contact with the Earth and to provide power levels for space experiments. Where higher power levels are necessary one possibility being vigorously explored for energizing remote spacecraft is to use laser beam transmission. This would supply power by optical beam that could be collected, absorbed, and converted to heat and hence using a Stirling engine to electrical energy. A laser is a device which can highly concentrate light energy to a coherent beam or ray.

STIRLING ENGINES FOR UNDERWATER POWER

The situation with regard to Stirling engines for underwater power is much like that for space power—lots of promise and potential but little actual application experience.

A major United States report (Chapman 1968) on underwater power systems in the 1 to 100 kW (1.4 to 136 hp) power range recommended that significant effort be invested in Stirling engine development in preference to other dynamic converters and fuel cells. The particular advantages were that the reciprocating Stirling engine was admirably suited for the modular concept. There was little penalty in system weight, volume, or fuel consumption if four 10 kW (13.6 hp) units were used instead of one 40 kW (54.4 hp) unit. The modular system would allow the use of a small inventory of parts and components to fulfill a broad power range.

They judged the reciprocating Stirling engine to have a higher thermal efficiency potential than any of the rotary engines, and to have the favourable characteristic of operating on any source of heat at different temperature levels without noise and without the need to dump waste products overboard.

The Stirling dynamic converter system was judged to have a weight and volume advantage over fuel cell systems up to 3000 kWh (4079 hp hour) capacity. It was thought to have the advantage at all levels of energy requirement when fuelled with a halogen-metal reaction. Cost comparisons of the fuel cell and Stirling engine were established in favour of the Stirling engine.

The isotope-powered Stirling engine appeared particularly attractive in the power range from 1 to 15 kW (1.4 to 20.4 hp). It was found that a high proportion of U.S. Navy requirements for future deep-sea activities could be adequately served by three sizes of power generators of about 3, 10, and 25 kW (4, 13.6, and 34 hp) coupled in various combinations with three sizes of energy storage containers of 40, 100, and 1000 kWh (54.4, 136, and 1360 hp hours).

Despite these favourable recommendations no substantial activity in Stirling engine development was undertaken by the U.S. Navy. Indeed in a later study (McCartney and Cates 1975) of power sources for remote ocean-oriented applications Stirling-engine systems *were not even mentioned* despite the inclusion of other highly esoteric concepts.

During the tenure of their licence agreement with Philips, 1958–1970, General Motors invested considerable effort in studies of Stirling engines for underwater power systems. Percival (1967) reviewed some of this effort with particular reference to thermal energy storage systems and liquid-metal combustion. General Motors' interest in thermal-energy storage systems predates the Philips licence, for in 1957 a proposal was

made to the U.S. Navy for a Rankine-cycle system combined with lithium hydroxide thermal storage. Later in 1959 the proposal was converted to lithium fluoride thermal storage combined with a Stirling engine.

THERMAL STORAGE SYSTEMS

A thermal-storage system consists essentially of an insulated tank containing some material having a high heat capacity. The material can be heated by combustion, by electric heating, or by nuclear source. When the temperature of the material increases on heating the heat is said to be 'sensible' heat. When during heating the material melts at constant temperature, changing phase from solid to liquid, the heat is said to be 'latent' heat. Energy-storage materials should have a high heat capacity, low vapour pressure, and high density. They should be chemically stable and compatible with the container and heat transfer surface materials.

The heat capacity on a weight and volume basis of possible energy-storage materials is shown in Fig. 18.3. This figure was prepared for the energy transfer per unit mass or volume to effect a change in temperature from 538°C (1000°F) to the maximum temperature shown alongside each material. The maximum temperatures were selected on considerations relating to the heat-storage materials and without regard to containment or insulation characteristics.

The first four materials are molten salts in which a high proportion of the heat transfer is latent heat associated with a phase change occurring at the maximum cycle temperature. Lithium hydride has an exceptionally

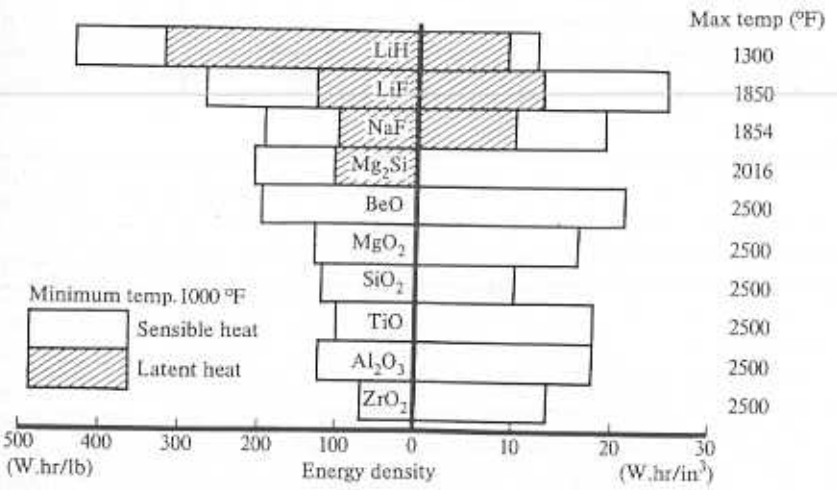


FIG. 18.3. Energy density of thermal energy-storage materials on a weight and volume basis (minimum temperature 1000 °F, after Mattavi *et al.* 1969).

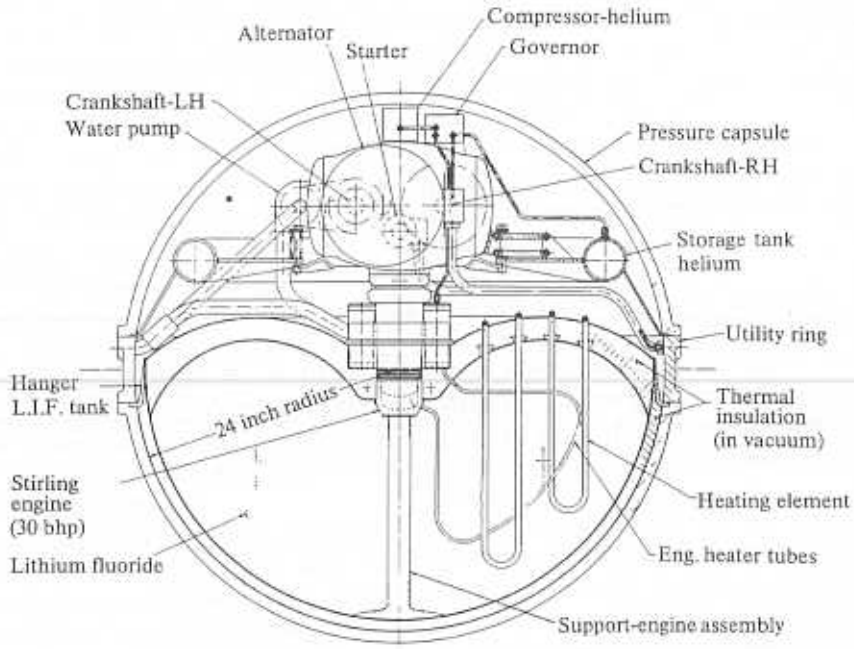


FIG. 18.4. Cross-section of spherical underwater power-system module including a 30 hp Stirling engine and a lithium fluoride thermal reservoir (after Percival 1967).

high heat capacity on a mass basis but this salt readily dissociates at temperatures only slightly above its melting point. A preferred material is lithium fluoridé which has a higher melting temperature and a better volumetric heat capacity. Unfortunately the latent heat capacity on a mass basis is less than half that for the hydride. High latent heat capacity is favourable to maintain a constant temperature as heat is withdrawn to operate the engine.

Most of the General Motors work on thermal-energy storage was done with lithium fluoride but other materials were studied, particularly aluminum oxide. Percival (1967) describes one experimental unit using 60 000 hexagonal pellets of aluminum oxide in an insulated tank. The system operated between temperature limits of 1482.2 °C (2700 °F) and 815.6 °C (1500 °F) with a storage capacity of approximately 73.6 kWh brake power (100 brake hp hours) when used in conjunction with a 22 kW (30 hp) Stirling engine.

Fig. 18.4 is a cross section of a 22 kW (30 hp) Stirling engine underwater power-plant with lithium fluoride thermal energy-storage, (Percival

1967). Approximately 453.6 kg (1000 lb) of lithium fluoride was contained in the lower hemisphere of the spherical capsule, 106.7 cm (42 in) in diameter. The engine heater tubes were immersed directly in the molten salt, heated initially by electric-resistance elements. In this system, the heat losses through the 2.54 cm (1 in) of multi-layer insulation were said to be 0.1 per cent of the stored heat per hour. A pump was used to circulate engine coolant to an ambient-temperature, sea-water heat sink located outside the pressure hull to eliminate high-pressure sea-water inside the hull. Power output was regulated by varying the pressure of the working fluid, hydrogen, or helium in the engine.

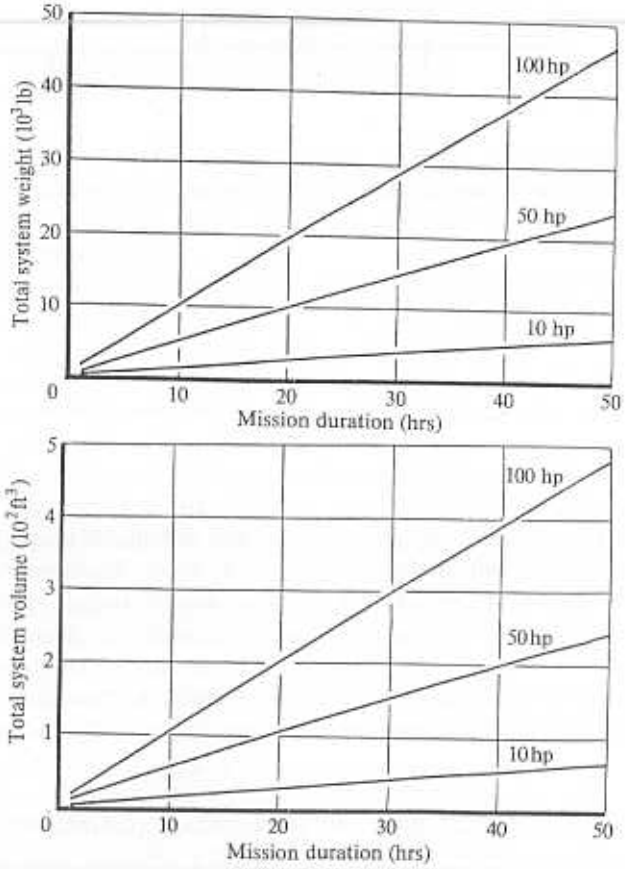


FIG. 18.5. Weight and volume requirements as a function of power level and mission duration for thermal energy storage and Stirling-engine underwater power systems (after Mattavi *et al.* 1967).

Mattavi *et al.* (1969) provided the data shown in Fig. 18.5 for the thermal energy-storage/Stirling-engine system weight and volume requirements as a function of the mission duration. Three different power levels, 7.3, 37, and 74 kW (10, 50, and 100 hp) were considered.

Other data included that reproduced in Fig. 18.6 showing the weight and volume capacities for a thermal-energy storage system as a function of storage time with two different types of insulation. Table 18.1 is a weight estimate and heat balance given by Mattavi *et al.* (1969) for a 11 kW (15 hp) Stirling engine and accessories for an underwater power system. This unit was basically a modified G.M. Type GPU-3 engine for which performance data is given in Chapter 13.

It should be noted that none of the above data included allocations for the pressure hull to house the power system or the material necessary to achieve a neutral buoyancy.

METAL COMBUSTION

The combustion of hydrocarbon fuels such as gasoline in air with consequent release of heat is a matter of common experience. In a similar way most metals can be combined with an oxidant to release heat. The advantages of metal combustion for underwater power systems are the relatively high heat of reaction and the absence of gaseous products of combustion. The reaction products are either solid or liquid depending on the temperature and occupy practically the same space as the original fuel. The products can therefore be retained on board. This eliminates both the problems of compressing the products to a sufficiently high pressure for expulsion to the marine environment and of vapour bubbles trailing from the vehicle.

Fig. 18.7 shows the energy density, on a weight and volume basis, for several metal/oxidant combinations (Mattavi *et al.* 1969). The desirable characteristics of a fuel/oxidant combination include:

1. A high heat of reaction on both a weight and volumetric basis.
2. The reaction should be controllable.
3. Reaction temperatures should be within limitations associated with construction material properties.
4. The products of combustion should be non-gaseous at the desired reaction temperature. In general, the formation of solid products is also undesirable since they can produce a reaction-inhibiting layer in the reaction zone. In addition, solid products can also increase the viscosity of the liquid melt, thereby adversely affecting liquid circulation.
5. The products of reaction should be soluble in the parent metal to promote control during reaction depletion and product formation.

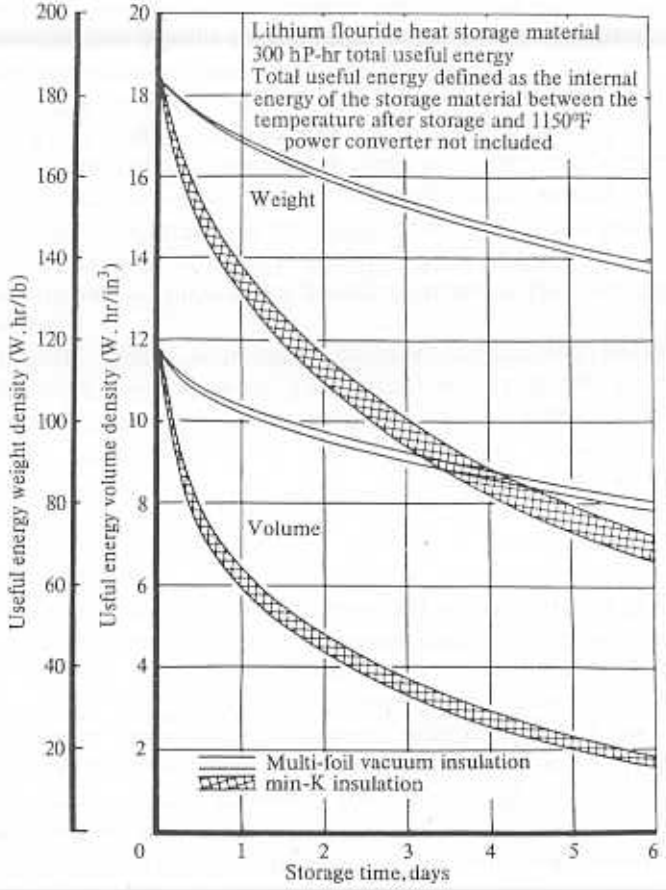


FIG. 18.6. Weight and volume requirements for a thermal energy-storage system using lithium fluoride, as a function of storage time, with different types of insulation (after Mattavi *et al.* 1969).

6. In a closed system, the products of reaction should occupy essentially the same volume as the metal fuel.
7. The reactants and products should be non-toxic. This is particularly important for the protection of personnel if they are located in the same closed environment of the underwater vehicle.
8. The fuel and oxidant should be compatible with materials of construction.
9. The reactants should be readily storable for extended periods of time. It is also desirable to be able to store the reactants in a soft package outside the pressure hull to reduce the overall system

Table 18.1. 15 hp Stirling engine and accessories weight

Item	Weight (lb)
Basic engine for use with heat storage	66.4
Required accessories:	
Water pump	2.6
Hydrogen reservoir	1.3
Hydrogen valve	1.5
Hydrogen compressor	2.5
Coolant loop	3.0
Instrumentation, wiring, plumbing	5.0
Total basic system	82.3
Other accessories for specific needs:	
Starter motor	6.2
Starter switch	0.8
Battery	14.5
Battery charger	9.7
Speed governor	6.0
Flywheel*	7.5
Voltage regulator	1.2
Generator drive coupling	2.0

* Flywheel requirements depend upon such particular integration considerations as inertia of generator or reduction gearing, etc. The above engine by itself requires shown flywheel weight for optimum performance.

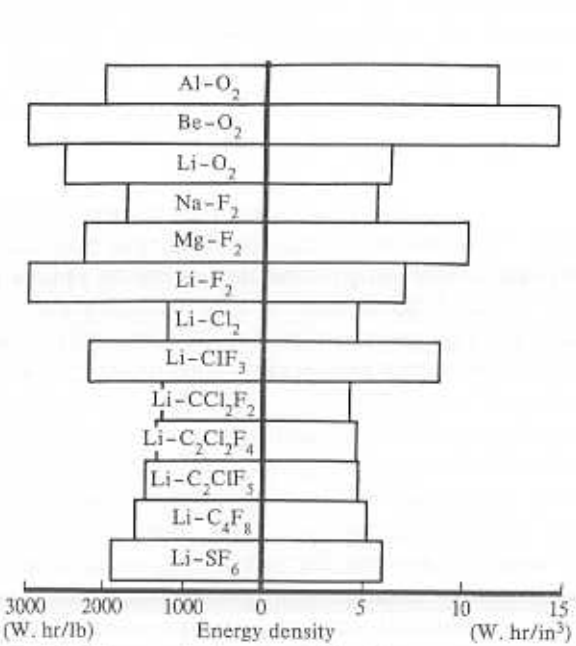


FIG. 18.7. Energy density of metal/oxidant combinations on a weight and volume basis (after Mattavi *et al.* 1969).

weight. In this case, the bulk modulus of the reactants should equal that of the ambient sea-water in order to minimize changes in buoyancy during storage periods.

10. The reactants should be economically feasible.

Unfortunately no combination meeting all these requirements has been located. A preferred combination is the metal lithium with various halogen compounds, sulphur hexafluoride or chlorine trifluoride. The halogen compounds, identified generally by the proprietory term 'Freons', and sulphur hexafluoride SF_6 are safe oxidants for storage and handling purposes. They combine with lithium in an intense but non-violent reaction accompanied by the release of heat. Some details of the experimental work on open and closed systems have been given by Mattavi *et al.* (1969). Average temperatures outside the reaction zone ranged from 650 to 1040 °C (1200 to 1900 °F) and were easily controlled by the admission of an oxidant.

Weight and volume estimates for Stirling-engine underwater power-systems energized by metal combustion are shown in Fig. 18.8 (Mattavi *et al.* 1969) as a function of the mission duration and for the three power levels 7.3, 37, and 74 kW (10, 50, and 100 hp). These curves are directly comparable with the corresponding data for thermal storage given in Fig. 18.5. Comparison of the data indicates that a metal-combustion system could be about half the size and one-third the weight of a thermal-storage system of equivalent power on a mission of similar duration.

Percival (1967) presented the interesting comparison reproduced in Fig. 18.9 of the weight per horsepower of various underwater propulsion systems as a function of the hours of operation. This comparison indicated the metal-combustion system to be far superior to any other candidate. In the same paper Percival also presented a comparison of torpedo propulsion systems. This is reproduced in Fig. 18.10 and again the Stirling engine with metal combustion emerges as the preferred candidate.

The interest in underwater power systems at General Motors stimulated a renewal of interest in double-acting engines to achieve a more compact design (Mattavi *et al.* 1969). In particular the Siemens arrangement of a multiple-cylinder double-acting engine driving a wobble-plate or swash-plate received substantial attention. This configuration, shown in Fig. 18.11, is clearly ideally suited for a torpedo drive. Substantial effort was invested in the development of swash-plate drive systems of improved performance (Maki and DeHart 1971).

Following the abrupt termination of the General Motors program in 1970 (Percival 1974), the General Motors work on underwater power systems was not entirely lost. Philips appears to have taken over much of it. As described in Chapter 12, all the Philips development effort in the

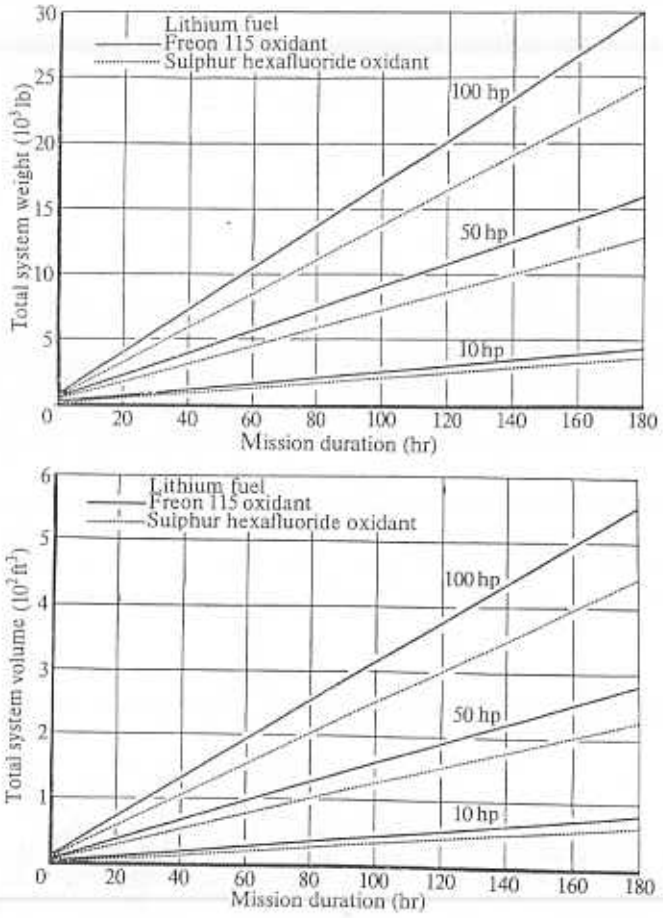


FIG. 18.8. Weight and volume requirements as a function of power level and mission duration for Stirling-engine underwater power-system with metal combustion (after Mattavi *et al.* 1969).

1970s has been devoted to Siemens-type swash-plate engines. The automotive engines being developed by Philips for Ford under Department of Energy sponsorship are swash-plate engines. It is ironic that an engine which started life as a torpedo motor at General Motors will most likely end up a decade later as the automotive engine of its principal competitor.

Philips also continued work on metal combustion as an energy source for underwater power systems. Two papers (van der Sluys 1975, and

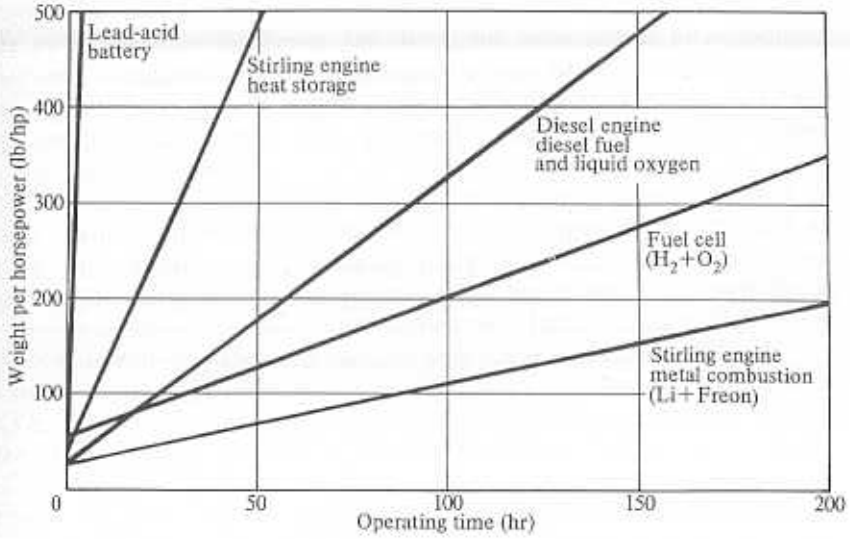


FIG. 18.9. Comparison of the specific power of different underwater power systems (after Percival 1967).

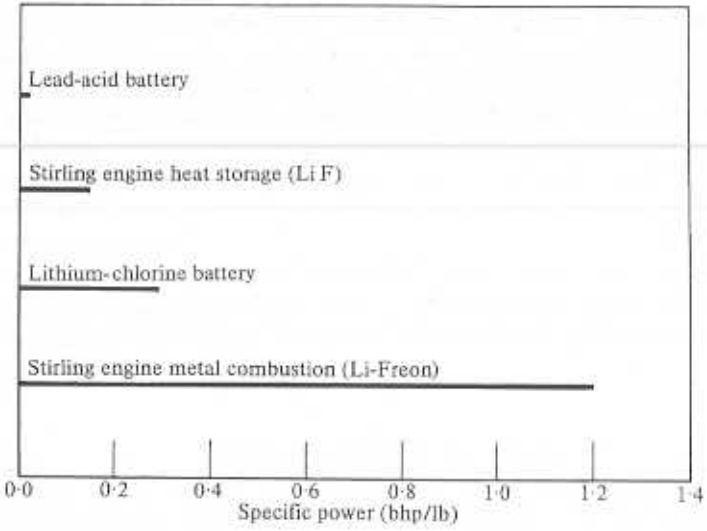


FIG. 18.10. Comparison of specific power for four torpedo propulsion systems (after Percival 1967).

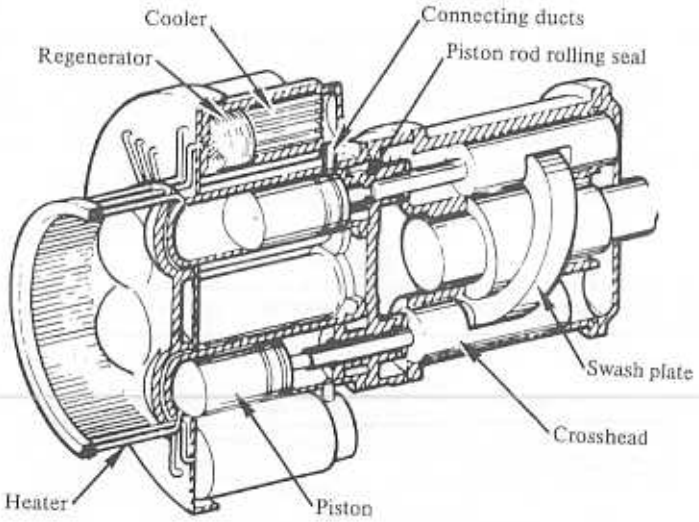


FIG. 18.11. Siemens swash-plate drive double-acting engine for torpedo propulsion (after Mattavi *et al.* 1969).

Biermann 1975) describe in considerable detail experimental and development work with lithium/sulphur hexafluoride heat sources for combination with Stirling-engine propulsion systems.

Substantial work on Stirling engines for underwater power systems is said to have been carried out by other Philips licensees, United Stirling, Malmo, and the West German consortium MAN/MWM, but no details of the scope or direction of this work has ever been publicly disclosed.

19 STIRLING ENGINES FOR HEAT PUMPS, STATIONARY POWER AND TOTAL-ENERGY SYSTEMS

INTRODUCTION

STIRLING engines are under study or development for a variety of non-automotive applications that can be broadly classified into three groups: heat pumps, stationary power generation and total energy or co-generation systems.

A *heat pump* is a device used for heating or cooling, usually as part of the air conditioning system of a building. In the heating mode it draws heat at ambient temperature from a thermal reservoir, such as a river, lake, or the ground around the building. It 'pumps' the heat to a higher temperature for rejection from the pump to heat the building. The energy is eventually dissipated by conduction or convection from the building at approximately ambient temperature. In the cooling mode, the situation is reversed. Heat is drawn from the building at a low temperature and rejected to a heat sink, the same river or lake that served earlier as the source.

Stationary power generation embraces a wide range of energy conversion activity but is usually interpreted to mean the production of electric power. The same term can also be applied to pneumatic (air compression) or hydraulic power systems operating in a fixed location or as the auxiliary power systems on board an automotive, flight, or marine vehicle. Power levels can range from the few watts of an unmanned navigation signal generator to the gigawatts of a base-load electric-power utility system. Current interest in Stirling engines for stationary power is concentrated in modular engines of 500 to 2000 horsepower utilizing municipal, agricultural and industrial wastes and small low-power engines.

A *total-energy or co-generation system* is a jargon term employed to describe an ensemble of machinery used principally in buildings to provide all or most of the services required, i.e. electricity for lighting and power purposes, heating, cooling, and hot and chilled water, utilizing an input energy source such as coal, gas, oil, or a waste product including an exhaust thermal stream.

A total-energy system will therefore include a stationary power generator and, in all probability, a sub-system operating as a heat pump. However, a stationary power generator or heat pump is not necessarily part of a total-energy system nor need the heat pump incorporate a stationary power generator. Nevertheless, they can be related and seem to fit together in a common grouping although from hereon we shall consider them all under separate headings.

HEAT PUMPS

A Stirling engine may be used in relation to a heat pump in two ways:

(a) as the heat pump itself;

(b) as the prime mover driving a heat pump.

When the Stirling engine is used as the heat pump it accepts heat at a low (probably) ambient temperature and rejects the heat at a higher temperature to a heating system. Such an arrangement is shown diagrammatically in Fig. 19.1. The system is illustrated by reference to the ideal Stirling cycle P–V diagram and a single-cylinder, piston-displacer, single-acting, Stirling engine. Heat is absorbed during the expansion process at ambient temperature (T_E) from an external source (shown as a lake or river). The heat is rejected from the engine during the compression process and is carried off to perform a useful heating function in the building.

Of course work must be supplied (equal to the shaded area on the P–V diagram) to allow the Stirling engine to work in this way. This external work may be supplied from any power source, including an electric motor or a heat engine, perhaps another Stirling engine.

The arrangement of a Stirling engine acting as a prime mover producing power to drive another Stirling engine acting as a heat pump of a cooling engine is illustrated in Fig. 19.2. This may be called the duplex Stirling arrangement or the Stirling–Stirling arrangement. Laboratory versions of such a combination have already been constructed (see Chapter 11) and commerical applications are anticipated in the foreseeable future.

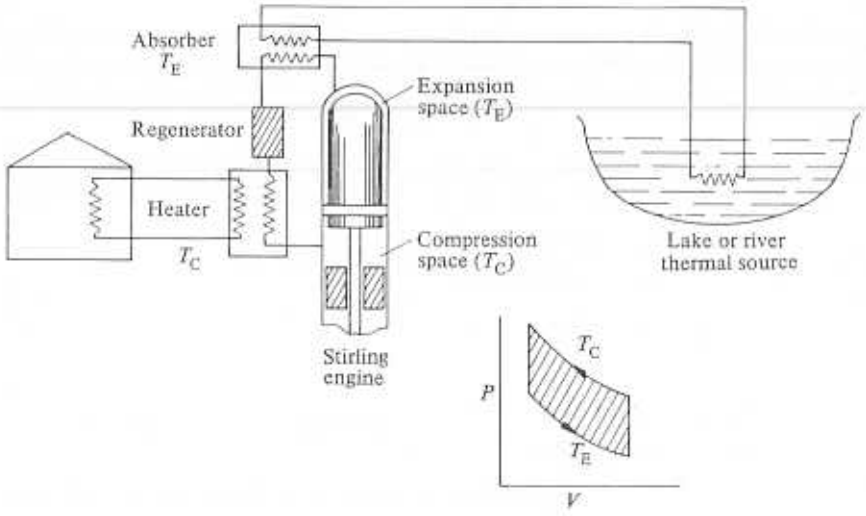


FIG. 19.1. Stirling-engine heat pump.

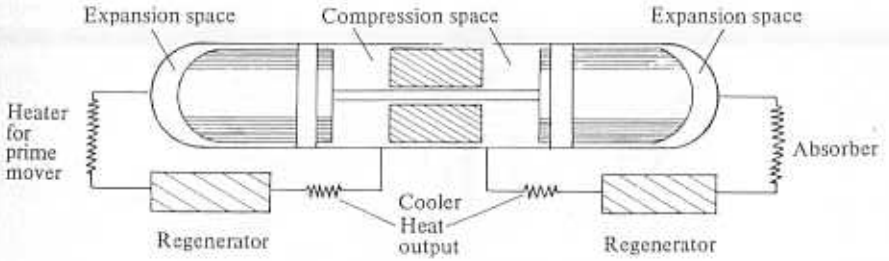


FIG. 19.2. Duplex Stirling-engine arrangement. A Stirling engine acting as a prime mover drives a Stirling engine acting as a cooling engine.

Energy flows in the duplex Stirling heat pump are illustrated in Fig. 19.3. Starting at the left of the diagram, air and fuel combine in the combustion chamber to produce high-temperature combustion products. The products leave the system passing through the air preheater where they heat the incoming air and also through a further heat exchanger incorporated in the building heating system.

Heat from the hot combustion products is transferred in the heater of the prime mover to the working fluid. Some of the heat is converted in the Stirling engine prime mover to work and the remainder is rejected in the cooler to the building heating system.

The work produced by the prime mover is consumed to drive the heat pump. The heat pump operates as described above absorbing heat from a thermal source at ambient temperatures and rejecting heat to the building heating system at a higher temperature.

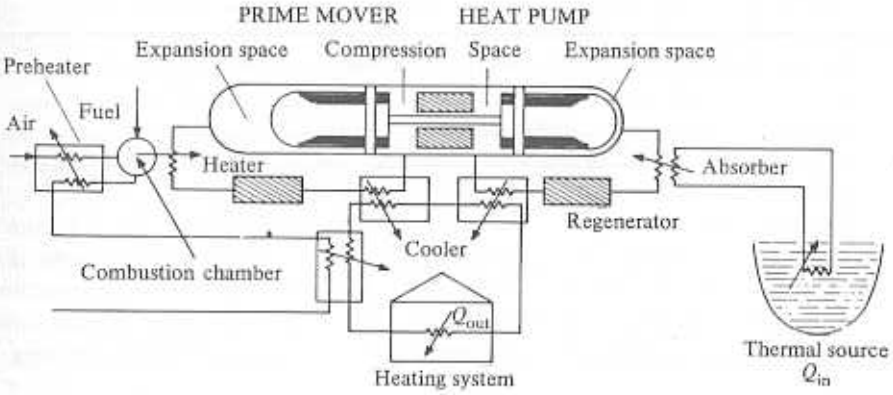


FIG. 19.3. Energy flows in the Duplex Stirling engine.

The performance of a heat pump is defined as:

$$\text{Coefficient of performance} = \frac{\text{Heat rejected to heater}}{\text{Work supplied}} = \frac{Q_R}{W_H} \tag{19.1}$$

Similarly the thermal efficiency of the prime mover is defined as:

$$\text{Thermal efficiency } \eta = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{W_P}{Q_S} \tag{19.2}$$

In the prime mover, the work done is the difference between the heat supplied to the engine and the heat rejected to the cooler, i.e.

$$W_P = Q_R - Q_C \tag{19.3}$$

Similarly in the heat pump, the work supplied is equal to the difference between the heat absorbed and the heat rejected to the heating system, i.e.

$$W_H = Q_R - Q_A \tag{19.4}$$

Now in the duplex Stirling arrangement the work done by the prime mover equals the work absorbed by the heat pump.

Therefore, since $W_P = W_H = W$, say

$$Q_S - Q_C = W = Q_R - Q_A \tag{19.5}$$

The total heat rejected to the heating system is

$$Q_C + Q_R \tag{19.6}$$

or, from (19.5), $= (Q_S - W) + (W + Q_A)$

$$= Q_S + Q_A \tag{19.7}$$

If the same energy *input*, Q_S, were consumed in electric resistance heating or in a gas or oil furnace the heat available for heating would simply be Q_S. Therefore, use of the heat pump has increased the heat available for heating to $Q_S + Q_A$. Depending on the conditions, the increase in *effective* heating available may range from 40 to 100 per cent.

Of course the increase in effective heating has been gained at the expense of a considerable involvement in machinery. The capital cost of a heat pump system would inevitably be much greater than a simple furnace. As fuel costs increase, the economy of heat pumps is enhanced.

Fortunately the same machinery used in the above heat pump for heating may also be deployed as the refrigerator cooling unit in an air conditioning system for summer use.

The machinery works in exactly the same way as described above. The only difference, as illustrated in Fig. 19.4, is that now the expansion-space heat exchanger is connected to the conditioned-space heat exchanger so that heat is *absorbed from* the building at a temperature less than the ambient value. Similarly the compression-space heat exchanger is linked with the heat exchanger of the thermal source (now acting as a thermal dump). The lake which provides heat for space heating in winter may also be used as the thermal reservoir for cooling applications in the summer.

This reversal of the role of the heat exchanger between winter and summer operation may be effected by simply reversing the direction of rotation of the Stirling engine. This would be a simple matter indeed, if the Stirling engine were driven by an electric motor. In the Stirling–Stirling arrangement a reversal in the direction of rotation might be more difficult to achieve.

The preferred method for reversing the role of the heat exchangers would likely be some form of flow switching, probably of the intermediate fluids connecting the conditioned space and the thermal source/dump reservoir and the engine.

An early attempt to develop a duplex Stirling engine cooling unit was made by Walker (1968a) under contract to the British Ministry of

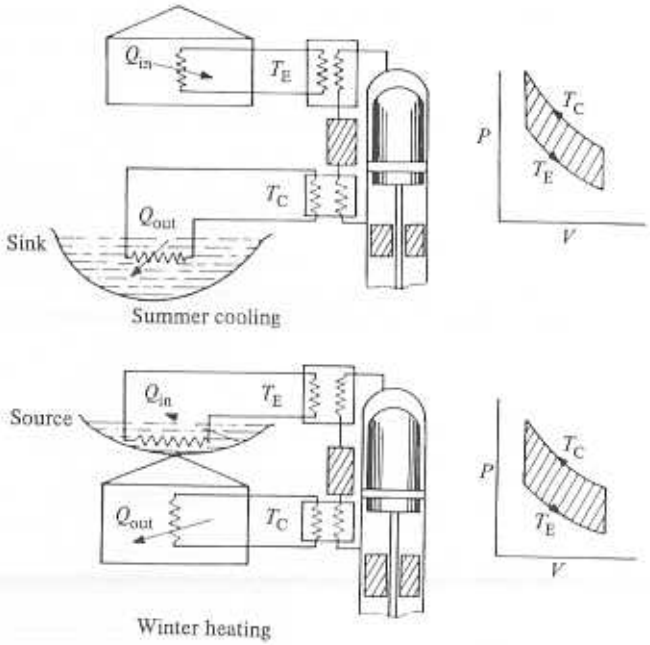


FIG. 19.4. Stirling-engine heat pump for summer cooling or winter heating.

Technology. Martini (1975a) has well outlined some of the principal considerations in a duplex Stirling scheme. Similarly Benson (1978) has outlined some developments in progress at ERG Inc., San Francisco. Beale, in Chapter 11, refers to a prototype unit operated at Sunpower Inc., Athens, Ohio. Other developments are known to be in progress and commercial application appears likely in the future.

One of the principal advantages of the duplex Stirling arrangement is the use of a common fluid for the combined Stirling engines. This can be most attractive in free-piston engines where the use of a common fluid greatly relaxes the sealing arrangements.

Furthermore, in free-piston engines the use of gas bearings employing the working fluid as the bearing media eliminates the problem of oil or grease lubricants contaminating the working space. Finally, free-piston engines are or can be made self-starting. This is a most attractive feature for solar-powered air-conditioning units operating intermittently. They produce a cooling effect only when the sun is shining enough to induce the engine to start, a rare and happy combination of cause and effect.

STIRLING/RANKINE-CYCLE HEAT PUMPS

Most development work presently in progress is directed to the use of Stirling engines as prime movers driving Rankine-cycle heat pumps. This is despite the attractions of a common working fluid in the duplex Stirling heat pump outlined above. The reason for this is probably related to the historical factors which have led to the separate development of Stirling-cycle prime movers and Rankine-cycle heat pump and refrigeration systems, usually driven by electric motors. Furthermore, there is a profound lack of awareness in the engineering community about the capability of Stirling engines for cooling purposes. All the established applications are at the cryogenic temperature level. The prospects for Stirling-engine refrigerating engines at relatively high temperature levels are rarely addressed in the literature.

The Stirling/Rankine-cycle heat pump is illustrated diagramatically in Fig. 19.5. The system consists of a Beale-type single-cylinder free-piston Stirling engine. The work developed in the engine is used to drive the compressor of a Rankine-cycle refrigeration circuit. Typically this would use a Freon working fluid and consist basically of four elements, a compressor increasing the pressure from p_l to p_h, a J-T (Joule-Thompson) valve where the fluid expands from p_h to p_l and two heat exchangers where the fluid changes phase (a) evaporating from liquid to vapour in absorbing heat at low pressure and low temperature and (b) condensing gas to liquid at high temperature and high pressure.

The system illustrated in Fig. 19.5 is the system used in the Beale Sunpower engine where an inertia compressor is used in the Freon circuit.

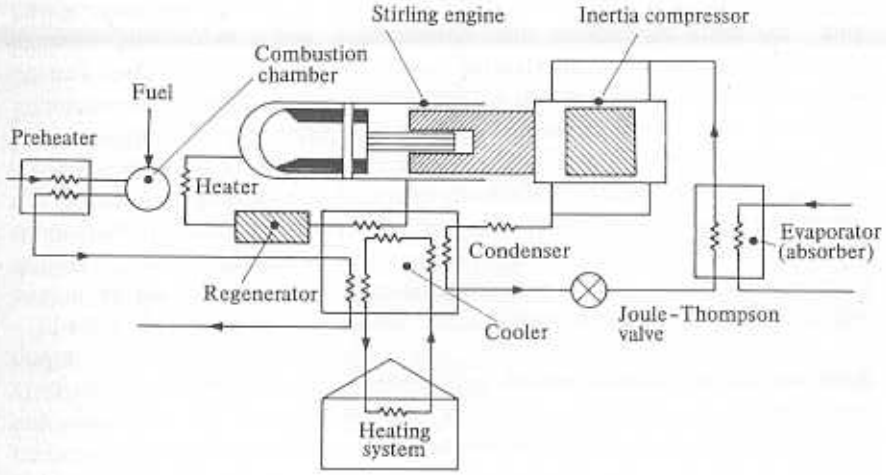


FIG. 19.5. Stirling-engine prime mover driving a Rankine-cycle heat pump. The Stirling engine shown is a Beale free-piston engine driving the inertia pump of the Rankine cycle.

The inertia compressor consists of a heavy mass, which remains substantially fixed in space, within an enclosure rigidly connected to the engine piston and which oscillates along the axis of the fixed mass so that fluid is pumped alternately from the left and right hand sides of the pump.

The cooler of the Stirling engine and the condenser of the Rankine circuit both transfer heat to the heating system of the building. Heat is absorbed from a thermal source at ambient temperature in the evaporator of the Rankine circuit.

Energy to drive the system is provided by burning fuel with air in a combustion chamber and heat is transferred in the heater to the working fluid of the Stirling engine. Heat rejected in the cooler of the engine can be included in the output of the space heater. Further output may be obtained by another heat exchanger downstream of the air preheater on the exhaust side.

In all this utilization of 'waste' it must be recalled that it is advantageous from the power and efficiency aspect to operated a Stirling engine at a *low* cooler temperature, probably below the temperature at which the space heating might be required. Furthermore successive cooling of the exhaust stream can be unproductive as the temperature decreases towards ambient. Overcooling can result in the condensation of combustion products that may be corrosive or lead to the build-up of surface deposits that interfere with the flow.

Substantial work has been carried out by Sunpower Inc. on the development of gas-fired free-piston Stirling engines of the Beale type used for driving the inertia Freon compressors of Rankine-cycle refrigeration

and heat pump systems. This work was sponsored by the American Gas Association and was variously reported by Beale *et al* (1975). The work centred on the development of a Beale single-cylinder free-piston engine of 3 kW capacity in combination with an inertia pump supplied by the Eaton Corp.

After proceeding through successive generations of prototype machines this development was then redirected by the sponsor, A.G.A., to the General Electric Space Division for pre-production prototype development. Recent work on the G.E. Stirling-engine gas-fired heat pump has been reviewed by Auxer (1977), Richards and Auxer (1978), Marusak and Chiu (1978).

Fortunately the years of experience accumulated at Sunpower were not dissipated by the surprising change in direction by A.G.A. A major U.S. oil company recognizing the inherent potential of the thermally-activated heat pump rapidly assumed financial responsibility for the programme of prototype development. Public disclosure of this program has not yet been made.

It is understood that a major part (one-third) of the total Philips effort on Stirling engines at Eindhoven was, in 1978, directed to the development of small Stirling engines for inclusion as the prime movers in thermally-activated heat pump systems. It is further understood the engine is of the two-cylinder two-piston Rider form of about 10 horsepower for use in domestic heat pump systems. This hearsay should be confirmed by reference to the Philips Naturdig Laboratorium or by reference to publicity or informational material that may be published subsequent to this work.

Benson (1978), in a general review of work on advanced heat pumps at Energy Research and Generation Inc. (ERG) Oakland, Ca., discussed a number of concepts including duplex Stirling and Stirling/Rankine systems. The degree to which these concepts have been reduced to practice was not made clear. It is not known how far ERG has proceeded with hardware development.

Interesting and extensive theoretical studies, reported on a comparative basis, of different heat pump systems including some Stirling engines have been summarized by Wurm and Staats (1977). This work was carried out at the Institute of Gas Technology for the American Gas Association.

STATIONARY POWER GENERATION

Stationary power generation is a term embracing the widest possibilities. Engines that may be stationary power plants in one application may be adapted as the auxiliary equipment or even the main propulsion machinery in space, aeronautic, marine, railway, automotive, or recreational vehicle applications.

Stirling engines have attractive characteristics for stationary power applications. They have a wide multifuel capability, operate without noise, have excellent part-load performance, and respond fast to sudden changes in load. They have the potential to operate for very long periods with minimal maintenance and low lubricant-oil consumption.

In the closing years of the twentieth century, the multifuel capability of the Stirling engine will most likely become important. The engine can operate on any source of heat, and so as oil and gas become increasingly valuable, more and more use will be made in power generation of solid fuel like coal, industrial wastes such as wood bark, forestry trimmings, agricultural wastes, and municipal wastes. Anything that is combustible can be consumed in high-efficiency fluidized beds, or other advanced combustors to produce the hot gases for heating Stirling-engine systems.

Similarly the combination of a free-piston Stirling-engine/linear alternator with an absorber and solar concentrator may be used as illustrated in Fig. 19.6 to produce electricity from a solar input (Pilar 1978).

A major solar energy conversion demonstration is in progress for the U.S. Department of Energy in a program managed by the California Institute of Technology, Jet Propulsion Laboratory. This program includes the construction and evaluation in California of twenty-three 50 kW Stirling engines used in conjunction with a similar number of solar collectors each 16 m (52 ft) in diameter. The engines to be used for this program are United Stirling P75, Siemens-type, four-cylinder machines derated to 50 kW. The program contractor is Ford Aeroneutronics. No doubt technical papers on this program will be appearing in the technical press in due course. The use of 23 engines provides a good probability for 20 systems to be operating continuously at any one time to provide a $20 \times 50 = 1$ mW capacity. It is said that electrical storage is to be included for power supply during the dark hours.

In solar systems with a tracking concentrator it is possible to locate the engine at the focus of the concentrator but this is not desirable as it requires the concentrator and tracking mechanism to be capable of supporting the relatively heavy engine and generator system. The preferred alternative therefore is to locate the engine remote from the focus of the concentrator in a fixed, secure, location and to indirectly heat the engine with a liquid metal heat pipe or transfer loop. This has some important advantageous characteristics for the engine (see Chapter 16 on Stirling engines for automotive use) principally to eliminate hot spots, increase the upper cycle temperature and hence the efficiency, and, finally, decrease the internal engine dead space with consequent gain in power output and efficiency.

In addition to all these advantages the introduction of indirect heating

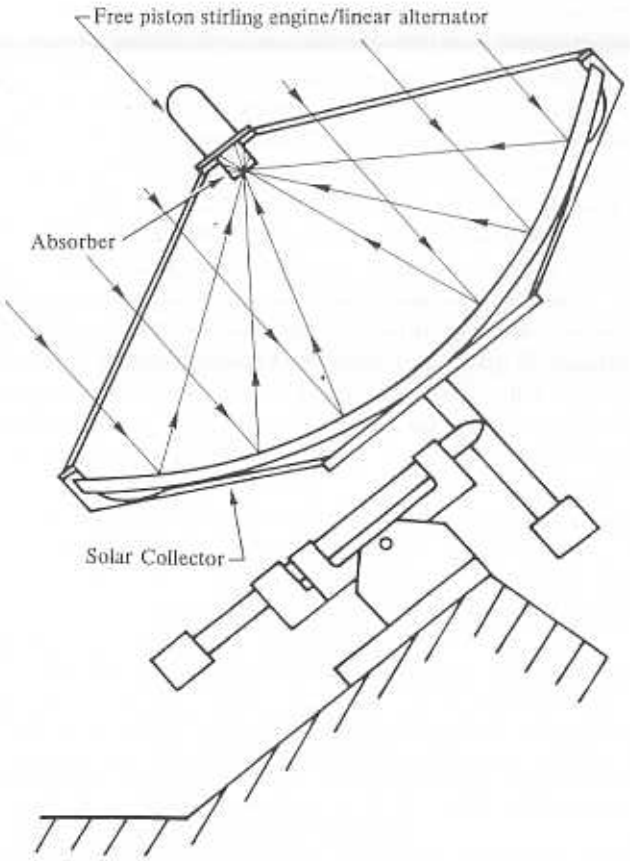


FIG. 19.6. Stirling engine with solar energy input.

with solar power systems offers a number of possibilities for power supply during the dark hours. A thermal storage system (thermal battery) of lithium fluoride may be incorporated in the circuit to be recharged by the thermal energy of concentrated sunlight during the sunlit hours. This would permit 24 hour (or as required) operation of the engine/generator system. Another possibility is that with indirect heating (and possibly thermal storage) a supplementary heat source may be added at very low cost to keep the engine/generator system going whether the sun is shining or not. Thus a fluidized bed combustor providing energy to a liquid sodium heat pipe may be the only additional equipment necessary to keep a large solar-concentrator/Stirling-generator active during the dark hours. The combustor may consume garbage, used tires, coal, or wood waste. As an accessory to an existing or forthcoming solar plant the cost would be minimal.

Fluidized bed combustion systems offer the possibility of intense thermal flux, in relatively moderate and well-controlled temperature combustion systems able to operate with a variety of solid fuels over a wide range of load. Their use in conjunction with Stirling engines has been studied and reported by Asselman (1976) and by Dunn and Rice (1975). They may be used to heat the working fluid in the heater tubes directly but the preferred use is with indirect heating using a sodium heat pipe or sodium/potassium eutectic transfer loop.

In 1978 the U.S. Department of Energy initiated the External Combustion Engine project under the management of the University of Chicago Argonne National Laboratory in Illinois. This program will explore the feasibility of Stirling-engine power generators of 500 to 2000 kW for modular application in situations where appreciable combustible wastes do exist but in quantities which do not justify the installation of a full-blown, base-load, Rankine-cycle steam-turbine plant. There are thousands of local municipalities which fall into this category. Publication of the Program Opportunity Annoucement attracted a response by over sixty U.S. corporations. The Request for Proposal and subsequent concept design and prototype stages are expected to initiate innovative engine arrangements previously unexplored, but which may prove particularly appropriate for large Stirling engines.

Earlier, Hoagland and Percival (1978) have reported on a very comprehensive technology evaluation of Stirling engines for stationary power generation in the 375 to 1500 kW (500 to 2000 hp) range. They concluded that the Stirling engine was well suited, principally on account of the multifuel capability, for commercial development as an alternative to diesel, gas turbine, and steam plants. On a considerably larger scale Stirling engines of megawatt capacity have been envisaged by staff of the Atomic Energy of Canada. Such engines would be associated with advanced nuclear reactors.†

Small stationary power generators, say up to 10 kW capacity are of intense interest on a broad front for a variety of military and civil applications. In many cases quiet operation is the attractive feature, particularly in small power generators for the Army or as propulsion motors for rubber assault boats or clandestine operations. Quiet low-power generators are attractive for recreational vehicles, camp-grounds, and similar situations.

F.F.V., the Swedish defence-related company, half owner with Kockums of United Stirling (see Chapter 15), has carried out a decade of

† Private discussions with J. Bradley, Chalk River Laboratories, Canada.

independent development of a 10 kW generator. This is near the point of introduction to commercial application (Johansson 1978) and may well be the first commercial multiple-horsepower Stirling engine outside of existing cryogenic cooling systems. It is understood the engine has a moderate efficiency (20 per cent) to allow for the use of non-exotic materials for the hot parts.

In yet smaller sizes (up to 200 watts) there appears to be a substantial market for small thermally-activated power generators for navigation signal generators, particularly light-ships and buoys.

Present systems on buoys utilize acetylene gas supplied in bottles as the energy source for the periodic emission of light. The timing and flow control device is activated by the acetylene gas pressure and extremely reliable systems have been developed. Unfortunately there is little that can be done to optically magnify or concentrate the emission of an acetylene flame so that visibility is limited.

The size of large marine tankers is now so great and their inertia so high that the limit of visibility of an acetylene flame is in fact less than the distance travelled by the ship between the start of a change of course and a response by the vessel. Radar systems are of course used in the main but these have proved fallible and an alternative supplementary warning system of unfailing reliability is required.

The preferred system would appear to be periodic flashes from an electric Xenon lamp. The flash can be tailored to provide intense energy emission for an extremely short interval and can be further concentrated optically to effect an order-of-magnitude improvement in visibility. Electric batteries are presently used and a variety of turbo-power generators using wave action are being evaluated. An alternative would be a small Stirling-engine power generator operating on diesel oil, liquid petroleum gas, or radioisotope. Such a unit capable of operating six months (diesel oil) or two years (radioisotope) unattended and with guaranteed reliability would most likely find a substantial market in the international lighthouse and buoy market of the maritime regulatory authorities.

One promising development in this field at the University of Calgary, for a generator operating with cobalt 60 radioisotope, was abandoned when the isotope power program of the Atomic Energy of Canada Ltd. was cancelled in the Canadian recession of 1971.

A similar program carried out at the British Atomic Energy Authority Harwell Research Centre resulted in the successful development of prototype generators using strontium 90 radioisotope or liquid petroleum gas as the energy source (Cooke-Yarborough and Yeats 1975b). This unit has passed the laboratory prototype stage and is presently being evaluated in a navigation buoy by the Irish Lighthouse Authority. It is projected for production by the A.G.A. Ltd., a British affiliate of the celebrated Swedish industrial and maritime manufacturer.

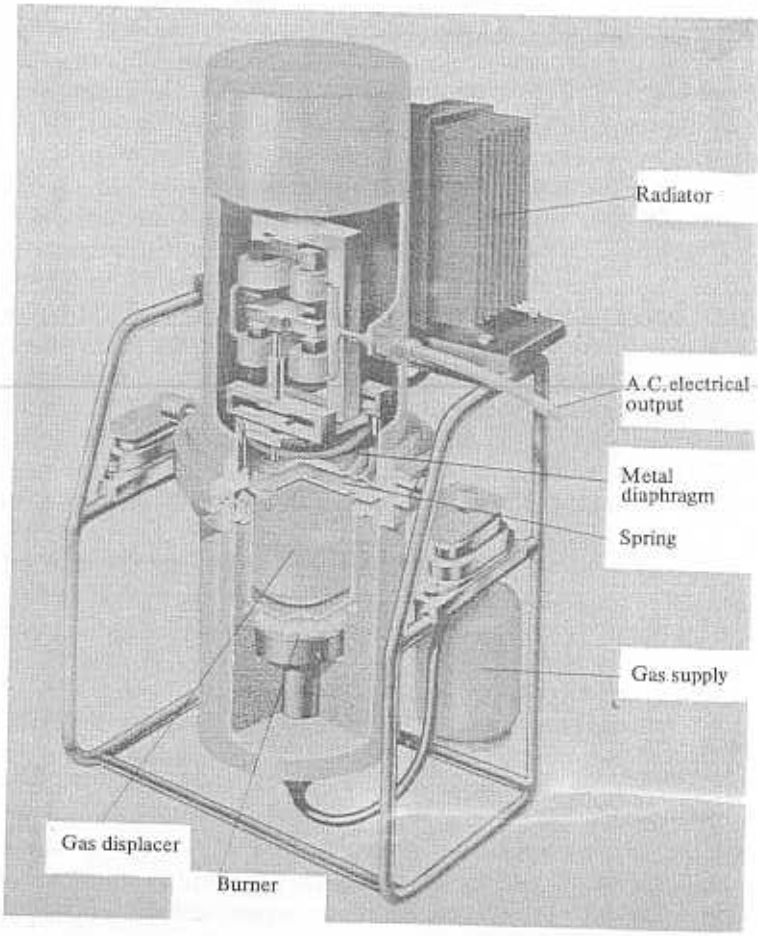


FIG. 19.7. Harwell free-piston Stirling engine power generator.

The Harwell engine, shown in Fig. 19.7, is a large, heavy, free-piston Stirling engine utilizing a flexing diaphragm coupled to the generator unit. It has demonstrated long life with excellent reliability.

Another interesting development at Harwell is the Fluidyne engine (West 1971), a free-piston Stirling engine with liquid pistons used for water pumping. A water column acting as the piston resonates at natural frequency to operate a Stirling engine system contained in the cylinder above the liquid piston. Arrangements can be made for the resonant water column acting as the piston to pump water. The engine can be heated by combustion, by electric heating or by solar heat. Elrod (1974) has given a good description of the Fluidyne engine and instructions for construction of a prototype. Rallis *et al* (1977) have described their

experience with experimental Fluidyne engines at the University of Witwatersrand.

SYNCHRONOUS OPERATION

Many stationary power engines are required (a) to operate at constant speed and (b) to be relatively unaffected by sudden changes in the load. Utility power companies are required by the regulatory agencies to provide electric power at a near constant frequency with close limitations in the permissible frequency variation.

The control devices for Stirling engine stationary power systems will include (a) a fuel/air regulator to increase or decrease the fuel supply so as to maintain a constant maximum temperature in the combustion region, (b) an engine torque control so as to vary the engine power and hence maintain constant speed during variation in the load.

The various torque-control systems available are described in Chapter 10 and include:(a) mean-pressure control, (b) pressure-amplitude control, (c) phase-angle control, and (d) stroke control.

Martini (1978)† has described another type of control system well-suited to the constant-speed regulation required for stationary power systems with close frequency stability. The Martini control is illustrated in Fig. 19.8. The engine is a piston/displacer system driving an electric alternator. The displacer is driven by a synchronous motor consuming a fraction of the output of the alternator and operating at a speed corresponding exactly to the speed of the alternator. Motion of the displacer induces flow of the working fluid between the hot space and cold space so the pressure varies cyclically at the same frequency as the motion of the piston-displacer. The engine pressure variations act upon the piston causing it to reciprocate in the cylinder and so driving the alternator to produce power, part of which is consumed to drive the displacer. Thus the *frequency* of reciprocation of the piston in the cylinder is determined by the frequency of reciprocation of the displacer. This in turn is determined by the speed of the synchronous drive motor which itself is determined by the speed of the alternator driven by the piston. The system is therefore completely self-stabilizing with regard to speed control. The *phasing* of the piston and displacer motion will automatically adjust to assume the phasing required to produce the *exact* amount of torque required to generate the load demanded from the engine.

Starting would, of course, present a special problem. One way would be to start the displacer motor using battery power passing through an inverter set to produce alternating current at the frequency eventually required. Once up to speed, the displacer drive would automatically be transferred to the alternator output.

† Private communication

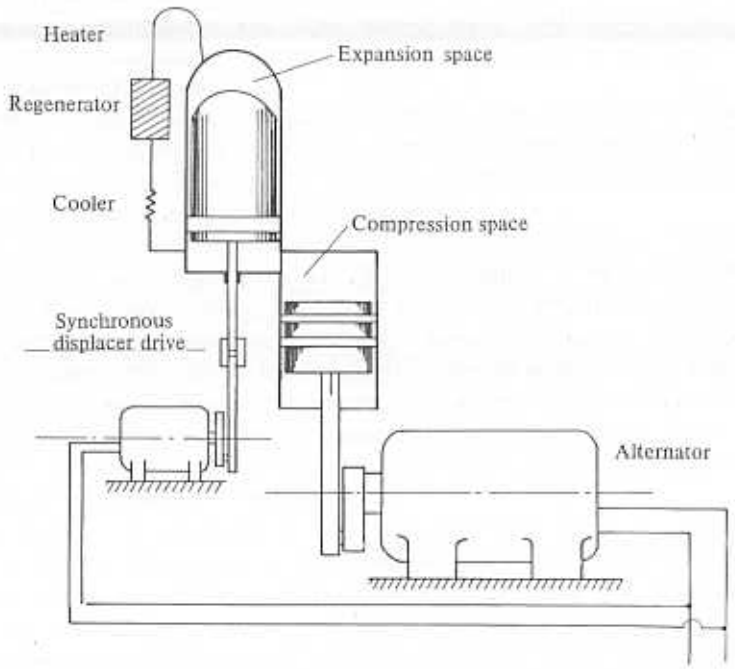


FIG. 19.8. Constant-speed control system for piston/displacer Stirling engine (after Martini, private communication 1978).

In most instances the displacer drive would consume a small fraction (maybe one per cent) of the total electric power developed. In other cases, the alternator would be simply sufficient in size to provide power for the displacer drive and for battery charging. The principal output of the engine would be used as mechanical work for driving a gas compressor, fluid pump or some other mechanical system.

TOTAL-ENERGY SYSTEMS

A total-energy or co-generation system is an ensemble of machinery utilizing a single or varied energy source to provide a range of utilities in a building or plant. Typically the total energy plant in the basement of an office building is supplied with natural gas fuel. Combustion of the gas produces heat, part of which is used in an engine to drive electric generators providing high frequency (400 cycle per second) power for lighting and low frequency (60 cycles per second) power for other purposes. The waste heat may be utilized to generate high-pressure steam for heating, for process purposes or, in an absorption chiller, to produce

chilled water. Similarly low pressure steam or hot water may be produced for laundry heating and bathrooms.

Stirling engines appear to be well suited for use in total energy systems as prime movers, heat pumps, or refrigerating engines. The particular characteristics of the Stirling engine which are advantageous in total energy applications are, primarily, the multifuel capability, quiet operation, minimal exhaust emissions, excellent part-load efficiency, and good starting, control, and torque characteristics.

Walker (1967) appears to have been the first to consider Stirling engines for total-energy systems in a survey carried out for the Institute of Gas Technology. Later Jaspers and du Pre (1973) assessed the prospects of the Stirling engine in total energy systems to be highly favourable.

Lehrfeld (1977a) analysed, very comprehensively, the use of Philips Stirling engines in total energy systems in a variety of applications, commercial and hospital buildings, residential apartment buildings, and offices. This study was summarized (Lehrfeld (1977b) and further referred to in a survey of the Stirling engine for co-generation applications. Co-generation is a term used interchangeably with total energy to describe an on-site electric power plant fulfilling electric power demands while utilizing waste heat from the prime mover to supply heating and/or cooling requirements. Marciniak *et al* (1978) of the Argonne National Laboratory carried out an assessment of potential applications of Stirling engines in total and integrated energy systems as part of the U.S. Department of Energy Total Energy Technology Alternatives Studies (TETAS). Simultaneously workers at NASA Lewis Research Centre were engaged in similar studies for industrial-plant applications of Stirling-engine total-energy systems as part of the U.S. Department of Energy Co-generation Energy Technology Alternatives Studies (CETAS).

Somewhat earlier Gadsby (1977) assessed the market potential for Stirling engines in the 750 kW (1000 hp) range in relation to industrial diesel and gas-turbine engines, in an effort to define limits and targets for U.S. government funding priorities.

20 MODEL STIRLING ENGINES

This chapter on model Stirling engines was contributed by Mr. Andrew Ross, of Columbus, Ohio. Mr. Ross is an attorney at law and an avid collector of historical Stirling engines and related memorabilia. He is a machinist par excellence and has an intuitive grasp of engineering fundamentals permitting the reduction to practice of his creative ingenuity. G.W.

INTRODUCTION

The increasing interest among professional engineers and scientists in Stirling-cycle machines has created a corresponding interest among model-building engineers. Such engineers are amateurs in the best sense of the word; they design and make their own engines, clocks, and other mechanical devices in home workshops for the sheer enjoyment of it. Although their experimental Stirling-engine work is still at an early stage of development, it is nevertheless worthwhile to review what they have accomplished on limited time and modest resources.

The 65 cm^3 (4 in^3) rhombic-drive engine illustrated in Fig. 20.1 was originally 'completed' in 1973 by Ross of Columbus, Ohio, after a year of spare-time effort. On its initial test it did not run. After helpful correspondence with several professional engineers in the field, Ross rebuilt this engine to reduce heat leaks and dead volume. In the revised configuration it immediately ran, although initially it produced a mere 1.5 watts at 750 rpm, at atmospheric pressure.

Various modifications were made in the burner and the regenerator, and vast improvements in performance resulted. The most important of these modifications was the development of a high-temperature annular propane burner. The engine now reliably produces well over 25 watts on air at atmospheric pressure, over 50 watts on air at 0.2 MN/m^2 (30lb per sq in), and over 75 watts on helium at just over 0.2 MN/m^2 (30lb per sq in), as is shown in Fig. 20.2. Although it has been run at internal pressures of up to 0.4 MN/m^2 (60lb per sq in), power tests have not been conducted at these higher pressures.

Typical efficiency figures are 4 per cent net thermal efficiency (power out/fuel in) on helium, 17 per cent internal thermal efficiency (power out/[heat into cooler + power out]) on helium and 14 per cent internal thermal efficiency on air.

Among the regenerator matrices tested were stainless wire, steel wool, stainless wool, stainless woven cable, stainless foil, and quartz wool. The metal wool, cable, and foil generally worked well, but the quartz wool was a disaster; its fibres were blown to and fro throughout the engine, necessitating a thorough clean-up.

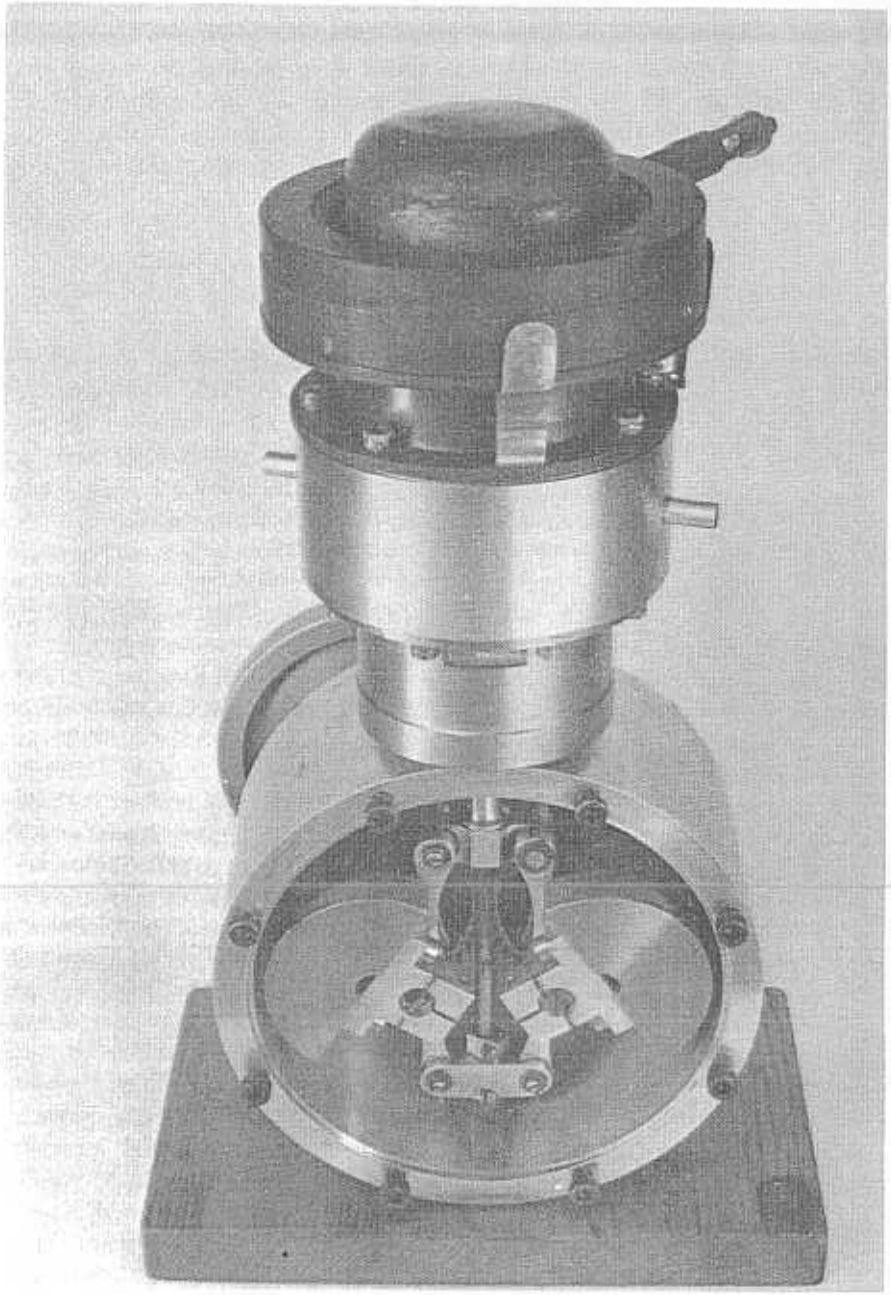


FIG. 20.1. Rhombic-drive model Stirling engine. Piston displacement 65 cm³. By A. Ross (1973).

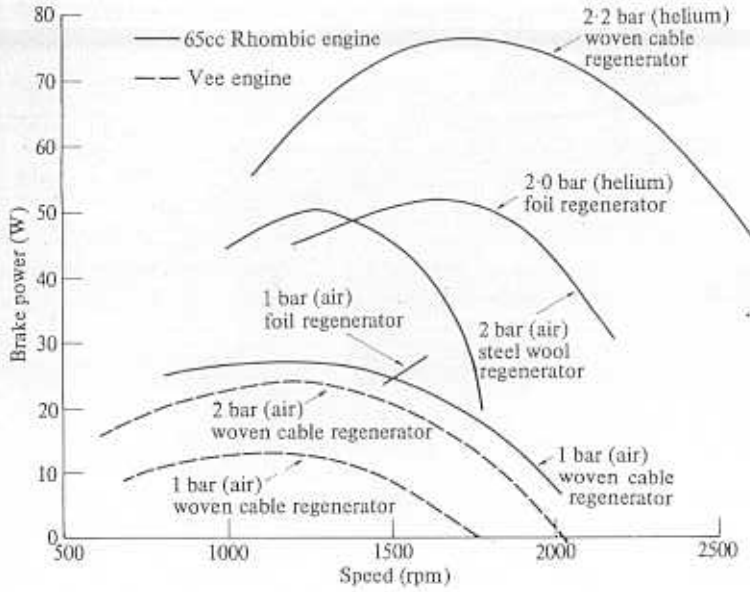


FIG. 20.2. Power/speed characteristics for 65 cm³ Ross rhombic-drive Stirling engine and 38 cm³ Vee Engine.

Nor were all the modifications on this engine successful. For example, a low dead-volume, high surface-area milled aluminum cooler was substituted for the original drilled cast-iron cooler, with essentially no improvement in performance. Eventually, work on this engine was discontinued in the belief that the bearings had about reached their limit.

A piston-displacer vee two cylinder engine of 38 cm³ (2.3 in³) piston displacement was then made by Ross. This design used an identical heater regenerator and cooler as the 65 cm³ (4 in³) rhombic. Needle bearings were used on the connecting rods' lower ends to ease bearing problems with the essentially dry crankcase. After certain pumping loss problems caused by a porous crankcase-casting were eliminated, power tests were satisfactory. As can be seen in Fig. 20.2, the engine produced 24 watts on air at 0.2 MN/m² (30lb per sq in), and up to 31 watts on air at 0.27 MN/m² (40lb per sq in). Subsequent efficiency tests, however, were very disappointing, showing only slightly over 1 per cent net thermal efficiency, with 5 per cent internal efficiency on air and 6 per cent on helium. During these tests, performance was not up to par, and it is quite likely elusive leaks were at least partly to blame for the poor figures.

Fig. 20.3 shows two simple engines, described by Ross (1976) and Ross (1977b). The engine on the left is a small 12 cm³ (0.73 in³) rhombic-drive

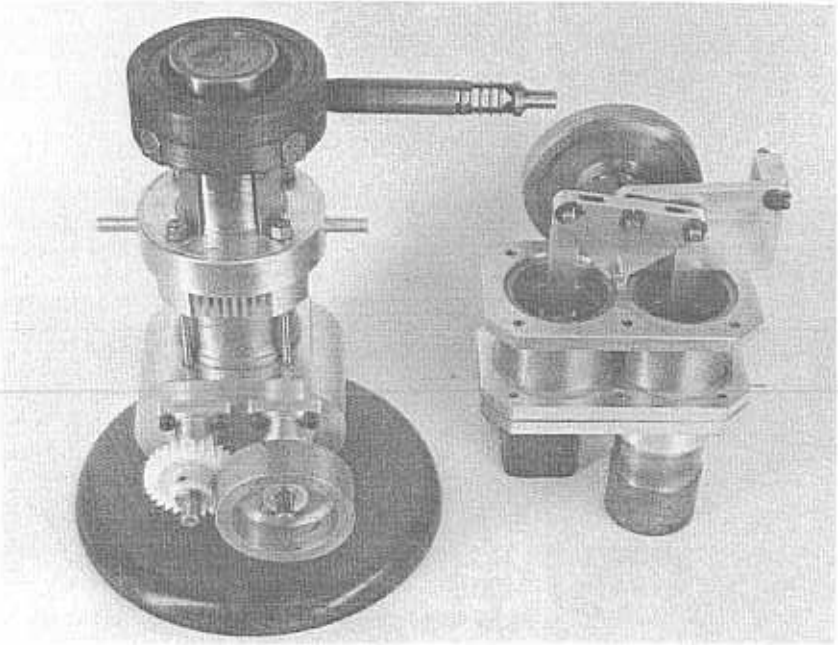


FIG. 20.3. Two Stirling engines by Ross. Engine at left is a 12 cm^3 piston displacement engine with rhombic drive. Engine at right is a two-cylinder Rider-type engine.

machine which, with its annular propane burner, turns a maximum of 3600 rpm on air at atmospheric pressure. A similar engine, built by Thomas, but incorporating his own external piston and a self-pressurizing pump, produced 4 watts at atmospheric pressure, and 11.5 watts at 0.3 MN/m^2 (45lb per sq in) pressure. The engine to the right is of the two-piston Rider type. The unusual crank geometry shown was added later, primarily as a means of reducing piston side forces.

REGENERATIVE DISPLACER ENGINES

Another model engineer interested in the Stirling engine is W. D. Urwick, of Malta, who has undertaken an extensive program of testing the regenerative displacer. In the Urwick design, the conventional displacer is replaced by a series of screen discs mounted on the displacer shaft, which act as a regenerator. Urwick (1975) reported various experiments in which the engine tested performed as well or better with this regenerative displacer than with a conventional displacer, despite the greatly reduced compression ratio accompanying the change. Discoloration of the screen discs suggested that a very even temperature gradient is maintained between the discs. Other experiments showed the effect of greatly increased dead volume on engines with very low compression ratios.

Later Urwick (1977) described a number of different engines incorporating his type of displacer, including one with a swash-plate and several with nutator (wobble-plate) drives. An Urwick engine of 5 cm^3 (0.3 in^3) design is shown in Figs. 20.4 and 20.5. He has resisted the temptation, sometimes strong among model engineers, to pursue every new idea. Rather, he has continued carefully and patiently to explore this regenerative displacer idea, in an effort to realize whatever potential it may have.

Collins (1977) has also developed a regenerative displacer engine. This very interesting 5.5 watt 5 cm^3 (0.3 in^3) engine operates on air at just over 0.6 MN/m^2 (90lb per sq in.), and incorporates a simple but effective shaft seal that permits runs of up to 56 hours without supplemental pressure. As with most model engines, this machine has been tested in several different configurations. Some tests were made with conventional displacers, and others with regenerative displacers.

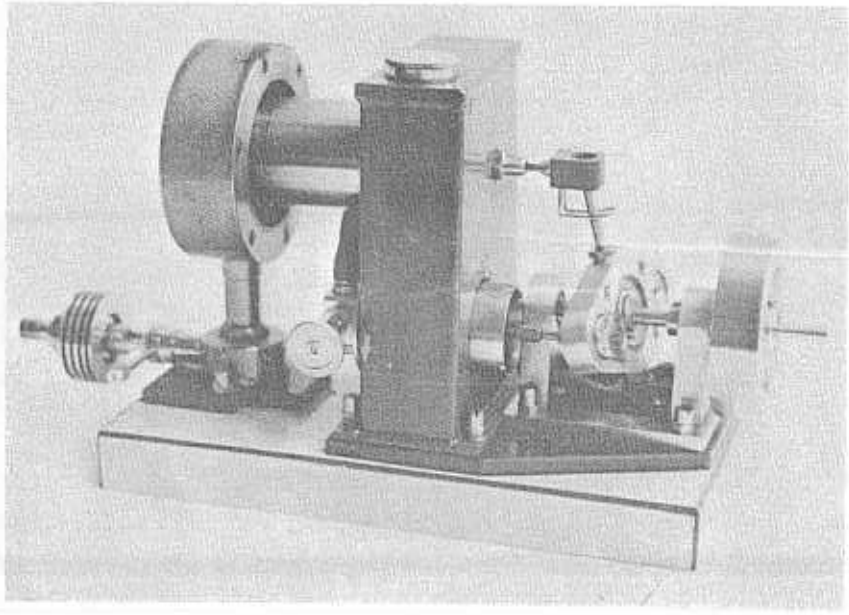


FIG. 20.4. Model Stirling engine with regenerative displacer by W. D. Urwick (1975).

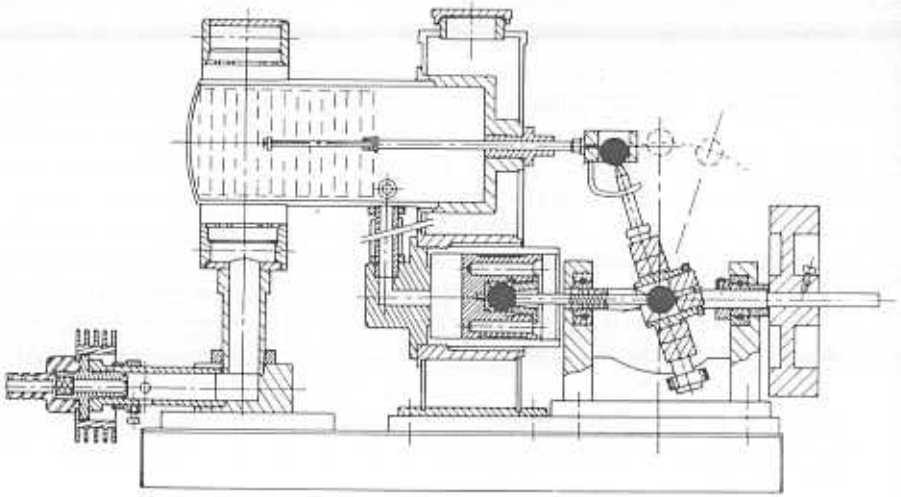


FIG. 20.5. Cross-section of Urwick engine with nutator drive.

HOT-AIR ENGINE COMPETITIONS

The 5 cm^3 (0.3 in^3) engines of Urwick and Collins were built for the first hot-air engine competition, held at the 1977 Model Engineer Exhibition in London. This competition was sponsored by A. N. Clark and promoted by *Model Engineer* magazine. In light of the increasing interest in model Stirling engines, it was correctly believed that the time was right for an international competition (Chaddock 1976).

Since originality was to be encouraged, the rules were few. The only significant restriction was that power piston displacement be limited to 5 cm^3 (0.3 in^3), and the only aim was to produce as much power with that volume as possible.

Seventeen people entered the competition, and the winner, E. F. Clapham of Bristol, produced an engine beyond the sponsors' wildest expectations. It was pressurized with air at over 6.7 MN/m^2 (1000 lb per sq in), and it produced 39.4 watts at 900 rpm. This machine was designed and built in 600 hours, and it was the first Stirling engine Clapham had ever built.

Clapham's engine, shown in Fig. 20.6, was a co-axial piston-displacer design with a pressurized crankcase. Tubular heat exchangers provided almost 122.6 cm^2 (19 sq in) of external heater surface area, and over 45.2 cm^2 (7 sq in) of cooler area. The crankshaft mechanism was a modified bell-crank type, designed to operate without oil lubrication. The crankcase was run dry to prevent oil contamination and any danger of explosion. A separate crosshead was used, with PTFE pads to absorb side

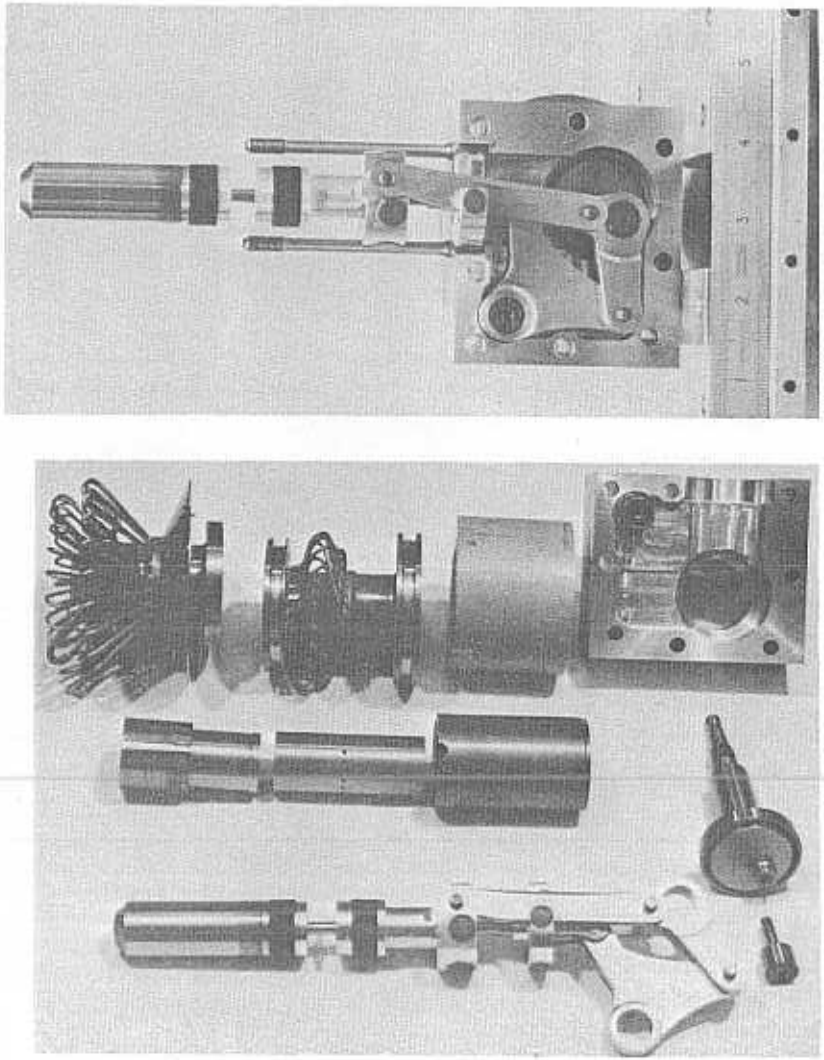


FIG. 20.6. Model Stirling engine by Clapham (1977). This high-pressure high-performance engine won the first hot-air engine competition for engines with 5 cm^3 piston displacement. Output was 95 watts at 2000 revolutions per minute with helium at 80 bar mean pressure.

loads. The piston rings were carbon, backed by 0-rings. A commercial carbon face-seal of the pressure-balanced type was used to seal the crankshaft as it emerged from the crankcase.

Clapham (1977) has described how, after the competition, further modifications were made to this most impressive engine, to increase the power to 55 watts at about 1350 rpm on air at 6.7 MN/m^2 (1000 lb per sq in) and 95 watts at about 2000 rpm on helium at 8 MN/m^2 (1200 lb per sq in).

It is interesting to compare Clapham's design with the 65 cm^3 (4 in^3) rhombic of Ross. Both are high-speed engines and both have produced 55 watts on air, yet Clapham's machine has only 1/13 the volume, but 34 times the pressure, of Ross's machine. The Clapham engine has a much higher dead-volume ratio, as would be expected of an engine of such high output per unit volume. The overall dimensions are comparable, the 5 cm^3 (0.3 in^3) engine standing about 24.1 cm (9.5 in) tall, the 65 cm^3 (4 in^3) engine being 29.2 cm (11.5 in) tall.

The second place engine at the *Model Engineer* competition was a beautiful rhombic-drive design of Dr. James Senft (1977), of Minot, North Dakota. He had previously built a number of small Stirling engines and (Senft 1976) had earlier written a detailed and extensive article for model engineers on Stirling engine theory. Senft is unusual among model engineers in that he also works professionally on Stirling engines. His engine, shown in Fig. 20.7, was designed to be a reliable and practical low-pressure machine suitable for model purposes. It employed a compact version of the rhombic-drive, with the connecting rod centre length only 2.5 times the crankpin radius. A machined teflon cup with a spring expander sealed the piston; a similar seal was used on the one shaft that emerges from the pressurized crankcase. The displacer was of the clearance type, with no regenerator.

This engine made 4.7 watts at 1260 rpm on air at 0.3 MN/m^2 (45lb per sq in) during the competition. Subsequent tests on helium at 0.4 MN/m^2 (60lb per sq in) produced 12.9 watts at 2920 rpm. The highest speed recorded on the engine is 3860 rmp. This was not the 'no load' speed, as the load then was still 8.6 watts.

Other interesting engines in this first competition were described in an article by Chaddock (1977) who, as technical consultant to the *Model Engineer*, conducted the power tests.

The subsequent 1978 competition was encumbered by considerably more complicated rules (*Model Engineer* 1977). To enable testing to be carried out within the exhibition hall, various restrictions on fuel tanks, supply pipes, gas fittings, etc. were specified. External flame guards were required so no external flame would be visible while the engines ran. Some sort of rapid extinguishment device was necessary. Pressure was

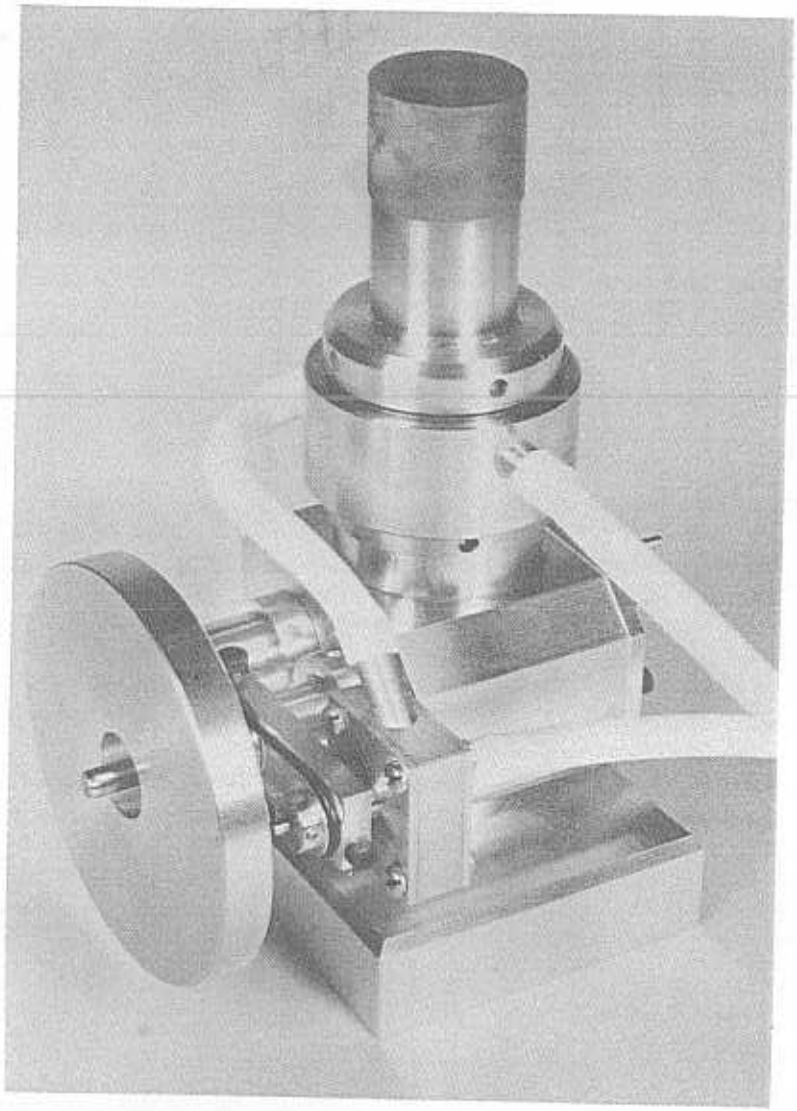


FIG. 20.7. Model Stirling engine by J. Senft (1977). This engine was second in first hot-air engines competition.

limited to 0.76 MN/m^2 (115 lb per sq in). Helium was permitted, and many potential competitors no doubt felt it would be essential to be competitive.

Although these rules were not unreasonable, they dampened enthusiasm to such a degree that only two builders entered the competition. The winner was F. R. Wilkinson, whose double-acting piston-displacer

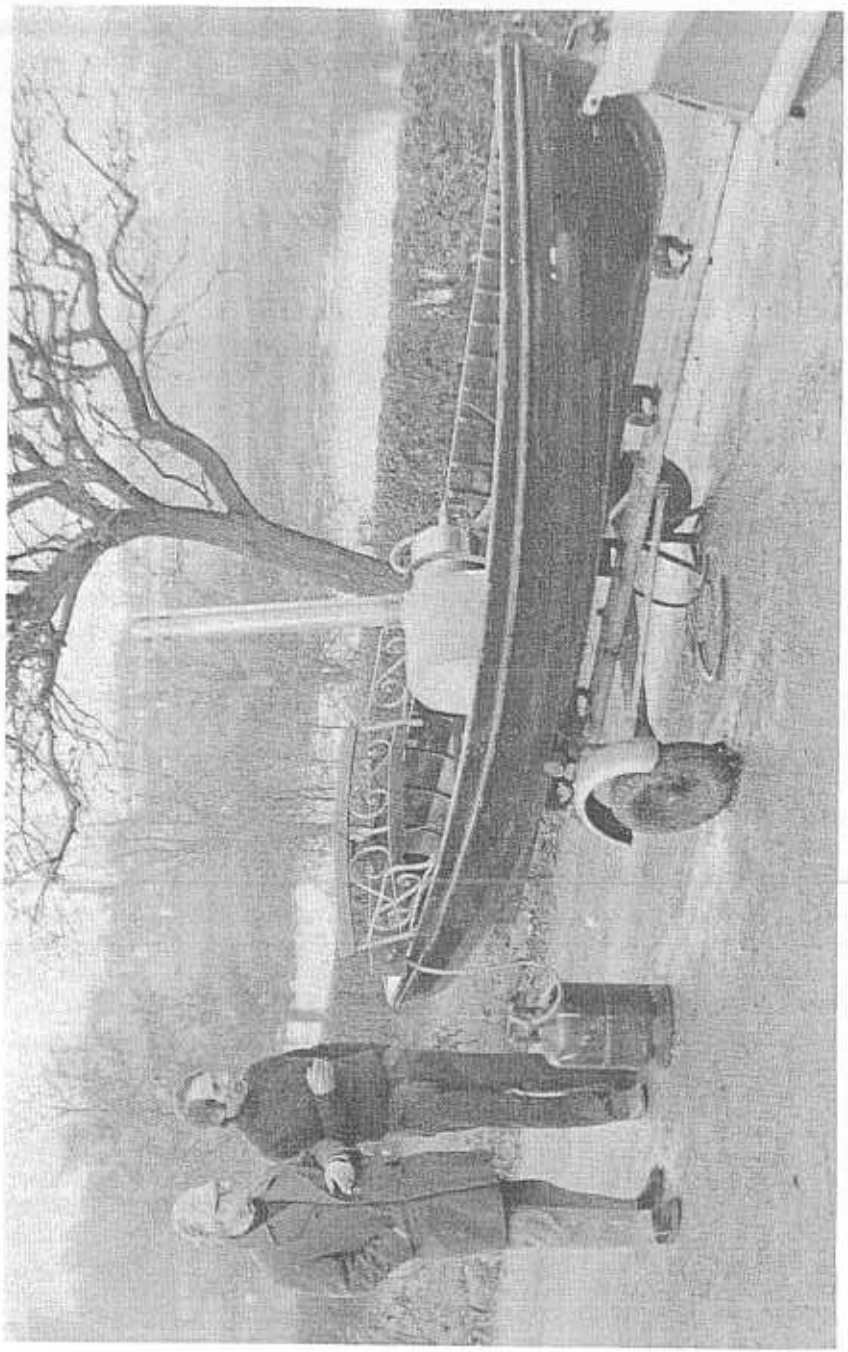


FIG. 20.8. Stirling-engine powered boat by M. Bomford (1975).

engine was similar to Robert Stirling's engine of 1843 for the Dundee foundry. It made 8 watts when pressurized to 0.76 MN/m^2 (115 lb per sq in). One interesting feature was a form of isothermalizer in the hot end which is said to have increased power by 2 watts.

The other contestant was Dr. Brian Thomas, who has built many small Stirling engines of imaginative and unusual design, including several with cam activated displacers. His competition engine was a rhombic-drive with an 'external piston'; that is, a piston with a skirt sealing against the outside of the cylinder. This fine engine had taken third place in the prior year's competition by producing 2.5 watts at 2000 rpm. For this year, Dr. Thomas had developed a self-pressurizing pump, but various difficulties kept power down to 3.7 watts.

FUTURE ACTIVITY: MODELS AND LARGER ENGINES

Regardless of the future of the competition it seems certain that development of the model Stirling will continue. Ross has recently completed a practical die to produce wax patterns for externally and internally finned heater heads; these have been successfully investment-cast in brass, and in Type 316 stainless steel. Senft and others are planning to incorporate regenerators in their small engines. Thomas is making progress on his self-pressurizing machines. and is also experimenting with a 'stuffed' heater, whereby a simple pressed heater can provide considerably extended heat transfer area. New designs continue to appear, such as the Henshall (1977) crank geometry linking two cylinders, each containing a co-axial piston and displacer. It seems at least possible that the history of the model engineers in developing flash steam hydroplane engines, or miniature two stroke internal combusion engines, could be repeated with the small Stirlings.

Many of these builders are also very interested in building practical engines of 100 to 800 W (1/8 to 1 hp) suitable for powering small skiffs or bicycles, perhaps on solid fuel. One builder who has already completed such a project is Morris Bomford (1975), shown with his Stirling-powered boat in Fig. 20.8. His large-volume, slow-turning engine propels the skiff at 1.3 to 1.8 m/s (3 to 4 mph). It is expected there will be much more amateur activity in this area, too, in the near future.

On all these matters, there exists an active written correspondence, not only among the model engineers themselves, but also between them and a number of the independent professionals in the field. Amateurs and professionals alike have in common both the rewards and the frustrations of working in a field where there are more questions than answers.

GLOSSARY

(Terms are defined as they are used in the context of Stirling engines)

Adiabatic compression and expansion: Thermodynamic process of volume, pressure, temperature change, and also adiabatic process change that occurs *without* heat transfer to or from the system.

Beale free-piston Stirling engine: A type of Stirling engine in which the piston and displacer move entirely under the action of fluid forces. There are no connecting mechanisms between the piston and displacer. The load is direct-coupled to the piston.

Clearance space: The minimum volume of the compression and expansion spaces.

Coefficient of performance: The ratio of heat transferred to input work. For a refrigerator the COP = Heat lifted (refrigeration effect)/Work supplied. For a heat pump the COP = Heat rejected/Work supplied (i.e. the inverse of thermal efficiency).

Compound working fluid: The working fluid of a Stirling engine that consists of two or more components and which may exist as a liquid, gas, vapour, or dissociated elements.

Compression space: The part of the working space in a Stirling engine where the working fluid is principally concentrated when the total system volume is decreased, the pressure rises, and heat is rejected to the cooling water. In a prime mover the compression space is cooler than the expansion space. In a refrigerator or heat pump the compression space is warmer than the expansion space.

Constant pressure process: Thermodynamic heating or cooling process that occurs at constant pressure. This may or may not be regenerative.

Constant temperature process: Thermodynamic heating or cooling process that occurs at constant temperature.

Constant volume process: Thermodynamic heating or cooling process that occurs at constant volume. This may or may not be regenerative.

Cooler: The heat exchanger provided to facilitate the transfer of thermal energy *from* the working fluid to the cooling medium, water, air, or some other fluid.

Crank drive: One form of kinematic drive consisting of a crank and connecting rod used to convert reciprocating to rotary motion and to convey power between pistons and drive shaft.

Cryogenerator: A cooling engine capable of achieving refrigeration at cryogenic temperatures (less than 100 K or 180 °R).

Dead-volume ratio: That part of the total working space not included in the variable volumes of the expansion and compression spaces, expressed in terms of the variable volume of the expansion space.

Direct heating: A system in which the hot products of combustion pass directly over the heater tubes in which the working fluid flows so that heat is transferred directly from the combustion products to the heater tube walls and hence to the working fluid.

Discontinuous piston motion: The non-sinusoidal motion of the piston and displacers required to achieve the necessary volume variations of the idealized thermodynamic cycles.

Displacer: A lightweight structural reciprocating element in a Stirling engine characterized by a large temperature difference but a negligible pressure difference between the upper and lower transverse faces.

Double-acting engines: A family of Stirling engines having a single reciprocating element per thermodynamic system. There is a minimum number of two cylinders but no maximum number.

Duplex Stirling engine: Two Stirling engines arranged so that one operating as a prime mover receives heat at a high temperature and produces work to drive the second Stirling engine acting as a cooling engine refrigerator or heat pump.

Emission products: The constituents of the exhaust products of an engine. With fossil fuel combustion these are principally water vapour, unburned hydrocarbon, carbon monoxide, nitrogen, and oxides of nitrogen.

Ericsson cycle: An idealized thermodynamic cycle consisting of isothermal compression and expansion processes at different temperatures bounded by constant pressure regenerative processes.

Exhaust-gas heat exchanger: See *Regenerative cycle.*

Exhaust-gas recirculation: A system whereby a sizeable fraction of the exhaust gas is circulated back through the combustion system. Used to minimize the quantity of oxides of nitrogen in the exhaust emissions.

Expansion space: The variable volume of the working space in a Stirling engine where the working fluid is principally concentrated when the total system volume is increased, the pressure falls and heat is absorbed. In a prime mover, the expansion space is hotter than the compression space. In a refrigerator or heat pump the expansion space is cooler than the compression space.

Finkelstein adiabatic cycle: An idealized thermodynamic cycle for Stirling engines with no heat transfer in the compression and expansion spaces and infinite rates of heat transfer in the heat exchangers.

Free-displacer engines: A form of Ericsson regenerative engine (Bush type) where the displacer moves under the action of fluid forces. Used principally as a pressure generator or pump.

Freezer: The heat exchanger provided in a refrigerator or heat pump to

facilitate the transfer of heat to the working fluid from an external low temperature source.

Harmonic piston motion: The near sinusoidal motion of the pistons and displacers used in practical Stirling engines.

Heat pipe: A device used in an indirect heating system in which an intermediate fluid is used to transfer heat from an external energy source to the working fluid. Usually the intermediate fluid (a liquid metal, usually sodium) is evaporated at the thermal inlet and condenses at the thermal outlet. Large rates of heat transfer can be effected with minimal temperature differences. Furthermore, large differences in the rates of heat transfer can be achieved. This provides the possibility for large areas for heat transfer from the combustion products and small areas for heat transfer to heat tubes. 'Hot spots' on the heater tubes are avoided and a significant improvement in heater temperature and cycle efficiency can be gained.

Heat pump: A machine driven from external power supply absorbing heat at ambient temperature and rejecting the heat at some higher temperature.

Heater: The heat exchanger provided in a prime mover to facilitate the transfer of thermal energy from an external source *to* the working fluid.

Hybrid free-displacer/crank-controlled piston engine: A form of Stirling engine where the reciprocating piston has kinematic coupling to a rotating shaft but the displacer is oscillated under the action of fluid forces.

Indirect heating: A system in which thermal energy from an outside source heats an intermediate fluid (sodium, say) which conveys the energy to the heater tubes and hence to the working fluid (see *Heat pipe*).

Isentropic process: Thermodynamic process of volume, pressure, and temperature change that takes place at constant entropy.

Isobaric process: See *Constant pressure process.*

Isometric process: See *Constant volume process.*

Isothermal compression and expansion: The process of volume and pressure change that occurs without change in the temperature of the system.

Isothermal process: See *Constant temperature process.*

Kinematic drive: A system of cranks, connecting rods, levers or swash-plates used to regulate and control the reciprocating motion of pistons or displacers and to convey power between the pistons and drive shafts.

Metallurgical limit: The maximum temperature of operation for the materials used in the hot spaces of the engine.

Multifuel capacity: The ability of an engine to operate on various fuels or energy sources.

Phase angle: The angle by which volume variations in the expansion space *lead* those in the compression space.

Piston: A heavy structural reciprocating element of a Stirling engine

characterized by a large pressure difference but a negligible temperature difference between the upper and lower transverse faces.

Porosity: The total void volume expressed as a fraction of the volume envelope of the porous solid (frequently expressed also as a percentage).

Pressure drop, Pressure loss: The difference in pressure that arises when fluid flows through a duct or heat exchanger because of aerodynamic-friction effects.

Pressure excursion: The range of variation of the cyclical pressure change of the working fluid in the cylinder.

Pressure ratio: The ratio of the maximum and minimum pressures of the working fluid.

Prime mover: A Stirling engine used to produce mechanical work from heat supplied at high temperatures.

Rallis cycle: An idealized thermodynamic cycle with regenerative processes that occur partly at constant volume and partly at constant pressure. The process of compression and expansion may occur isothermally or adiabatically.

Recuperator (Recuperative heat exchanger): A form of heat exchanger (tube and shell, or finned tube) with separate channels for the hot and cold fluids. Usually the flow is continuous and constant in the channels.

Regenerative annulus: A narrow annular gap between the displacer and cylinder through which the working fluid passes *en route* from the expansion or compression spaces. There is a temperature difference along the length of the annulus and as the gas passes through, a measure of regenerative heat exchange is accomplished.

Regenerative cycle: A thermodynamic cycle in which some attempt is made to utilize the heat in the fluid being rejected from the cycle at low temperatures to heat the incoming fluid and so reduce the amount of 'new' heat required and hence improve the efficiency of the cycle. The regenerative action may take place periodically as in the Stirling engine or continuously as in the Brayton-cycle gas turbine. In the latter case the heat transfer unit which accomplishes the regenerative action may be either a regenerative or a recuperative heat exchanger. Great care must be exercised to avoid confusion when discussing exhaust-gas heat exchangers for regenerative thermodynamic cycles.

Regenerative matrix: A porous volume of finely-divided material (usually metallic) contained in the working space between the compression and expansion spaces. It acts as a reservoir of thermal energy.

Regenerator (Regenerative heat exchanger): A form of heat exchanger consisting of a porous solid mass with a single set of flow passages through which pass periodic, alternate flows of hot and cold fluids.

Regulation: The process of temperature or power control used to regulate the output of a Stirling engine.
Reitlinger cycle: Generalized thermodynamic ideal cycle with isothermal compression and expansion processes at different temperatures bounded by regenerative processes of any nature.
Rhombic drive: A special kinematic drive for Stirling engines which regulates the motion of the piston and displacer in single-acting engines. It is possible to achieve perfect dynamic balance while operating the reciprocating elements at the required phase difference. There are no side forces on the cylinder walls.
Roll-sock seal: A rolling diaphragm seal developed by Philips for containing the working fluid in the working space.
Schmidt cycle: An idealized thermodynamic cycle for Stirling engines with sinusoidal volume variation of the isothermal compression and expansion spaces at different temperatures.
Single-acting engine: A family of Stirling engines with two reciprocating elements per thermodynamic system.
Space power system: An energy conversion device used to provide power for spacecraft.
Stirling cycle: An idealized thermodynamic cycle consisting of isothermal compression and expansion processes at different temperatures bounded by constant volume regenerative processes.
Swash-plate drive: A system used in double-acting Siemens-type Stirling engines for regulating the motion of the displacer-pistons and transmitting power to the drive shaft. The pistons are connected to an inclined disc on a rotating shaft which causes the pistons to reciprocate as the disc rotates.
Swept-volume ratio: The volume variation in the compression space expressed in terms of the volume variation in the expansion space.
Temperature ratio: The ratio of the temperatures of the working fluid in the compression and expansion space.
Thermal efficiency: The fraction of total heat supplied that is converted to useful work.
Total-energy system: An ensemble of machinery receiving a single external energy supply that is capable of providing all the utility needs of a hospital, or residential or commercial building.
Total working space: See *working space.*
Two-phase, two-component working fluid: See *Compound working fluid.*
Underwater power system: An energy conversion device used to provide power for underwater purposes.
Void volume: The total volume of the void spaces in the working space of a Stirling engine including the porous volume of the regenerator and the associated heat exchangers and connecting ducts or ports.

Volume compression ratio: The ratio of the maximum and minimum volumes of the total working space.
Wobble-plate drive: See *Swash-plate drive.*
Work done: The work done by or on the working fluid during a change in volume.
Working fluid: The gas, liquid or vapour which experiences periodic compression and expansion at different temperatures in the working space of a Stirling engine.
Working space: The ensemble of variable volumes and constant volumes comprising the Stirling engine system, including an expansion space, a compression space, void volumes of the regenerator, heater, cooler, and the volumes of clearance spaces and connecting ducts or ports.

BIBLIOGRAPHY AND REFERENCES

THE LITERATURE OF STIRLING ENGINES

THE following bibliography of Stirling engines includes books, papers, patents, and other material generally available from a good technical library. The material is arranged in alphabetical order of the prime author. The list is by no means complete. Additions are being entered constantly as new papers are written, and as others, previously unknown, are found. Many of these contribute further new references. All the material has been through the author's hand at one time or another and most is, euphemistically speaking, on file at the University of Calgary.

The literature of Stirling engines is surprisingly extensive. Some measure of the interest in a subject can be gained by the number of papers written about it. Fig. 22.1 shows the annual number of publications about Stirling engines in the forty-year period, 1940–80. These numbers were gleaned from Martini (1978a) who gives an extensive bibliography arranged chronologically, by subject and by author.

The present situation is perhaps akin to that celebrated by the Dutch legend of the little boy whose finger in the dyke stopped the flood and saved the town. In the case of Stirling engine literature, it would appear his finger is already out of the dyke and the flood is upon us.

In 1971, the author felt competent to review Stirling engines for both power and cooling applications in the same volume. Since that time an equally extensive bibliography for cooling engines has been assembled and separate volumes for the two distinct applications are clearly warranted. Plans are already in train for further separate volumes on free-piston machinery.

Newcomers to the field will find the volume of literature both daunting and challenging. The following guide to the *essential* literature is a good place to start:

1. Overall survey. Ross (1977a) has provided in a small pocket book a most readable, entertaining, and non-technical survey of the whole field including an interesting history, principles of operation, and recent developments.
2. Historical. Finkelstein (1959) has given an extensive historical account of air engines. The long article by Babcock (1885) is also recommended for those interested in the history of heat engines.
3. Philips engines. The paper by van Beukering and Fokker (1973) is the most recent survey of the Philips program.
4. United Stirling engines. Rosenqvist, Gummesson, and Lundholm (1977) gave a status report on the development of the automotive Stirling engine in Sweden.

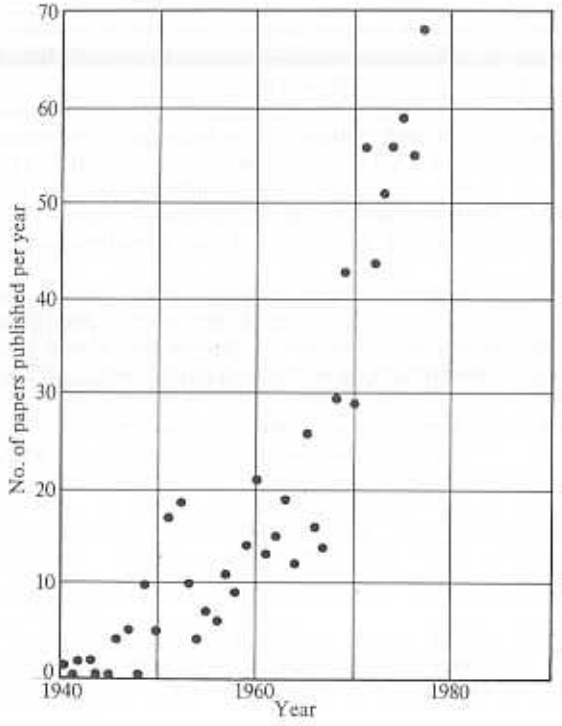


FIG. 22.1: Growth of interest in Stirling engines (after Martini 1978a).

5. MAN/MWM. A recent paper reviewing development of Stirling engines in West Germany was given by Zacharias (1974).
6. General Motors engines. Percival (1974) has provided a historical survey of the General Motors program from 1960 to 1970. It contains a wealth of data and information and is one of the most important survey documents in the open literature.
7. Ford/Philips engines. Progress reports in this program are presented semi-annually at the U.S. Department of Energy Automotive Highway Contractors Coordination Meetings. Kitzner (1977a) has given a comprehensive account of recent automotive Stirling engine activity.
8. Artificial hearts. Annual reports are filed to the U.S. National Institute of Health by McDonnel Douglas, Aerojet-General and Thermo-Electron. Quarterly progress reports are filed to the U.S. Department of Energy by Westinghouse. Papers are contributed annually by most of the above contractors to the Intersociety Energy Conversion Engineering Conference.

9. Design manual. Martini (1978a) assembled much material about Stirling engines with particular reference to different computational methods for design. This document is highly recommended. It contains an extensive bibliography arranged chronologically, by subject, and by author.
10. Contemporary developments. The annual Intersociety Energy Conversion Engineering Conference appears to have become the principal vehicle for publications on Stirling engines. One or two sessions each year are now devoted to Stirling engines. Other relevant papers may be found in the sessions on biomedical applications, heat pumps, energy conservation, and on-site power generation.
11. Automotive engines. The U.S. Department of Energy Stirling engine automotive development program is managed by the Stirling Engine Project Office, NASA/Lewis Research Centre. Various documents are circulated by the Office from time to time.
12. Newsletter. Martini (1977a) publishes an occasional newsletter of the Stirling Engine Research Institute. It contains interesting and informative news, and developments in the field that Martini has been advised about.
13. Cooling engines. Developments of Stirling and other types of regenerative cryogenic cooling engines are regularly reported at the International Cryogenic Conferences, at the U.S. Cryogenic International Cryogenic Conferences, and at the U.S. Cryogenic Engineering Conferences (presently biannual). Such material is also published as *Advances in Cryogenic Engineering*, Plenum Press, New York (now up to about 25 volumes). The most complete repository of information about cryogenics is undoubtedly the Cryogenic Information Center, National Bureau of Standards, U.S. Department of Commerce, Boulder, Co., U.S.A., (Director, Dr. N. Olean).

ABBREVIATIONS

In the following references, titles of journals are abbreviated as recommended in the *World list of scientific periodicals* (ed. P. Brown and G. B. Stratton, Butterworth, London, 1963–5 and later supplements).

However, research reports and other documents with restricted circulation, many of which are of United States origin, are quoted in the style used in journals published by the American Society of Mechanical Engineers (ASME). A number of abbreviations not found in the *World list*, but which occur frequently in these references, are as follows:

AERE—Atomic Energy Research Establishment, Harwell, United Kingdom.

DOE—United States Government, Department of Energy.
EPA—United States Government, Environmental Protection Agency.
ERDA—Energy Research and Development Administration (now replaced by DOE, see above).
I.E.C.E.C.—Intersociety Energy Conversion Engineering Conference.
MAN/MWM—Entwicklungsgruppe Stirlingmotor MAN-MWN, Augsburg, West Germany.
MTI—Mechanical Technology Inc., Latham, New York.
NASA—United States National Aeronautics and Space Administration.
NTIS—National Technical Information Service, United States Department of Commerce, 5285 Port Royal Road, Springfield, Virginia 22151, USA. (Many of the documents mentioned in these references can be obtained via this service).
S.A.E.—Society of Automotive Engineers.

BIBLIOGRAPHY

AGARWAL, P. D., MOONEY, R. J. *and* TOEPEL, R. P. (1969). Stirlec 1, a Stirling electric hybrid car. S.A.E. Paper No. 6900/74, Detroit, Michigan.

AGBI, B. (1971). *The Beale free-piston engine.* M.Sc. Thesis, University of Calgary, Alberta, Canada.

—— (1973). Theoretical and experimental performance of the Beale free-piston Stirling engine. *Proc. 8th I.E.C.E.C.*, Philadelphia, Pa., Aug. 13–17.

ALM, C. B. S., CARLQVIST, S. G., KUHLMANN, P. F., SILVEROVIST, K. H. *and* ZACHARIAS, F. A. (1973). Environmental characteristics of Stirling engines and their present state of development in Germany and Sweden. *10th Int. Cong. Combust. Eng.*, April 5–9, Washington, Paper No. 28.

AMANN, C. A. (1974). Why the piston engine lives on. *Mach. Des.*, **46** (5), Feb. 21.

ANDERSON, D. H. (1975). First quarterly progress report, automotive gas turbine ceramic regenerator design and reliability program. contract no. 68–03–2150, submitted by Ford Motor Co. to E.P.A. Jan.

ANDERSON, L. *and* ENGEL, E. F. (1913). *Caloric Engine.* U.S. Patent No. 1 073 065.

ANDRUS, S. (1976). Development and evaluation of a pneumatic left ventricle assist thermal power system. Annual Report Nat. Inst. of Health, Report No. NO1-HV-3-2930-1, NTIS, Springfield, Va., Sept.

—— BAZINET, G. D., FRASER, R. J., HOFFMAN, L. C. *and* RUDNICKI, M. I. (1972). Development and evaluation of a modified Stirling cycle heart engine. Semi-annual Report, Aerojet Liquid Rocket Co., Sacramento, Ca., June.

ANON, (1854). Napier and Rankine's patent hot-air machines. *Mechanics Magazine*, (1628), Oct. 21.

—— (1906). *The improved Rider and Ericsson hot-air pumping engines.* Rider-Ericsson Engine Co., Catalogue.

—— (1919). *Improvements in or relating to heat engines.* L'Air Liquide Societe Anonyme, British Patent No. 126 940.

—— (1947a). Caloric Engine. *Auto. Engr*, **37** (493), pp. 372-6.
—— (1947b). Philips Air-Engine. *Engineer, Lond.*, **184** (4794), Dec. 12, pp. 549–550; (4795), Dec. 19, pp. 572–574.
—— (1948a). Prime Movers in 1947. *Engineer, Lond.*, **185** (4798), Jan. 9, pp. 44–46; (4799), Jan. 16, pp. 71–72; (4800), Jan. 23, p. 95.
—— (1948b). Inventor of hot-air engine and engine-driven air pump. *Engineer, Lond.*, **186** (4829), Aug. 13, pp. 168–9.
—— (1949). Old hot-air engine. *Engineer, Lond.*, **187** (4862), April 1, pp. 365–6.
—— (1957a). Air engine and air motor. *Collier's Encyclopedia.*
—— (1957b). Air engine. *Encyclopedia Britannica.*
—— (1960). GMR Stirling thermal engine. *Mech. Engng*, **82,** pp. 88–89.
—— (1963). Mobility engineers test Stirling cycle engines. *Automot. Inds*, **129** (3), pp. 35–37.
—— (1965). Stirling machine developments. *Engineer, Lond.*, pp. 906–912, Nov. 26.
—— (1966). General Motors looks to future. *Engineering, Lond.*, **202** p. 927.
—— (1968a). Stirling engine—isotope power source. *Isotop. and Radiat. Technol.*, **6** (1).
—— (1968b). Smogless Stirling engine promises new versatility. *Product. Engng.*, **39,** pp. 31–2, Feb. 26.
—— (1968c). GM Stirlec—a Stirling engine electric hybrid car. GMR Laboratories, Warren, Mich.
—— (1968d). The General Motors Stirling engine. application from under the sea to outer space. GMR Laboratories, Warren, Mich.
—— (1969a). Stirling Engine—a new lease on life. *Mech. Engng* p. 52, July.
—— (1969b). *Study of unconventional thermal, mechanical and nuclear low-pollution-potential power sources for urban vehicles.* Battelle Columbus Labs, Ohio.
—— (1969c). Stirling engines vie for underwater-vehicle jobs. *Product. Engng*, **40** (24), Dec. 15.
—— (1969d). Metal-combustion energy drives Stirling engines under the sea. *Product. Engng*, **40** (24), Dec. 15, pp. 104–5.
—— (1970a). Supercharged and water-injected solar hot air engines. *Engng Prog.*, Univ. of Florida, **24** (2), pp. 1–16.
—— (1970b). Dutch on the road to a pollution-free engine. *Business Week*, Jan. 10, pp. 52–3.
—— (1971a). Low Pollution Engines. *Scientific American*, Sept.
—— (1971b). Developing the Stirling Engine. *Automot. Des. Eng., Lond.*, pp. 57–58, Oct.
—— (1972a). Ford buys license for old Stirling engine, eventual use is possible to fight pollution. *Wall Street Journal*, Aug,. 14.
—— (1972b). Ford will develop 'hot air' engines with Dutch partner. *L.A. times*, Aug. 9.
—— (1972c). Emissions from continuous combustion systems. *Symp. General Motors Research Lab.*, Sept. 27–28, 1971, Plenum Press.
—— (1973a). Evaluation of practicability of a radioisotope thermal converter for an artificial heart device. Westinghouse Electric Corp., Phase I and II Final Reports.
—— (1973b). Evaluation of alternative power sources for low-emission automobiles. Nat. Acad. Sci., Report, Washington D.C.
—— (1973c). Technological feasibility of 1975–76 motor vehicle emission standards. Nat. Acad. Sci., Panel Report, Washington, D.C.

—— (1973d). United Stirling–Stirling air cycle engine status. NATO, Brussels, Belgium.
—— (1973e). Ford is readying Stirling engine for Torino-II. *Auto. Engr*, **81** (8), pp. 42–45.
—— (1974a). A Stirling performance. *Time*, Sept., p. 61.
—— (1974b). Thermo-mechanical generator. *Engineering, Lond.*, **214** (9), Sept. 4.
—— (1974c). Heat regeneration by thermal wheel. Applegate, C. G. and Newberry Ltd., *Energy Dig.*, **3** (6), pp. 21–23.
—— (1975a). What price new engine designs? *New Scient. Lond.*, **65,** p. 267 (Jan. 30).
—— (1975b). British device runs TV on propane gas. *New York Times*, June 12.
—— (1975c). Stirling pump for India. *Atom*, (227), Sept.
—— (1975d). Alternate engines. *L.A. Times*, Sept. 4.
—— (1975e). Alternative automotive power plants. *Auto. Engr*, **83** (12), pp. 18–23, 57.
—— (1976a). Alternative powerplants and energy conservation. *Auto. Engr*, Vol. 84, No. 3, March.
—— (1976b). Ford shows latest Stirling. *Mach. Des.*
—— (1976c). *Stirling isotope power system program.* General Electric Company, Document No. GESP-7130, June–Dec.
—— (1976d). *A simplified heat engine (with pneumatic system). Compressed Air*, **81** No. (9), Sept.
—— (1976e). Stirling cycle applied to water pump. *Solar Energy Digest*, **8,** Sept.
—— (1976f). Century-old engine pumps heart. *New York Times*, Sunday, Dec. 5.
—— (1977). *The Swedish Stirling Engine Development.* Kommanditbolaget, United Stirling, BI.16.3000 12.76.
Anzelius, A. (1926). Uber Erwarmung Vermittels Durchstromender Medien. *Z. angew. Math. Mech.* **6,** pp. 291.
Arthur, J. (1965). Whispering engine. *Pop. Mech.*, Jan., pp. 118–129, 210.
Asselman, G. A. A. (1976a). Fluidized bed coal combustion as a heat source for Stirling engines. *Proc. 11th I.E.C.E.C.*, Lake Tahoe, Nev., Sept. 12–17.
—— (1976b). Thermal energy storage unit based on lithium fluoride. *Proc. 11th I.E.C.E.C.*, Lake Tahoe, Nev., Sept. 12–17.
—— *and* Green, D. B. (1973a). Heat pipes, pt. I operation and characteristics. *Philips Tech. Rev.*, **33** (4), pp. 104–113.
—— —— (1973b). Heat pipes, pt. II applications. *Philips Tech. Rev.*, **33** (5), pp. 138–148.
—— Mulder, J. *and* Meijer, R. J. (1972). A high performance radiator. *Proc. 7th I.E.C.E.C.*, Paper no. 729132, San Diego, Ca.
—— Spigt, C. L. *and* Meijer, R. J. (1977). Design considerations of a thermal storage Stirling engine automobile. S.A.E. Paper No. 770080, pp. 1–12, Detroit, Mich.
Auxer, W. L. (1977). Development of a Stirling engine powered heat activated heat pump. *Proc. 12th I.E.C.E.C.*, pp. 397–401, Washington, D.C., Aug. 28–Sept. 2.
—— (1978). Design study of a general purpose Stirling test engine. DOE Hwy Veh. Syst. Cont. Coord. Mtg, Troy, Mich., May.
Ayres, R. V. (1973). Alternative means of controlling air pollution from vehicular engines. *Inova. 73*, World Innov. Week Conf., Paper No. 10, pp. 148–159, June.
Baas, H. B. *and* Veldhuizen, E. J. J. (1963). *Hot gas reciprocating machine*

and system composed of a plurality of these machines. U.S. Patent 3 074 229, Jan. 22.

BABCOCK, G. H. (1885). Substitutes for steam. *Trans. Am. Soc. mech. Engrs*, **7**, pp. 680–741.

BAHNKE, G. D. *and* HOWARD, C. P. (1964). The effect of longitudinal heat conduction on periodic-flow heat-exchanger performance. *J. Engng Pwr*, **A86**, pp. 105–120.

BAKER, L., MAMIKAUSA, *and* PETROZZI, R. J. (1964). Physical properties of nitrogen tetroxide. Pubn. 870, Aerojet General Corp., Sacramento, Ca.

BALKAN, S. (1975). Thermal energy storage systems for pollutant-free operation of automobiles. *VDI Zeitshrift*, **117**, pp. 422–429, (in German).

BARBER, K. (1975). Introductory comments on Stirling engine program. Div. Transportation Energy Conservation, Automotive Power Systems Contractors Coordination Meeting, ERDA, Wash., D.C.

BARBER, T. A. (1975). Should we have a new engine? An automobile power systems evaluation. Vol. 2, Technical Reports, Calif. Inst. of Tech., S.A.E. Spec. Publ., SP399, Sp400.

BARKER, J. J. (1965). Heat transfer in packed beds. *Ind. & Engng Chem.*, **57** (4), pp. 43–51.

BAUMGARDNER, A. R., JOHNSTON, R. P., MARTINI, W. R. *and* WHITE, M. A. (1973). *Stirling cycle machine with self-oscillating regenerator.* U.S. Patent 27 567, reissued Jan. 23.

BAYLEY, F. J., EDWARDS, P. A. *and* SINGH, P. O. (1961). The effect of flow pulsations on heat transfer by forced convection from a flat plate. *Int. Heat Transfer Conf.*, ASME, Boulder, Colo., pp. 499–509.

—— *and* RAPLEY, C. (1965). Heat transfer and pressure loss characteristics of matrices for regenerative heat exchangers. ASME Paper no. 65-HT-35, Aug.

BAZINET, G. D., FRASER, R. J., HOFFMAN, L. C., MERCER, S. D. *and* RUDNICKI, M. (1971). Development and evaluation of a modified Stirling-cycle engine. Semi-annual report, June–Nov., Aerojet Liquid Rocket Co., Sacramento, Calif.

BEACHEM, C. D., (ed.). (1977). *Hydrogen damage.* Am. Soc. Metals, Metals Park, Ohio.

BEALE, W. T. (1969). Free-piston Stirling engines—some model tests and simulations. S.A.E. Paper No. 690203, Jan.

—— (1971). *Stirling cycle type thermal device.* U.S. Patent 3 552 120, Jan. 5.

—— (1972). *Stirling cycle-type thermal device servo pump.* U.S. Patent 3 645 649, Feb. 29.

—— (1975). A Stirling hydrostatic drive for small vehicles. *Proc. 10th I.E.C.E.C.*, pp. 958–960, Newark, N.J., Aug. 17–22.

—— (1978). New developments in free-piston Stirling engines. *Inst. of Gas Techh. Seminar Stirling-cycle Prime Movers*, Chicago, Ill., June.

—— GEDEON, D. *and* RANKIN, C. (1978). The optimization of Stirling-cycle machines, *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.

—— HOLMES, W., LEWIS, R. S. *and* CHENG, E. (1973). Free-piston Stirling engines—a progress report. S.A.E. Paper no. 730647.

—— *and* RANKIN, C. F. (1975). A 100-watt Stirling electric generator for solar or solid fuel heat sources. *Proc. 10th I.E.C.E.C.*, pp. 1020–22, Newark, N.J., Aug. 17–22.

—— RAUCH, J. S. *and* LEWIS, R. S. (1972). Free piston Stirling-engine-driven inertia compressor for gas-fired air conditioning. *Proc. 2nd Conf. Nat. Gas Res. and Tech.*, Session III, paper no. 5, Washington, June 5–7.

—— —— —— (1973). Design details and performance characteristics of some free-piston Stirling engies, *Proc. 8th I.E.C.E.C.*, pp. 190–7, Philadelphia, Pa., Aug. 13–17.

—— —— —— *and* MULEI, D. (1971). Free-cylinder Stirling engines for solar-powered water pumps, ASME Paper no. 71-WA/Sol-11, Aug.

BEGG, W. (1976). A hot-air engine with rhombic drive. *Model Engineer*, **142**, pp. 296ff and 349f.

BELDING, J. (1978). The role of Stirling engines in power systems. *Inst. of Gas Tech. Seminar, Stirling Cycle Prime Movers*, Chicago, Ill., June.

BELL, A. J. *and* GOLDBERG, L. F. (1976). *The fluidyne engine.* Student Project thesis, Univ. Witwatersrand, Johannesburg.

BENSON, G. M. (1950). Rotary engine hydrostatic drive system. A.A.A.S. Paper, Minneapolis.

—— (1973). Thermal oscillators. *Proc. 8th I.E.C.E.C.*, pp. 182–189, Philadelphia, Pa., Aug. 13–17.

—— (1975). *Thermal oscillators.* U.S. Patent no. 3 928 974.

—— (1977a). Free-piston heat pumps. *Proc. 12th I.E.C.E.C.*, Paper no. 779068, Washington, D.C., Aug. 28-Sept. 2.

—— (1977b). Thermal oscillators. *Proc. 12th I.E.C.E.C.*, pp. 1478–1487, Washington, D.C., Aug. 28-Sept. 2.

—— (1977c). Thermal oscillators. U.S. Patent No. 4 044 558.

—— (1978). Stirling-cycle advanced heat pumps, *Inst. of Gas Tech. Seminar, Stirling Cycle Prime Movers*, Chicago, June.

BERCHOWITZ, D. M., RALLIS, C. J. *and* URIELI, I. (1977). A new mathematical model for Stirling cycle machine. *Proc. 12th I.E.C.E.C.*, pp. 1522–1527, Washington, D.C., Aug. 28-Sept. 2.

BEREMAND, D. G.. (1978). Putting it all together. DOE Hwy Veh. Syst. Cont. Coord. Mtg, Troy, Mich., May.

BERGMAN, U. C. (1975). *Stirling engine power control means.* U.S. Patent 3 914 940, Oct. 28.

BIERMANN, U. K. P. (1975). The lithium/sulphurhexaflouride heat source in combination with a Stirling engine as an environmental independent underwater propulsion system. *Proc. 10th I.E.C.E.C.*, pp. 1023–1030, Newark, N.J., Aug. 17–22.

BIRYUKOV, V. I. *and* SERGEEV, P. V. (1975). Synthesis of three-parameter mechanisms of the rhombic drive of a Stirling engine. *Izv Vyssh Uchevn Zaved Masinostr*, No. 11. pp. 70–76.

BJERKLIE, J. W., CAIRNS, E. J., TOBIAS, C. W. *and* WILSON, D. G. (1975a). Alternative power sources for low-emission automobiles. *Auto. Engr*, **83** (10).

—— —— —— —— (1975b). Evaluation of alternative power sources for low-emission automobiles. S.A.E. Paper no. 750929.

BLANKENSHIP, C. P. *and* SCHULZ, R. B. (1977). Opportunities for ceramics in the ERDA/NASA continuous combination propulsion systems program. NASA. TM X-73597, Jan.

BOESTAD, G. (1938). Die Warmeubertragung im Ljungstrom Luftwarmer. *Feuerungstecknik*, **26**, p. 282.

BOHR, E. (1948). Den Moderna Varmluftsmotorn. *Teknisk Tidskrift*, No. 18, pp. 595–599.

BOLTZ, C. L. (1948). New research work regarding an old machine. *Antriebstechnik*, **13**, pp. 3–4, March–April.

BOMFORD, M. (1975). Duet for hot-air and gramophone. *Motor Boat and Yachting*, Aug. 1, pp. 30–31.

BOSER, O. (1977). Safety considerations for high temperature thermal energy storage in flouride salts. *Proc. 12th I.E.C.E.C.*, pp. 575–582, Washington, D.C.
BOUGARD, J. (1975). Importance of kinematics in the Stirling engines. *Rev. M. Mech.*, Belgium, **21,** pp. 185–190.
BROGAN, J. J. (1973). Alternative powerplants. S.A.E. Spec. Publ. No. SP-383, Paper no. 730519, pp. 31–36, July.
—— (1974). Developments in power systems for transportation. *Proc. 14th A.S.M.E. Annual Symposium*, pp. 45–57, Feb. 28–March 1.
—— (1975). United States research and development program. NATO Committee on the Challenges of Modern Society (ERDA), Washington, D.C.
BUCK, K. E. (1968). Experimental Efforts in Stirling Engine Development. ASME Paper No. 68-WA/Ener-3.
—— (1969a). Development of a Stirling cycle power system for artificial hearts. *Proc. 4th I.E.C.E.C.*, Washington, D.C., Sept. 21–26.
—— (1969b). An Implantable Artificial Heart. *Mech. Engng*, pp. 20–25, Sept.
—— (1971). *Artificial heart pumping system powered by a modified Stirling cycle engine-compressor having a freely reciprocable displacer piston.* U.S. Patent 3 597 766, Aug.
—— (1972). *Modified Stirling cycle engine-compressor having a freely reciprocable displacer piston.* U.S. Patent 3 678 686, July 25.
—— FORREST, D. L. *and* TAMAI, H. W. (1968). A radioisotope-powered Stirling engine for circulatory support. *Proc. 3rd I.E.C.E.C.*, Paper no. 68910, pp. 723–732, Boulder, Col., Aug. 14–16.
BURN, K. S. *and* WALKER, G. (1976). Exploratory study of the rainbow variant Stirling cycle engine. *Proc. 11th I.E.C.E.C.*, Paper no. 769262, Vol. 2, Lake Tahoe, Nev., Sept.
BURWELL, C. C. (1975). Is the energy problem solvable? Oak Ridge Nat. Lab., Tenn.
BUSH, V. (1938). *Apparatus for transferring heat.* U.S. Patent 2 127 286, Aug. 16.
—— (1939). *Apparatus for compressing gases.* U.S. Patent 2 157 229, May.
—— (1949). *Thermal apparatus for compressing gases.* U.S. Patent 2 461 032, Feb. 8.
—— (1969). *Hot gas engines method and apparatus.* U.S. Patent 3 457 722, July 29.
—— (1970). *Compound Stirling cycle engines.* U.S. Patent 3 527 049, Sept. 8.
BYER, R. L. (1976). Initial experiments with a laser-driven Stirling engine. NASA-SP-395, *Conf. on Laser Energy Conversion*, pp. 181–188.
CAIRELLI, J. E. *and* THIEME, L. G. (1977). Initial test results with a single-cylinder rhombic-drive Stirling engine. Proc ERDA Highway Veh. Syst. Cont. Coord. Mtg, Dearborn, Mich., Oct. 4–6.
CARLQVIST, S. G., LIA, T. *and* LUNDHOLM, G. S. K. (1975). Stirling engines: their potential use in commercial vehicles and their impact of fuel utilization. *Proc. Conf. Power Plants and Future Fuels*, I. Mech. E., London, pp. 35–46, Jan.
—— *and* ORTEGREN, L. G. H. (1974). The potential impact of the Stirling engine on environmental issues. Inst. Road Transport Engines, Jan. 17.
—— ROSEQVIST, K. G. *and* GUMMESSON, S. G. (1977). Developing the Stirling engine for fuel economy in marine, industrial and medium duty automotive applications. *12th Int. Cong. Combust. Eng.* Tokyo.
CARRETT, K. (1975). Stirling engine is a hot contender for car power-plants. *Engineer, Lond.*, **240,** March.
CAYLEY, G. (1807). *Nicholson's Journal*, p. 206 (letter).
CERQUEIRA-LIMA, A. R. (1969). *Dynamic performance of the thermal regenerator (analysed by a digital model).* Master's Thesis, Instituto Techologico de Aero-Nautica, Sas Jose dos Compos.
CHADDOCK, D. H. (1976). The hot-air engine competition. *Model Engr*, **142,** pp. 914–918, Sept. 17.
—— (1977). The hot-air engine competition at the Wembley Model Engineer Exhibition. *Model Engr*, **143,** pp. 558–568, May 20.
CHAPMAN, R. (1968). *Energy systems of extended endurance in the 1–100 kW Range for undersea applications.* Publication 1072, Nat. Acad. Sc., Washington, D.C.
CHIRONIS, N. P. (1968). Smogless Stirling engine promises new versatility. *Prod. Engng*, **39,** pp. 30–33, Feb. 26.
CLAPHAM, E. F. (1977). Hot-air engine, *Model Engr*, **143,** pp. 852–856, Aug. 5, pp. 908–912, Aug. 19.
COLE, D. W., HOLEMAN, W. S. *and* MOTT, W. E. (1973). Status of the U.S.A.E.C.'s nuclear-power artificial heart. *Trans. Soc. Artif. Int. Organs*, **19,** p. 537.
COLLINS, F. M. (1977). A pressurized hot-air engine. *Model Engr*, **143,** pp. 822–875, July 15.
COLOSIMO, D. D. (1976). Analysis of portential temperature discontinuities in Stirling cycle heat exchanger system. Dynatech Report No. 1388, Project No. HC-81-48, Amer. Gas. Ass., May.
—— (1976). On-site heat-activated heat pumps. *Tech. Oportunities for Energy Conservation in Appliances* (Conference), Boston, Mass., May 11.
COMBES, Par M. (1853). Sur des documents relatifs à la machine à air chaud du Capitaine Ericsson. *Annales des Mines*, **3.**
CONLIN, D. M. *and* REED, L. H. K. (1973). The Performance of a modified Stirling engine with exhaust gas recuperator. Project Report, Sch. of Eng., Univ. of Bath, U.K.
COOKE-YARBOROUGH, E. H. (1967). A proposal for a heat-powered non-rotating electrical alternator. Harwell AERE-M1881.
—— (1970). *Heat engines.* U.S. Patent 3 548 589, Dec. 22.
—— (1974a). Simplified expressions for the power output of a lossless Stirling engine. Harwell, AERE-M2437, March.
—— (1974b). Thermo-mechanical generator: an efficient means of converting heat to electricity at low power levels. *Proc. Instn elect. Engrs*, **121,** pp. 749–751.
—— FRANKLIN, E., GLEISOW, J., HOWLETT, R. *and* WEST, C. D. (1974a). The Harwell thermo-mechanical generator. Harwell, AERE-R7714, March.
—— —— —— —— —— (1974b). Harwell thermo-mechanical generator. *Proc. 9th I.E.C.E.C.*, Paper no. 749156, San Francisco, Cal., Aug. 26–30.
COOKE-YARBOROUGH, E. H. *and* YEATS, F. W. (1975A). Efficient thermo-mechanical generation of electricity from the heat of radioisotopes. *Proc. 10th I.E.C.E.C.*, Paper no. 759150, pp. 1003–1011, Newark, N.J., Aug. 17–22.
—— —— (1975b). Efficient thermo-mechanical generation of electricity from the heat of radioisotopes. Harwell, AERE-R8036, May.
COPPAGE, J. E. (1952). *Heat -transfer and flow friction characteristics of porous media.* Thesis, Stanford University.
—— *and* LONDON, A. L. (1953). The periodic-flow regenerator—a summary of design theory. *Trans. Am. Soc. mech. Engrs*, **75,** pp. 779–787.
—— —— (1956). Heat-transfer and flow-friction characteristics of porous media. *Chem. Eng. Prog.*, **52,** pp. 56–7.
COWANS, K. W. (1968). *Heat powered engine.* U.S. Patent 3 379 026, April 23.

CRABTREE, L. F. (1976). Engineering—art and science. *R.Ae.S Aerospace*, **3** (7), pp. 22–25.
CRANDALL, S. H. (1956). *Engineering analysis*. McGraw-Hill Book Company, New York, p. 177.
CRESWICK, F. A. (1957). A digital computer solution of the equation for transient heating of a porous solid, including the effects of longitudinal conduction. *Ind. Math.*, **8**, pp. 61–68.
—— (1965). Thermal design of Stirling-cycle machines. S.A.E. Paper No. 949C, Detroit.
— *and* MITCHELL, R. K. (1962). Design and construction of a Stirling-cycle engine to drive a gas-furnace circulating blower. Battelle Research Report, Dec. 28.
CROSSLAND, J. (1974). Cars, fuel and pollution. *Environment*, March.
CROUTHAMEL, M. S. *and* SHELPUK, B. (1972). A combustion-heated, thermally-actuated vuilleumier refrigerator. *Cryog. Eng. Conf.*, pp. 339–351, Aug. 9–11.
DANIELS, A. (1973). Stirling engines—capabilities and prospects. *Proc. 6th Cryog. Symp. and Expo*, pp. 190–210, Oct. 2–4.
—— (1974a). The Stirling engine as a total energy system prime mover. Philips Labs., Briarcliff Manor, N.Y.
—— (1974b). Stirling engine, a promising power source for communications equipment. Philips Labs., Briarcliff Manor, N.Y.
—— (1978). Advanced developments in Stirling machines. *Inst. of Gas Tech. Seminar, Stirling Cycle Prime Movers*, Chicago, Ill., June.
—— *and* DU PRE, F. K. (1966). Closed cycle cryogenic refrigerators as integrated cold sources for infrared detectors. Applied Optics, **5** (9).
—— —— (1971a). Miniature refrigerators for electronic devices. *Philips Tech. Rev.*, **32** (2), pp. 49–56.
—— —— (1971b). Triple-expansion Stirling cycle refrigerator. *Adv. Cryog. Eng.*, **16**, pp. 178–184.
—— —— (1973). Miniature refrigerators for electronic devices. *Cryogenics*, **13** (3), pp. 134–140, March.
DANIELS, F. (1971). Power production with assemblies of small solar engines. ASME paper no. 71-WA/Sol-5, Nov. 28–Dec. 2.
DARK, H. E. (1975). *Auto engines of tomorrow: power alternatives for cars to come*. Indiana University Press, Bloomington, Ind.
DARLING, G. B. (1959). Heat transfer to liquids in intermittent flow. *Petroleum*, pp. 177–178, May.
DAVIES, S. J. *and* SINGHAM, J. R. (1951). Experiments on a small thermal regenerator. General Discussion on Heat Transfer, Inst. of Mech. Eng., London, pp. 434–435.
DAVIS, S. R. *and* HENEIN, N. A. (1973a). Comparative analysis of Stirling and other combustion engines. S.A.E. Spec. Publn, March.
—— —— (1973b). Controlling emissions with the Stirling engine. *Auto. Engr*, **81** (4), pp. 32–35, April.
—— —— *and* LUNDSTROM, R. R. (1971). Combustion and emission formation in the Stirling engine with exhaust gas recirculation. S.A.E. Paper No. 710824, Truck, Powerplant, Fuels and Lubricants Meeting, St. Louis, Missouri, Oct. 26–29.
—— —— *and* SINGH, T. (1972). Emission characteristics of Stirling engines. *Proc. 7th I.E.C.E.C.*, pp. 887–895, San Diego, Ca.
DAVOUD, J. G. *and* BURKE, J. A. (1977). D-cycle vapor Stirling engine. Abstract, *4th Inter. Symp. on Auto. Prop. Systems*, Vol. 3, Washington, April 20.

DE BENIE, G. *and* MARCHETTI, C. (1970). Hydrogen, key to the energy market. *Eurospectra*, **9** (2).
DEBONO, A. N. (1975). A swash-plate hot-air engine. *Model Engineer*, **141**, pp. 905f, Sept. 19.
DE BREY, H. (1952). *Hot-gas reciprocating engine of the kind comprising one or more closed cycles*. U.S. Patent 2 616 249.
—— RINIA, H. *and* VAN WEENEN, F. L. (1947). Fundamentals for the Development of the Philips Air Engine. *Philips Tech. Rev.*, **9**, pp. 97–104.
DECKER, O. (1978). MTI Stirling engine powertrain development. DOE Hwy. Veh. Syst. Cont. Coord. Mtg., Troy, Mich., May.
DE LANGE, L. (1956). *Cooler construction in a hot-gas engine*. U.S. Patent 2 764 879, Oct.
DE WILDE DE LIGNE, J. H. (1971). The Philips 4-235 Heavy-duty Stirling engine, a progress report. *6th I.E.C.E.C.*, Boston, Mass., Aug. 5, (paper not included in proceedings but available on request from author).
DEN HAAN, J. J. W. (1958). *Regenertor for use in hot-gas reciprocating engine*. U.S. Patent 2 833 523, May.
DENHAM, F. R. (1953). *A study of the reciprocating hot-air engine*. Ph.D. Thesis, Durham University.
DENTON, W. H. (1951). The heat transfer and flow resistance for fluid flow through randomly packed Spheres. Proc. Gen. Disc. on Heat Trans., I. Mech. E., London, pp. 370–373.
DIDION, D., MAXWELL, B. *and* WARD, D. (1977). A laboratory investigation of a Stirling engine driven heat pump. International Seminar on Heat Transfer in Buildings, Dubrovnik, Yugoslavia.
DINEEN, J. J. (1965). *Heat Engine*. U.S. Patent 3 220 178.
DODGE, D. D. (1978). Stirling-cycle engines: design, projection, application and measurement. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.
DOODY, R. D. (1975). Long-life, high-capacity vuilleumier refrigerator for space applications. AFFDL-TR-75-108, W.P.A.F.B., Ohio, Sept.
DROS, A. A. (1951). *Combination comprising a hot-gas engine and a piston machine driven thereby*. U.S. Patent 2 558 481.
—— (1965a). An industrial gas refrigerating machine with hydraulic piston drive. *Philips Tech. Rev.*, **26**, pp. 297–308.
—— (1965b). An industrial gas refrigerating Stirling machine. *Adv. Cryog. Eng.*, **10**.
—— (1966). Large-capacity industrial Stirling machine. *Adv. Cryog. Eng.* **10.**
—— *and* KOOPMANS, A. (1952). *Multitube Heater for Hot-Gas Reciprocating Engines*. U.S. Patent 2 746 474.
—— YZER, J. A. L. *and* HELLINGMAN, E. (1956). *Hot-Gas Reciprocating Engine*. U.S. Patent 2 746 241, May 22.
DUNN, P. D., RICE, G. *and* THRING, R. H. (1975). Hydraulic and rotary drive Stirling engines with fluidised bed combustion/heat pipe system. *Proc. 10th I.E.C.E.C.*, pp. 942–947, Newark, N.J., Aug. 17–22.
DUNNE, J. (1968). Test driving GM's hybrid electric car—a Stirling engine and an electric motor working together—a promising idea on the way to a smogfree car. *Pop. Sci.*, pp. 116–119, Dec.
DU PRE, F. K. (1952). *Hot-Gas Engine or Refrigerator*. U.S. Patent 2 590 519.
—— *and* DANIELS, A. (1965). Miniature refrigerator opens new possibilities for cryo-electronics. North Amer. Philips Co., reprinted from *Signal* magazine, Sept.

(1974). An implantable thermal converter as power source for an artificial heart. *Proc. 9th I.E.C.E.C.*, pp. 775–781, San Francisco, Ca., Aug. 26–30.

FRITZ, B. (1875). Ueber die Ausnutzung der Brennftoffe, (Utilization of Fuel). *Dingler's Polyt. Jnl.*

FRYER, B. *and* LEACH, C. (1968). A radioisotope energized undersea Stirling engine. *Proc. 3rd I.E.C.E.C.*, p. 830, Boulder, Col., Aug. 14–16.

FURNAS C. C. (1930). Heat transfer from a gas stream to a bed of broken solids. *Indus. Engng Chem.* **22,** p. 26.

—— (1932). Heat transfer from a gas stream to a bed of broken solids. *Bulletin*, U.S. Bureau of Mines, No. 361.

GABRIELSSON, R. G. *and* LIA, T. A. (1975). Low-emission combustors for Stirling engines. *Proc. 10th I.E.C.E.C.*, Newark, N.J., pp. 927–932, Aug. 17–22.

GADSBY, G. N. (1977). A preliminary study of the market potential of the Stirling engine vis-a-vis industrial diesel and gas turbine engines. Mitre Corp. Tech. Report no. 766S; also DOE/NTIS.

GAMSON, B. W. (1951). Heat and mass transfer, Fluid solid systems. *Chem. Eng. Prog.*, **47,** pp. 19–28.

—— THODOS, G. *and* HOUGEN, O. A. (1963). Heat, mass and momentum transfer in the flow of gases through granular solids. *Trans. A.I.C.E.*, **39,** pp. 1–35.

GARAY, P. N. (1960). Stirling engine betters efficiency. *Power*, Sept.

GARBUNY, M. *and* PECHERSKY, M. J. (1976). Optimization of engines operated remotely by laser power. Conf. on Laser Energy Conversion, NASA SP-395, pp. 173–180.

—— —— (1976). Laser engines operating by resonance absorption. *Appl. Opt.*, **15** (5) May.

GASPAROVIC, N. (1972). Engines with Rhombic Drive Mechanism. *Marine Eng. Rev.*, **77,** pp. 25–27.

GAWRON, K. *and* SCHRODER, J. (1972). Heat storage properties of eutectic fluoride mixtures. 4th European Symp. on Flourine Chemistry, Lhubljana, Sept.

GEDEON, D., BEALE, W. T. *and* RANKIN, C. (1978). The optimization of Stirling-cycle machines. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.

GIBSON, B. M., HORNBECK, C. J., LONGWORTH, R. C. *and* HARMISON, L. T. (1971). Bypass gas-actuated thermocompressor as an implantable artificial heart power source. *Proc. 6th I.E.C.E.C.*, pp. 310–316, Boston, Mass., Aug. 2–6.

GIFFORD, W. E. *and* LONGWORTH, R. C. (1963). Pulse-tube refrigeration. ASME Paper no. 63-WA-290.

—— —— (1964). Pulse-tube refrigeration progress. *Adv. Cryog. Eng.*, **10,** Section M–U, pp. 69–79.

—— —— (1965). Surface heat pumping. Cryog. Eng. Conf., Rice University, Houston, Texas.

GLASSFORD, A. P. M. (1962). *An oil-free compressor, based on the Stirling cycle.* M.Sc. Thesis, M.I.T.

GODOY, J. V. (1914). *Improvements Relating to Heat Engines.* British Patent.

GOLDBERG, L. F. RALLIS, C. J., BELL, A. J. *and* URIELI, I. (1977). Some experimental results on laboratory model fluidyne engines. *Proc. 12th I.E.C.E.C.*, pp. 1528–1532, Washington, D.C., Aug. 28–Sept. 2.

GOLDOWSKY, M. *and* LEHRFELD, D. (1977). Artificial heart thermal converter component research and development. *Proc. 12th I.E.C.E.C.*, pp. 126–133, Washington, D.C., Aug. 28–Sept. 2.

GOLDWATER, B. *and* MORROW, R. B. (1977). Demonstration of a free-piston Stirling linear alternator power conversion system. *Proc. 12th I.E.C.E.C.*, pp. 1488–1495, Washington, D.C., Aug. 28–Sept. 2.

—— PILLER, S., RAUCH, J. *and* CELLA, A. (1977). Demonstration of a free-piston Stirling engine driven linear alternator. Phase I Report No. 77TR40, Mech. Tech. Inc.

GOODALE, T. C. *and* WALTER, D. (1976). Hydrogen safety tests of the Stirling engine. Stanford Research Institute Report, No. S.R.I., Project PYC-2696, to the Ford Motor Co., (included as Appendix A in Kitzer 1977b).

GORANSON, R. B. (1968). Application of the raioisotope-fuelled Stirling engine to circulatory support systems. MDAC Final Report, No. DAC-60742, June.

—— *et al.* (1968). Development of a simplified Stirllng engine to power circulatory assist devices. *Proc. 3rd I.E.C.E.C.*, Boulder, Colorado, Aug.

GRASHOF, F. (1890). *Theorie der Kraftmaschinen.* Hamburg.

GRATCH, S. (1976). *Advanced Automotive Propulsion.* AIChE, New York, Vol. 1, pp. 2–5.

GRIGORENKO, N. M., SAVCHENKO, V. I. *and* PRUSMAN, Y. O. (1975). Results of Test of a Heat-Using Cryogenic Machine. *Khim. Neft. Mashinostr.* **11** (9).

GROLL, M. *and* ZIMMERMANN, P. (1969). Kenngrößen zum Beurteilen von Wärmeträgern fur Wârmerohre. *Chem. Ing. Tech.* **41** (24).

GUILFOY, R. J., Jr. (1973). Refrigeration systems for transporting frozen foods. *ASHRAE Jnl.*, **15** (5), pp. 58–60.

HAHNEMANN, H. (1948). Approximate calculation of thermal ratios in heat-exchangers including heat conduction in the direction of flow. National Gas Turbine Establishment, Memorandum 36.

HAKANSSON, S. A. S. (1974). *Heat exchangers for Stirling-cycle engines.* U.S. Patent 3 834 455.

—— (1975). *Multi-Cylinder Double-Acting Stirling Cycle Engines.* U.S. Patent 3 927 529.

HALL, W. B. (1958). *Reactor Heat Transfer.* Temple Press Ltd., London.

HALLARE, B. (1975). Stirlingmotorn i undervattenstjänst. *Industriell Tecknik*, **7–8,** pp. 48–51.

—— *and* ROSENQVIST, K. (1977). The development of 40–150 kW Stirling engines in Sweden and their application in mining equipment, total energy systems and road vehicles. *Proc. 4th Int. Symp. Auto. Prop. Syst.*, Vol. 3, Sess. 8/9, Washington, D.C., March/April.

HALLEY, J. A. (1958). The Robinson-type air engine. *Jnl Stephenson Engng Soc.*, King's College, Newcastle, **2** (2), p. 49.

HAMERAK, K. (1971). Der Heissgasmotor—eine Interessante Hubkolben-kraftmaschine mit Ausserer Vebrennung. *Energie und Technik*, **23** (5), pp. 175–178.

HANOLD, R. J. *and* JOHNSTON, R. D. (1962). *Power Plant Heat Storage Arrangement.* U.S. Patent 3 029 596.

HAPKE, H. (1973). The influence of flow pattern and heat-transfer in the heat-exchanger unit of Stirling machines on the thermodynamic cycle. *Brennst.-Waerme-Kraft*, **25** (10), pp. 389, 392–394.

HARKLESS, L. B. (1974). Demonstration of advanced cryogenic cooler—Infared detector assembly. Air Force Flight Dynamics Lab., AFFDL-TR-74-15.

HARLEY, J. (1974a). Ford leads in the race to find a cleaner yet economic engine. *Engineer, Lond.*, 238, May.

—— (1974b). Engines to stretch the world's oil. *New Scient. Lind.*, **63,** p. 334.

—— (1974c). Stirling set for 1980. Automot. Des. Eng., **13,** pp. 27, 29, Sept.

HARMISON, L. T. *and* HASTINGS, F. W. (1969). The artificial heart program. *Proc. First Artificial Heart Conf.*, pp. 1–16, Washington, d.C., June.

—— MARTINI, W. R., RUDNICKI, M. I. *and* RUFFMAN, F. N. (1972). Experience with implanted radioisotope-fuelled artificial hearts. *Proc. 2nd Int. Symp. on Power for Radioisotopes*, Paper EN/IB/10, pp. 730–761, May 29–June 1, Madrid.

HARREWIJNE, A., MICHELS, A. P. J. *and* GASSELING, F. W. E. (1975). *Device for converting heat energy into mechanical energy.* German Patent 2 427 819.

HARRIS, W. S., RIOS, P. A. *and* SMITH, J. L. (1971). The design of thermal regenerators for Stirling-type refrigerators. *Adv. Cryog. Eng.*, **16,** pp. 312–323.

HARVEY, D. C. *and* MENCHEN, W. R. (1974). *The automobile: Energy and the environment.* Hittman Associates, Inc., Columbia, Md.

HAUSEN, H. (1929). Warmeaustauch in Regeneratoren. *VDI Zeitschrift*, **73,** p. 432.

—— (1929). Uber die Theorie des Warmeaustausches in Regneratoren. *Z. angew Math. Mech.*, **9,** pp. 173–200.

—— (1931). Naherungsverfahren zur Berechnung des Warmeaustausches in Regeneratoren. (A Approximate Method of Dimensioning Regenerative Heat-Exchangers). *Z. angew. Math. Mech.*, **11,** pp. 105–114, April.

—— (1942). Vervollstandigte Berechnung des Warmeaustauches in Regeneratoren. Z. Ver. Dt. Ing. Beiheft Verfahrenstechnik, No. 2, p. 31.

HAVEMANN, H. A. *and* NARAYAN, R. N. (1954). Heat transfer in pulsating flow. *Nature, (Lond.)*, **174,** p. 41.

—— —— (1955). Studies for new hot-air engine. *J. Indian Inst. Sci.*, Sec. B, **37,** pp. 38, 172, 224.

—— —— JAYACHANDRA, P. *and* GARG, G. C. (1959). Studies for new hot-air engine. *J. Indian Inst. Sci.*, Sec. B., **38,** pp. 172–202.

HAYS, D. F. *and* MAKI, E. R. (1971). The crowned circular slider bearing: analysis and test. *J. Lubric. Technol.* (ASME), **94,** pp. 280–6.

HAZARD, H. R. (1964). Multifuel thermal energy converters. Proc. 18th Annual Power Sources Conf., Fort Monmouth, May 19–21.

HEFFNER, F. E. (1966). Highlights from 6500 hours of Stirling engine operation. S.A.E. Paper No. 949D, pp. 33–54, and GMR Publication No. GMR-456.

—— (1969). GMR Stirling Engines-G.P.U.3. Information Sheet, G. M.R.L., Jan.

—— (1968). A collection of Stirling engine reports from General Motors research 1958 to 1970. G.M. Research Publication GMR 2690. General Motors Technical Center, Warren, Michigan.

HENDERSON, R. E. *and* DRESSER, D. L. (1960). Solar concentration associated with the Stirling engine. Amer. Rocket Soc., Space Power Systems Corp., Santa Monica, Cal., Sept.

HENSHALL, J. (1977). A Stirling cycle engine, parts I and II. *Model Engr*, **142** (1976), pp. 1259–1265, and **143** (1977) pp. 25–27, Jan. 7.

HERMANS, M. L., UHLEMANN, H. *and* SPIGT, C. L. (1972). The combination of a radioisotopic heat source and a Stirling cycle conversion system. *Proc. 2nd Int. Conf. on Power from Radioisotopes*, pp. 445–466, Madrid, May/June.

—— *and* ASSELMAN, G. A. A. (1978). The Stirling cycle in climate control systems. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.

HERSCHEL, J. (1850). Making ice. *The Athenaeum*, p. 22, Jan. 5.

HIGA, W. H. (1965). Practical Philips cycle for low temperature refrigeration. Cryogenic Tech., p. 203.

—— (1967). Simplifed heat engine J.P.L. Calif. Report No. 30-3297/NPO-13613.

—— (1975). *Stirling cycle engine and refrigeration systems.* U.S. Patent 3 971 230.

HINTON, M. G. Jr. LURA, T., ROESSLER, W. U. *and* SAMPSON, H. T. (1971). Exhaust emission characteristics of hybrid heat engine/electric vehicles. S.A.E. Paper 710825, Oct. 26–29.

HISE, E. C. *and* WILSON, J. V. (1977). A heat pump cycle with an air-water working fluid. *Proc. 12th I.E.C.E.C.*, Report No. 779066, Washington, D.C., Aug. 28–Sept. 2.

HOAGLAND, L. C. *and* PERCIVAL, W. H. (1978). Potential of the Stirling engine for stationary power applications in the 500–2000 hp range. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.

HOEHN, F. (1978). Stirling laboratory research engine: Report on the design and fabrication. JPL Report No. 5030-178, Jet Propulsion Lab., Pasadena, Ca.

—— *and* MCDOUGAL, A. R. (1978). Design of a preprototype Stirling Laboratory Research Engine *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.

HOESS, J. A. *and* STAHMAN, R. C. (1969). Unconventional thermal mechanical and nuclear low-pollution—potential power sources for urban vehicles. *Proc. 4th I.E.C.E.C.*, pp. 959–980, Washington, D.C., Sept. 21–26.

—— CHEANEY, E. S., CRESWICK, F. A., TRAYSER, D. A., FISCHER, R., TIMBERLAKE, A. B., BASHAM, S. J., HERRIDGE, J. T. *and* WILCOS, J. P. (1969). Study of unconventional thermal, mechanical and nuclear low-pollution-potential power sources for urban vehicles. Battelle Columbus Labs., Ohio, Oct.

HOFFMAN, L. C. (1976). Engine Performance. Site Review Sub-Committee for Pneumatic Left Ventricle Assist Thermal Power System, Program Review, May. Sacramento, Ca.

HOGAN, W. H. (1963). *Method and apparatus for employing fluids in a closed cycle.* U.S. Patent 3 115 014, Dec. 24.

—— (1964). *Closed-Cycle Cryogenic Refrigerator and Apparatus Embodying Same.* U.S. Patent 3 151 466, Oct. 6.

—— *and* STUART, R. W. (1963). Design considerations for cryogenic refrigerators. ASME Paper No. 63-WA-292.

HOLMAN, R. R. *and* JASPERS, H. A. (1974). A Stirling cycle 400 W(e) economical radioisotope generator. Proc. Am. Nuc. Soc. Ann Mtg., Phila.

HOLMGREN, J. S. (1970). Implanted energy conversion system. MDAC Annual Report, No. PH43-67-1408-3, June 29, 1969–July 7, 1970.

HORN, S. B. (1973). *Pneumatic Stirling cycle cooler with non-contaminating compressor.* U.S. Patent 3 765 197, Oct.

HORTON, J. H. (1966). Military engines for ground power. *Automot. Ind.*, pp. 65–67, May 1.

HOUGEN, J. O. *and* PIRET, E. L. (1951). Effective thermal conductivity of granular solids through which gases are flowing. *Chem. Eng. Prog.*, **47,** pp. 295–303.

HOWARD, C. P. (1963). Heat-transfer and flow-friction characteristics of skewed-passage and glass-ceramic heat-transfer surfaces. ASME Paper No. 63-WA-115.

—— (1964), The single-blow problem including the effects of longitudinal conduction. ASME Paper No. 63-GTP-11.

HUFFMAN, F. N., HARVEY, R. J. *and* KITRILAKIS, S. S. (1968). Design of an implantable Rankine-cycle radioisotope power source. *Proc. 3rd I.E.C.E.C.*, p. 750, Boulder, Col., Aug. 14–16.

HURLEY, E. G. (1954). Tests on a twin-piston Stirling-cycle engine using internal combustion. Shell Thornton Report K. 121.

HURN, R. W. (1978). Fuel tolerance in the Stirling engine. *Proc. 13th I.E.C.E.C.*, San Diego. Ca., Aug. 20–25.

HUTCHINSON, F. W. (1955). *Thermodynamics of heat power systems.* Addison Wesley Publ. Co. Inc., Reading, Mass., Chapter 16, pp. 373–376.
ILIFFE, C. E. (1948). Thermal Analysis of the Contra-Flow Regenerative Heat-Exchanger. *Proc. I. Mech. E.*, **159,** pp. 363–372.
JAKEMAN, R. W. (1966). *The construction and testing of the Stirling universal research engine.* B.Sc. Thesis, Durham Univ.
JANICKI, E. (1976). Which Auto Engine Next? *Auto. Engr,* **1,** (7).
JASPERS, H. A. (1975). *Power-Control System for Stirling Engines.* U.S. Patent 3 886 744.
—— *and* DU PRE, F. K. (1973). Stirling engine design studies of an underwater power system and a total energy system. *Proc. 8th I.E.C.E.C.,* pp. 588–593, Philadelphia, Pa., Aug. 13–17.
JOHANSSON, L. (1978). Small Stirling machines for stationary applications. *Inst. of Gas Tech. Seminar, Stirling Cycle Prime Movers,* Chicago, Ill., June.
JOHNSON, J. E. (1952). Regenerator heat-exchangers for gas turbines. Aero Research Council Technical Report, R and M No. 2630.
JOHNSON, O. (1946). Civilization, to John Ericsson, debtor. *The Scientific Monthly,* pp. 101–106, Jan.
JOHNSTON, R. P. (1972–3). Implanted Energy Conversion System. MDAC Annual Reports 1971–2, 1972–3. Reports. Nos. PH43-67-1408-5, and PH43-67-1408-6.
—— BAKER, L. P., BENNETT, A., BLAIR, C. R. *and* EMIGH, S. G. (1968). Implanted energy conversion system. MDA Annual Report no. MDC-G4448, 1975–6.
—— BENNETT, A., EMIGH, S. G., GRIFFITH, W. R., NOBLE, J. E., PERRONE, R. E. *and* WHITE, M. A. (1977). Stirling/hydraulic artificial heart power source. *Proc. 12th I.E.C.E.C.,* pp. 104–111, Washington, D.C., Aug. 28–Sept. 2.
—— —— —— —— —— —— —— MARTINI, W. R., *and* NICCOLI, L. G. (1976). A Stirling hydraulic power source for artificial hearts. *Proc. 11th I.E.C.E.C.,* p. 143, Lake Tahoe, Sept. 12–17.
—— EMIGH, S. G., GRIFFITH, W. R., NOBLE, J. E. *and* PERRONE, R. E. (1975). Implanted energy conversion system. MDA Annual Report no. MDC-G4444, 1974–5.
—— GRIFFITH, W. R., PERRONE, R. E., MARTINI, W. R. *and* EMIGH, S. G. (1974). Implanted energy conversion system. MDAC Annual Report no. 1-HV-4-2901-1, 1973–4.
—— NOBLE, J. E., EMIGH, S. G., WHITE, M. A., GRIFFITH, W. R. *and* PERRONE, R. E. (1975). A Stirling engine with hydraulic power output for powering artificial hearts, *Proc. 10th I.E.C.E.C.,* pp. 1448–55, Newark, N.J., Aug. 17–22.
—— *and* WHITE, M. A. (1971). Simulation of an artificial heart system, MDAC Paper no. WD 1589.
JONES, L. L. Jr. *and* FAX, D. H. (1954). Perturbation solutions for the periodic-flow thermal regenerator. ASME Paper no. 54-A-130.
JONKERS, C. O. (1958). *Hot-Gas Reciprocating Engine.* U.S. Patent no. 2 857 220.
—— *and* KOHLER, J. W. L. (1960). *Gaseous medium leakage prevention arrangement for a hot-gas reciprocating machine.* U.S. Patent No. 2 943 453.
JOULE, J. (1852). On the Air Engine. *Phil. Trans. R. Soc.* **142,** pp. 65–77.
KARAVANSKY, I. I. *and* MELTSER, L. Z. (1958). Thermodynamic investigations of the working cycle of the Philips machine. *Proc. 10th Int. Cong. Refrig.,* pp. 3–29, 209.

KAYS, W. M. *and* LONDON, A. L. (1958). *Compact Heat-Exchangers.* McGraw Hill, New York; 2nd edn, McGraw Hill, 1964.
KEITH, T. (1978). Pumping Ring Analysis. DOE Hwy. Veh. Syst. Cont. Coord. Mtg., Troy, Mich., May.
KELLY, D. A. (1976). *Rotary Closed Parallel Cycle Engine Systems.* U.S. Patent 3 958 421.
—— (1976). *Rotary Stirling Cycle Engine Systems.* U.S. Patent 3 958 422.
KETTLER, J. R. (1975). The thermal vehicle—A pollution-free concept. *Proc. 10th I.E.C.E.C.,* pp. 548–553, Newark, N.J., Aug. 17–22.
KHAN, M. I. (1962). *The application of computer techniques to the general analysis of the Stirling cycle.* M.Sc. Thesis, Univ. of Durham.
KIM, J. C. (1970). *An analytical and experimental study of heat transfer and flow-friction characteristics for periodically reversing flow through the porous matrix of thermal regenerators.* Ph.D. Thesis, Purdue Univ.
—— (1973). An analytical and experimental study of flow friction characteristics of periodically reversing flow. ASME Paper no. 73-WA/FE-13, pp. 1–8.
—— *and* QVALE, E. B. (1971). Analytical and Experimental Studies of Compact Wire-Screen Heat Exchanger. *Adv. Cryog. Eng.,* **16,** pp. 302–311.
—— —— *and* HELMER, W. A. (1971). Apparatus for studies of regenerators and heat exchangers for pulse tube, vuillemier and Stirling-type refrigerators. 8th Int. Cong. of Refrig., Paper no. 1:46, Aug.
KIRK, A. (1874). On the mechanical production of cold. *Proc.* of the *Inst. Civil Eng.* **37,** pp. 244–315.
KIRKLEY, D. W. (1959). *Continued work on the hot-air engine.* B.Sc. Hons. Thesis, Durham Univ.
—— (1962). *An investigation of the losses occurring in reciprocating hot-air engines.* Ph.D. Thesis, Durham Univ.
—— (1962). Determination of the optimum configuration for a Stirling engine. *J. mech. Engng Sci.,* **4,** pp. 203–12.
—— (1965). A thermodynamic analysis of the Stirling cycle and a comparison with experiment. S.A.E. Paper No. 949B, Int. Auto. Eng. Congress, Detroit, Michigan, Jan.
KITZNER, E. W. (1977a). Stirling engine feasibility study of an 80–100 hp engine and of improvement potential for emissions and fuel economy. NASA/DOE Report No. COO/2631-22, NTIS, Springfield, Va.
—— (1977b). The Ford/Philips Stirling engine programme. Proc. ERDA Adv. Auto Power Syst. Cont. Coord. Mtg., Dearborn, Mich., Oct., NTIS, Springfield, Va.
KNOOS, S. (1972). *Method and device for hot gas engine or gas refrigeration machine.* U.S. Patent 3 698 182.
KOENIG, K. (1966). *An investigation into a means of approaching isothermal processes applicable in Stirling cycle machines.* Master's Thesis, M.I.T.
KÖHLER, J. W. L. (1956). *Refrigerator gas liquification device,* U.S. Patent 2 734 354.
—— (1957). *Hot-gas reciprocating engine for refrigerating.* U.S. Patent 2 784 570.
—— (1959). *Single-acting hot-gas reciprocating engine.* U.S. Patent 2 872 779.
—— (1960). The gas refrigerating machine and its position in cryogenic technique. *Prog. Cryog.,* **2,** pp. 41–67.
—— (1965). The Stirling refrigeration cycle. *Scientific American* **212,** No. 4, pp. 119–127.
—— (1968). Computation of the temperature field of regenerators with temperature-dependent parameters. Proc. Int. Cryogenic Conf., Brighton.

—— *and* BLOEM, A.T. (1957). *Cold gas refrigerator.* U.S. Patent 2 781 647.
—— *and* JONKERS, C. O. (1954a). Gas refrigerating machine. *Philips Tech. Rev.*, **16,** pp. 69–78.
—— —— (1954b). Construction of a gas refrigerating machine. *Philips Tech. Rev.*, **16,** pp. 105–15.
—— —— (1955). Fundamentals of the gas refrigerating machine. *Philips Tech. Rev.*, **16,** pp. 69–78.
—— *and* SCHALKWIJK, W. F. (1956). *Cold gas refrigerator.* U.S. Patent 2 750 765.
KOHLMAYER, G. F. (1967). Extension of the maximum slope method for arbitrary upstream fluid temperature changes. ASME Paper No. 67-HT-79, Aug.
KOIZUMI, I. (1976). Development of Stirling engines. *Jnl Jap. Soc. Mech. Eng.*, Vol. **79,** No. 693, Aug.
KOLIN, I. (1968). The Stirling cycle with nuclear fuel. *Nuclear Eng.*, **13,** pp. 1027–1034.
KOOPMANS, A. (1952). *Hot-Gas Reciprocating Engine.* U.S. Patent 2 618 923.
KORTH, M. W., ASHBY, H. A. *and* STAHMAN, R. C. (1972). Emission measurement techniques for non-conventional powerplants. *Proc. 7th I.E.C.E.C.*, Ann Arbor, Mich.
KOVTON, I. M., NAUMOV, A. M. *and* NESTERENKO, V. B. (1967). The Stirling cycle in dissociating gas. *Int. Chem. Eng.*, **7,** pp. 608–610.
KRASICKI, B. R. *and* PIERCE, B. L. (1977). Heat transport research and development for the nuclear-powered artificial heart. *Proc. 12th I.E.C.E.C.*, pp. 119–125, Washington, D.C., Aug. 28–Sept. 2.
KRASIN, A. K. (1971). Physical and technical principles of creating atomic power stations with fast neutron gas reactors cooled by dissociating N_2O_4. Proc. 4th U.N. Int. Conf. on Peaceful Uses of Atomic Energy, Geneva, Sept.
—— *and* NESTERENKO, V. B. (1967). Thermodynamic and transfer properties of chemically reacting gas systems. Nauka & Technica, Minsk.
KRAUTER, A. I. (1978). Hydrodynamic lubrication of Stirling engine rod seals, DOE Hwy. Veh. Syst. Cont. Coord. Mtg., Troy, Mich.
KUHLMAN, P. (1973). *MTZ Motortech Z.*, **34,** pp. 135–9.
—— (1974). Stirling engine for vehicle propulsion. *VDI-Berichte.*
—— *and* HORST, Z. (1970). The Stirling engine—A new prime mover. MAN Research Eng. Manuf., pp. 56–60, Sept.
—— *and* ZAPF, H. (1970). Kraftsmaschine Stirlingmotor. *VDI-Nachrichten*, Nr. 12.25.3.70.5.18.
LAING, N. (1975). *Thermal power plant.* U.S. Patent 3 894 395.
LAMBECK, A. J. J. (1955). *Hot-Gas Reciprocating Engine and Refrigerator or Heat Pump Operating on the Reversed Hot-Gas Reciprocating Engine Principle.* U.S. Patent 2 709 334.
LAMBERTSON, T. J. (1958). Performance factors of a periodic-flow heat exchanger. *Trans. Am. Soc. mech. Engrs*, **80,** pp. 586–592.
LANCE, J. R. *and* SELZ, A. (1968). An Implantable Artificial Heart power source. *Proc. 3rd I.E.C.E.C.*, pp. 758–765, Boulder, Col., Aug. 14–16.
LANCHESTER, F. W. (1898). *Improvements in Fluid Pressure Engines.* British Patent 10836.
LAPEDES, D. E., HINTON, M. G. *and* MELTZER, J. (1974). Current status of alternative automotive power systems and fuels. Aerospace Corp., El Segundo, Cal.
— *and* MELTZER, J. (1974). Status review of hybrid heat engine/battery and heat engine. Aerospace Corp., El Segundo, Cal.

LAVIGNE, P. (1973). *Driving device of the Stirling-cycle relaxation type for an implantable artificial heart.* U.S. Patent 3 766 568.
LEACH, C. E. *and* FRYER, B. C. (1968). Radioisotope energized undersea Stirling engine. *Proc. 3rd I.E.C.E.C.*, pp. 830–844, Boulder, Col., Aug. 14–16.
LEE, H. (1969). Development of satisfactory long term percutaneous leads. *Proc. First Artificial Heart Conf.*, pp. 793–808, Washington, D.C., June.
LEE, K. (1976). *The Stirling cycle with adiabatic compression and expansion.* M.Sc. Thesis, University of Calgary.
LEE, R. (1937). *Heat engine.* U.S. Patent 2 067 453.
LEEDER, W. (1975). *Simple and cheap heat pump for heating.* German Patent No. 2 360 585.
LEETH, G. G. (1969). Energy conversion devices for ground transportation. *Proc. 4th I.E.C.E.C.*, pp. 933–939, Washington, D.C., Sept. 21–26.
LEFEBVRE, A. H. (1974). Pollution control in continuous combustion engines. *Proc. 15th Int. Sym. on Combust.*, pp. 1169–1180, Aug. 25–31.
LEHRFELD, D. (1977a). System analysis design and proof of concept experiment on a total-energy system, DOE Report no. COO-2947-3 (available from NTIS).
—— (1977b). Practicability study of Stirling total energy systems. *Proc. 12th I.E.C.E.C.*, Washington, D.C., pp. 1504–1511, Aug. 26–Sept. 2.
—— *and* GOLDOWSKY, M. (1977). Artificial heart thermal converter component research and development. *Proc. 12th I.E.C.E.C.*, Paper No. 779019, Washington, D.C., Aug. 28–Sept. 2.
LIA, T. A. (1971). Stirlingmotoren-Miljovennlig, Energibesparende-et Alternative Til Dagens Diesel—Og Ottomotorer. *Masken, Norway*, Vol. **42,** pp. 23–27.
—— (1973). The Stirling engine. *Combustion Engine Progress*, pp. 44–47.
—— *and* LAGEROVIST, R. S. G. (1973). Stirling Engine with Uncónventional Heating System. *Proc. 8th I.E.C.E.C.* pp. 165–173, Philadelphia, Pa., Aug. 13–17.
LIANG, C. Y. *and* YANG, W. J. (1975). Modified Single-Blow Technique for Performance Evaluation on Heat Transfer Surfaces. *J. Heat Transfer* (ASME), **97,** pp. 16–21.
LIENESCH, J. H. *and* WADE, W. R. (1968). Stirling engine progress report—Smoke, odor, noise and exhaust emission. S.A.E. Paper No. 680081, pp. 292–307, Jan. 8–12.
—— —— (1969). Stirling engine operating quietly with almost no smoke and odor, *S.A.E. Jnl*, **77,** pp. 40–44, Jan.
LINDSLEY, E. F. (1975). Air-conditioning cold from any source of heat. *Popular Science.*
LIVINGSTON, R. P., FRYKBERG, D. G. *and* RALLIS, C. J. (1978). A generalized computer based kinematic dynamic simulation of single degree of freedom Stirling cycle planar mechanims. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.
LOCKE, G. L. (1950). Heat transfer and flow-friction characteristics of porous solids. Dept. of Mech. Eng., Stanford Univ., Technical Report No. 10.
LOFTUS, G. H. (1964). Performance of model 4-S1210 Stirling cycle engine—Assignment 71–101. U.S. Navy MEL Evaluation Report No. 158/64. Oct.
LONGMORE, D. (1971). *The heart.* World Univ. Library, Weidenfield, and Nicolson, London.
LONGSWORTH, R. C. (1966). *An analytical and experimental investigation of pulse tube refrigeration.* Ph.D. Thesis, Syracuse Univ., June.

LOWE, J. F. (1976). Liquid piston oscillates naturally. *Design News*, **32,** Feb. 23.
LUDVIGSEN, K. (1972a). The Stirling: Ford's engine for the eighties? *Motor*, Sept. 9.
—— (1972b). The engine of the 1980s—Stirling's 'Mr Clean' images lies behind Ford-Philips deal. *Ward's Auto. World*, pp. 41–45, Sept.
—— (1973). Stirling engine—History and current development of another possible alternative to the internal combustion engine. *Road & Track*, **24,** No. 7, p. 83, Mar.
LUECK, R., DAMSZ, G. *and* DANIELS, A. (1967). Adaptation of rolling-type seal diaphragms to miniature Stirling-cycle refrigerators. Report No. AFFDL-TR-67-96, WPAFB, Ohio.
LYAPIN, V. I., PRUSMAN, Y. O. *and* BAKHNEV, V. G. (1975). Effect of efficiency of the end heat exchanger on the start-up period of a helium cooler. *Khim. and Neft. Mashinostr, U.S.S.R.*, **11** (9), Sept; also *Chem. Pet. Eng.*, **11** (9–10), Sept.
MCCARTNEY, J. F. *and* CATES, M. A. (1975). Power sources for remote ocean-oriented applications. *Proc. 10th I.E.C.E.C.*, Newark, N.J., pp. 1318–1327.
MCLEAN, A. F. (1975). Brittle materials design for high temperature gas turbine. AMMRC CTR 75-8, by Ford Motor Co. on Contract no. DAAG-46-71-C-0162.
MCMAHON, H. O. *and* GIFFORD, W. E. (1959). *Fluid expansion refrigeration method and apparatus.* U.S. Patent 2 906 101.
—— —— (1960). A new low-temperature gas expansion cycle. *Adv. Cryog. Eng.*, **5,** pp. 345-72.
MAGEE, F. N. *and* DOERING, R. D. (1968). Vuilleumier-cycle cryogenic refrigerator development. Report no. AFFDL-TR-68-67.
MAKI, E. R. *and* DEHART, A. O. (1971). A new look at swash-plate drive mechanism. S.A.E. Paper no. 710829.
MALAKER, S. F. *and* DAUNT, J. G. (1963). *Miniature cryogenic engine.* U.S. Patent 2 074 244.
MALIK, M. J. (1968). *Stirling-cycle drive for an electrokinetic transducer.* U.S. Patent 2 074 244.
MALLETT, T. (1973). The Robinson hot air engine. *Model Engr.*, **139,** p. 160.
MALONE, J. F. J. (1930). A new prime mover. *Jnl roy. Soc. Arts*, **79,** pp. 679–709.
—— (1931). A new prime mover. *Engineer, Lond.*, pp. 97–101.
MARCINIAK, T. J., BRATIS, J. C. *and* DAVIS, A. (1978). Total energy technology alternative studies. Argonne National Lab. Report, DOE, available NTIS.
MARTINI, W. R. (1968a). A Stirling engine module to power circulatory-assist devices. ASME Paper no. 68-WA/Ener-2.
—— (1968b). Stirling-cycle energy converter to power circulatory support systems. MDAC Annual Report no. DAC-60763.
—— (1969). Implanted energy conversion system. MDAC Annual Report, No. MDC G2025, 1968–9.
—— (1971). Implanted energy conversion system. MDAC Annual Report, No. PH43-67-1408-4, 1970–71.
—— (1972). Developments in Stirling engines. ASME Paper No. 72-WA/Ener-9, or MDAC Paper WD 1833, Nov.
—— (1974). Development and evaluation of a modified Stirling-cycle Engine. MDAC Quarterly Report, No. MDC G4438, April.
—— (1975a). The free-displacer, free-piston Stirling engine—Potential Energy Conserver. *Proc. 10th I.E.C.E.C.*, pp. 995–1002, Newark, N.J., Aug. 17–22.
—— (1975b). An efficient Stirling heat engine—heat pump. *Proc. 10th I.E.C.E.C.*, Paper No. 7506130, Newark, N.J., Aug. 17–22.

—— (1977a). Stirling Engine Research Institute Newsletter. No. 1, Feb. Subsequently issued at Irregular Intervals, Publ. Joint Centre for Grad. Studies, Richland, Wash.
—— (1977b). Biomedical power. *Proc. 12th I.E.C.E.C.*, Washington, D.C., pp. 102–3.
—— (1978a). Design manual for Stirling engines. DOE/NASA Report, available from NTIS.
—— (1978b). Comparison of calculation procedures for Stirling engines with experimental measurements. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–3.
—— (1978c). Status of Stirling machine design techniques. *Inst. of Gas. Tech. Seminar, Stirling Cycle Prime Movers*, Chicago, June.
—— (1978d). A Stirling engine design analysis manual. DOE Hwy Veh. Syst. Cont. Coord. Mtg., Troy, Mich.
—— HAUSER, S. G. *and* MARTINI, M. W. (1977). Evaluation of isothermalized Stirling engines. *Proc. 12th I.E.C.E.C.*, Washington, D.C., pp. 1496–1503.
—— EMIGH, S. G., WHITE, M. A., GRIFFITH, W. R., HINDERMAN, J. D., JOHNSTON, R. P. *and* PERRONE, R. E. (1974). Unconventional Stirling engines for the artifical heart application. *Proc. 9th I.E.C.E.C.*, San Francisco, p. 791.
—— JOHNSTON, R. P., GORANSON, R. B. *and* WHITE, M. A. (1968). Stirling engine to power circulatory-assist devices. *Proc. 3rd I.E.C.E.C.*, Boulder, Col., p. 733.
—— —— —— —— (1970). A simplified Stirlng engine to power circulatory-assist devices. *Isotope Rad. Tech.*, **7,** pp. 145–60.
—— —— *and* NOBLE, J. E. (1969). Mechanical engineering problems in energetics—Stirling engines. ASME paper no. 69-WA/Ener-15, or MDAC paper WD 1109.
—— —— —— (1968b). The thermocompressor and artificial heart power. MDAC paper no. 10, p. 177.
—— RIGGLE, P. *and* HARMISON, L. T. (1972). Radioisotope-fuelled Stirling engine artificial heart system. *Nucl. Technol.*, **13,** pp. 194–208.
—— WHITE, M. A. *and* DESTEESE, J. G. (1974). How unconventional Stirling engines can help conserve energy. *Proc. 9th I.E.C.E.C.*, San Francisco, pp. 1092–9.
—— —— *and* GASPER, K. A. (1972). Stepz power source concept for space applications. *Trans. Am. Nucl. Soc.*
MARUSAK, T. J. *and* CHIU, W. S. (1978). The performance of a free piston Stirling engine coupled with a free piston linear compressor. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.
MASSA, D. J. (1974). High efficiency compressors—Design and application. Proc. Conf. on Improving Efficiency on Equipment for Residential and Small Commercial Buildings, Purdue Univ., Oct. 7–8.
MATTAVI, J. N., HEFFNER, F. E. *and* MIKLOS, A. A. (1969). The Stirling engine for underwater vehicle applications. S.A.E. Paper No. 690731, pp. 2376–2400, or General Motors Research Publ. No. GMR-936.
MEEK, R. M. G. (1961). The measurement of heat-transfer coefficients in packed beds by the cyclic method. *Int. Heat-Trans. Conf.*, ASME, Boulder, Colorado, pp. 770–780.
MEIJER, R. J. (1957a). *Device for Circulating a Medium in a Reciprocating Engine.* U.S. Patent 2 781 632.
—— (1957b). *Hot-Gas Reciprocating Apparatus.* U.S. Patent 2 794 315.
—— (1958). *Hot-Gas Reciprocating Engine.* U.S. Patent 2 828 601.

—— (1959a). The Philips hot-gas engine with rhombic drive mechanism. *Philips Tech. Rev.*, **20,** pp. 245–276.
—— (1959b). *Hot-gas reciprocating apparatus.* U.S. Patent 2 867 973, Jan. 13.
—— (1959c). *Hot-gas reciprocating machine of the displacer-piston type.* U.S. Patent 2 885 855, May 12.
—— (1960a). *The Philips Stirling thermal engine.* Ph.D. Thesis, Technical University, Delft.
—— (1960b). *Thermo-dynamic reciprocating apparatus.* U.S. Patent 2 951 334, Sept. 6.
—— (1960c). *Thermo-dynamic reciprocating apparatus.* U.S. Patent 2 959 019, Nov. 8.
—— (1960d). *Hot-gas reciprocating engine.* U.S. Patent 2 963 854, Dec. 13.
—— (1960e). *Thermo-dynamic reciprocating apparatus.* U.S. Patent 2 963 871, Dec. 13.
—— (1961). The Philips thermal engine. Philips Research Reports Supplements, No. 1, Eindhoven.
—— (1965). Philips Stirling engine activities. S.A.E. Paper No. 949E, Int. Auto. Eng. Cong., Detroit, Michigan.
—— (1966). Philips Stirling Engine Activities, *S.A.E. Transactions*, pp. 18–32.
—— (1968). Der Philips-Stirlingmotor. *MTZ Motortechnische Zeitschrift*, **29,** pp. 284–298.
—— (1969a). Combination of electric heat battery and Stirling engine. *Denkschrift Elektrospeicherfahrzeuge*, **11,** pp. 143–164.
—— (1969b). Mit Elektro-Warme Speicher und Stirlingmotor. *Denkschrift*, **11,** pp. 143–164, Elektrofahrzeuge, Deutsche Forschungsgemeinschaft.
—— (1969c). Philips Stirling Motor—Varmgasmotor med Utveckling-smojligheter. *Teknisk Tidskrift*, **99,** pp. 373–378.
—— (1969d). The Philips Stirling engine. *De Ingenieur*, **8,** No. 18, pp. W69–W79, and pp. W81–W93.
—— (1969e). Rebirth of the Stirling engine. *Science Journal*, **5A,** No. 2, pp. 31–39.
—— (1970a). Prospects of the Stirling engine for vehicular propulsion. *Philips Tech. Rev.*, **31,** pp. 168–185.
—— (1970b). The Philips Stirling engine as a propulsion engine. *Proc. 5th I.E.C.E.C.*, Las Vegas, Nevada, Sept. 20–25.
—— (1971). Towards cleaner air—A passenger coach powered by a Philips Stirling engine. Information Sheet, Philips Research Labs., Eindhoven, Netherlands, Jan.
—— (1972). Moglichkeiten des Stirling-Fahrzeugmotors in unserer kunftigen Gesellschaft. *Schwezerische Technische Zeitung*, STA **69,** 31/32, pp. 649–660.
—— (1978). Advanced automotive Stirling concepts. DOE Hwy Veh. Syst. Cont. Coord. Mtg., Troy, Mich., May.
—— *and* Michels, A. P. J. (1978). A variable displacement Stirling engine for automotive propulsion. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.
—— *and* Spigt, C. L. (1974). The potential of the Philips Stirling engine for pollution reduction and energy conservation. *Second Symposium on Low Pollution Power Sys. Devel.*, pp. 1–12, Dusseldorf, W. Germany, Nov. 4–8.
Meltzer, J. *and* Lapedes, D. (1971). Hybrid heat engine/electric systems study. Aerospace Corp., El Segundo, Cal., June 1.
Michels, A. P. J. (1971). The NO_x content in the exhaust gases of Stirling engine. *Proc. 6th I.E.C.E.C.*, Paper No. 719134, Boston, Mass., August 2–6.

—— (1972). C.V.S. test simulation of a 128 kW Stirling passenger car engine. *Proc. I.E.C.E.C.*, pp. 875–886, San Diego, Ca.
—— (1976). The Philips Stirling engine: a study of its efficiency as a function of operating temperatures and working fluids. *Proc. 11th I.E.C.E.C.*, Paper No. 769258, Lake Tahoe, Nev., Sept. 12–17.
—— *and* Philips, N. J. (1972). C.V.S. test simulation of a 128 kW Stirling and other passenger car engines. Glocilampen Fabrieken, Eindhoven, Netherlands.
Minaichev, V. E. *and* Zukov, V. M. (1974). Cryocondensation Booster Pump. Instrum. Exp. Tech., Vol. 17, No. 3, Part 2, pp. 795–798, May–June.
Model Engineer. (1977). Hot-air engine competition. *Model Engr*, **143,** pp. 708–809.
Moise, J. C. *and* Faeser, R. J. (1977). Thermocompressor powered artificial heart, *Proc. 12th I.E.C.E.C.*, Washington, D.C., pp. 112–18.
—— —— *and* Rudnicki, M. I. (1974). Status of a thermocompressor artificial heart system. *Proc. 9th I.E.C.E.C.*, San Francisco, pp. 799–804.
—— —— —— (1973). Development of a thermocompressor power system for artificial heart. *Proc. 8th I.E.C.E.C.*, Philadelphia, pp. 511–35.
—— —— —— (1975). Thermocompressor artificial heart system. *Proc. 10th I.E.C.E.C.*, Newark, N.J., pp. 1242–5.
—— —— *and* Russo, V. F. (1976). Thermocompressor powered artificial heart assist system. *Proc. 11th I.E.C.E.C.*, Lake Tahoe, p. 150.
Mondt, J. R. (1964). Vehicular gas turbine heat exchanger temperature distributions *J. Engng Pwr* (ASME), **86,** series A, pp. 121–6.
Monson, D. S. *and* Welsh, H. W. (1962). Allison adapting Stirling engine to one-year-in-space operation. *S.A.E. Jl* No. 70, pp. 44–51.
Moon, J. R. (1972). European progress with Stirling engines. *Diesel and Gas Turbine Progress*, pp. 74–77, Sept–Oct.
Morash, R. T. *and* Marshall, O. W. (1974). The Roesel closed cycle heat engine. *Proc. 9th I.E.C.E.C.*, pp. 1117–1124, San Francisco, Cal., Aug. 26–30.
Morgan, D. (1978). Thermo Electron conceptual design study of thermal energy storage for a Stirling car. DOE Hwy. Veh. Syst. Cont. Coord. Mtg., Troy, Mich., May.
Morgan, N. E. (1972). Analysis and preliminary design of airborne air liquefiers. Report No. AFFDL-TR-71-171, Feb.
Morgenroth, H. (1966). *Stirling Cycle Engine Divided into a Pressure Generating Unit and Energy Converting Unit.* U.S. Patent 3 248 870.
Moritz, K. *and* Pruscheck, R., (1969). Grenzen des Energietransports in Warmerohren. *Chemie Ing. Technik*, **45,** No. 1–2.
Mortimer, J. (1975). Low Cost, Long Life (Stirling) Engine May be Ideal for the Third World. *Engineer, Lond.*, **240**, March.
—— (1976). Alternative Engines for Tomorrow's Car. *Engineer, Lond.*, **242,** May.
Mott, W. E. (1975). Nuclear power for the artificial heart. *Biomater. Med. Dev. Artif. Organs*, **3,** No. 2, pp. 181–191.
—— Cole, D. W. *and* Holman, W. S. (1976). The U.S. Atomic Energy Commission nuclear-powered artificial heart programme. *Proc. 2nd Int. Symp. on Power from Radioisotopes*, pp. 713–730, May 20–June 1, Madrid.
Mullins, P. J. (1975). Progress on the Stirling engine. *Automot. Inds*, **152,** pp. 32–34, Feb. 1.
Murray, J. A., Martin, B. W., Bayley, F. J. *and* Rapley, C. W. (1961). Performance of thermal regenerators under sinusoidal flow conditions. *Int. Heat-Trans. Conf.*, ASME, Boulder, Colorado, pp. 781–796.

NASA (1977). A Second Stirling Engine Development Team. U.S. Govt. *Commerce Business Daily*, Issue No. PSA-6921, Sept. 29.

NEELEN, G. T. M. (1967). De Philips-Stirling Motor. Van Klassiek Idee/Tot Moderne Kractbron, Ship en werf, **17**.

—— (1970). Vacuum brazing of complex heat exchangers for the Stirling engine. *Welding Jnl*, **49**, pp. 381–386, May.

—— (1971). Precision castings advance the development of the Stirling engine. *Giesserei*, **58**, pp. 166–170, April 8.

—— ORTEGREN, L. G. H., KUHLMANN, P. *and* ZACHARIAS., F. (1971). Stirling engines in traction applications. C.I.M.A.C., 9th Int. Congress on Combus. Eng., Paper A26, Stockholm, Sweden.

NEWGARD, P. M. *and* EILERS, F. J. (1969). Skin transformer and power conditioning components. *Proc. First Artificial Heart Conf.*, pp. 927–936, Washington, D.C., June.

NEWHALL, H. K. (1974). Low emissions combustion engines for motor vehicles. *Am. Chem. Soc., Div. Fuel Chem.*, **19**, pp. 15–37.

NEWTON, A. .B (1957). *Refrigeration compressor.* U.S. Patent 2 803 951.

—— (1959). *Hot fluid engine with movable regenerator.* U.S. Patent 2 907 169.

—— (1959). *Refrigerant Engine and Work Device.* U.S. Patent 2 909 902.

—— (1961). *Hot Gas Refrigeration System.* U.S. Patent 2 993 341.

NOBLE, J. E., RIGGLE, P., EMIGH, S. G. *and* MARTINI, W. R. (1974). *Heat engine.* U.S. Patent 3 855 795.

—— *and* JOHNSTON, R. P. (1974). *Energy converter to power circulatory support systems.* U.S. Patent 3 788 772.

—— RIGGLE, P., EMIGH, S. G. *and* MARTINI, W. R. (1976). *Heat Engine.* U.S. Patents 3 949 554 and 3 956 895.

NOBREGA, A. C. (1965). *A harmonic analysis of Stirling engines.* Thesis, M.I.T.

NORBYE, J. P. *and* DUNNE, J. (1973). Ford's gamble—Stirling-cycle engine promises low emissions without add-ons. *Popular Science*, pp. 72–75 & 154, Feb.

NORMAN, J. C., HARMISON, L. T. *and* HUFFMAN, F. N. (1972). Nuclear-fueled circulatory support systems. *Arch. Surg.*, **105**, Oct.

NUSSELT, W. (1927). Die Theorie des Winderhitzers. *Z. Verb. Dt. Ing.*, **71**, p.85.

—— (1928). Der Beharrungszustand im Winderhitzer. *Z. Verb. Dt. Ing.*, **72**, p. 1052.

NYSTROEM, P. H. G. (1975). *Method and system to control the output of a Stirling motor.* German Patent 2 449 742.

ORGAN, A. J. (1970). Strling engine power and transmission. *Jnl Auto. Eng.*, pp. 9–16, July.

—— (1971a). Analytical optimization of specific performance of a Stirling cycle machine. Unpublished note, King's College, University of London.

—— (1971b). The Stirling-cycle regenerator thermal machine as a low-pollution prime mover. 2nd Brasilian Symposium on Heat Transfer and Fluid Mechanics, Beb Horiteute (obtainable from Universidade Federal de Minar Gerais, Beb Horiteute, Brazil).

—— (1973a). Some theoretical and practical aspects of the design of Stirling cycle engines. Proc. Stirling Engine Seminar, Lecture Series No. 53, Von Karmen Institute for Fluid Mechanics, Brussels, Belg., Feb. 12–16.

—— (1973b). Stirling engine beats pollution's threat. *Design Engineering*, pp. 43–46, May.

—— (1975). The concept of 'critical length ratio' in heat exchangers for Stirling cycle machines. *Proc. 10th I.E.C.E.C.*, pp. 1012, Neward, N.J., Aug. 17–22.

—— (1976). Fluid particle trajectories in Stirling cycle machines. Research Report, King's College, London.

—— (1977a). Latent pressure waves in Stirling engines. Research Report, King's College, London, Sept.

—— (1977b). The isothermal Stirling cycle with arbitrary equation of state. Research Report. King's College, London.

—— (1977c). The use of bulk coefficients of convective heat transfer in the simulation of the practical Stirling cycle by digital computer. Research Report, King's College, London.

—— (1978). Mechanical efficiency of a rhombic-drive Stirling cycle machine. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.

ORTEGREN, L. G. (1971a). Stirling engines activities at United Stirling (Sweden). Tech. Report Conf. on Low Pollution Power Systems Development, Eindhoven, Netherlands.

—— (1971b). Svsnsk Stirlingmotor i Produktion 1976. *Teknisk Tidskrift*, **101** (3), pp. 44–50.

—— (1973a). Development of Stirling Engines in Sweden. United Stirling Conf. on Low Pollution Power Systems Development, Ann Arbor, Mich., Oct. 14–191.

—— (1973b). Noise Emission of Stirling Engines and Overall Vehicle Noise. Conf. on Low Pollution Power Systems Development, Ann Arbor, Mich., Oct. 14–19.

—— (1974). Stirling Motorn-realistiskt alternativ fr 80-talets Jordon. *Tecknisk Tidskrijt*, **104** (6), pp. 24–27, 40.

—— HENRIKSSON, L. *and* LIA, T. (1971). Stirlingmotorn och dess potential i militara System. *Militarteknisk Tidskrift*, **40** (2), pp. 5–19.

OTTEN, E. H. (1956). Tests on a displacer-type Stirling engine using internal combustion. Shell Thornton Report K140.

PAKULA, A. (1950). Kylmailmakoneet U U dessa Kchitysvaiheessa. *Teknillinen Aikakausleptie*, **40** (6), pp. 123–127.

PARKER, M. D. *and* MALIK, M. J. (1962). *Investigation of a 3 kW Stirling cycle solar power system.* Vol. 1–10, W.A.D.D. TR-61-122, Vol. 1 Engine Design, Vol. 10 Experimental Evaluation.

—— *and* SMITH, C. L. (1960). Stirling engine development for space power. ARS Paper No. 1315–60, Space Power Systems Conf., Santa Monica, Ca., Sept.

PATTERSON, D. J. *and* BOLT, J. A. (1968). Low pollution heat engines. *Proc. 3rd I.E.C.E.C.*, pp. 779–787, Boulder, Col., Aug, 14–16.

PATTERSON, M. F., *et al.* (1975). Improved multi-layer insulation for compact high temperature power source. *Proc. 10th I.E.C.E.C.*, pp. 1554, Newark, N.J., Aug. 17–22.

PEDROSO, R. I. (1974). Control of elongation in the rolling diaphragm seal under cyclic pressure loading. *Proc. Joint Automatic Control Conf.*, pp. 539–548, Univ. Texas, Austin, Texas.

—— (1976). Stirling engine: engineering considerations in view of future needs. *Proc. 11th I.E.C.E.C.*, Paper No. 769257, Lake Tahoe, Nev., Sept. 12–17.

PENN, A. W. (1974). Small Electrical Power Sources *Phys. Tech.*, May.

PERCIVAL, W. H. (1967). The Stirling engine for naval applications. *Proc. Conf. on Energy Sources*, NRC/CUW, 0340, pp. 196–209, Washington, April.

—— (1974). Historical review of Stirling engine development in the United States from 1960 to 1970. ERDA, No. NASA CR-121097.

—— (1976). Stirling cycle engines. *Proc. 11th I.E.C.E.C.*, pp. 1496–1497, (Chairman introduction), Lake Tahoe, Nev., Sept. 12–17.

—— (1978). United Stirling program for power generation and automotive applications. *Inst. of Gas Tech. Seminar, Stirling Cycle Prime Movers*, Chicago, Ill., June.
Philips Co. (1943). *Improvements in or relating to hot-gas engines.* British Patent 697 157.
—— (1946). *Improvements in or relating to hot-gas engines.* British Patent 630 429.
—— (1948a). *Improvements in or relating to hot-gas reciprocating engines.* British Patent 605 992.
—— (1948b). *Improvements in or relating to closed cycle hot-gas engines.* British Patent 606 758.
—— (1949a). *Improvements in or relating to cylinder heads for hot-gas engines.* British Patent 615 260.
—— (1949b). *Improvements in or relating to hot-gas reciprocating engines.* British Patent 617 850.
—— (1949c). *Improvements in or relating to hot-gas reciprocating engines.* British Patent 618 266.
—— (1949d). *Improvements in or relating to hot-gas engines.* British Patent 619 277.
—— (1949e). *Improvements in systems comprising a hot-gas reciprocating engine.* British Patent 623 090.
—— (1949f). *Improvements in or relating to hot-gas engines.* British Patent 630 428.
—— (1949g). *Improvements in or relating to hot-gas reciprocating engines.* British Patent 632 669.
—— (1950a). *Improvements in or relating to heat-exchanging apparatus.* British Patent 635 691.
—— (1950b). *Improvements in or relating to hot-gas reciprocating engines.* British Patent 637 719.
—— (1950c). *Improvements in or relating to hot-gas engines.* British Patent 645 934.
—— (1951a). *Improvements in and relating to hot-gas engines.* British Patent 648 742.
—— (1951b). *Improvements in hot-gas engines.* British Patent, 654 625.
—— (1951c). *Improvements in or relating to hot-gas reciprocating engines.* British Patent 654 936.
—— (1951d). *Improvements in or relating to reciprocating engines.* British Patent 654 940.
—— (1951e). *Improvements in or relating to hot-gas reciprocating engines.* British Patent 655 565.
—— (1951f). *Improvements in or relating to the control of hot-gas reciprocating engines.* British Patent 655 935.
—— (1951g). *Improvements in or relating to hot-gas reciprocating engines and reciprocating refrigerators.* British Patent 656 250.
—— (1951h). *Improvements in or relating to multi-cylinder piston machines.* British Patent 656 252.
—— (1951j). *Improvements in and relating to thermal regenerators.* British Patent 657 472.
—— (1951k). *Improvements in reciprocating hot-gas engines.* British Patent 657 743.
—— (1952a). *Improvements in or relating to hot-gas engines.* British Patent 665 735.

—— (1952b). *Improvements relating to hot-gas reciprocating engines and refrigerating engines.* British Patent 665 989.
—— (1952c). *Improvements in or relating to hot-gas engines or engines operating on the reverse hot-gas engine principle.* British Patent 666 313.
—— (1952d). *Improvements in or relating to hot-gas engines.* British Patent 668 061.
—— (1952e). *Improvements in or relating to refrigeration machines operating on the reversed hot-gas engine principle.* British Patent 668 621.
—— (1952f). *Improvements in or relating to hot-gas engines.* British Patent 669 891.
—— (1952g). *Improvements in or relating to hot-gas reciprocating engines.* British Patent 675 161.
—— (1952h). *Improvements in refrigerating systems.* British Patent 680 241.
—— (1952i). *Improvements in or relating to reciprocating apparatus.* British Patent 684 394.
—— (1952j). *Improvements in or relating to hot-gas reciprocating engines and refrigerating engines.* British Patent 684 600.
—— (1953a). *Improvements in or relating to hot-gas reciprocating engines and engines operating on the reversed hot-gas engine principle.* British Patent 690 026.
—— (1953b). *Improvements in hot-gas engines.* British Patent 691 781.
—— (1953c). *Improvements in hot-gas engines.* British Patent 691 784.
—— (1953d). *Improvements in hot-gas reciprocating engines.* British Patent 691 785.
—— (1953e). *Improvements in heat regenerator filling material consisting of wire material.* British Patent 695 271.
—— (1953f). *Improvements in or relating to hot-gas engines.* British Patent 697 082.
—— (1953g). *Improvements in or relating to heat exchangers.* British Patent 702 132.
Phillips, J. B. *and* Reid, T. J. (1974). An investigation of the effect of an exhaust gas recuperator on the performance of a Stirling engine. Project Report No. 278, Sch. Engineering, Univ. of Bath, U.K.
Pillar, S. (1978). Free-piston Stirling systems for power generation and heat pumps. *Inst. of Gas Tech. Seminar, Stirling Cycle Prome Movers*, Chicago, Ill., June.
Pitcher, G. K. (1975). Spacecraft Vuilleumier cryogenic refrigerator development. AFFDL Report No. AFFDL-TR-75-114, W.P.A.F.B., Ohio, Sept.
—— *and* du Pre, F. K. (1970). Miniature Vuilleumier-cycle refrigerator. Proc. Cryog. Eng. Conf., June.
Pitrolo, A. A., Morrow, R. B. *and* Arker, A. J. (1971). Multihundred Watt radioisotope thermoelectric generator. *Proc. 6th I.E.C.E.C.*, Boston, Mass., Aug. 2–6.
Polster, N. E. (1976). *Hot gas engine.* U.S. Patent 3 994 136.
—— *and* Martini, W. R. (1976). Self-starting, intrinsically controlled Stirling engine. *Proc. 11th I.E.C.E.C.*, pp. 1511–1518, Lake Tahoe, Nev., Sept. 17–19.
Postma, N. P. (1975). Review of Ford/Philips program. Highway Vehicle System Contractors Coordination Meeting, ERDA, Washington, D.C.
—— (1976). Stirling engine program. Highway Vehicle System Contractors Coordination Meeting, ERDA, Washington, D.C.
—— (1978). Ford Stirling engine powertrain development. DOE Hwy. Veh. Syst. Cont. Coord. Mtg., Troy, Mich.

—— VAN GIESSEL, R. *and* REININK, F. (1973). The Stirling Engine for Passenger Car Application. S.A.E. Paper no. 730648, Chicago.
POUCHOT, W. D. *and* DANIELS, A. (1974). Nuclear artificial heart bench model. *Proc. 9th I.E.C.E.C.*, pp. 782–790. San Francisco, Ca., Aug. 26–30.
—— BIFANO, N. J. *and* HANSON, J. P. (1975). Artificial heart system thermal converter and blood pump component research and development. *Proc. 10th I.E.C.E.C.*, pp. 1223–1231, Newark, N.J., Aug. 17–22.
—— *and* LEHRFELD, D. (1976). Nuclear-powered artificial heart system. *Proc. 11th I.E.C.E.C.*, pp. 157–162, Lake Tahoe, Nev., Sept. 12–17.
PRAST, G. (1963). A Philips gas refrigerating machine for 20 °K. *Cryogenics*, **3,** pp. 156–160.
—— (1964). A modified Philips-Stirling cycle for very low temperatures. *Int. Adv. Cryog. Eng.*, **10,** pp. 40–45.
—— (1965). A gas refrigerating machine for temperatures down to 20 °K and lower. *Philips Tech. Rev.*, Vol. **26,** pp. 1–11.
—— *and* REITDIJK, J. A. (1970). *Device for converting mechanical energy into heat energy or conversely.* U.S. Patent 3 487 635.
—— *and* DE JONGE, A. K. (1978). A free piston Stirling engine for small solar power plants. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.
PRON'KO, V. G., AMMAMCHYAN, R. G., GUILMAN, I. I. *and* RAYGORODSKY, A. E. (1976). Some problems of using absorbents as a matrix material for low-temperature regenerators of cryogenic refrigerators.
QVALE, E. B. (1963). *An analytical model of Stirling-type engines.* D.Sc. Thesis, M.I.T.
—— *and* SMITH, J. L. Jr. (1968). A mathematical model for steady operation of Stirling-type engines. *J. Engng Pwr* (ASME), **90,** pp. 45–50.
—— —— (1969). An approximate solution for the thermal performance of a Stirling engine regenerator. *J. Engng Pwr* (ASME), **91,** pp. 109–112.
RAAB, B., SCHOCK, A. *and* KING, W. G. (1975). Nuclear heat source for cryogenic refrigerators in space. *Proc. 10th I.E.C.E.C.*, Newark, N.J., Aug. 17–22.
RAETZ, K. (1974). Development and application of a Stirling heat pump for heating. Brauschweig, PTB-FMRB-57, Sept.
—— (1975). *Stirling heat pump for heating.* German Patent 2 400 256.
RAGSDALE, R. G. (1977). Stirling engine project status. Proc. ERDA Adv. Auto Power Syst. Cont. Coord. Mtg., Dearborn, Mich., Oct., NTIS, Springfield, Va.
—— (1978). Stirling engine project status. DOE Hwy. Veh. Syst. Cont. Coord. Mtg., Troy, Mich., May.
RALLIS, C. J. *and* URIELI, I. (1976). Optimum compression ratios of Stirling cycle machines. Univ. of Witwatersrand, Dept. of Mechanical Engineering, Report No. 68, p. 17, June.
—— URIELI, I. *and* BERCHOWITZ, D. M. (1977). A new ported constant volume external heat supply regenerative cycle. *Proc. 12th I.E.C.E.C.*, pp. 1534–1537, Aug. 28–Sept. 2, Wash., D.C.
—— *and* BERCHOWITZ, D. M. (1978). A computer and experimental simulation of Stirling cycle machines. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.
—— —— (1978). Optimized sizing of Stirling cycle machines based on a comprehensive second order analysis. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.
RANKINE, M. (1854). On the means of realizing the advantages of air engines. British Association, Proceedings of Annual Meeting.

RAPLEY, C. (1960). *Heat transfer in thermal regenerators.* M.Sc. Thesis, Durham University.
RAUCH, J. S. (1975). Steady state analysis of free-piston Stirling engine dynamics. *Proc. 10th I.E.C.E.C.*, pp. 961–965, Newark, N.J., Aug. 17–22.
—— (1978). Stirling cycle plant demonstrator. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.
REA, S. N. (1966). *A study of thermal regenerators subjected to rapid pressure and flow cycling.* Ph.D. Thesis, M.I.T.
—— *and* SMITH, J. L. Jr. (1967). The Influence of Pressure Cycling on Thermal Regenerators. *J. Eng. Industry* (ASME), **89,** Series B., pp. 563–569.
READER, G. T. (1978). The pseudo-Stirling cycle—A suitable performance criterion! *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.
REDSHAW, C. G. (1976). *Rotary Stirling Engine.* U.S. Patent 3 984 981.
REED, B. (1968). Hot-Gas Engines. *Gas and Oil Power*, **64,** pp. 8–11.
REES, R. A. (1920). *Improvements in hot-air engines.* British Patent 146 620.
REICH, S. A. (1978). Rotary piston configuration suitable for operation as a Stirling cycle machine. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.
REITLINGER, J. (1876). Uber Kreisprozesse Zwishchen Zwei Isothermen. *Zeit., des Ost. Ing, U. Arch. Ver.*
REULEAUX, F. (1876). *The kinematics of machinery.* (trans. A. B. W. Kennedy), London; new edn, Dover Books, New York, 1963.
RICARDO, H. Sir. (1966). Stirling Cycle Hot Air Engine. *Engineer, Lond.*, **221.** April 8.
RICE, G., DUNN, P.D. *and* THRING, R. H. (1975). Hydraulic and rotary drive Stirling engines with fluidised bed combustion/heat pipe systems. *Proc. 10th I.E.C.E.C.*, Paper No. 759141, Newark, N.J., Aug. 17–22.
RICHARDS, W. D. *and* AUXER, W. L. (1978). Performance of a Stirling engine powered, heat activated heat pump. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.
RICHARDSON, R. W. (1974). Automotive engines for the 1980s. *Am. Chem. Soc., Div. Fuel Chem.*, **19,** pp. 40–45.
RIETDIJK, J. A., VAN BEUKERING, H. C. J., VAN DER AA, H. H. M. *and* MEIJER, R. J. (1965). A positive rod or piston seal for large pressure differences. *Philips Tech. Rev.*, **26,** pp. 287–296.
RIGGLE, P., NOBLE, J., EMIGH, S. G., MARTINI, W. R. *and* HARMISON, L. T. (1971). Development of a Stirling engine power source for artificial heart application. MDAC Paper No. WD 1610, Sept.
RINIA, H. (1946a). New possibilities for the air engine. *Proc. Kon. Ned. Akad. Wet.*, Amsterdam, **49,** pp. 150–155.
—— (1946b). New possibilities for the air engine. Philips Gloeilampenfabrieken, Paper No. 1684.
—— *and* DU PRE, F. K. (1946). Air engines. *Philips Tech. Rev.*, **8,** pp. 129–136.
RIOS, P. A. (1970). *An analytical and experimental investigation of the Stirling cycle.* Ph.D. Thesis, M.I.T.
—— (1971). An approximate solution to the shuttle heat-transfer losses in a reciprocating machine. *J. Engng Pwr* (ASME), **93,** pp. 177–82.
—— QVALE, E. B. *and* SMITH, J. L. Jr. (1968). An analysis of the Stirling cycle refrigerator. *Cryog. Eng. Conf.*, pp. 332–342.
—— *and* SMITH, J. L. Jr. (1969). An analytical and experimental evaluation of the pressure-drop losses in the Stirling cycle. ASME Paper No. 69-WA/Ener-8, Vol. 92, Ser. A, No. 2, pp. 182–8, April.

ROGERS, C. F. C. *and* MAYHEW, Y. R. (1967). *Engineering thermodynamics: work and heat transfer.* Longman. 2nd Edition.
ROMIE, F. E. *and* AMBROSIO, A. (1966). Heat transfer to fluids flowing with velocity pulsations in a pipe. In *Heat Transfer, Thermodynamics and Education* (ed. H. Adelman), pp. 273–294, McGraw-Hill.
ROSENQVIST, K. (1978). In-vehicle Stirling engine operational experience. DOE Hwy. Veh. Syst. Cont. Coord. Mtg., Troy, Mich., May.
—— GUMMESSON, S. G. *and* LUNDHOLM, S. G. K. (1977). The development of a 150 kW (200 hp) Stirling engine for medium duty automotive application—A status report. S.A.E. Paper No. 770081.
ROSS, M. A. (1973). *Speed control apparatus for hot gas engine.* U.S. Patent 3 724 206.
—— (1976). Stirling performance. *The Columbus Dispatch Magazine,* July 18.
—— (1976). A rhombic drive Stirling engine. *Model Engr,* **142,** Parts I and II, pp. 760–762, Aug. 6, and pp. 796–799, Aug. 20.
—— (1977a). *Stirling cycle engines.* Solar Engines, Phoenix, p. 121.
—— (1977b). A Rider Stirling engine. *Model Engr,* **143,** pp. 160–161, Feb. 4.
RUSSER, W. A. *and* WISE, H. (1956). Thermal decomposition of nitrogen dioxide. *Jnl Chem. Phy.,* **25,** pp. 493–494.
SADVSKII, M. R., MAIKOV, V. P. *and* DLUKHIN, N. K. (1969). Investigation of the heat transfer and hydraulic resistance of gauze packing. *Int. Chem. Eng.,* **9,** pp. 592–595.
SANDQVIST, M., SMITH, L. M. *and* KOFF, W. J. (1975). Plutonium 238 as a heat source for the artificial heart. *Proc. 10th I.E.C.E.C.,* pp. 1473–1481, Newark, N.J., Aug. 17–22.
SARKES, L. A., NICHOLLS, J. A. *and* MENZER, M. S. (1977). Gas-fired heat pumps: An emerging technology. ASHRAE Jnl., Mar.
SAUNDERS, O. A. (1954). Heat transfer in regenerators. General Discussion on Heat Transfer, I. Mech. E., London.
—— *and* FORD, H. (1940). Heat transfer in the flow of gas through a bed of solid particles. *Jnl Iron Steel Inst.,* No. 1, p. 291.
—— *and* SMOLENIEC, S. (1948). Heat regenerators. *Proc. 7th Int. Cong. Appl. Mech.,* Vol. 3, pp. 91–105.
SCARINGE, R. P. (1978). *A theoretical analysis and experimental investigation of the Stirling cycle, with emphasis on the system as a heat pump.* Ph.D. thesis, Rensselaer Polytechnic Inst., N.Y., March.
SCHAB, H. A. (1964). Noise studies of Stirling cycle engines—assignment 71–101. U.S. Navy MEL R and D Report 152/64, Sept.
SCHALKWIJK, W. F. (1958). A simplified regenerator theory. ASME Paper No. 58-A-135.
—— DELANGE, L. *and* KOOPMANS, A. (1957). *Regenerator construction of a cold-gas refrigerator.* U.S. Patent 2 775 876.
SCHMIDT, G. (1861). Theore der Geschlossen Calorischen Maschine Laubroy und Schwatskopff in Berlin. *Z. ver Oster Ing.,* p. 79.
—— (1871). Theore der Lehmannschen Calorischen Maschine. *Z. Verb. dt Ing.,* **15,** No. 1.
SCHOCK, A. (1976). Isotope heat source for dynamic power systems. S.A.E. Paper No. 769198.
—— (1978a). SNAP, Stirling numerical analysis programme DOE Report (Preliminary Draft), Available NTIS, Springfield, Va.
—— (1978b). Nodal analysis of Stirling cycle devices. *Proc. 13th I.E.C.E.C.,* San Diego, Ca., Aug. 20–25.

SCHRADER, A. R. (1951). 1015 hr. endurance test of Philips model 1/4 D external combustion engine. Naval Eng. Experiment Station, EES Report No. C-3599-A(3), NTIS #AD 494926, Feb. 1.
SCHROEDER, J. (1973). Thermal energy storage. Invited Paper, Jozef Stefan Institut, Univ. of Ljubljana, (Philips, Eindhoven).
—— (1974). Thermal energy storage control. Trans. ASME.
—— (1975). Thermal energy storage and control. *J. Eng. Industry* (ASME), pp. 893–896.
SCHULTZ, B. H. (1951). Regenerators with longitudinal heat conduction. General Discussion on Heat Transfer, I. Mech. E., London.
—— (1953). Approximate formulae in the theory of thermal regenerators. *Appl. Scient. Res.* **A., 3,** pp. 165–173.
SCHUMAN, M. (1975). Energy converter utilizing a modified Stirling cycle.
SCHUMANN, F. A. *and* OSMEYER, W. E. (1975). Advanced heat source development for static and dynamic radioisotope space power systems. *Proc. 10th I.E.C.E.C.,* pp. 911–915, Newark, N.J., Aug. 18–22.
SCHUMANN, T. E. W. (1929). Heat transfer to a liquid flowing through a porous prism. *Jnl Franklin Inst.,* **208,** pp. 405–416.
—— *and* VOSS, V. (1934). Heat flow through granulated material. *Fuel,* **13,** pp. 249–256.
SCOTT, D. (1971a). European roundup. *Auto. Ind.,* p. 24, Feb. 15.
—— (1971b). Amazing hot-gas engine powers clean-air bus. *Popular Science,* pp. 54–56, June.
—— (1971c). Stirling engine development continues. *Auto. Ind.,* pp. 22–23, July 15.
—— (1974a). New Stirling-cycle zero-pollution car runs on stored heat. *Popular Science,* pp. 66–68, 148, June.
—— (1975b). Stirling-cycle liquid-piston engine with no moving parts. *Popular Science,* Jan.
—— (1975). Flame-powered push-pull generator runs a year without maintenance, refueling. *Popular Science,* Feb.
SELCUK, M. K., WU, Y. C., MOYNIHAN, P. I. *and* DAY, F. D. (1977). Solar Stirling power generation; systems analysis and preliminary tests. International Solar Energy Society Solar World Conference, Orlando, Florida, June 6–9.
SENFT, J. R. (1973). A Small Hot Air Fan. *Model Engr,* **139,** p. 1017, Oct. 19.
—— (1974). Moriya a 10 inch Stirling powered fan. *Live Steam,* **8,** No. 12, p. 10; **9,** No. 1, p. 28; **9,** No. 2, p. 8.
—— (1976). The design of Stirling cycle engines. *Live Steam,* **10,** Nov. & Dec.
—— (1977). Delta. *Model Engr,* **143,** pp. 1292–1295.
SERRUYS, M. (1973). Concerning Ericsson's and Stirling's cycles. *Rev. Gen. Therm.* **12,** pp. 1125–1134.
SHELPUK, B. (1974). A solar Vuilleumier system. Workshop Proc. Solar Cooling for Buildings, NSF-RA-N-74-063, Feb.
SHERMAN, A. (1971). Mathematical analysis of a Vuilleumier refrigerator. ASME Paper No. 71-WA/HT-33, Nov. 28–Dec. 2.
SHUTTLEWORTH, P. (1958). *An experimental investigation of a Stirling engine.* M.Sc. Thesis, Durham University.
SIEGEL, R. *and* PERLMUTTER, M. (1961). Two-dimensional pulsating laminar flow in a duct with a constant wall temperature. *Int. heat-Trans. Conf.* (ASME), Boulder, Colorado, pp. 517–535.
SIER, R. (1973). Hot Air Engines. *Model Engr,* **139,** pp. 298, 376, 444.
SLABY, A. (1879). *Die Luftmaschine von D. W. van Rennes.*

—— (1880). Ueber Neuerungen an Luft—und Gasmachinen. (Innovations of air and gas machines). *Dingler's Polt. Jnl.*, Bd **236,** H.1.
—— (1889). Die Feuerluftmaschine. *Z. Verb. dt Ing.*, **33,** No. 5. Sonnabend, Feb. 2.
SLACK, A. (1973). A hot air engine suitable for powering a small boat. *Model Engr,* **139,** p. 1072, Nov. 2.
SMAL, P. (1905). *Improved motor using hot and cold compressed air.* British Patent 29 002.
SMITH, H. F. (1932). *Heat Engine.* U.S. Patent 1 879 563.
—— (1942). *Refrigerating Apparatus.* U.S. Patent 2 272 925.
SMITH, L., SANDQUIST, G., OLSEN, D. B., ARNETT, G., GENTRY, S. *and* KOLFF, W. J. (1975). Power requirements for the A.E.C., artificial heart. *Trans. Amer. Soc. Artif. Int. Organs,* **21,** pp. 540–544.
SMITH, J. L. (1975). Development of the valved hot-gas engine.
—— *and* LEE, K. P. (1978). Influence of cyclic wall-to-gas heat transfer in the cylinders of the valved hot gas engine. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.
SPIGT, C. L. *and* DANIELS, A. (1975). The Philips Stirling engine: A progress report. *Proc. 10th I.E.C.E.C.*, pp. 919–926, Newark, N.J., Aug. 17–22.
—— *and* PHILIPS, N. V. (1978). Work on the 4-215 engine related to the Ford Stirling program. DOE Hwy. Veh. Syst. Cont. Coord. Mtg., Troy, Mich., May.
STARR, M. D. *and* HAGEY, G. L. (1968). Navy two-to-ten kW(e) radioisotope power system for undersea applications. *Proc. 3rd I.E.C.E.C.*, Boulder, Col., Aug. 14–16.
STEPHENS, C. W., SPIES, R. *and* MENETREY, W. R. (1960). Dynamic thermal converters. Electro-Optical Systems, Inc., Pasadena, Calif., Sept.
STEPHENS, J. R. (1978). Materials characterization program. DOE Hwy Veh. Syst. Cont. Coord. Mtg., Troy, Mich., May.
—— WITZKE, W. R., WATSON, G. K., JOHNSTON, J. R. *and* CROFT, W. J. (1977). Materials technology assessment for Stirling engines. NASA/DOE Report No. CONS/1011–22, NASA TM-73789, NTIS, Springfield, Va.
STEPHENSON, R. R. (1975). *Should we have a new engine? An automobile power systems evaluation.* 2 vols. S.A.E. (JPL report SP43-17).
STERNLICHT, B. (1974). Which automotive engines in the future? ASME New York.
STIRLING, R. (1817). *Improvements for diminishing the consumption of fuel and in particular, an engine capable of being applied to the moving of machinery on a principle entirely new.* British Patent 4081.
—— *and* STIRLING, J. (1827). *Air engines.* British Patent 5456.
—— —— (1840). *Air engines.* British Patent 8652.
STODDART, D. (1960). *Generalized thermodynamic analysis of Stirling engines.* B.Sc. Thesis, Durham University.
STORACE, A. (1971). A miniature, vibration-free rhombic-drive Stirling cycle cooler. *Adv. Cryog. Eng.*, **16,** pp. 185–194.
SUMMERS, J. L. (1975). Hot gas machine. U.S. Patent 3 879 945.
SVEDBERG, R. C. *and* BUCKMAN, Jr., R. W. (1975). Artificial heart system thermal insulation component development. *Proc. 10th I.E.C.E.C.*, p. 1489, Newark, Aug. 17–22.
TABOR, H. Z. (1967). Power for remote areas. *Int. Sci. & Technol.*, pp. 52–59, May.

TAKASE, C. N. (1972). *Optimization of the specific performance of Stirling engines.* M.Sc. Thesis, Instituto Techologico de Aero-Nautica, Sas Jose dos Campos.
TERRY, R. D. (1966). Ocean engineering. Report submitted by National Security Association, Vol. III, Part 1; Energy Sources and Energy Conservation, Western Periodicals Co., North Hollywood.
TEW, R. (1977). Comments at Stirling engine symposium. Joint Centre for Graduate Study, Richland, Wash., Aug.
THUMIN, A. K. (1969). Power transformer for artificial heart. *Proc. First Artificial Heart Conf.*, pp. 915–926, Washington, D.C. June.
TIPLER, W. (1947). A simple theory of the heat regenerator. Technical Report No. ICT/14, Shell Petroleum Co. Ltd.
—— (1948). An Electrical Analogue to the Heat Regenerator. *Proc. Int. Cong. of Appl. Mech.*, Vol. 3. pp. 196–210.
—— (1975). Energy Economics of Automotive Power Generation. S.A.E., New ork.
TOMAZIC, W. (1978). Stirling engine test program at the Lewis Research Center. DOE Hwy. Veh. Syst. Cont. Coord. Mtg., Troy, Mich., May.
—— *and* CAIRELLI, J. E. (1977). Ceramic applications in the advanced Stirling automotive engine. Fifth Army Materials Technology Conference, Mar. 21–25.
TONELLI, A. D., BLANKENSHIP, S. G. *and* SHINBROT, C. H. (1963). Auxiliary power generating system for manned Mars mission. AIAA, New York.
TRAYSER, D. A., BLOEMER, J. W., THORSON, J. T. *and* EIBLING, J. A. (1965). Feasibility of a 50-Watt portable generator using a solar powered Stirling-cycle engine. RACIC Report, Battelle, Jan. 18.
—— *and* EIBLING, J. A. (1966). A 50-Watt portable generator employing a solar-powered Stirling engine. *Proc. 2nd I.E.C.E.C.*, pp. 238–244; also *Solar Power,* **11,** pp. 153–9.
UHLEMANN, H., SPIGT, C. L. *and* HERMANS, M. L. (1974). The combination of a Stirling engine with a remotely placed heat source. *Proc. 9th I.E.C.E.C.*, pp. 620–627, San Francisco, Ca.,
UMAROV, G. Ya., ORUNOV, B. B., KLYUCHEVSKII, Yu. E., TURSUNBAEV, I. A. *and* TTUKHOV, V. S. (1976). Thermodynamic analysis and selection of optical parameters of a dynamic converter for a solar power plant. *Geliotkhnika,* No. 5, pp. 31–34.
—— TURSUNBAEV, I. A. *and* TRUKHOV, V. S. (1972). Influence of Heat Exchanger Inefficiency on the Efficiency and Power Characteristics of a Solar Powerplant Dynamic Converter. *Applied Solar Energy,* U.S.S.R., Jan/Feb.
—— —— LASHKAREVA, T. P. *and* TRUKNOV, V. S. (1973). Influence of regenerator efficiency on the thermal efficiency of a Stirling engine dynamic energy converter. *Geliotekhnika,* **9,** No. 3, pp. 58–61.
—— TRUKHOV, V. S. *and* TURSENBAEV, I. A. (1974). Prospects for using dynamic thermocompression converter in solar power plants. *Applied Solar Energy,* **10,** No. 1–2, pp. 53–56.
—— —— —— *and* ORUNOV, B. B. (1976). Method of optimizing heat exchangers for a Stirling engine. *Geliotekhnika,* No. 6, pp. 18–23, (in Russian).
UNDERWOOD, A. F. (1970). Requiem for the piston engine? *Machine Design,* pp. 20–34, Aug. 6.
—— (1976). Comments on heat engine developments. ERDA Highway Veh. Syst. Cont. Coord. Mtg., Ann Arbor, Mich., Oct.
URIELI, I. (1977). *A Computer Simulation of Stirling Cycle Machines.* Ph.D. Thesis, University of Witwatersrand, Johannesburg.

—— *and* RALLIS, C. J. (1975). Stirling cycle engine development—A review. Univ. of Witwatersrand, Dept. of Mech. Eng., Report No. 61.
—— —— (1976). A new regenerator model for Stirling cycle machines. Univ. of Witwatersrand, Dept. Mech. Eng., Report No. 67.
—— —— *and* BERCHOWITZ, D. M. (1977). Computer simulation of Stirling cycle machines. *Proc. 12th I.E.C.E.C.*, Washington, D.C., pp. 1512–1521, Aug. 28–Sept. 2.
—— HOEHN, F. *and* FINEGOLD, J. G. (1978). A computer simulation of the JPL Stirling engine. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.
URWICK, W. D. (1975). Hot air engines—the regenerator. *Model Engr*, **141,** Parts I and II, p. 140, Feb. 7 and p. 179, Feb. 21.
—— (1977). Stirling engines—more research and development. *Model Engr*, **143,** pp. 220–226, Feb. 18.
UTZ, J. A. *and* BRAUN, R. A. (1960). Design and initial tests of a Stirling Engine for solar energy applications. Univ. of Wisconsin, Dept. of Mech. Eng., Wis.
VALENTINE, H. (1978). Engine performance analysis development. DOE Hwy. Veh. Syst. Cont. Coord. Mtg., Troy, Mich., May.
VAN BEUKERING, H. C. J. *and* VAN DER AA, H. H. M. (1967). A rolling diaphragm seal for high pressures and high speeds. 3rd Int. Conf. on Fluid Sealing, Paper G421, Canbridge, England, April.
—— *and* FOKKER, H. (1973a). Present state-of-the-art of the Philips Stirling engine. S.A.E. Paper No. 730646; also in *Auto. Engr*, **81,** No. 7, pp. 37–43, July.
—— —— (1973b). Where Philips stands on the Stirling engine. *Auto. Engr*, **81** (7), pp. 37–43.
VAN DER SLUYS, W. L. N. (1975). A lithium/sodium/sulphurhexaflouride heat source in combination with a Stirling Engine. *Proc. 10th I.E.C.E.C.*, pp. 1031–1037, Newark, N.J., Aug. 17–22.
VAN DER STER, (1960). *The production of liquid nitrogen from atmospheric air using a gas refrigerating machine.* Delft Technische Hoschschule, Thesis.
VAN GIESSEL, R. *and* REININK, F. (1977). Design of the 4-215 D.A. automotive Stirling engine. *S.A.E. Int. Auto eng. Cong. Exp.* pp. 1–16, Detroit, Mich., Feb.
VAN HEECKEREN, W. J. (1949). *Hot-air engine actuated refrigerating apparatus.* U.S. Patent 2 484 392.
—— (1952). *Hot-gas motor with means for controlling the heat supply therefor.* U.S. Patent 2 583 311.
—— (1959). *Hot-gas engine comprising more than one device for the supply of heat.* U.S. Patent 2 894 368.
VAN NEDERVEEN, H. B. (1966). The nuclear Stirling engine. Proc. First Inter. Conf. on the Use of Radioisotopes, Paper 35, Harwell.
VAN VUCHT, J. H. N., KUIJPERS, F. A. *and* BRUNING, H. C. A. M. (1970). Reversible room temperature absorption of large quantities of hydrogen by intermetallic compounds. Philips Research Reports, Vol. 25, No. 2.
VAN WEENEN, F. L. (1947). The construction of the Philips air engine. *Philips Tech. Rev.*, **9,** pp. 125–134.
—— *and* DROS, A. A. (1957). *Hot-gas reciprocating engine construction.* U.S. Patent 2 817 950.
VAN WITTEVEEN, R. A. J. O. (1962). Experiments on high power hot gas engines. *Philips Tech. Rev.*, **24,** p. 409.
—— (1966). The Stirling engine, present and future. UKAEA/ENEA Symp. on Industrial Appl. for Isotropic Power Generators, Harwell, England, Sept.

—— (1971). The Stirling cycle engine. Technical Report Conf. on Low Pollution Power Systems Develop., Eindhoven, Netherlands, Feb.
VARNEY, F. M. (1977). Solar dome house and automobile with self-contained energy system. Jan.
VASISHTA, V. (1969). *Heat-transfer and flow-friction characteristics of compact matrix surfaces for Stirling-cycle regenerators.* M.Sc. Thesis, University of Calgary.
VEDIN, B. A. (1968). Stirlingmotorn. *Ny Teknik*, **2,** No. 34, pp. 16–17, 20.
VER BEEK, H. J. (1969). A two-stage compressor with rolling diaphragm seals. *Philips Tech. Rev.*, **30,** pp. 51–54.
VERNTE-LOZET, M. (1967). *Thermodynamics of piston engines for space applications.* Gordon and Breach, Science Publishers, Inc., New York.
VICKERS, P. T. (1970). Unconventional power plants. Proc. Air Pollution Control Assoc., Conf., Harrisburg.
VIKLUND, G. D. (1972). Ny Svensk Stirlingmotor Premiakord i Malmo. *Ny Teknik*, **5,** No. 1.
VON RETH, R. D., FRANK, G., KELLER, H., PLITZ, W., RICHTER, C. *and* SCHMID, P. (1974). An implantable thermal converter as power source for an artifical heart. *Proc. 9th I.E.C.E.C.*, p. 775, San Francisco, Ca., Aug. 26–30.
—— HAERTEN, R., NEMSMANN, U., HENNING, E. *and* BUCHERL, E. S. (1975). Development of power sources for blood pump applications. *Proc. 10th I.E.C.E.C.*, pp. 1214–1222, Newark, N.J., Aug. 17–22.
VONK, G. (1962). Regenerators for Stirling machines. *Philips Tech. Rev.*, **24,** p. 406.
VUILLEUMIER, R. (1918). *Method and apparatus for inducing heat changes.* U.S. Patent 1 275 507.
WAALWIJK, J. M. *and* WIDENHOF, N. (1974). The Ford-Philips Stirling engine programme. Philips Information No. 6519E, Oct.
WADSWORTH, J. (1961). An experimental investigation of the local packing and heat-transfer processes in packed beds of homogeneous spheres. *Int. Heat-Trans. Conf.* (ASME), Boulder, Colorado, pp. 760–769.
WAGATSUMA, S. (1973). *Development of a Stirling cycle machine.* M.Sc. Thesis, Instituto Techologico de Aero-nautica, Sas Jose dos Campos.
WALKER, G. (1958). Stirling Cycle Engines. *Jnl Stephenson Soc. Newcastle*, **2,** No. 2.
—— (1961a). The operational cycle of the Stirling engine with particular reference to the function of the regenerator. *J. mech. Eng. Sci.*, **3,** pp. 394–408.
—— (1961b). *Some aspects of the design of Stirling-cycle machines.* Ph.D. Thesis, Durham University.
—— (1961c). Machining internal fins in components for heat exchangers. *Machinery, London*, **101,** No. 2590.
—— (1962). An optimization of the principal design parameters of Stirling-cycle machines. *J. mech. Eng. Sci.*, **4,** No. 3.
—— (1963a). Regeneration in Stirling engines. *Engineer, Lond.*, **216,** pp. 1097–1103.
—— (1963b). Density and frequency effects on the pressure drop across the regenerator of a Stirling-cycle machine. *Engineer, Lond.*, **216,** pp. 1063.
—— (1965). Some aspects of the design of reversed Stirling-cycle machines. ASHRAE Paper No. 231, Summer Meeting, Portland.
—— (1965). Regenerative thermal machines—A status survey. *Proc. 27th Am. Power Conf.*, pp. 530, Chicago, Ill.

—— (1967). Stirling-cycle engines for total-energy systems Report Institute of Gas Technology, Chicago.
—— (1968). Military Applications of Stirling-Cycle machines. *Proc. 3rd I.E.C.E.C.*, pp. 1008–1016, Boulder, Col., Aug. 14–16.
—— (1969). Dynamical Effects of the Rhombic Drive for Miniature Cooling Engines. *Adv. Cryog. Eng.*, **14,** pp. 370–377.
—— (1971). Stirling cycle machines. Presentation Note to Stirling Cycle Machine Seminar, University of Bath, Dec. 14.
—— (1972a). Stirling engines for isotope power systems. *Proc. 2nd Int. Symp. of Power from Radioisotopes*, pp. 467–493, Madrid.
—— (1972b). Stirling engines—The second coming? *Chartered Mech. Eng.*, I. Mech. E., pp. 54–57, London, April.
—— (1973c). *Stirling cycle machines.* Clarendon Press, Oxford, 156 pgs.
—— (1973d). Stirling engine power supplies for remote unattended sites. *Proc. 8th I.E.C.E.C.*, pp. 594–600, Philadelphia, Pa., Aug. 13–17.
—— (1973e). The Stirling engine *Sci. Am.*, **229** (2), pp. 80–87, Aug.
—— (1973f). Stirling engines. Proc. 2nd Nat. Tech. Conf., Canadian Gas Assn., Oct.
—— (1974a). Optimization of Stirling engines. *The Big Byte*, **7** (1), pp. 1–8 (Univ. of Calgary).
—— (1974b). Stirling-cycle cooling engine. *Cryogenics*, **14,** pp. 458–62.
—— (1976). Exploratory study of the Rainbow-variant Stirling-cycle engine. *Proc. 11th I.E.C.E.C.*, Lake Tahoe.
—— (1978a). Recent history of developments in Stirling machines. *Inst. of Gas. Tech. Seminar, Stirling Cycle Prime Movers*, Chicago, Ill., June.
—— (1978b). Double-acting free-piston Stirling engines. Invention disclosure, Feb. (patent pending).
—— *and* Agbi, B. (1973). Optimum design configuration for Stirling engines with two-phase, two-component working fluids. ASME Paper no. 73-WA/DGP-1, Winter Ann. Mtg.
—— —— (1974). Thermodynamic aspects of Stirling engines with two-phase, two-component working fluids. *Trnas. Can. Soc. mech. Eng.*, **2** (1), pp. 1–8.
—— Finkelstein, T. *and* Joshi, T. (1970). Design optimization of Stirling-cycle cryogenic cooling engines. Cryog. Eng. Conf., Paper no. K4, Boulder, Col.
—— *and* Khan, M. (1965). The theoretical performance of Stirling-cycle machines. S.A.E. Paper no. 949A, Int. Auto. Eng. Congress, Detroit, Michigan.
—— *and* Metwally, M. (1977). Stirling engines with a chemically active working fluid—some thermodynamic effects. *J. Engng Pwr* (ASME), **99,** pp. 284–7.
—— *and* Vasishta, V. (1971). Heat transfer and friction characteristics of wire-screen Stirling engine regenerators. *Adv. Cryog. Eng.*, **16,** pp. 324–32.
—— *and* Wan, W. K. (1972). Heat transfer and fluid-friction characteristics of dense-mesh wire screen Proc. 4th Int. Cryog. Eng. Conf., Eindhoven, Netherlands.
—— Ward, G. L. *and* Slowly, J. (1978). Operating characteristics of a small Stirling engine. *Trans. Can. Soc. mech. Eng.*
Walton, H. (1965). Amazing no-fuel 'space' engine you can build. *Pop. Sci.*, **176,** pp. 106–110, July.
Wan, W. K. (1971). *The heat-transfer and friction-flow characteristics of dense-mesh wire-screen regenerator matrices.* M.Sc. Thesis, University of Calgary.
Ward, E. J., Spriggs, J. O. *and* Varney, F. M. (1972). New prime movers for ground transportation—Low-pollution, low-fuel consumption. *Proc. 7th I.E.C.E.C.*, pp. 1013–1021, San Diego, Ca.
Ward, G. L. (1972). *Performance characteristics of the Stirling engine.* M.Sc. Thesis, Univ. of Bath, U.K.
Watelet, R. P., Ruggles, A. E. *and* Hagen, K. G. (1976). Vapour cycle energy system for implantable circulatory assist devices. Annual Report, Nat. Inst. of Health, Report No. N01-HV-4-2909-3, NTIS, Springfield, Va., May.
—— —— *and* Torti, V. (1976). Status of the tidal regenerator engine for nuclear circulatory support systems. *Proc. 11th I.E.C.E.C.*, pp. 136–142, Lake Tahoe, Nev., Sept. 12–17.
Waterman, A. (1978). Advanced sliding seal development. DOE Hwy. Veh. Syst. Cont. Coord. Mtg., Troy, Mich., May.
Waters, E. D. (1978). Sigma research conceptual design study of energy storage for a Stirling car. DOE. Hwy. Veh. Syst. Cont. Coord. Mtg., Troy, Mich., May.
Weimer, G. A. (1976). Stirling engine: solution to the U.S. auto dilemma? *Iron Age*, Nov. 22.
Weissler, P. (1965). G.M.'s amazing new 2-piston engine. *Sci. & Mech.*, pp. 76–79, 104–105, March.
Welsh, H. W., Harp, J. L. Jr., Yand, R. A., Ottway, T. P., Riley, C. T. *and* Nawroczynski, L. (1972). Study of low emission vehicle power plants using gaseous working fluids. Thermo Mechanical Systems Co., Canoga Park, Calif, Aug.
—— *and* Monson, D. S. (1962). The Stirling engine for space power S.A.E. Paper 594 C, Nat. Aero. Eng. and Manuf. Mtg., Los Angeles, Ca., Oct.
—— Poste, E. A. *and* Wright, R. B. (1959). The advanced Stirling engine for space power. ARS Paper No. 1033–49, Nov.
West, C. (1971). The fluidyne heat engine. Report No. AERE-R 6776, Harwell.
West, C. D. (1976). Solar power and the Stirling engine. *Solar Energy Digest*, 4, March.
Westbury, E. T. (1970). A Robinson-type hot-air engine. *Model Engr*, **136,** pp. 164, 216, 268, 320, 372.
Westinghouse Corp., Quarterly Progress Reports. Artificial heart development programme. Report No. COO-3043 (Nos. 1 through 29), U.S. DOE, Washington, D.C. See in particular: COO-3043-22, July/Sept., 1975; COO-3043-25, April/June, 1976.
White, M. A. (1973). Proof-of-principle investigation of 300 W(e) Stirling engine piezoelectric (STEPZ) generator. MDAC Final Report, No. MDC G4420, Sept.
—— Martini, W. R. *and* Gasper, K. A. (1972). A Stirling engine piezoelectric (STEPZ) power source. 25th Power Sources Symposium, May, or MDAC Paper WD 1897.
White, R. (1976). Vuilleumier-cycle cryogenic refrigeration. AFFDL-TR-76-17, April.
White, W. D. (1961). Lithium and sodium for underwater propulsion. *Astronautics*, April, pp. 38–39, 78, 79.
Wiggers, C. J. (1957). The heart. *Scientific American*, **196,** pp. 74–78, May.
Wilding, T. (1971). Stirling-Engine Coach at Brussels. *Commercial Motor*, Jan. 22.
Wile, D. D. *and* Brainard, D. S. (1960). Control system for winter operation of air-cooled condensers. U.S. Patent 2 943 457.
Wiley, R. L. *and* Lehrfeld, D. (1978). Development of a 1 kW(e) isotope

fueled Stirling cycle power system. *Proc. 13th I.E.C.E.C.*, San Diego, Ca., Aug. 20–25.
WILKINS, G. (1971). Hot air engine runs quietly and cleanly. *Mechanix Illustrated*, pp. 68–71, Oct.
WILLIAM, C. G. (1973). Alternative to the motor car—or alternative fuels. Natl. Soc. Clean Air, Brighton, England.
WILLIAMSON, J. M. (1959). The effectiveness of the periodic-flow heat exchanger. English Electric Report, No. W/M(4B).
WILSON, S. S. (1975). Possible developments in transportation. Proc. of a Summer School, Lincoln Coll., Oxford, Engl., July 14–25.
WINTRINGHAM, J. S. (1960). Potential passenger car power plants. S.A.E. Paper No. S275.
—— (1961). Promising methods of generating electricity. S.A.E., New York.
WOLGEMUTH, C. H. (1968). *The equilibrium performance of the theoretical Stirling cycle with a chemically reactive gas as the working fluid.* Ph.D. Thesis, Ohio St. University.
—— (1969a). The equilibrium performance of the theoretical Stirling cycle with chemically reactive gas as the working fluid. *The performance of high temperature systems.* Vol. 2, Paper 20, pp. 371–387.
—— (1969b). Dynamic performance of a thermodynamic cycle using a chemically reactive gas. *Proc. 4th I.E.C.E.C.*, pp. 599–605.
WUOLIOKI, J. R. (1948). Kuumailmakoneen Renessanssi. *Teknillinen Aikakausleptie*, **38,** pp. 241–246.
WURM, J. (1975). Assessment of selected engine-driven heat pumps. Solar Energy Heat Pump Systems for Heating and Cooling Buildings, Paper no. 7506130.
—— *and* STAATS, W. R. (1977). An assessment of selected heat pump systems. Project HC-4-20. American Gas Association/Institute of Gas Technology, Chicago.
YAGI, S., KUNII, D. *and* WAKAO, N. (1961). Radially Effective Thermal Conductivities in Packed Beds. *Int. Heat-Trans. Conf.*, ASME, Boulder, Colorado, pp. 742–749.
YATES, D. (1978). Hydrogen permeability of ceramics and metals. DOE Hwy Veh. Syst. Cont. Coord. Mtg., Troy, Mich., May.
YENDALL, E. F. (1958). A novel refrigerating machine. *Adv. Cryog. Eng.* **2,** pp. 188–196.
YZER, J. A. L. (1952). *Cylinder head for hot-gas reciprocating engines.* U.S. Patent 2 616 250.
ZACHARIAS, F. A. (1970). Mollier-1, S-Diagramme fur Verbennungsgase in der Datenverarbeitung. *MTZ Motortechnische Zeitschrift*, **31,** pp. 296–303.
—— (1971). Betrachtungen zum Ausseren Verbrennungssytem des Stirling Heissgasmotors. (Considerations on external combustion Stirling engines). *MTZ, Motortechnische Zeitschrift*, **32,** pp. 1–5.
—— (1971). Der Stirlingmotor fur Aufgaben der Meerestechnik. *Sonderdruck aus der Zeitschrift Schiffestechnik Heft*, **92,** pp. 39–45.
—— (1973). Unique requirements of the cooperation of computation and design in the development of Stirling engines. Von Karman Institute for Fluid Dynamics, Lecture Series 53, Feb. 12–16.
—— (1974). Advanced development of Stirling engines at MAN/MWM. 2nd Symp. on Low Pollution Power Syst. Development, Dusseldorf, Sept.
—— (1977). Further Stirling Engine Development Work—Parts 1 and 2. *MTZ, Motortechnische Zeitschrift*, **38,** pp. 371–378.

—— *and* GARTNER, F. (1975). Industrieentwicklungen am Stirlingmotor. Vortrag an der Universitat Stuttgart.
ZANZIG, J. (1965). Stirling-cycle engine. S.A.E. Paper no. 733B.
ZARINCHANG, J. (1972). *The Stirling engine* (an annotated bibliography). Intermediate Technology Publications, London.
—— (1975). Some considerations on design of Stirling engines. *Proc. 10th I.E.C.E.C.*, Newark, N.J., Paper no. 759142.
ZEUNER, G. (1887). *Technische Thermodynamik*. Leipzig, Vol. 1, pp. 347–357.
ZIMMERMAN, F. J. and LONGSWORTH, R. C. (1971). Shuttle Heat Transfer. *Adv. Cryog. Eng.* **16,** pp. 342–349.

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