THERMODYNAMICS OF STIRLING TYPE ENGINES FOR THE ARTIFICIAL HEART

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by

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Abstract

The thesis commences by tracing the underlying research work that established the feasibility and technically justified the current programmes for the development of the implantable artificial heart, more particularly those programmes based upon the use of a heat engine as the power unit of the artificial heart.

The provision of a heat source for the heat engine, whether it be a continuously generating source, such as a mass of a suitable radioisotope, or a thermally rechargeable material, adds to the burden of weight and bulk that the user must carry. This inconvenience could be reduced if the thermal efficiency of the engine can be increased. Thus, in the light of a heavy and probably an unending and continuously increasing demand for a practicable artificial heart the author considers that further work is justified if the engine efficiency can be raised, albeit by only a small amount.

The author's main endeavour is to identify an ideal engine cycle that appears to promise the most likelihood of high efficiency. It seems remarkable that there is little record of similar investigations readily available in the literature, and therefore, the present author has taken the approach to develop and evaluate parameters for a number of ideal cycles: Carnot, Stirling, Ericsson and the regenerative Otto cycle. The parameters for these cycles are compared, together with the published results for the Thermocompressor, the Schmidt isothermal and the Schmidt adiabatic cycles.

Working fluid mass distribution between the hot and cold spaces of the Schmidt isothermal engine is investigated, revealing the low utilization of the working fluid when large dead volumes occur in the engine.

The hot volume versus cold volume relationships of the Schmidt and rhombic drive engines are analyzed and compared with the relationship for the true Stirling cycle.

Illustrations of the arrangement principles of historical and current "Stirling" engines, as well as the constructions of contemporary artificial heart engines are presented to indicate how problems of arrangement have been answered.

The main findings are:

(a) Of the isothermal cycles considered, i.e. cycles of potential Carnot efficiency, the true Stirling cycle is significantly the best and better than the adiabatic cycles.

(b) The ideal adiabatic cycles achieve efficiencies tending to the Carnot level as the compression ratio tends to unity. Working fluid utilization is least when the ideal efficiency is maximum, that is at unity compression ratio. Inevitable losses in the practical engine indicate that the maximum efficiency will be attained at a compression ratio significantly greater than unity.

(c) The effect of regenerator inefficiency in the Stirling engine can be partly compensated by increasing the compression ratio.

(d) The hot space-cold space relationships of the Schmidt and rhombic drive engines deviate significantly from those of the ideal Stirling cycle.

It is concluded that an engine built to run as closely as possible to the true Stirling cycle is the most promising route to the best attainable efficiency, also that in practice the compression ratio is likely to strongly influence the achievable efficiency.

The author recommends building an initial research engine with its hot-cold spatial relationships cam controlled to the requirements of the true Stirling cycle, also to be adjustable for compression ratio. The major purpose of this initial engine would be to prove the veracity or otherwise of the cycle selection and to determine how the efficiency varies in practice with compression ratio. For comparison, cams would also be made to simulate other cycles, e.g. the Schmidt cycle. Suggestions are given for features that would facilitate the operation and performance measurement of the engine.

In presenting this work the author also considers that he has compiled a useful bibliography which contains the more pertinent references in the field.

ACKNOWLEDGMENTS

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Fig. Nos.	
21	Thermo Electron Corporation
22) 23) 24) 25)	Donald W. Douglas Astronautic Corporation
26) 27)	Aerojet General Corporation
28) 29)	North American Philips Corporation

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NOTATION

A	area
C_P	specific heat, constant pressure
C_V	specific heat, constant volume
d	linkage dimension
J	mechanical equivalent of heat
k	piston area correction factor for rod
l	linkage dimension
P	pressure
Q	heat quantity
r	compression ratio, V_{max}/V_{min}
r	linkage dimension
T	temperature, absolute
V	volume
w	fluid mass
W	total mass of working fluid
W	power
MEP	mean effective pressure
α	phase angle
β	angle defining linkage geometry
γ	ratio of specific heats C_P/C_V
η	efficiency
θ	shaft rotation
τ	temperature ratio T_C/T_H

Subscripts:

C	cold side
H	hot side
max	maximum
min	minimum
o	original
f	final

CHAPTER 1

INTRODUCTION

It may have been the dramatized publicity of heart transplant operations from humans to humans that caused the National Heart Institute of the United States in 1966 to adopt a systems approach to direct and coordinate the Artificial Heart Program, which had its inception in 1965.

A major objective of the program is the development of an artificial heart[1] that can completely or partially (when it is known as an "assist" heart) replace the natural heart, being implanted in the thoracic cavity of the body itself. This could give new hope to about one third of the six hundred thousand people, in the United States alone, who now die each year of heart attacks. It has been estimated that one in three of these people could have their lives usefully extended if artificial hearts were available.

In recent historical times there were total replacement attempts in Russia in 1937 and 1958 by Demikhov[2]. There seems to be current interest in the artificial heart concept in many countries, yet it appears that the major portion of published work stems from the efforts of a few American teams holding contracts with the United States government through the National Heart and Lung Institute (NHLI). The various teams doing this work are mainly comprised of medical, mechanical engineering and bioengineering personnel. There are many problems in many related fields to be solved and the end of the road is not yet in sight. However, although there are as yet no humans equipped with artificial hearts and leading active lives, much progress has been made with animals.

Contemporary artifical heart builds are comprised of four major parts: a source of energy and an engine to convert the energy to a fluidic or mechanical form, a suitable transforming device for applying that energy to a blood pump, the blood pump and a control system for regulating the operation of the heart in accordance with the needs of the body.

To narrow the field let us consider the engine itself. Several types of engines received early investigation, including electrical, compressed air and thermodynamic devices. Of these the thermodynamic machines have been found to be the most promising with electric motors obtaining power by transcutaneous transformation, a rather distant second choice. Consequently most of the development effort today is in the thermodynamic direction.

It might be noted that the provision of power from natural body functions by fuel cells has been studied but found impracticable for the heart application, which requires one hundred times the power of a pacemaker. There is potentially sufficient power for a pacemaker, but not enough for a heart.

The thermodynamic devices have come out ahead of electric motors on the grounds of portability: a charge of heat at high temperature may be stored within the body. The electric motor on the other hand needs percutaneous leads and a power source external to the body. Batteries are not yet good enough for implanting in the body.

Thermodynamic devices considered, built and tested both in vitro and in vivo, include Rankine and compression-expansion regenerative type engines. The latter have generally been termed, rather loosely, "Stirling" engines, "Modified Stirling" engines, etc.

It is essential that a certain minimum thermal efficiency be achieved by any heat engine in an implanted artifical heart, for reasons that will become clear in the text.

This report records a review of Stirling type engines in the Artificial Heart Program. In the course of this review a doubt has developed in my mind as to whether current developmental heart engines are proceeding along the optimum lines, or at least as to whether another variant of the Stirling engine should not be tried.

The question arose because of the paramount need for thermal efficiency in a heart engine and it has not quite been dissolved in spite of the most recent publications describing the excellent engineering and performance of contemporary developmental engines. Admittedly the implied criticism is not supported by practical experience of these, or similar, engines. It arises rather from the facts of basic thermodynamics which I have developed to show why engine development should be attempted with control of the working fluid flow different from that of any known contemporary engines.

Stirling engines have been receiving attention for many years. It is unlikely that a single development of detail will give earth-shaking efficiency improvements, but rather that the frontier will be advanced by minor gains at various points along the line. With this outlook this thesis concludes with a proposal that outlines some features that are recommended for use in a research engine to be used for developing a heart engine. Some of these features are also applicable to very much larger engines, covering power ranges where efficiency improvements are becoming necessary because of the rising cost of fuel. These proposed features are also applicable to the reversed Stirling engines, ranging from the very low powered devices used to establish low reference temperatures for laboratory instrumentation to the larger machines suitable for gas liquefaction plant.

Stirling engines as a class have had an interrupted career. Invented and built and developed by the Scottish Rev. Stirling and his brother, and by the Swede Ericsson early in the nineteenth century, they were popular for a time because they were far more efficient than the steam engine, the only other heat engine of industrial note. However, the steam engine efficiency improved while Stirling engine progress was impeded by, essentially, heat transfer problems. Consequently Stirling engines lost ground and although some small ones were still made, for example fractional horsepower ceiling fan units, laboratory demonstration units and toys, the Stirling idea slept for a hundred years until the Dutch N.V. Philips organization devoted a great deal of effort in exploring the practical possibilities of the engine. By this time the sciences of thermodynamics and metallurgy had developed. Thus in a very short time the power output of the engine on a cylinder volume basis had risen a hundredfold and the Stirling engine was on its way and in some important aspects it now compares favourably with other contemporary engines. The Philips organization of Eindhoven, Holland, has made great contributions to the practical development of Stirling engines. Other large organizations have more recently become involved, including General Motors and Ford of the U.S.A., M.A.N. of Germany and United Stirling in Sweden. Nevertheless, in spite of the great efforts of these people, to my knowledge there is not yet a commercially successful Stirling engine on the market, although Philips have sold a number of reversed Stirling engines for gas liquefaction duties. The significant advantages of Stirling engines are: smoothness, quietness, low chemical pollution, high thermal efficiency and the ability to be designed for almost any source of thermal energy. Increasing demands by society that powered equipment shall possess all of these qualities, together with inevitable improvements of manufacturing technologies are expected sooner or later to make the Stirling engine economically acceptable. In the meantime Stirling engines have received much attention for boat, bus and ship propulsion, as power units for the military (suitable for dropping from the air), for submarines and torpedoes, for small reliable electric power units in inaccessible or remote locations, for solar/electric converters in space. For most of these examples hardware is known to have been made.

I also feel that the Stirling engine is uniquely suitable for the following applications:

(a) As part of an energy converter using burnable garbage and/or the purer petroleum fuel for use in heat pumps at the institutional, commercial and perhaps household sizes.

(b) To replace some of the steam and diesel power equipment in industries where large amounts of mechanical power are required and relatively inexpensive burnable organic material is available, for example in forest products.

(c) For small and medium sized power units in developing countries and in countries without natural petroleum or uranium sources. Solar energy and, especially, energy from burning vegetable material are envisaged here. Application of a wood-burning Stirling engine to railroad locomotives is an intriguing possibility: an engine of much better efficiency than the old steam locomotive, but without the disadvantages of the diesel electric locomotive should be possible.

These applications are expected to become feasible, if not already so, before general acceptance of the Stirling engine to replace conventional engines in North America.

In these applications the engine powers involved vary widely from a small fraction of a horsepower to many thousands. Energy sources may be light or heavy petroleum, reacting chemicals, the sun or burning garbage, but in all instances it will be found that many of the design and development problems originate in heat transfer requirements. Evidence of this is given in the figures below which indicate, typically, the number of units of heat per unit of work output that must be transferred between the working fluid and some (solid) interface in typical power plants:

Diesel, gasoline engines, liquid cooled	1
Gas turbine, simple	0
Gas turbine, regenerative	2
Modern steam plant	4
"Stirling" type engines	20

The relative difficulties of effecting heat transference in the "Stirling" engines are even greater than are probably suggested by these figures: in the other power plants heat transfer is either incurred in preventing the overheating of engine components (diesel and gasoline) or is achieved (regenerative gas turbine and steam plant) using whatever volumetric space is necessary to provide for mechanically convenient heat transfer surfaces. In "Stirling" engines heat transfer occurs throughout a cycle in which the working fluid pressure is continuously changing. Due to varying pressures, if too much otherwise useless or "dead" volume accrues in providing heat transfer surfaces the engine performance is compromised. This is the origin of a major optimizing problem in Stirling type engines.

Before commencing with engine cycle studies related to the implantable artificial heart we should consider the implications in engineering terms of some of the operational requirements and conditions of the heart. These will be reviewed in the following chapter.

CHAPTER 2

CONTEXT OF THE ENGINE PROBLEM

Introduction

In developing an implantable artificial heart one of the earlier problems to be solved must be to arrive at a design specification. If the final objective is to make an artificial heart equal to the natural heart in all its relationships to the body we might well tremble to approach this problem. The poetic words from the Bible "... for behold I am fearfully and wonderfully made..." may bespeak even more extensive fields of knowledge of ourselves than those already discovered. We may wonder if it is ever possible to develop the design specification.

To be practical the development of a heart substitute that provides the main haemodynamic functions of the natural heart, without too much forseeable danger to the body and is reasonably tolerable to the user can be the only realistic approach to the heart and, in particular, to its power plant component. It is on this basis that the current artificial heart programs are proceeding. This chapter reviews the main recognized parameters and the underlying preparatory work that has lead to the present promising achievements of artificial hearts.

Physical Requirements Influencing Engine Design

The theoretical considerations for an implantable power source include:

(a) Cardiac pumping power.

(b) Additional endogenous heat loads and mechanisms of dissipation.

(c) Energy storage requirements.

(d) Configurational limitations.

(e) Possible shielding requirements.

These influences on engine design are considered.

Cardiac Pumping Power

The power actually delivered to the blood, haemodynamic power, in a healthy person depends upon some basal value and on the degree of body activity. The basal value is a function of body surface area, time of day, time during the cardiac cycle and age. Analysis during an average day shows peak cardiac power levels two to five times the basal level while during an individual cycle the peak instantaneous power is approximately five times the average power through the cycle. Some extreme figures are given for three different ages, from reference [3]:

Age, Years	Minimum Biological Cardiac Power, Watts	Maximum Biological Cardiac Power, Watts
20	1.11	6.53
50	1.03	6.48
80	0.75	4.22

These studies suggest that a range of 1.5 W to 4 W should be adequate for a sedentary man. More recent German studies by G. Frank et al [61] give average values and a 24-hour demand profile for people in the age group 20 and 50 years of age that tend to support these figures. Evidently no single maximum/minimum power level will perfectly suit all ages and degrees of activity. However, this is not yet a pressing problem as the artificial hearts currently under development are still in the stage of animal in vivo testing. Furthermore the design and construction of the components of the heart appear to be amenable to scaling up or down slightly. Thus the main concerns in proliferating a range of cardiac powers would be those of production.

The haemodynamic output powers of all the artificial hearts now under development are about 5 W maximum and they all use the TECO (Thermo Electron Corporation) blood pump. To achieve the required cardiac output the engine must have a somewhat higher power to allow for transmission and conversion losses between the engine and blood pump.

From considerations of either the total volume of the systems, or from the need to minimize heat losses, i.e. to keep the engine dimensions compact, the peak pressure of the engine must be very much higher than the peak systolic pressure of the normal circulatory system. Therefore a pressure-volume transformation must be used between the engine and the blood pump. A study of the basic relationships of this is given by Frank et al [61], who have analyzed transmission trend efficiencies and show in effect that low pressure liquid transmissions have large losses due to friction and therefore have low efficiencies. Higher pressures are better because of the smaller flow volumes. Except for the Philips-Westinghouse engine which has a mechanical drive, the current developmental engines use high pressure hydraulic systems (these engines are described in Chapter 5). The earlier models had overall transmission efficiencies of the order of 65%, but today's engines are considerably better. For example, McDonnell Douglas Astronautics Co. claim 89% efficiency for their pending System 6 implantable engine. It should be noted that hydraulic power transmissions are very convenient for transporting waste heat from the engine to the heat exchanger which is integral with one wall of the blood pump.

"Synchronous" engines, operating at the blood pump frequency, have been built. One might expect these engines to have a simpler and more efficient power transmission system with advantage to the overall efficiency. But the engines were evidently very massive and with one exception all current development is with high frequency engines. The exception is a new patented synchronous design termed a "tidal regenerator engine" [22,23] and this engine is under development. Its operation is described in Chapter 5. Medical studies show the desirability of reproducing the natural cardiac output pulse shape, this characteristic accounting for the high ratio of peak power to average power. The hydraulic transmissions of the high frequency engines are able to provide for energy storage so that engine design and performance are not compromised in designing and controlling the blood pump and its actuator to provide this desirable pulse characteristic. Whether, or how, this pulse characteristic is actually achieved is not evident in the literature. However, this is not of too much concern to the engine developer because of the energy storage feature. The significant effect on the engine is that of the energy conversion efficiency of the actuator and blood pump.

Dispersion of Waste Heat

An artificial heart using a heat engine as its energy converter has the problem of dissipating the portion of the engine supply heat that is not converted into mechanical work. This waste heat is large, of the order of six or eight times the mechanical output of the engine. For the heat source temperatures proposed and used in current development programmes the waste heat is several times that necessitated by a Carnot efficiency engine. Thus in spite of the great practical efforts of the engine developers there is room for improvement of engine efficiency and therefore for the reduction of waste heat. The waste heat problem was forseen from the beginning of the artificial heart program and various methods considered for disposal of the waste heat. Of these the only practicable sink for heat rejection is the blood stream itself and all current developments use engine waste heat exchangers integral with one wall of the elastomeric blood pump. Hydraulic fluid carries waste heat from the engine working fluid, is cooled in passing through the blood pump heat exchanger and returns to the engine. Compared with the more direct discharge of waste heat from the cold side of the engine working fluid to the blood stream the indirect hydraulic fluid method has the great advantages of non-interference with the blood flow and it obviates difficulties arising from hot spots due to flow stagnation. Investigations [3,4,5] show that the amount of heat that can be dissipated by the body on a long term basis is not known with certainty. Assuming an overall efficiency of 15%, fairly representative of today's engines, the imposed thermal load on the blood will be 20 W to 30 W, corresponding to .29 W to .43 W per kilogram of body weight for a 70 kilogram person. In vivo tests in animals are encouraging in that the heat loads can be carried, the body, after a time, seeming to adjust itself to this unnatural load. In addition to the main flow of heat from the engine to the blood stream, some heat is lost directly through the heat source insulation. In spite of high quality multifoil vacuum insulation [6] this loss is significant. It also causes increased tissue temperatures in those parts of the body that touch the insulation, though these temperatures are down to a level (about 2°C above normal) that are considered satisfactory.

Energy sources for the engine are (a) radioactive isotopes, the radiation being captured mainly in the fuel matrix and surrounding material with the evolution of heat, (b) a less controversial source consisting of a mass of suitable material that is thermally rechargeable. Current artificial hearts favour the radioisotope fuel, namely Plutonium 238, a by-product of reactors. This obviates the complexities of energy recharging devices, but may not be economically attractive if the heart is successful and develops to mass production, due to the limitation of supplies. As mass production approaches one would expect to see more emphasis on rechargeable heat sources utilizing the latent heat of fusion of salts and hydrides, for example. In any event these materials may be useful as [illegible] thermal capacitors in isotope powered engines, permitting an isotope inventory based upon average rather than peak power demand and imposing lower mean thermal loading on the body. The feasibility of these materials as the bases of thermal capacitors has been tentatively established by Bluestein and Huffman [9].

A rechargeable heat source is essentially a mass of suitable material from which heat may be continuously withdrawn by the engine and which itself is occasionally replenished. Obviously isothermal operation is ideal, leading to the interest in fusion-solidification phase changes. Development is along the lines of electric resistance wires to heat the mass, electric power being derived from the secondary winding of a transformer. This secondary winding is implanted beneath the skin of the chest and when a recharge is required, say every eight hours, the primary winding unit is placed (outside the chest) over the secondary unit and power is transferred electromagnetically and transcutaneously to the transformer secondary unit, thence to the heat source heater. Bluestein and Huffman's work included studies of thermal gradients in the heat source material mass, and presumably accounts for the use of heat pipes in some engine heat source units (Chapter 5) in order to reduce the temperature loss between the heat source and engine hot space. Thermal storage materials that have been considered include:

Material	Melting Point °C	Latent Heat cal/gm	Potential Carnot Efficiency
Lithium chloride/lithium fluoride eutectic	499	165	.60
Lithium fluoride/sodium fluoride	650	app. 172	.66
Lithium hydride	677	610	.67

Engine bench development, typically using electric heaters, has used peak temperatures in the approximate range 430°C to 500°C.

Biological problems arising from the radiation emanating from isotopes have been studied by several groups [3,4,7,8]. The studies, covering not only normal daily living but also the possibility of catastrophic accident, indicates that the radiation probably would be tolerable to the user of an isotope powered heart, as well as to society generally. In vivo tests in animals support these opinions, over the limited time durations that have been possible for these tests.

Configurational Limitations

The original targets for a complete artificial heart package were 2,000 ml total volume and 3,000 gm total weight, rather round-looking numbers! For the "assist heart" variant, which takes over the function of the left ventricle only, there seem to be no published targets. Although artificial hearts have not yet been implanted in humans and therefore there is no certainty that these limitations will not have to be readjusted, current designs not only meet, but improve upon these limitations. However, considering that the average human heart weighs about 250 grams for the female and 310 grams for the male, with corresponding volumes of 475 ml and 650 ml respectively there is still much room for improvement in this respect. For the artificial assist heart application the natural heart remains within the body so it is even more desirable to minimize the intrusion of weight and volume into the body.

Much of the power unit weight must be ascribed to the thermal energy source and, in the case of radioactive sources, its screening. Engine component weights are not given in the literature but they are certainly relatively low due to each component being pared down to the minimum to reduce thermal conduction losses. For example, one engine had a cylinder wall .010 inch thick and a displacer-regenerator wall .005 inch thick. Saving in power source weight then must come mainly from improvements of the engine thermal efficiency and of the engine power transmission conversion efficiency, both improvements ultimately reducing the energy source inventory. A constant requirement for low weight is the use of compact geometry of the power source and the hot part of the engine cylinder, both to minimize radioactive shielding requirements and to minimize the area for heat leakage into the surrounding body organs.

Engine Sealing

Most engine development work has used helium as the working fluid. In other fields of engineering this has already become known as a difficult gas to contain without leakage. In the artificial heart engine the working charge is only very small, the potential required life of the engine could conceivably be many years, the engine operates under fluctuating pressure and part of the engine is at a high temperature. For the relatively short life bench tests and animal implant tests the helium leakage has been acceptable using conventional seals. For the long term, however, the establishment of sound engine seal designs and of the technology for translating these into production quantities constitute a recognized requirement of the successful engine.

Summary

The more important recognized constraints to the development of the artificial heart engine have been considered. These constraints all originate from the physical needs of the body. For each of these constraints basic investigations have been done by teams of engineering and medical practitioners to establish practical feasibilities following which engine design and development have progressed. In addition to many thousands of hours of bench testing, some testing has been performed using animals with promising results. Thus the status is now generally well beyond the "proof of principle" stage and significantly into life and reliability improvement, although there are a few details that still require major improvements, for example the various engine helium seals and the engine power control. It is noted, however, that published literature does not seem to mention any requirements for quietness or vibration limits. In these respects perhaps we have to await the first transplants in humans before satisfactory feedback is gained in this important area. For the potential user of an artificial heart the prospective volume and weight burdens have eased from the early target of 2 litres and 3 kilograms to, typically achieved, 1 litre and 2.4 kilograms.

CHAPTER 3

OPERATIONAL CHARACTERISTICS OF CARNOT, STIRLING AND ERICSSON CYCLES

Introduction

The purpose of this chapter is to examine certain characteristics of the Carnot, Stirling and Ericcson cycles, each of which potentially gives the Carnot efficiency. It is recognized that no engine will be built to yield the ideal Carnot efficiency, but it is hoped that this study will reveal the strengths and weaknesses of these cycles. An open minded approach is taken: it is not accepted that a practical "Stirling type" engine for the artificial heart has to be of the free piston variety (Chapter 5) or such that its piston motions are simple harmonic (Schmidt) or the same as those of any other "Stirling" engine. Thus it is hoped to find directions to an engine design that can be developed to the highest efficiency.

In any type of "Stirling" engine the amount of heat transferred from all the internal surfaces to and from the working fluid is several times the work output of the cycle, most of this heat being transferred by forced convection. This is at the cost of internal pumping losses. Thus one criterion for comparing cycles is the ratio of net cycle output to the internal heat transfer load. The effects of regenerator thermal deficiencies on the cycle efficiencies, the mean effective pressures and the work ratios of these cycles are also important.

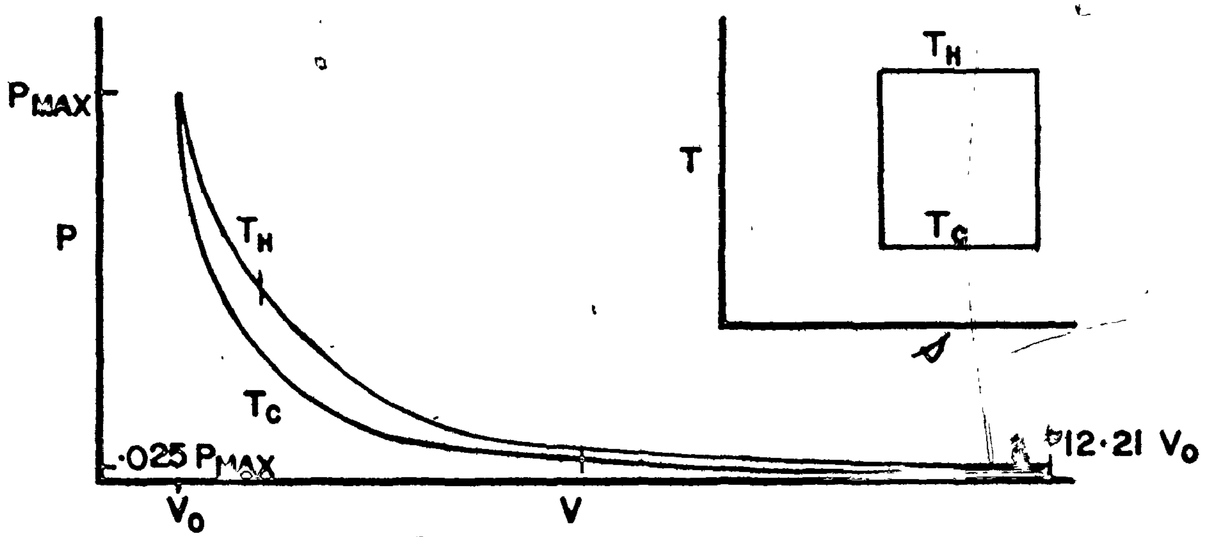
Although much analysis of Schmidt and more sophisticated cycles has been done, rather remarkably, important characteristics of the simple classical cycles seem hardly to be found in the literature. In this chapter, I have therefore developed and compared, for the true Stirling and Ericsson cycles, the following characteristics: (a) heat transfer across the working fluid boundaries during regeneration, (b) relation of cycle work output to regenerator heat exchange, (c) relation of boundary heat flow to regenerator heat flow, (d) effect of regenerator inefficiency, (e) effective pressure to maximum pressure ratio, (f) work ratio, (g) work output to total reversing heat exchange.

Description of Cycles

In each of the cycles a fluid is compressed isothermally while cold and expanded isothermally while hot. These two elements (only) are common to the Carnot, Stirling and Ericsson cycles. The input work of compression being less than the output work of expansion, a net work output results. The fluid used is almost invariably a gas though in the industrial field there is record [35] of an attempt by Malone to establish an engine (not a true Stirling engine) using water near the critical point. The engine was developed with a view to marine propulsion and working pressures of about 12,000 lb/in^2. Also, more recently Beale and Walker have been considering "two phase" working fluids [19,20].

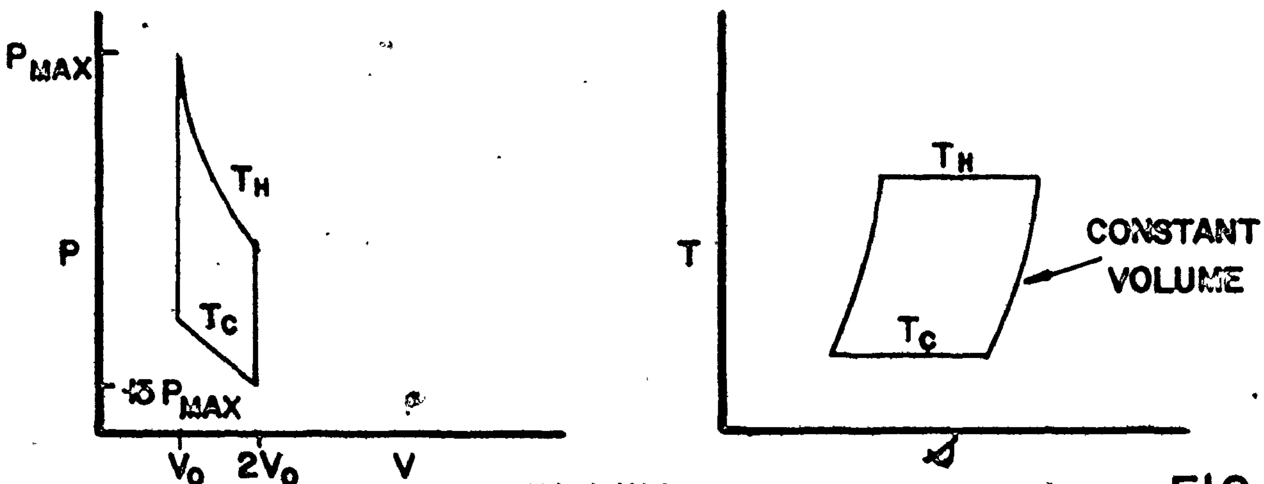
In the ideal cycles the compression and expansion elements are those through which waste low temperature heat and incoming high temperature heat are transmitted from or to the working fluid. These elements are isothermal. Samples of the cycles are shown in Figs. 1, 2 and 3 which have been drawn to the same scale for equal work outputs, peak pressures and temperature ratios. They show the considerable differences between the relative displacement volumes required and the mean effective pressures obtained.

Actual "Stirling" cycles have resemblances to the true Stirling and



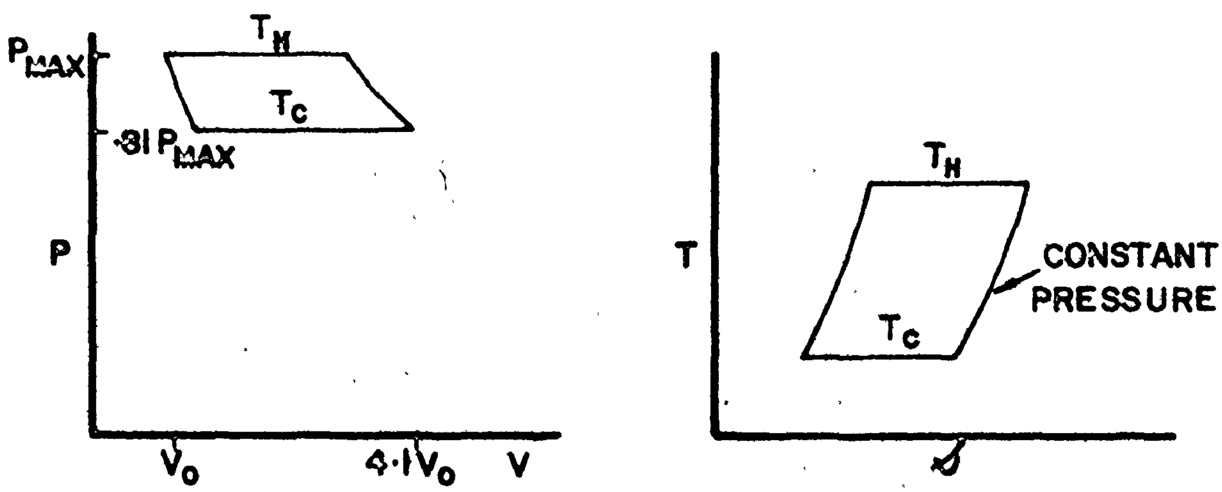
CARNOT

FIG. 1



STIRLING

FIG. 2



ERICSSON

CYCLES OF EQUAL WORK

FIG. 3

Ericsson cycles. They are very complex, but Creswick feels that the classical theory of Schmidt [18] which is a development of the ideal Stirling cycle theory allowing for the type of piston movements occurring in some contemporary Stirling engines (this is considered later), can be used as a basis for the design of Stirling engines. The ideal engines have heat reception at uniform temperature T_H and rejection at T_C. Provided these isothermals are connected by reversible processes during which there is no heat flow to or from the system, that is by adiabatics (Carnot), constant volume regeneration (Stirling) or constant pressure regeneration (Ericsson), we obtain the Carnot efficiency of $1 - \frac{T_C}{T_H}$, i.e. $1 - \tau$. Entropy diagrams are satisfactory for illustrating these cycles but practical engines have continuous variation of conditions between parts of the engine, so that entropy diagrams are of little real help.

As was clearly enunciated by Schüle [25], the reversible processes between the heat supply and rejection processes may be regarded as stepping stones between isothermal operations. If these "stepping stones", whatever they may be, are reversible and do not allow heat input or rejection at other than T_H or T_C the efficiency must be that of Carnot. Thus a cycle having the isothermals connected by perfectly reversible regeneration conditions somewhere between constant volume and constant pressure would also have the Carnot efficiency. Such a cycle is very roughly approximated by the modern "Stirling" engine, as being developed for commercial power units rather than the artificial heart, with one exception. These engines show on their pressure-volume indicator cards line elements of both constant pressure and constant volume. Obviously, there is an infinite number of ideally, Carnot efficiency cycles. In the true Stirling cycle, unlike the Carnot and Ericsson cycles, no work is done between the isothermal operations, these being constant volume processes.

For the Carnot and Ericsson cycles work is indeed done between the isothermals but in each of these cycles the work quantities are equal and opposite, cancelling out through the cycle and resulting in low mean pressures. Thus in any of these cycles the net work may be considered as the differences of areas below the isothermals only. In comparison with the Stirling cycle the Carnot and Ericsson are serious swept volume wasters and the lower the value of τ, the more serious does this waste become. For the Carnot cycle there are further facts that quite remove it from the realm of practical approximation. These have been recognized before but are given here for completeness:

(a) During the isothermal processes the piston-cylinder surfaces defining the fluid boundaries are expected to be perfectly conducting, while during the adiabatic processes they must be perfectly insulating; certainly not an approachable situation.

(b) The work ratio, defined as net output work divided by the sum of all component terms (all taken positive) is only a very small fraction of that achieved by other cycles.

Therefore the Carnot cycle is unattractive on thermodynamic and mechanical grounds.

The regenerative operations of the Stirling and Ericsson cycles are not as simple in practice as theoretical pressure-volume diagrams would suggest. In reality the only way to actually achieve heat transfer to or from the fluid is to transfer the gas from one volume to another, the regenerator lying in the path of gas flow, while the sum of the two volumes either remains constant (Stirling) or changes gradually in such a way that the pressure remains constant (Ericsson). Thus points in the regenerator lines of the entropy and pressure-volume diagrams for these two cycles do not represent a homogeneous state except at the ends of the lines. This is in contradistinction to the Carnot cycle where any point on the diagrams represents a homogeneous state. The Stirling and Ericsson cycles differ between themselves in the nature of the heat transfer phenomena that occur during regenerative processes. In the Ericsson cycle, since the pressure is constant during regeneration there is no tendency for the temperature of the fluid on either side of the regenerator to change and therefore no heat flow to or from the boundaries of the hot and cold spaces. In the Stirling cycle, since the regenerator processes are at constant total volume and the fluid is being heated or cooled the pressure changes progressively during the operation, therefore there is a tendency for its temperature to rise or fall on each side of the regenerator. For reversibility, the hot and cold space boundaries must be perfectly insulating or perfectly conducting, but the former is inconsistent with isothermalcy, so the latter condition must apply. Thus the constant volume regenerative process requires perfect heat transfer between the working fluid and its boundaries at all points in the cycle. Also it is evident that all boundary surfaces in the system are, simultaneously, heat absorbers or heat emitters. The magnitude of this heat transfer is calculated.

1. <u>Heat transfer between working fluid and boundary walls during regeneration in the Stirling cycle</u>

Assume the conditions of the ideal cycle, i.e. isothermalcy and no pressure losses, also that the working fluid obeys the gas laws.

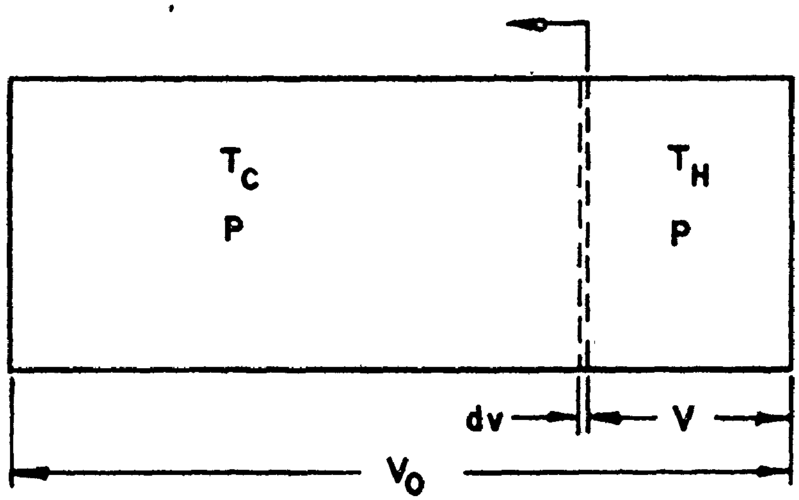
Fig. 4 represents a regenerator matrix (dotted lines) moving relatively through a fixed volume V_o of the working fluid.

(a) Considering the heating process, initially the pressure is

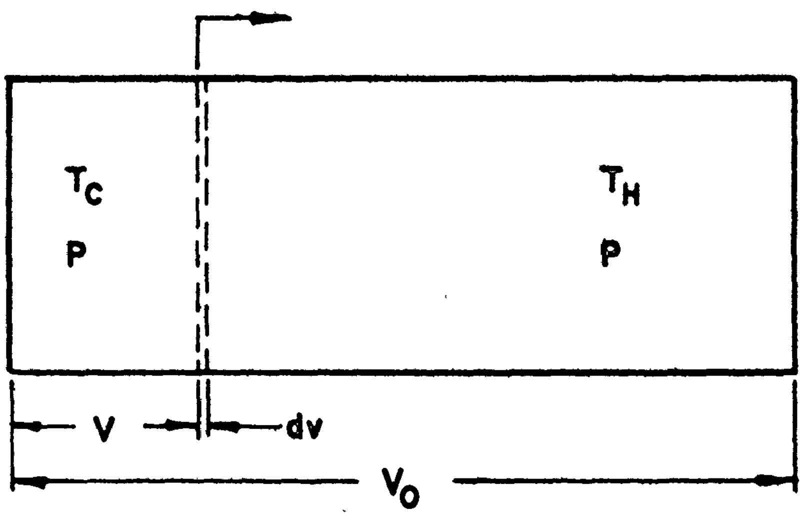
$$P_o = \frac{wRT_C}{V_o} \qquad 3.1$$

After the entire mass of fluid has been traversed the final pressure is

$$P_f = \frac{P_o}{\tau} \qquad 3.2$$



HEATING



COOLING

DIAG. ILLUSTRATING
REGENERATIVE PROCESS

FIG. 4

For intermediate positions when the regenerator has transversed V, from the constancy of the total fluid mass

$$w = w_C + w_H \qquad 3.3$$

$$\frac{P_o V_o}{RT_C} = \frac{P(V_o - V)}{RT_C} + \frac{PV}{RT_H}$$

$$\frac{P}{P_o} = \frac{\frac{V_o}{RT_C}}{\left(\frac{V_o - V}{RT_C}\right) + \frac{V}{RT_H}}$$

$$= \frac{1}{1-(1-\tau)\frac{V}{V_o}} \quad . \qquad 3.4$$

In the cold space the heat already given up by volume dV which is about to enter the regenerator is

$$dQ = \frac{PdV}{J} \ln \frac{P}{P_o} \qquad 3.5$$

or

$$\frac{J}{P_o V_o} dQ = \frac{P}{P_o} \cdot \frac{dV}{V_o} \cdot \ln \frac{P}{P_o}$$

$$= \frac{1}{1-(1-\tau)\frac{V}{V_o}} \cdot d\left(\frac{V}{V_o}\right) \cdot \ln\left(\frac{1}{1-(1-\tau)\frac{V}{V_o}}\right) \ .$$

Initially and finally, $\frac{V}{V_o} = 0$ and $\frac{V}{V_o} = 1$ respectively, thus the expression integrates to

$$Q = \tfrac{1}{2} \frac{P_o V_o}{J} \cdot \frac{1}{1-\tau} \left[\ln\left(\frac{1}{\tau}\right)\right]^2$$

$$= \tfrac{1}{2} \frac{wRT_C}{J} \cdot \frac{1}{1-\tau} \left[\ln\left(\frac{1}{\tau}\right)\right]^2 \ . \qquad 3.6$$

Similarly in the hot space the heat that will be given up by a volume dV that has just traversed the regenerator is

$$dQ = \frac{PdV}{J} \ln\left(\frac{P_o/\tau}{P}\right) \ , \qquad 3.7$$

or

$$\frac{JdQ}{P_oV_o} = \frac{P}{P_o} \cdot \frac{dV}{V_o} \ln\left[\frac{1-(1-\tau)\frac{V}{V_o}}{\tau}\right] .$$

This integrates to

$$Q = \tfrac{1}{2}\,\frac{P_oV_o}{J} \quad \frac{1}{1-\tau}\,[\ln\tau]^2 \qquad 3.8$$

Since $\left|\ln(\frac{1}{\tau})\right| = \left|\ln\tau\right|$ the sum of the heats transferred to the hot and cold space boundaries, taking all quantities positive, is

$$\frac{P_oV_o}{J} \cdot \frac{1}{1-\tau}\,[\ln(\tfrac{1}{\tau})]^2 \qquad 3.9$$

or

$$\frac{wRT_C}{J} \cdot \frac{1}{1-\tau}\,[\ln(\tfrac{1}{\tau})]^2 . \qquad 3.10$$

(b) Considering the cooling process,

Initial pressure $P_o = \dfrac{wRT_H}{V_o}$. 3.11

Final pressure $P_f = P_o\tau$. 3.12

Intermediate pressure after traversing a volume V is

$$P = \frac{P_o}{1-(1-\frac{1}{\tau})\frac{V}{V_o}} , \qquad 3.13$$

or

$$\frac{P}{P_o} = \frac{1}{1-(1-\frac{1}{\tau})\frac{V}{V_o}} .$$

Volume dV in the hot space about to enter the regenerator will have absorbed heat

$$dQ = \frac{PdV}{J}\ln(\frac{P_o}{P}) , \qquad 3.14$$

$$\frac{J}{P_oV_o}\,dQ = \frac{1}{1-(1-\frac{1}{\tau})\frac{V}{V_o}} \cdot \frac{dV}{V_o} \cdot \ln[1-(1-\tfrac{1}{\tau})\tfrac{V}{V_o}] .$$

This integrates over the full volume to

$$Q = \tfrac{1}{2}\frac{wRT_H}{J}\;\frac{\tau}{1-\tau}\,[\ln\tfrac{1}{\tau}]^2 \qquad 3.15$$

$$= \tfrac{1}{2}\,\frac{wRT_H}{J} \cdot \frac{1}{1-\tau}\,[\ln\tfrac{1}{\tau}]^2 \qquad 3.16$$

Similarly the heat supplied to the gas on the cold side is

$$Q = \frac{1}{2} \frac{wRT_C}{J} \cdot \frac{1}{1-\tau} [\ln\tau]^2 \quad . \qquad 3.17$$

The total heat exchange for all boundary surfaces through one cycle, taking all quantities positive is thus

$$2 \frac{wRT_C}{J} \cdot \frac{1}{1-\tau} [\ln\tau]^2 \qquad 3.18$$

In contrast to this the Ericsson cycle has simpler heat transfer processes in as far as the regenerator processes are at constant pressure. There is therefore no tendency to temperature change due to pressure change, thus no requirement for heat interchange with the boundary walls during regeneration. The heat exchange within the regenerator is greater than for the Stirling cycle by the factor of $\frac{C_P}{C_V}$ or γ.

2. Relation between cycle work output and regenerator heat transfer

The heat exchanged between the working fluid and the regenerator matrix is many times the work output of the cycle, whether Stirling or Ericsson. Since heat exchange by forced convection is achieved only at the cost of mechanical energy, the ratio of net work (expressed in heat units) to regenerator heat exchange is likely to influence strongly the internal efficiency of the engine. For the ideal Stirling cycle, with reference to Fig. 5:

(a) The work done per cycle expressed in heat units is:

$$W = \frac{P_1V_1}{J} (1-\tau)\ln r \ , \qquad 3.19$$

$$= \frac{wRT_H}{J} (1-\tau)\ln r \ , \qquad 3.20$$

$$\text{where } \tau = \frac{T_C}{T_H} \text{ and } r = \frac{V_2}{V_1} .$$

The total heat transferred to and from the regenerator, through a cycle, all quantities being positive, is

$$Q = 2wC_V(T_H - T_C) \qquad 3.21$$

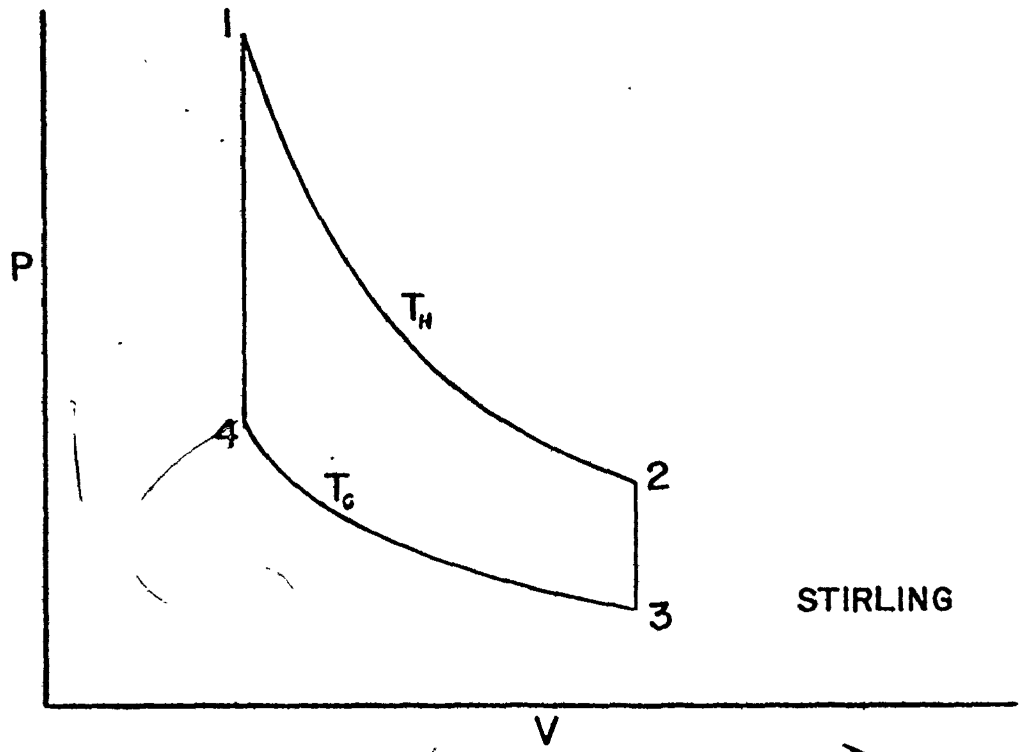


FIG. 5

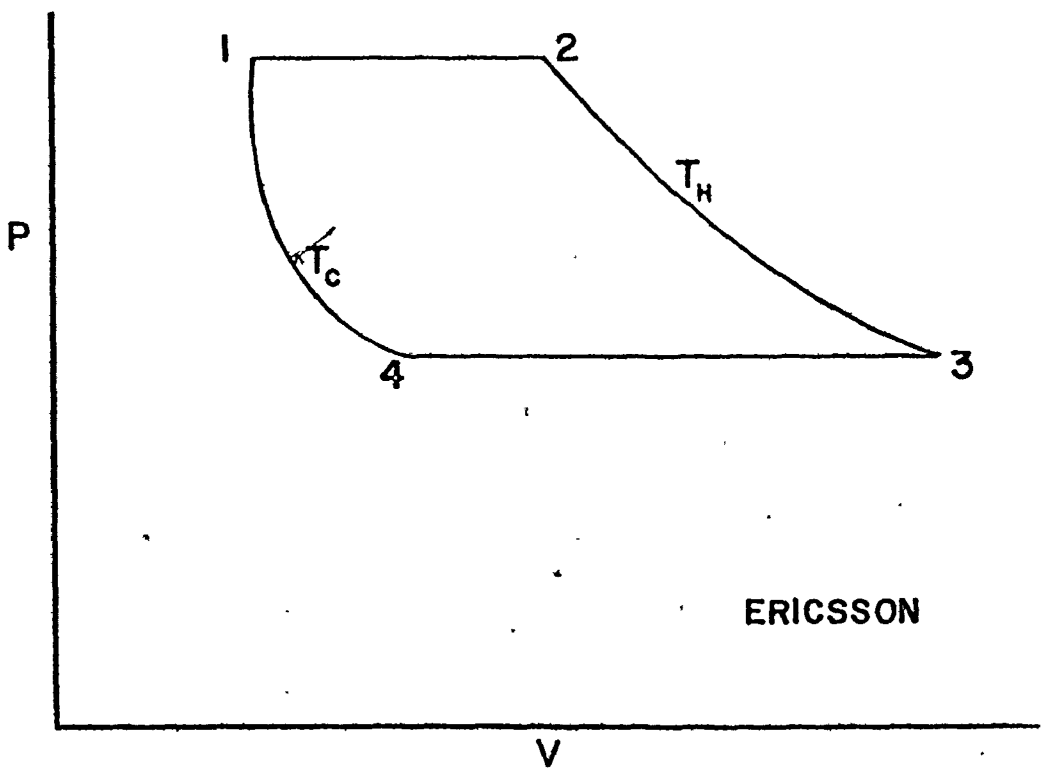


FIG. 6

BASIC CYCLES

$$= 2wC_VT_H(1-\tau) \quad ,$$

or

$$\frac{W}{Q} = \frac{wRT_H(1-\tau)\ln r}{2wJC_VT_H(1-\tau)} \quad ,$$

$$= \frac{(\gamma - 1)}{2} \ln r \quad . \qquad 3.22$$

(b) Ericsson Cycle

Referring to Fig. 6 the work done per cycle is essentially the difference of the areas below the isothermals and is

$$W = \frac{P_2V_2}{J} \ln \frac{V_3}{V_2} - \frac{P_1V_1}{J} \ln \frac{V_4}{V_1} \quad , \qquad 3.23$$

$$= \frac{wRT_C}{J} \ln (r\tau)\left[\frac{T_H}{T_C} - 1\right] \quad ,$$

$$= \frac{wRT_C}{J} \left(\frac{1-\tau}{\tau}\right) \ln(r\tau) \quad . \qquad 3.24$$

The regenerator heat transfer per cycle is

$$Q = 2wC_p(T_H - T_C)$$

$$= 2wC_pT_H(1-\tau) \quad . \qquad 3.25$$

$$\therefore \frac{W}{Q} = \frac{\frac{wRT_C}{J}\left(\frac{1-\tau}{\tau}\right) \ln(r\tau)}{2wC_pT_H(1-\tau)}$$

$$= \left(\frac{\gamma-1}{2\gamma}\right) \ln(r\tau) \quad . \qquad 3.26$$

During regenerator heating the working fluid expands by a ratio $1/\tau$. Thus a minimum overall expansion (compression) ratio of $1/\tau$ is required for this cycle and at this expansion ratio the power output is zero. Expansion ratios greater than $1/\tau$ result in a net power output.

It is notable that the work/regenerator heat ratio is independent of temperature ratio τ for the Stirling cycle but dependent upon τ for the Ericsson. Both cycles benefit considerably by using high gamma gases.

These work/regenerator heat ratios are shown in Fig. 7 (Stirling) and Figs. 8 and 9 (Ericsson) for hydrogen (gamma = 1.404) and helium (gamma = 1.667).

3. Heat exchange between gas and boundary surfaces compared with regeneration

The heat exchange with the hot and cold space boundaries during regenerative processes in comparison with that of the regenerator matrix is interesting.

For the Stirling cycle, from the quantities derived previously, for a complete cycle

$$\frac{\text{Boundary heat exchange}}{\text{Regenerator heat exchange}} = \frac{2wRT_C \cdot (\frac{1}{1-\tau})(\ln\tau)^2}{2wC_V T_H J(1-\tau)}$$

$$= (\gamma-1)\tau\left(\frac{\ln\tau}{1-\tau}\right)^2 \qquad 3.27$$

This ratio is plotted against τ in Fig. 10. The ratio eases slightly as engines become "hotter", that is as τ decreases. This function is also independent of the compression ratio. However, since the regenerator heat exchange per unit of work output decreases with increasing compression ratio, the boundary heat exchange improves in the same proportion.

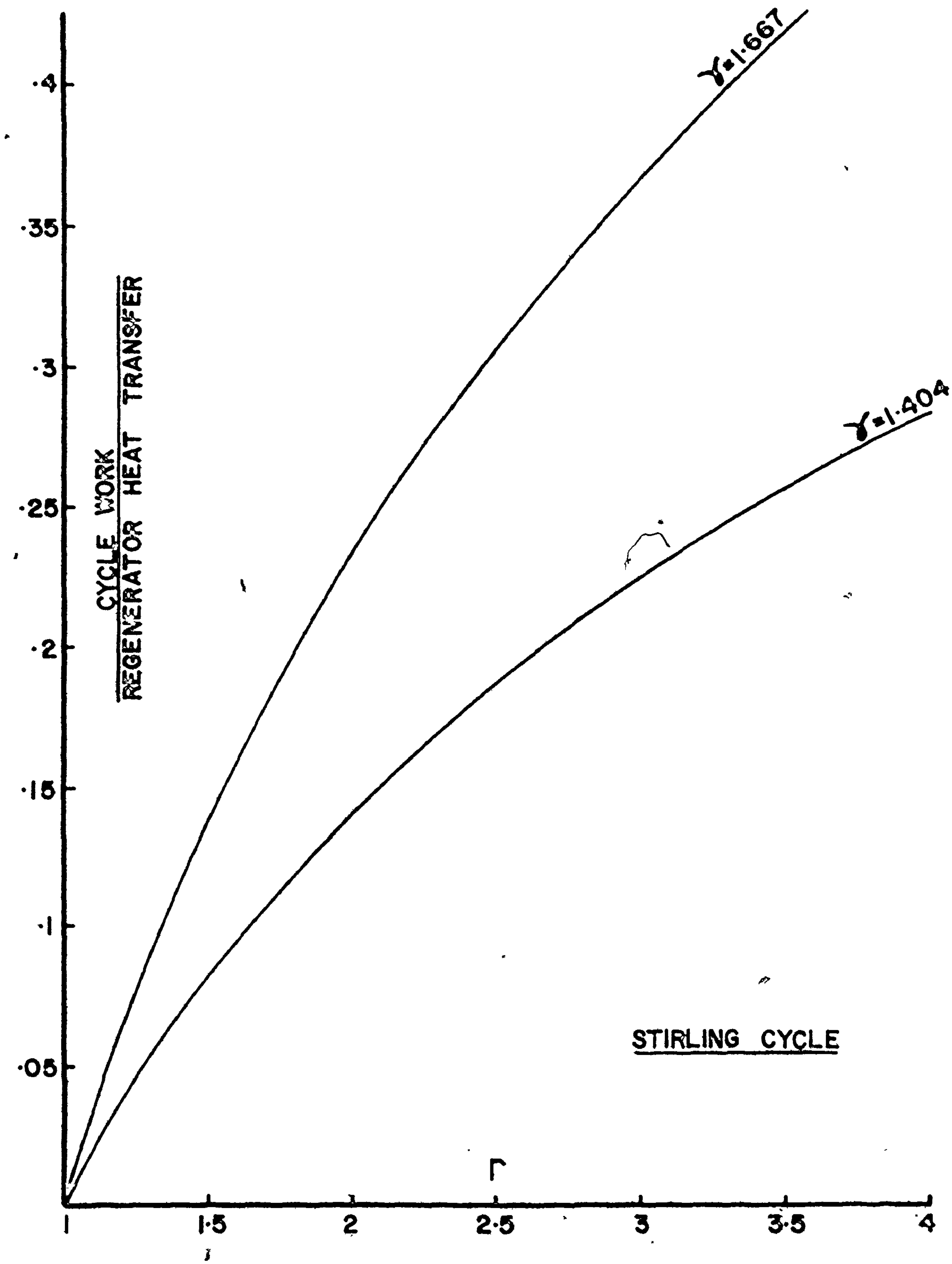
It should be noted that for the Ericsson cycle, due to constant pressure during regeneration this ratio is zero.

4. Effect of regenerator efficiency

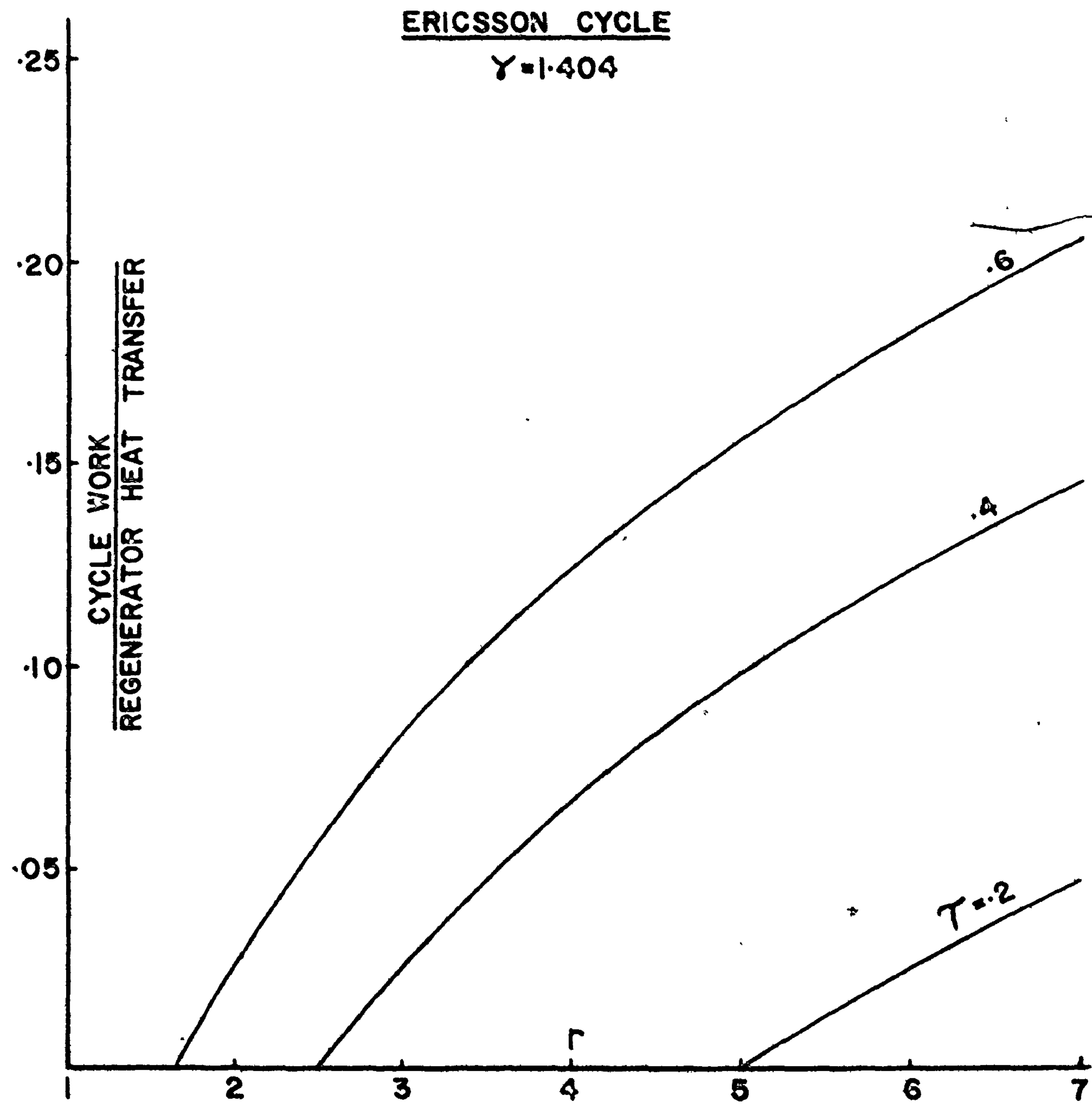
In the preceding analysis it was assumed that the regenerator efficiency was unity. This value is unlikely to be approached in practice due to the large surface areas required and accompanying void volume. Defining the regenerator efficiency η_R as the heat recovered on the cold blow per unit of heat potentially depositable in the regenerator on the hot blow:

(a) Stirling cycle (see Fig. 5)

Cycle heat input $= \frac{P_1V_1 \ln r}{J} + C_V \frac{P_1V_1}{RT_H}(T_H - T_C)(1-\eta_R)$, with the first term being the Carnot heat input as for the perfect regenerator cycle and

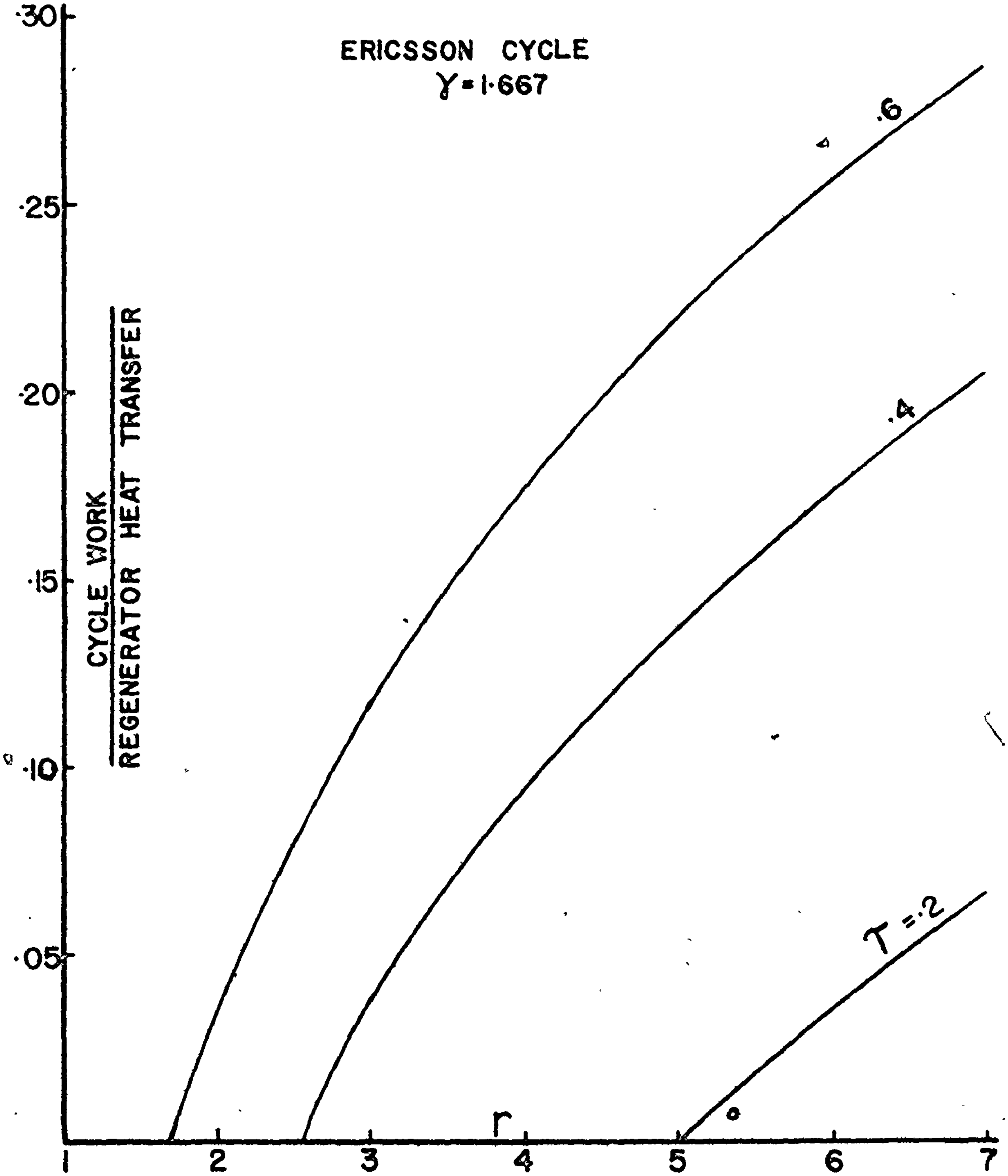


CYCLE WORK / REGENERATOR HEAT TRANSFER V.S. r FIG. 7



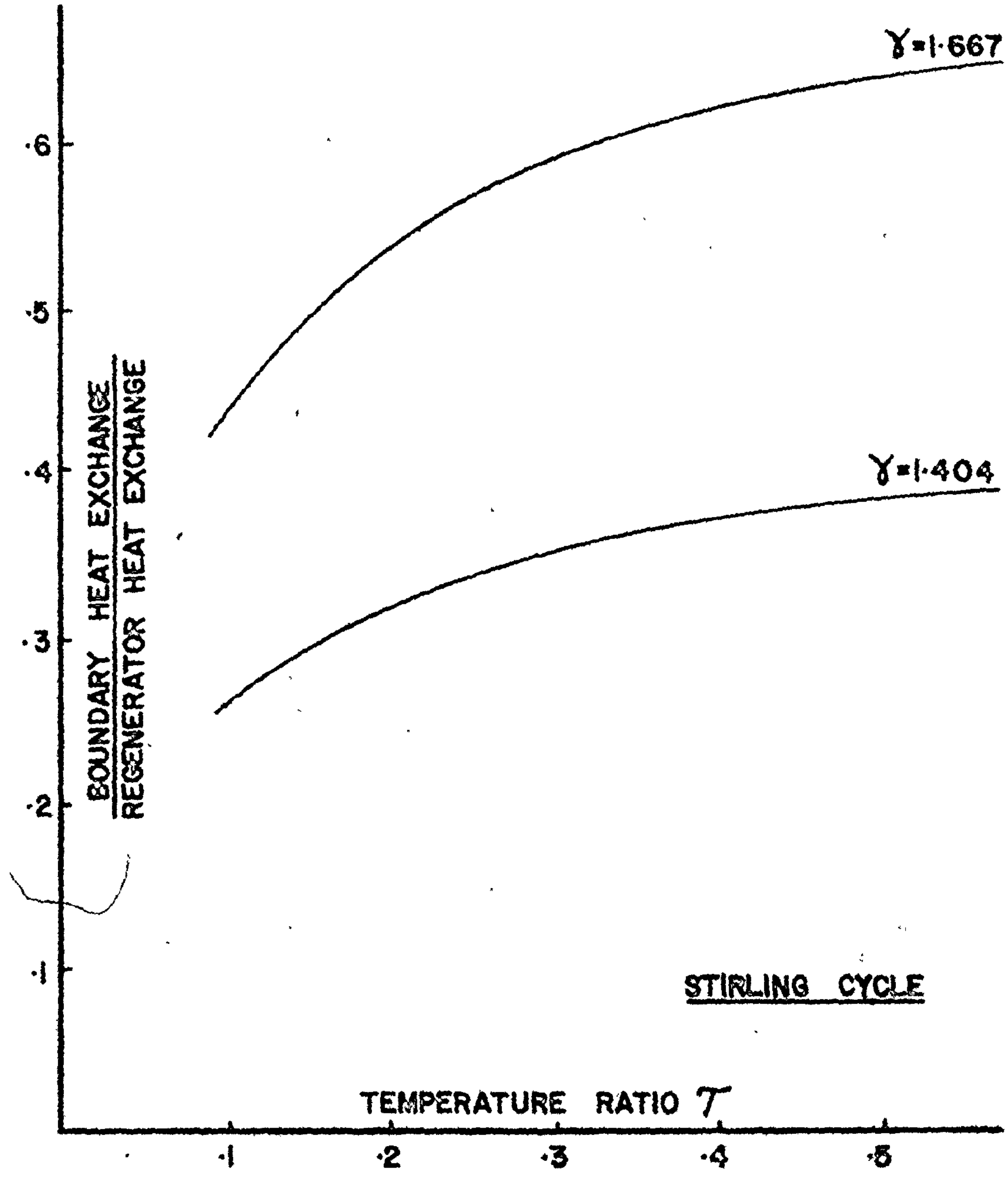
$\dfrac{\text{CYCLE WORK}}{\text{REGENERATOR HEAT TRANSFER}}$ V.S. r

FIG. 8



$\dfrac{\text{CYCLE WORK}}{\text{REGENERATOR HEAT TRANSFER}}$ V.S. r

FIG. 9



$\dfrac{\text{BOUNDARY HEAT EXCHANGE}}{\text{REGENERATOR HEAT EXCHANGE}}$ V.S. τ FIG. 10

the second term representing the additional heat due to the regenerator imperfection. As before the cycle work done is $\frac{P_1V_1}{J}(1-\tau)\ln r$ when expressed in heat units. Hence the cycle efficiency is expressed as

$$\eta = \frac{\frac{P_1V_1}{J}(1-\tau)\ln r}{\frac{P_1V_1}{J}\ln r + C_V\frac{P_1V_1}{RT_H}(T_H - T_C)(1-\eta_R)},$$

$$= \frac{(1-\tau)}{1+\frac{(1-\tau)(1-\eta_R)}{(\gamma-1)\ln r}} \qquad 3.28$$

A similar expression is given by Inchley [66].

(b) Ericsson Cycle (see Fig. 6)

By similar reasoning, for this cycle

$$\eta = \frac{(1-\tau)}{1+\frac{\gamma(1-\tau)(1-\eta_R)}{(\gamma-1)\ln(\tau r)}}, \quad \tau r > 1 \qquad 3.29$$

We see that both cycles change from independence to dependence on the compression ratio. The efficiencies also become dependent upon γ. Rather extreme combinations of temperature and regenerator efficiency are shown in Figs. 11 and 12 for the Stirling cycle. Essentially the same information is contained in Fig. 13 which shows the efficiency relative to the Carnot efficiency. The great help given by increasing the compression ratio at the lower values of r is evident.

5. The mean effective pressure to the maximum pressure ratio

This has two main influences:

(i) For a given maximum cycle pressure it controls the cylinder volumes required for a given power and frequency.

(ii) For a given engine power output, the economy of material usage for the engine cylinders is controlled by this factor. Thus a

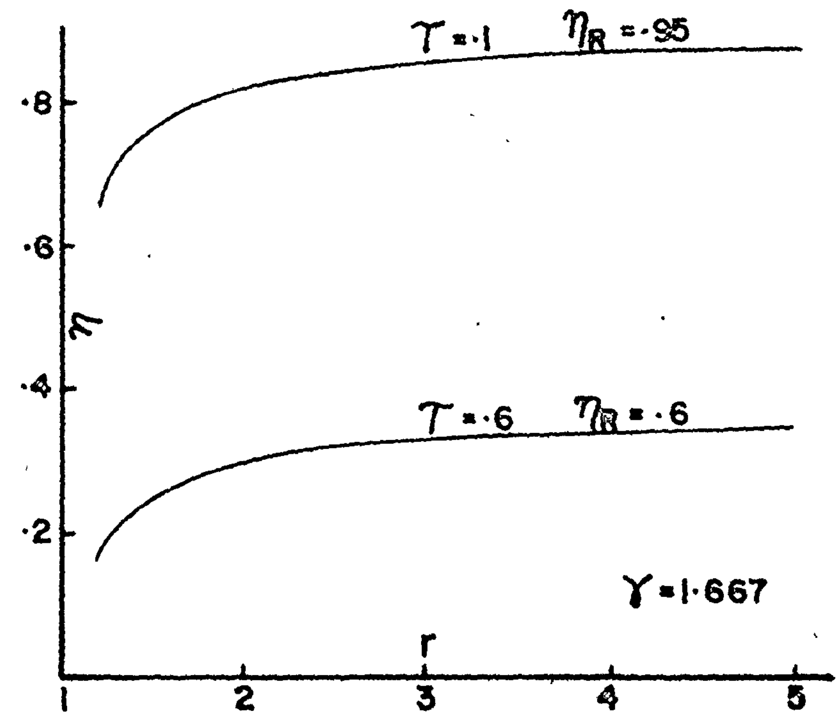


FIG. 11

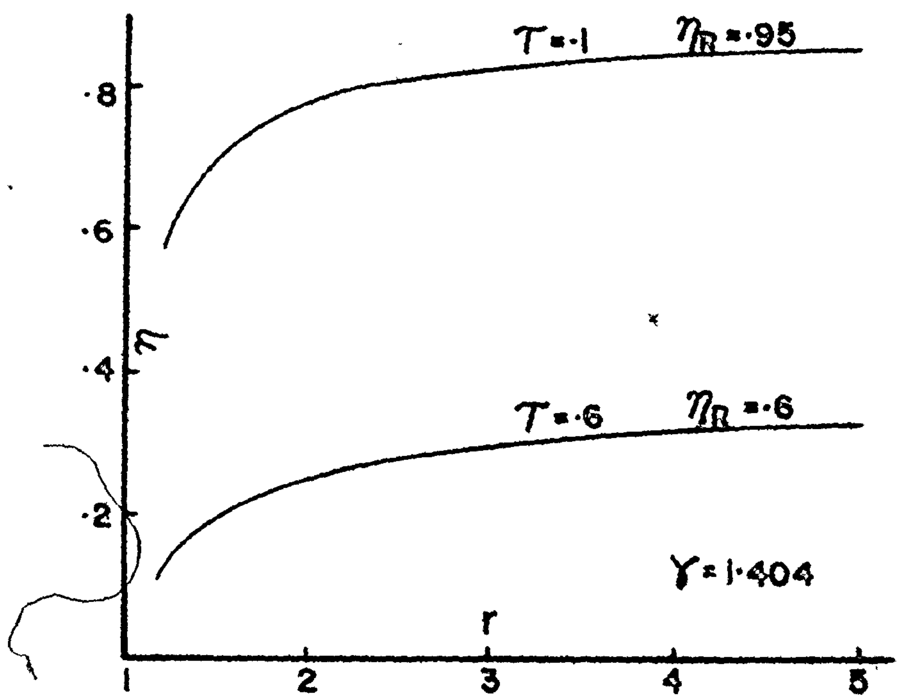
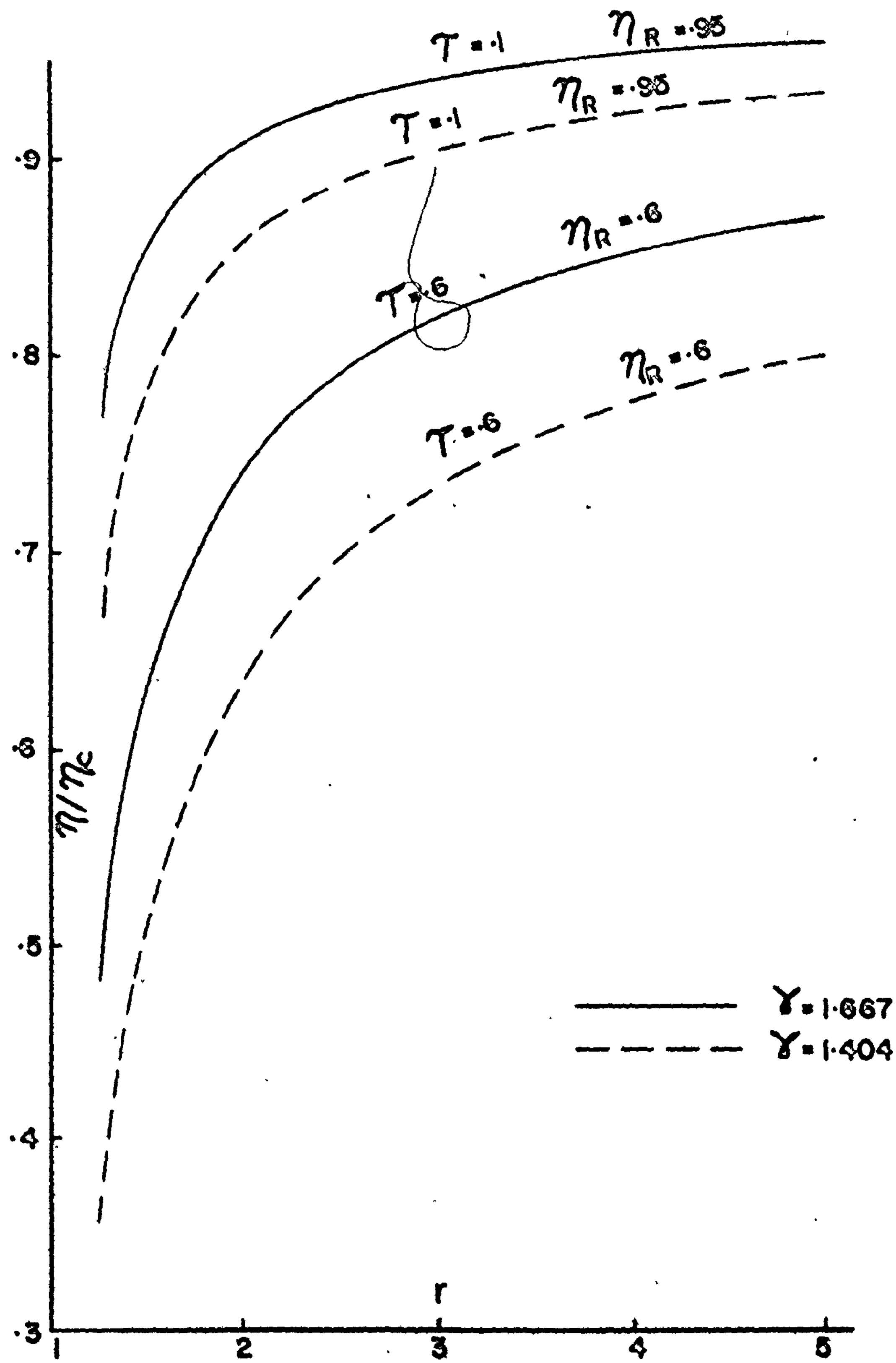


FIG. 12

EFFECT OF REGENERATOR EFFICIENCY
ON CYCLE EFFICIENCY



EFFECT OF REGENERATION EFFICIENCY ON CYCLE EFFICIENCY / CARNOT EFFICIENCY

FIG. 13

high value leads to minimum cylinder wall material as a result of minimum volume, and much more important, to minimum parasitic heat loss from cylinder temperature gradients.

(a) Stirling Cycle (Fig. 5)

$$\text{Mean effective pressure (MEP)} = \frac{P_1V_1\ln(\frac{V_2}{V_1}) - P_4V_4\ln(\frac{V_4}{V_3})}{V_2 - V_1}$$

$$= P_1 \frac{(1-\tau)\ln r}{(r-1)} .$$

$$\therefore \frac{MEP}{P_{max}} = \frac{(1-\tau)\ln r}{(r-1)} . \qquad 3.30$$

(b) Ericsson Cycle (Fig. 6)

$$\text{Mean effective pressure MEP} = \frac{P_2V_2\ln(\frac{V_3}{V_2}) - P_1V_1\ln(\frac{V_4}{V_1})}{V_3 - V_1}$$

$$= P_1 \frac{(\frac{1}{\tau} - 1)\ln(r\tau)}{(r-1)} .$$

$$\therefore \frac{MEP}{P_{max}} = \frac{(1-\tau)\ln(r\tau)}{\tau(r-1)} , \quad r\tau > 1 \qquad 3.31$$

For the Ericsson cycle the mean effective pressure ratio is only zero at the minimum compression ratio for this cycle, i.e. at $r = 1/\tau$. It climbs slowly to a maximum (.292 for $\tau = .3$ at $r \approx 8.1$) which although it is greater than for the Stirling (.206) at this compression ratio it does not seem to be exploitable due to the large void volumes of regenerators (evidence for this is given in Chapter 6).

6. Work ratio

The work ratio is defined as the net cycle work divided by the total work of the cycle elements, all work quantities being positive. Internally it is an indicator of the insensitivity of the output to departures from the ideal cycle, externally it indicates the insensitivity to mechanical losses.

(a) Stirling Cycle (Fig. 5)

$$\text{Work ratio} = \frac{P_1V_1 \ln r - P_4V_4 \ln r}{P_1V_1 \ln r + P_4V_4 \ln r}$$

$$= \frac{1-\tau}{1+\tau}\,. \qquad 3.32$$

Although this is based upon the P-V diagram which does not show the transfer of gas from one volume to another during the regenerative process the pressure difference necessary for this is ideally zero, so that no work is done and the expression is valid. For $\tau = .3$ the work ratio is thus .538.

(b) Ericsson Cycle (Fig. 6)

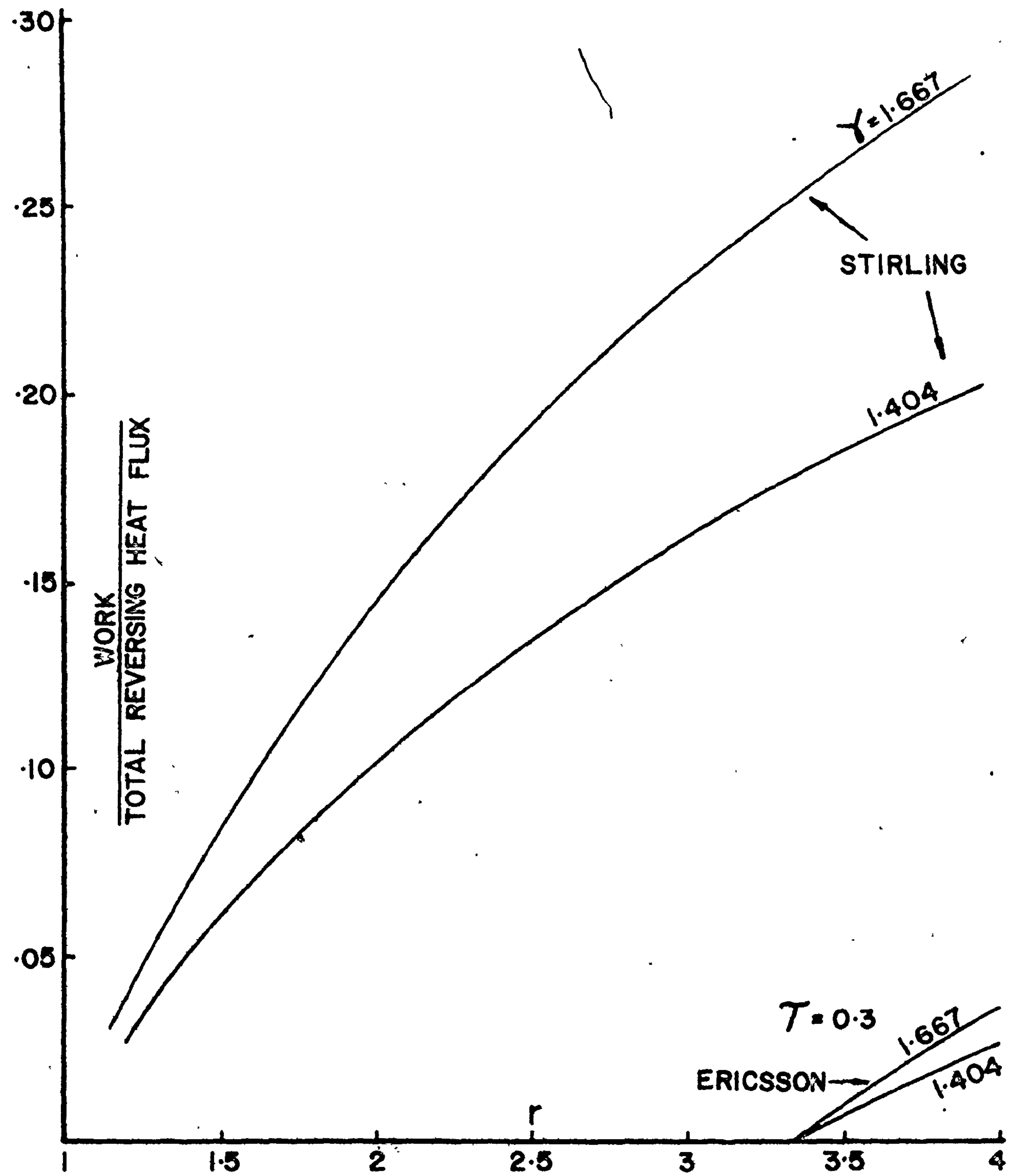
$$\text{Work ratio} = \frac{P_2V_2 \ln\left(\frac{V_3}{V_2}\right) - P_4V_4 \ln\left(\frac{V_4}{V_1}\right)}{P_2V_2 \ln\left(\frac{V_3}{V_2}\right) + P_4V_4 \ln\left(\frac{V_4}{V_1}\right) + (P_2V_2 - P_1V_1) + (P_3V_3 - P_4V_4)}$$

$$= \frac{(1-\tau)\ln(r\tau)}{(1+\tau)\ln(r\tau) + 1 + r\tau - 2\tau}\,, \quad r\tau > 1 \qquad 3.33$$

This expression has zero value at the minimum compression ratio of $1/\tau$, thereafter increasing to a low maximum (.159 for $\tau = .3$ at $r \approx 10$) and then declines continuously.

7. Work output/total reversing heat

We have considered the reversible heat flows for the boundary surfaces and the regenerator matrix within the Stirling and Ericsson engines. When these are totalled for each engine and related to the work output the characteristics of Fig. 14 result, again taking $\tau = .3$. For compression ratios likely to be practical the Stirling cycle is the only feasible one. The high gamma gas is still superior, but its advantage is not as marked as it is when considering the regenerator heat transfer alone (Fig. 7), having been offset somewhat by its relative disadvantage for heat transfer requirements across the hot and cold space boundaries. Nonetheless, a great improvement with increasing compression ratio still remains.



$\frac{\text{WORK}}{\text{TOTAL REVERSING HEAT FLUX}}$ V.S. r FIG. 14

Summary

Of the ideal cycles considered the classical Carnot reference cycle is seen to be quite impossible to translate into a working engine. This leaves the Stirling and Ericsson cycles for further consideration.

These two cycles have been examined in terms of work-regenerator heat flow, the effect of imperfect regeneration, mean effective pressure and work ratio. Generally, the Stirling cycle is superior to the Ericsson. Using helium (gamma = 1.667) and taking $\tau = .3$, fairly representative of near future and current work in the artificial heart field, from the relationships developed, it follows that;

	Work/Regenerator Heat Flow	Cycle Efficiency $\eta_R = .95$	$\eta_R = .80$	M.E.P. / P_{max}	Work Ratio
Stirling*					
r = 1.5	.14	.62	.46	.57	.54
r = 2.5	.31	.66	.57	.43	.54
r = 6.66	.63	.68	.63	.23	.54
Ericsson*					
r = 3.33	0	0	0	0	0
r = 6.66	.14	.62	.47	.29	.15

* Compression ratios selected:

1.5 ≃ compression ratio of contemporary "Stirling" engines, estimated
2.5 = improved ratio, arbitrary
6.66 = same as Ericssson for comparison
3.33 = Ericsson minimum ratio; i.e. starting point
6.66 = Twice Ericsson minimum, arbitrary

These figures show the clear superiority of the Stirling cycle in terms of these thermodynamic parameters.

CHAPTER 4

OTHER CYCLES

Regenerative Otto Cycle

In many "Stirling" engines, especially those of several horsepower upwards, the supply of heat to the working fluid is through tubular heaters located in series between the regenerator and the hot cylinder. The fluid space within the heaters together with the space within the cylinder comprise the hot space. With this arrangement the heat is supplied mainly during fluid transfer from the cold to the hot side. Therefore, relatively little heat can be supplied to an element of the fluid once it leaves the heater and enters the cylinder. A similar situation applies to the cold side where heat rejected from the cycle is absorbed in a series-connected cooler between the cold cylinder and the regenerator. (These remarks do not apply to current heart engines, where the heat source is thermally integral with the cylinder.) The cycle therefore approximates more to the Otto than to the true Stirling cycle, except that the regenerator substantially reduces the net heat input to the engine. The performance of this cycle, which I term a "regenerative Otto cycle" does not appear to be given in readily available literature, so I investigate this cycle for ultimate comparison with the Carnot potential cycles.

In this investigation the simplifying assumption has been made that the fluid volume within the heat exchangers is negligible in comparison with the swept volumes of the pistons, so that the expansion and compression pressures are not influenced by isothermal conditions that would probably be approximated in the heat exchangers, but would be determined only by adiabatic relations. This condition would not of course apply in practice but the assumption of a true regenerative Otto cycle does show what happens in the extreme case to the engine performance when all of the heat supply and rejection occurs only during the workless constant volume elements, instead of only during the expansion and compression elements as was the case for the engines of potential Carnot efficiency.

1. Cycle efficiency

With reference to Fig. 15 which shows the cycle on log temperature ratio - log volume ratio coordinates, useful regeneration is possible when the temperature at the end of expansion is greater than at the end of compression, i.e. $T_2 > T_4$. (The volume ratio at the end of expansion, i.e. the compression ratio shown has no particular significance.)

The cycle work in heat units is given by:

$$\frac{(P_1V_1 - P_2V_2) - (P_4V_4 - P_3V_3)}{J(\gamma-1)} \qquad 4.1$$

Heat input, allowing for a regenerator efficiency η_R is

$$= wC_V\{T_1 - [T_4 + \eta_R(T_2 - T_4)]\} \qquad 4.2$$

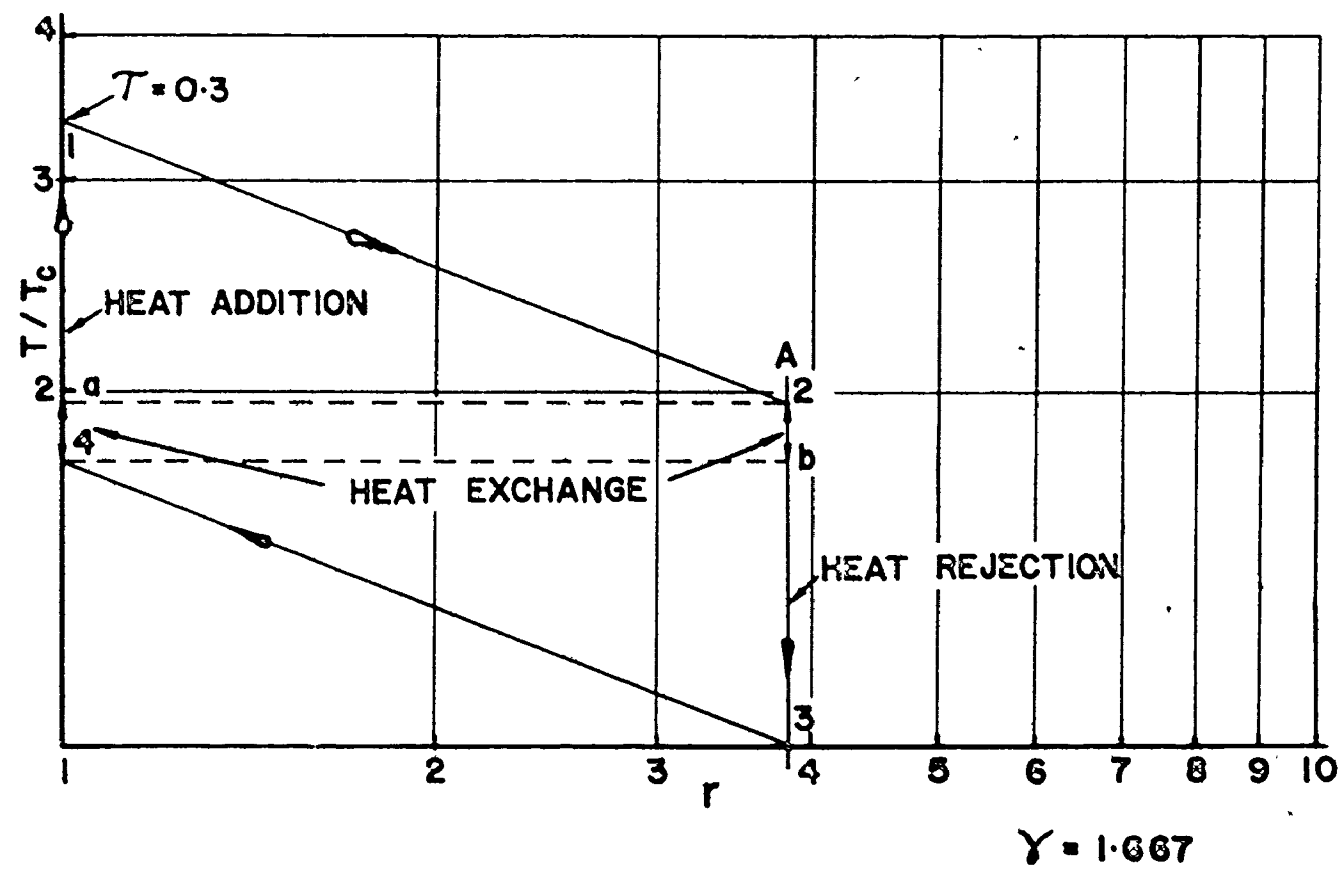
$$\text{Cycle efficiency } \eta = \frac{(P_1V_1 - P_2V_2) - (P_4V_4 - P_3V_3)}{J(\gamma-1)wC_V\{T_1 - [T_4 + \eta_R(T_2 - T_4)]\}}$$

$$= \frac{1 - \frac{1}{r^{\gamma-1}} - \tau r^{\gamma-1} + \tau}{1 - \tau r^{\gamma-1} + \eta_R(\tau r^{\gamma-1} - \frac{1}{r^{\gamma-1}})} \qquad 4.3$$

Regeneration is effective for compression ratios up to the value given when $T_2 = T_4$, i.e.

$$\frac{T_1}{r^{\gamma-1}} = T_1\, r^{\gamma-1}\cdot \tau$$

$$\text{or } r = (\frac{1}{\tau})^{\frac{1}{2\gamma-2}} \qquad 4.4$$



REGENERATIVE OTTO CYCLE FIG. 15

For a non-regenerative cycle this would be the optimum compression ratio. If the above expression for η is plotted against r for given η_R and τ a clear maximum efficiency η is obtained. The compression ratio for this maximum becomes more critical as τ increases.

If the maximum efficiencies are plotted against r for the selected values of τ and η_R the charts of Figs. 16a and 16b are obtained. On these charts the intersection of the η_R and τ lines defines the compression ratio that must be used to achieve the best possible cycle efficiency. Any other compression ratio will reduce the efficiency. The most striking feature is that low compression ratios and very high regenerator efficiences must be adopted to achieve high cycle efficiencies. For example, for $\tau = .3$ a regenerator imperfection of only .02, i.e. $\eta_R = .98$, will lower the cycle efficiency from .7 to .61.

2. Work/regenerator heat transfer

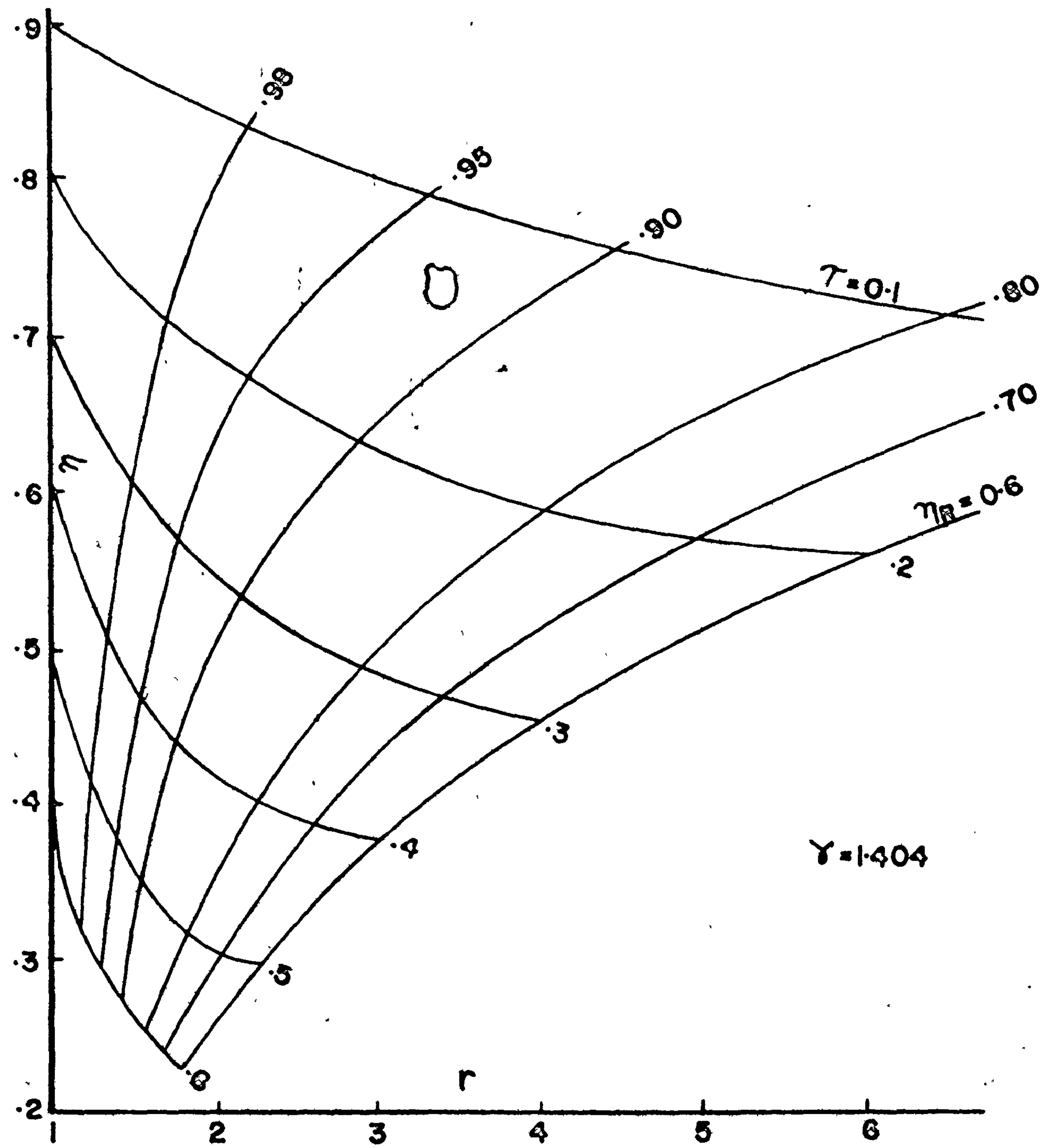
In view of the very high regenerator efficiencies required for an efficient engine operating on this cycle the work/regenerator heat transfer ratio will be calculated. This ratio is an indicator of the relative losses incurred when the working fluid passes through the regenerator. From the salient point numbering of Fig. 15,

$$\text{Cycle work} = \frac{(P_1V_1 - P_2V_2) - (P_4V_4 - P_3V_3)}{J(\gamma-1)} \; .$$

$$\text{Regenerator heat transfer} = 2wC_V\eta_R(T_2 - T_4) \; .$$

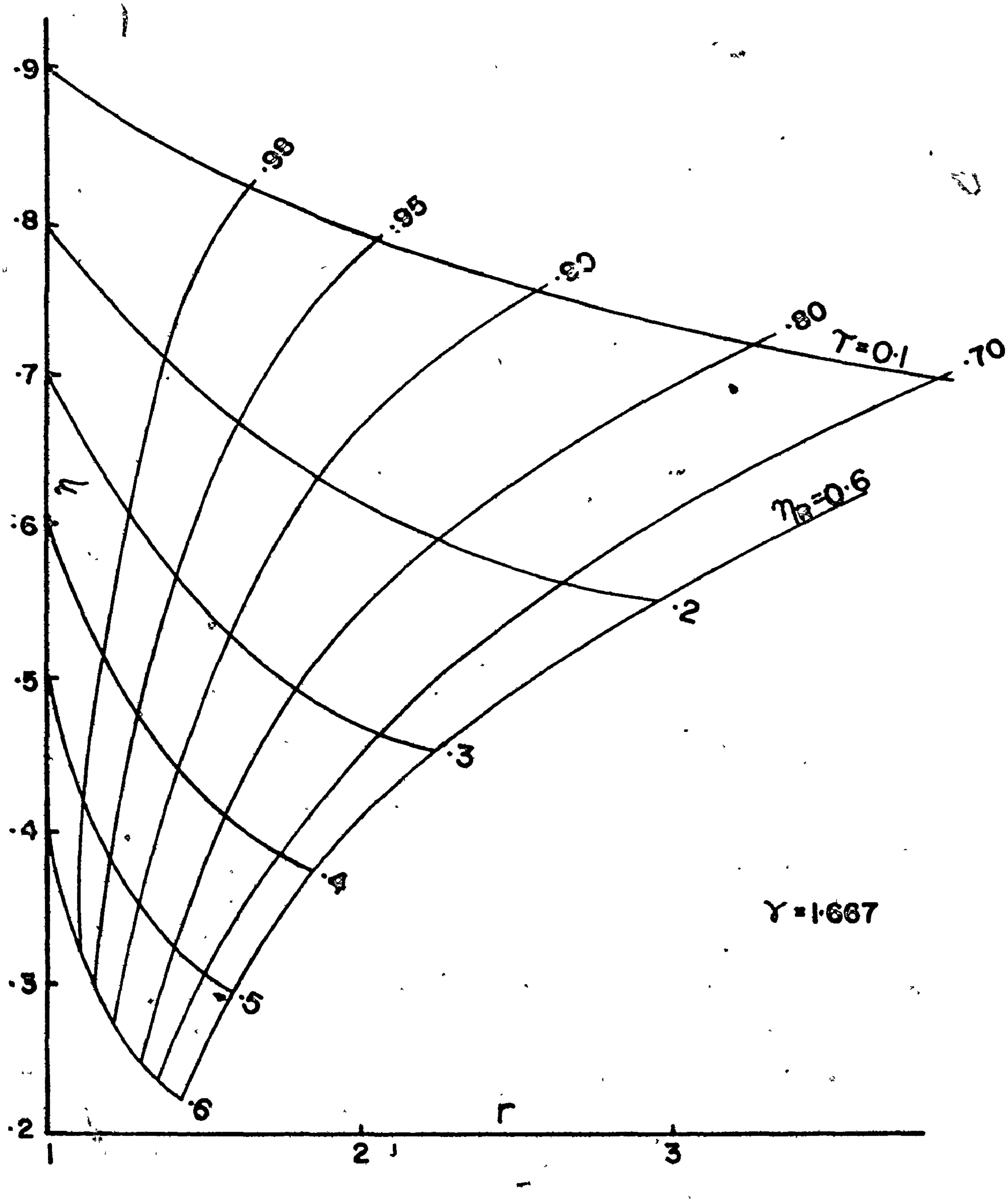
$$\text{Work/heat transfer} = \frac{(P_1V_1 - P_2V_2) - (P_4V_4 - P_3V_3)}{(\gamma-1)2wC_V\eta_R(T_2 - T_4)J} \; ,$$

$$= \frac{1 - \frac{1}{r^{\gamma-1}} - \tau r^{\gamma-1} + \tau}{2\eta_R\left[\frac{1}{r^{\gamma-1}} - \tau r^{\gamma-1}\right]} \; . \qquad 4.5$$



REGENERATIVE OTTO CYCLE
MAXIMUM IDEAL EFFICIENCIES

FIG. 16a



REGENERATIVE OTTO CYCLE
MAXIMUM IDEAL EFFICIENCIES

FIG. 16b

Applying this to an optimum helium cycle operating at $\tau = .3$ and $\eta_R = .98$ and therefore having $r = 1.24$, for example:

$$\text{Work/heat transfer} = \frac{1 - \dfrac{1}{1.24^{.667}} - .3 \cdot 1.24^{.667} + .3}{2 \times .98\left[\dfrac{1}{1.24^{.667}} - .3 \cdot 1.24^{.667}\right]}$$

$$= .085$$

This is slightly better than the Stirling ratio .073 for the same compression ratio.

3. Mean effective pressure

$$\frac{\text{Mean effective pressure}}{\text{Maximum pressure}} = \frac{(P_1V_1 - P_2V_2) - (P_4V_4 - P_3V_3)}{(\gamma-1)(V_2 - V_1)},$$

$$= \frac{(1 - \dfrac{1}{r^{\gamma-1}} - r^{\gamma-1} + \tau)}{(\gamma-1)(r-1)}, \qquad 4.6$$

showing that for this cycle the lower gamma gases have the advantage.

The non-dimensional mean effective pressure is plotted for $\tau = .3$ in Fig. 17, together with that of the Stirling cycle.

4. Work ratio

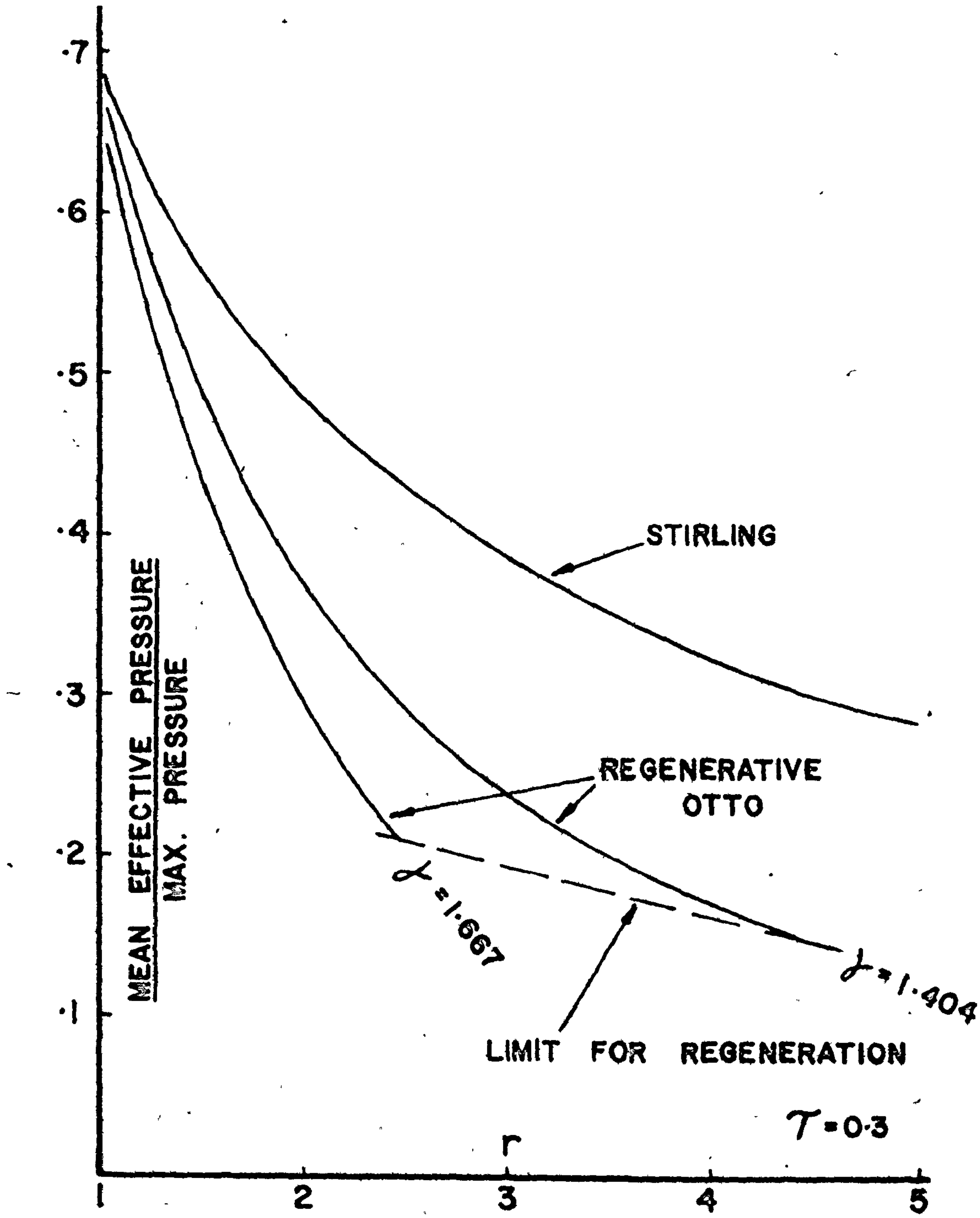
$$\text{Work ratio} = \frac{\dfrac{P_1V_1 - P_2V_2}{\gamma-1} - \dfrac{P_4V_4 - P_3V_3}{\gamma-1}}{\dfrac{P_1V_1 - P_2V_2}{\gamma-1} + \dfrac{P_4V_4 - P_3V_3}{\gamma-1}},$$

$$= \frac{T_1 - T_2 - T_4 + T_3}{T_1 - T_2 + T_4 - T_3},$$

$$= \frac{1 - \dfrac{1}{r^{\gamma-1}} - \tau \cdot r^{\gamma-1} + \tau}{1 - \dfrac{1}{r^{\gamma-1}} + \tau \cdot r^{\gamma-1} - \tau}.$$

$$= \frac{1 - \tau r^{\gamma-1}}{1 + \tau r^{\gamma-1}} \qquad 4.7$$

For the same conditions as above,



$\frac{\text{MEAN EFFECTIVE PRESSURE}}{\text{MAX. PRESSURE}}$ VS. r

FIG. 17

$$\text{Work ratio} = \frac{1 - 0.3 \times 1.24^{.667}}{1 + 0.3 \times 1.24^{.667}}$$

$$= .49$$

which compares with .54 for the Stirling.

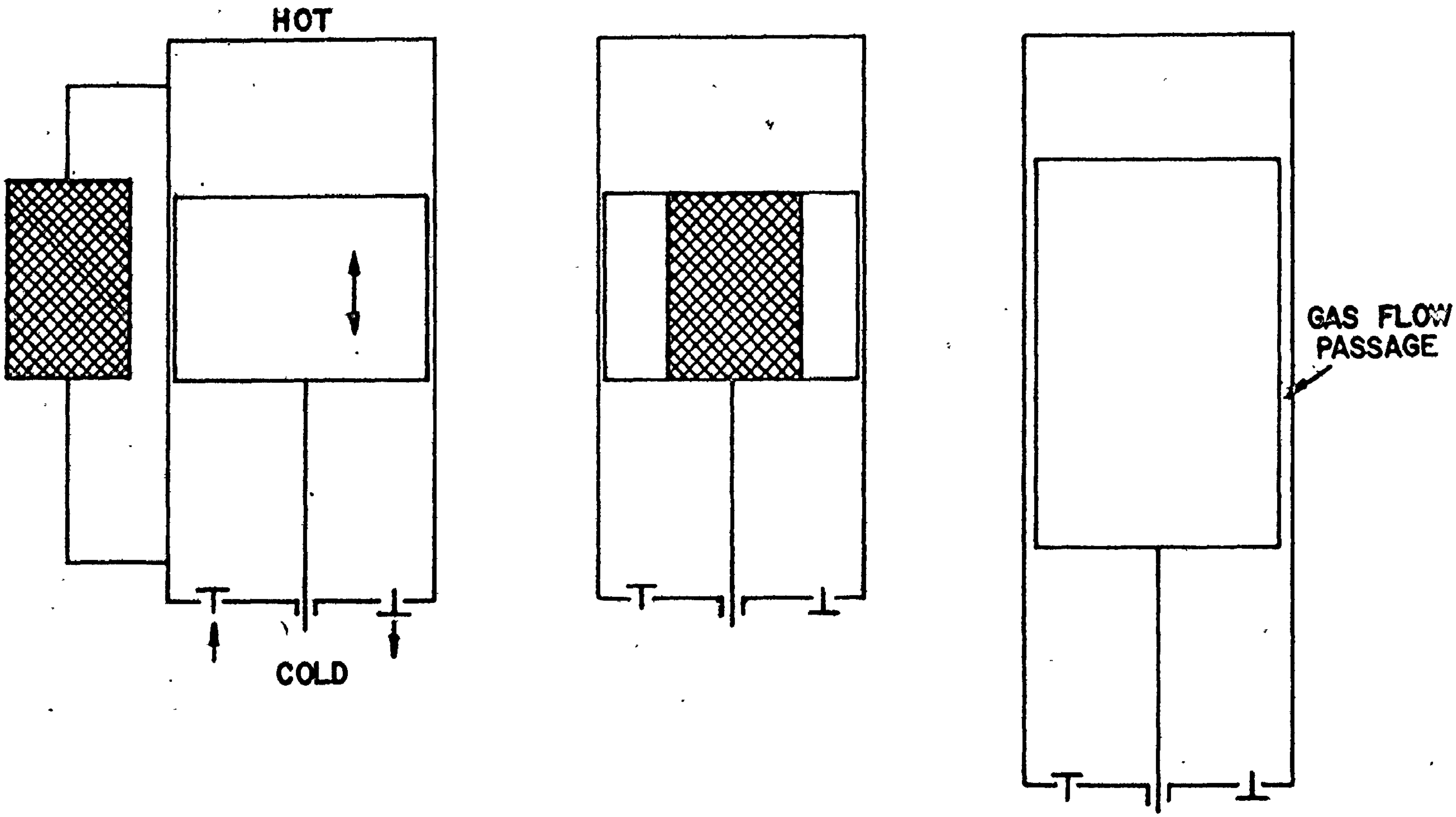
Cycle With Harmonic Piston Motions and Adiabatic Compression and Expansion

This cycle, termed the "Adiabatic Schimdt", in which heat is supplied and removed by heat exchangers between the regenerator and the compression and expansion spaces has been analyzed by Walker and Khan [34] for the case of perfectly efficient regeneration. They find the amount by which the efficiency falls below that for the isothermal (Carnot) cycle to be less for large void volumes. Large void volumes are equivalent to low compression ratios. Recalling the regenerative Otto cycle from which this cycle differs mainly in the "overlapping" of the processes, this result is not surprising.

The Thermocompressor

Invented in 1930 by Bush, the Thermocompressor and its variants have received much attention in the development of heart engines [10,21,22,33, 46 to 50]. Its great mechanical simplicity has a natural appeal. It may be significant that the relatively recent German start in the implantable artificial heart field [61,62] favours the same type of engine as that upon which a great deal of American effort has been expended over the last several years. Ideally the Thermocompressor receives and rejects heat at constant temperatures. With perfect regeneration and no losses this would result in Carnot efficiency.

Fig. 18 shows the principles of three variants of the Thermocompressor, all of which have been or are active in the artificial heart program. In each case, as the displacer reciprocates in the cylinder the pressure is



(a) FIXED REGENERATOR (b) MOVING REGENERATOR (c) REGENERATOR DISPLACER

THERMOCOMPRESSORS

FIG. 18

lowered or raised due to cooling or heating of the fluid. This, together with the one way valves, causes pumping of the working fluid through the engine. For clarity the displacer reversing devices are not shown. This form of the engine is known as a "cold expansion engine". If the valves were situated at the hot end it would be a "hot expansion engine". Constructions (a) and (b) are thermodynamically equal. Construction (c) is nearly so, but not quite, since the proximate surfaces of the displacer and cylinder walls, both of which are heat transfer surfaces, are not generally at the same temperature due to their relative motion. In the McDonnell Douglas engine (described later) the stroke is only a few percent of the displacer length; in this case one would expect substantially the same type of performance as for Constructions (a) and (b). The porous regenerators of (a) and (b), using stacked wire matrices would appear to give more efficient regeneration, due to their larger possible surface areas, than the hollow plug regenerator/displacer of type (c). Nevertheless the latter superseded the former in the development of the above-mentioned engine. A modification of this engine that does not essentially change the thermodynamic principles is the elimination of the valves by replacing them by a flexible diaphragm across which the engine power is transmitted to a (different) fluid that ultimately activates the blood pump. This modification is used in the McDonnell Douglas engine.

Martini et aI [47] show that when the Thermocompressor is driving its optimum load, the work done per unit of heat transferred in the regenerator per cycle for the ideal engine is approximately .035 (actually Martini gives .070: the factor of 2 is due to our different definitions of heat processed). This ratio corresponds to a Stirling cycle compression ratio of only 1.11, viewing the Thermocompressor as an equivalent volume change type of engine.

The mean effective pressure ratio for the ideal engine (no clearance volume) may be derived from the operating pressure ratio of $\frac{P_{max}}{P_{min}}$.

Referring to Fig. 19, V is the total swept volume, i.e. the maximum volume that alternately appears at each end of the cylinder. Volumes V_C, V_H are the volumes as the pressure falls to P_{min} and rises to P_{max} respectively and, with variables v are measured from each end as appropriate.

When the displacer has just absorbed all of the cold space, i.e. all of the gas is hot, the mass of gas is

$$\frac{P_{max}V}{RT_H}$$

where V is the total gas volume, which is constant. At the point when the displacer has moved towards the hot end and the pressure has just fallen to P_{min}, the cold volume is V_C and the hot volume is $(V - V_C)$. The total gas mass is the same as before, no new gas having yet entered the cylinder, thus

$$\frac{P_{max}V}{RT_H} = \frac{P_{min}(V - V_C)}{RT_H} + \frac{P_{min}V_C}{RT_C}$$

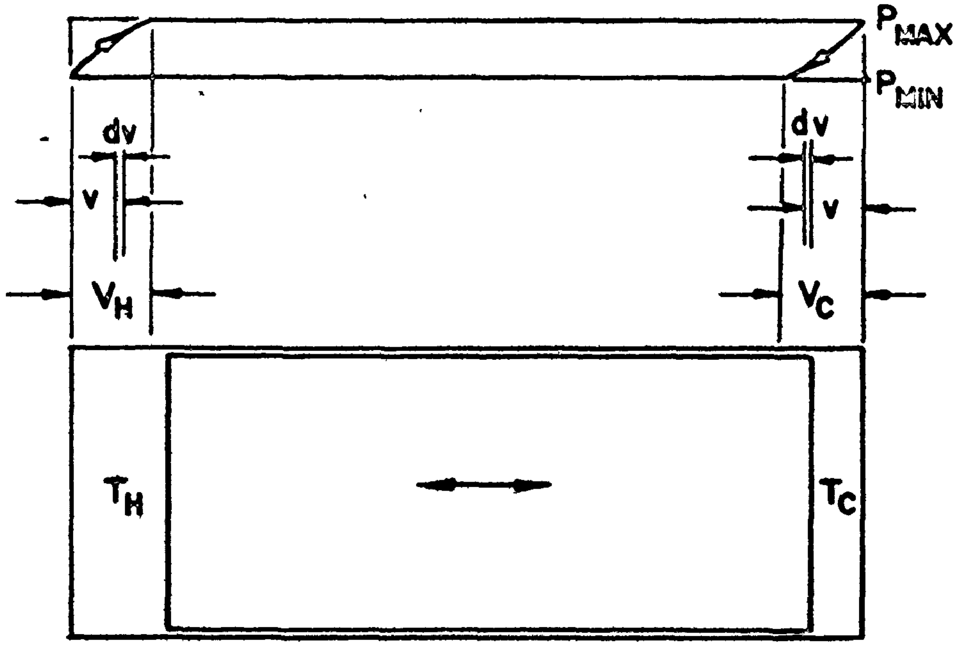
from which $$\frac{P_{max}}{P_{min}} = 1 + \frac{V_C}{V}\left(\frac{1-\tau}{\tau}\right) \qquad 4.8$$

or $$\frac{V_C}{V} = \left(\frac{P_{max}}{P_{min}} - 1\right)\left(\frac{\tau}{1-\tau}\right) \qquad 4.9$$

At this point new gas is about to enter and the process becomes one of constant pressure until the end of the stroke. Subsequent movement of the displacer towards the cold end raises the pressure until at volume V_H for the hot end and $(V - V_H)$ for the cold end the maximum pressure is again reached, V_H being given by

$$\frac{P_{min}V}{RT_C} = \frac{P_{max}V_H}{RT_H} + \frac{P_{max}(V - V_H)}{RT_C},$$

from which $$\frac{V_H}{V} = \frac{1 - \frac{P_{min}}{P_{max}}}{(1-\tau)} \qquad 4.10$$



PRESSURE — VOLUME CHANGES
IN THERMOCOMPRESSOR

FIG. 19

Representative values are $\tau \approx .4$ and $\frac{P_{max}}{P_{min}} = 1.2$. These give $\frac{V_C}{V} = .3$ and $\frac{V_H}{V} = .28$. The work done per cycle may be obtained by integrating to obtain the area of the P-V diagram. In the above expressions for $\frac{P_{max}}{P_{min}}$ we can obtain expressions by substituting P for P_{max} or P_{min}, v for V_C, V_H, to give

$$\text{Work/Cycle} = \int_{\frac{v}{V}=0}^{\frac{v}{V}=.28} \frac{-P_{min}\, d(\frac{v}{V})}{1-(1-\tau)\frac{v}{V}} + [P_{max} \times .3V] - \int_{\frac{v}{V}=0}^{\frac{v}{V}=.3} \frac{P_{max}\, d(\frac{v}{V})}{1+(\frac{1-\tau}{\tau})\frac{v}{V}}$$

$$+ (P_{max} - P_{min})(V - .3V - .28V) - [P_{min} \times .28V]$$

$$\text{or MEP} \times V = 0.12\, P_{max} V$$

$$\text{from which } \frac{MEP}{P_{max}} = .12$$

If the temperature ratio τ is taken as 0.3 the mean effective pressure ratio becomes .152. Also, from these integrals the work ratio, defined as net work divided by total work (all elements positive) is calculated as .0647 and .0590 for $\tau = 0.4$ and 0.3 respectively.

Summary

For the regenerative Otto cycle I have derived the efficiency vs compression ratio characteristics and derived expressions for the work/regenerator heat transfer, mean effective pressure ratio and work ratio.

The Thermocompressor has also been examined. Great mechanical attractiveness is offset to some degree by a low work/regenerator heat ratio, a low mean effective pressure ratio and a low work ratio.

The performance of these cycles will be compared with that of other cycles in Chapter 7.

CHAPTER 5

PRACTICAL ENGINE ARRANGEMENTS

Introduction

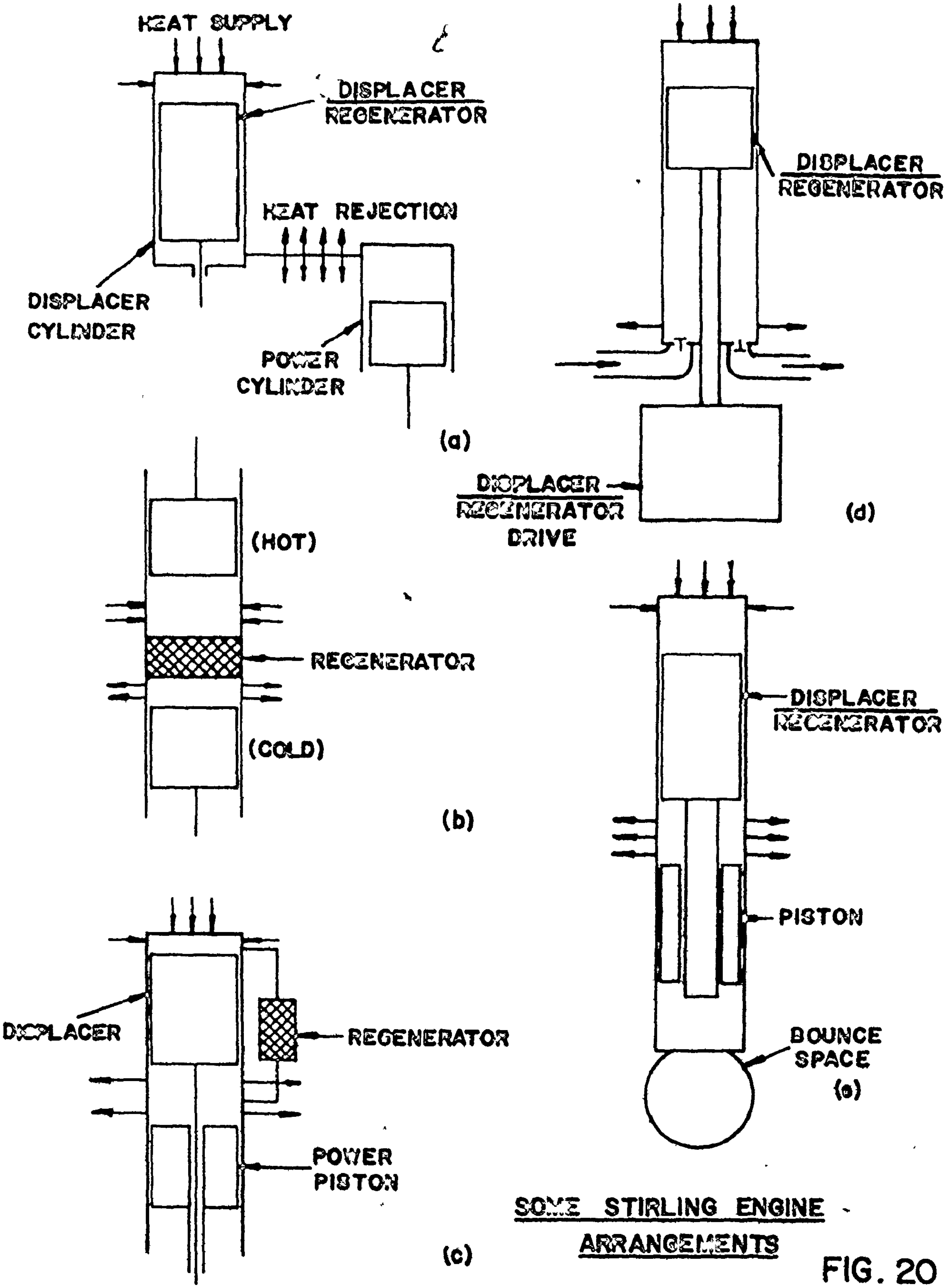
In Chapters 3 and 4 the theoretical characteristics of ideal cycles were discussed. These findings will be used in an attempt to identify the cycle that has the greatest promise for the achievement of maximum efficiency in a practical engine. To this end this chapter gives some of the constructional arrangements that have developed into actual engines, both for general use and for the artificial heart application. Reasons for first discussing mechanical arrangements are that problems due to high temperatures, lubrication, pressure sealing and dead spaces may be accentuated or diminished according to the engine layout. Very many Stirling engines with widely differing configurations have been proposed and built. If any one or two of them were markedly superior to the remainder it is doubtful if so many variations would have appeared. Even so, for a specific application we can expect a reduction in the number of suitable engine configurations.

The many "Stirling" engines that have been built and proposed in the past have been classified by Walker [19]. The majority, however, may be recognized as conforming in principle to one of the schematic arrangements shown in Fig. 20. Emphasis here is placed on cylinder arrangements rather than mechanical linkages. The cylinder arrangement partly controls the minimum possible dead space, defining dead space as that space unswept by pistons. Dead space will be recognized as a necessary evil in "Stirling" engines. The cylinder, piston and displacer (when used) relationship also determines whether seals are necessary, whether they operate at high or low temperatures and the general nature of thermal gradients in the engine components.

The schematic arrangements shown in Fig. 20 may be manifest in different ways, for example in (a) the displacer is separated from the cylinder by a narrow annular space, the bounding surfaces of which constitute the regenerator heat transfer surface. The regenerator operation would have been substantially the same if the displacer was relatively closely fitted and sealed to the cylinder, while bypass ducts guided the flow through a stationary pot type regenerator external to the cylinder, as shown in arrangement (c). Engine type (a) is the only one of those shown that seems not to be of interest today. The large dead space involved prevents the achievement of good pressure variation, thus the power output is low. The two seals required, one at the displacer rod and one for the power piston, both operate at the cold side temperature;thus there is no lubrication problem with hot parts.

Engine type (b) was once built in fair numbers for operating overhead fans at fractional horsepower. It offers simple sandwich type construction. However, both pistons carry the full fluid load and there is a hot sealing problem; steep temperature gradients along the regenerator matrix would require careful design of surrounding components. All heat supplied and rejected must pass directly through the cylinder walls, a difficulty for anything other than the very low powers for which this engine was built.

For engine type (c), having a coaxial displacer and power piston in one cylinder, it is possible for their strokes to overlap so that a higher compression ratio is possible than for type (a) or (b). The degree of over-



SOME STIRLING ENGINE ARRANGEMENTS

FIG. 20

lap may be limited by the kinematics employed; using cranks and connecting rods a useful improvement on (b) results. Philips have spent many years developing this type of engine, even to the extent of developing a special "rhombic" drive to control the piston and displacer motions (the rhombic drive is discussed in Chapter 6). This type of engine has been named after Dr. Meijer of the Philips organization, who lead in its development.

Stirling's original engine was also of type (c), except that a displacer-regenerator was used instead of the external pot-type regenerator. It should also be stated that in the Philips engines heat supply is mainly through heaters in series with the hot cylinder and regenerator (likewise heat rejection) whereas in the original Stirling engine heat supply and rejection was directly through the cylinders. One of the candidate engines for the artificial heart is of this type, being in fact a miniature rhombic drive engine using a displacer-regenerator. This engine, designed and built by the North American Philips Corporation is described later. Another pertinent observation of engines having revolving output shafts, i.e. types (a), (b) and (c): in types (a) and (c) the total internal volume is controlled by the power piston only; the distribution of volume into hot and cold portions is a function of both piston positions. In type (b) the total volume and its distribution is a function of the position of both piston positions. It should be noted that (a), (b) and (c) could be "Schmidt" engines, if mechanically their pistons have simple harmonic motions.

Arrangement (d) is the Thermocompressor, using in the thermodynamic sense only one moving part. Two heart engine developments of this cycle are described later in the chapter. Arrangement (e) having two free pistons has been suggested as the basis for a heart engine though it does not appear to have been developed for this purpose. The arrangement is the result of work by Beale [20] on free piston, hermetically sealed engines. In this engine the piston and displacer have different masses, different effective areas and different net pressures acting upon them, causing oscillations with a phase difference, thus producing work. The work may be taken off by fluid pumping or from the reactive movement of the engine casing. Except that the pistons are free, this engine has similarities to type (c).

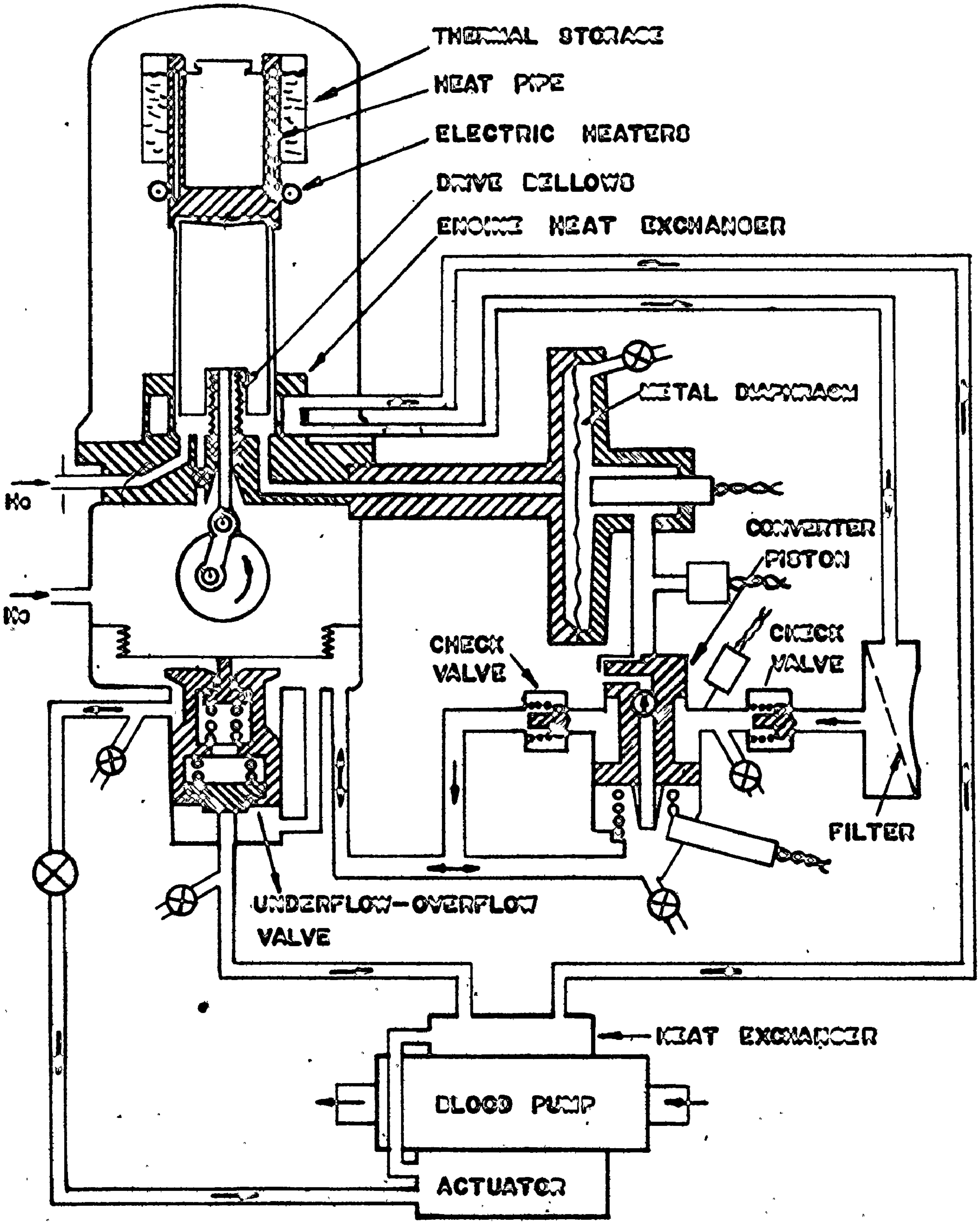
Benson [65] also has done much work on free piston engines, which he terms "thermal regenerative oscillators", including practical construction and testing. Benson's smallest engine appears to be one of .067 indicated horsepower, about eight times the power requirement of the artificial heart. Benson states that according to the engine arrangement and the working fluid selected the operating cycles may be made to approximate the regenerative Brayton, Rankine, Feher, Stirling or Ericsson cycles. Benson considers that the Ericsson cycle should be superior to the Stirling, which is the opposite of my conclusion drawn as a result of the work shown in Chapter 3. However, Benson's opinion follows from his proposal for avoiding one of the largest irreversibilities of conventional Stirling engines, i.e. the mixing losses in the hot and cold spaces, by using constant pressure rather than constant volume processes. However, in those engines in which heat supply and rejection is through the cylinder walls rather than through series-connected heat exchangers, heat transfer conditions that tend to maintain isothermalcy in the cylinder spaces, e.g. strong fluid movements, will also prevail during the regenerator processes too, thus continuing to promote isothermalcy and minimizing mixing losses.

Current Development Engines

1. Donald W. Douglas Laboratories (DWDL)

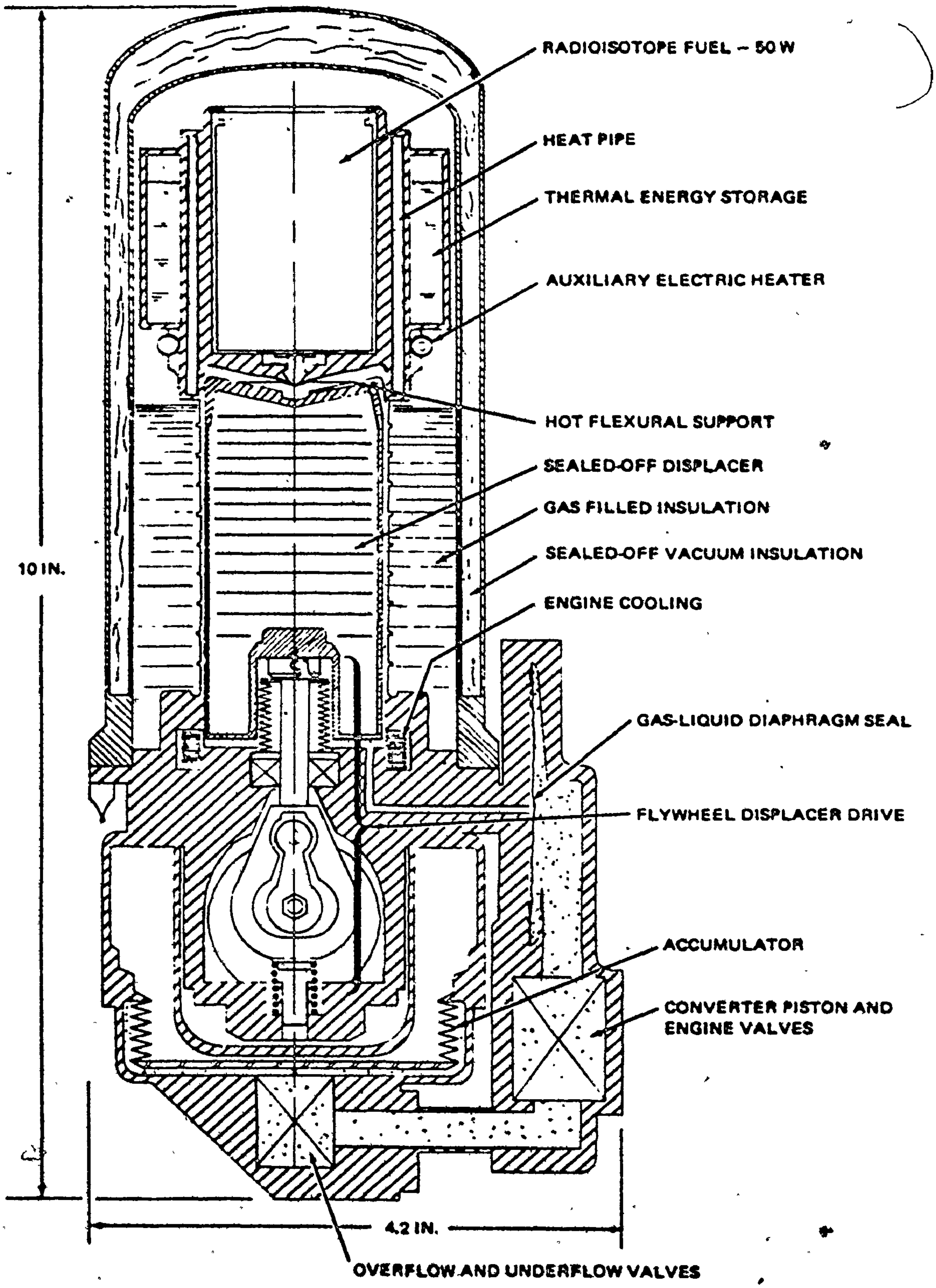
This company has been very active in the field since the inception of the artificial heart program. Their achievements are represented by their "System 5" and "System 6" constructions. These are well described in [49,51,52]. Fig. 21 shows schematically the relationship of the thermo-compressor type System 5 engine to other components of the complete heart. In the DWDL heart concept the heat source, engine and converter-accumulator are combined into an "engine module". The actuator/controller and the blood pump are combined into a "pump module". The two modules are flexibly connected by hydraulic lines which transmit both hydraulic power and waste heat. In Fig. 22 some details of the engine construction are shown. There are many interesting features of this engine which are described in the various DWDL references but a few of them may be mentioned here.

The displacer kinematics are not based on purely reciprocating motion but involve a small flywheel and connecting rod-crank mechanism. DWDL have used this construction for the past few years, although in their earliest engines the displacer mechanism was purely reciprocating, springs and pneumatic loads alone being used to control the displacer. The engine operates at the seemingly high rate of 1800 cycles/minute. The insulation of the engine has received much development, very high thermal resistivities being required both to ensure not only high thermal efficiency but also to obviate thermal damage to body tissues touching the engine module. This has resulted in multilayer foil type insulation sealed under vacuum. Support of the displacer at the hot end is by a hot flexural support. This eliminates the use of a sliding bearing and the consequent rubbing friction and the possibility of detritus in the engine cylinder. Elimination of the rubbing of the hot, unlubricated cylinder surface by use of the flexural support



SCHEMATIC OF TECO SYSTEM 5 DEVELOPMENT ENGINE MODULE
AND ASSIST HEART PUMP COMPONENTS

FIG. 21

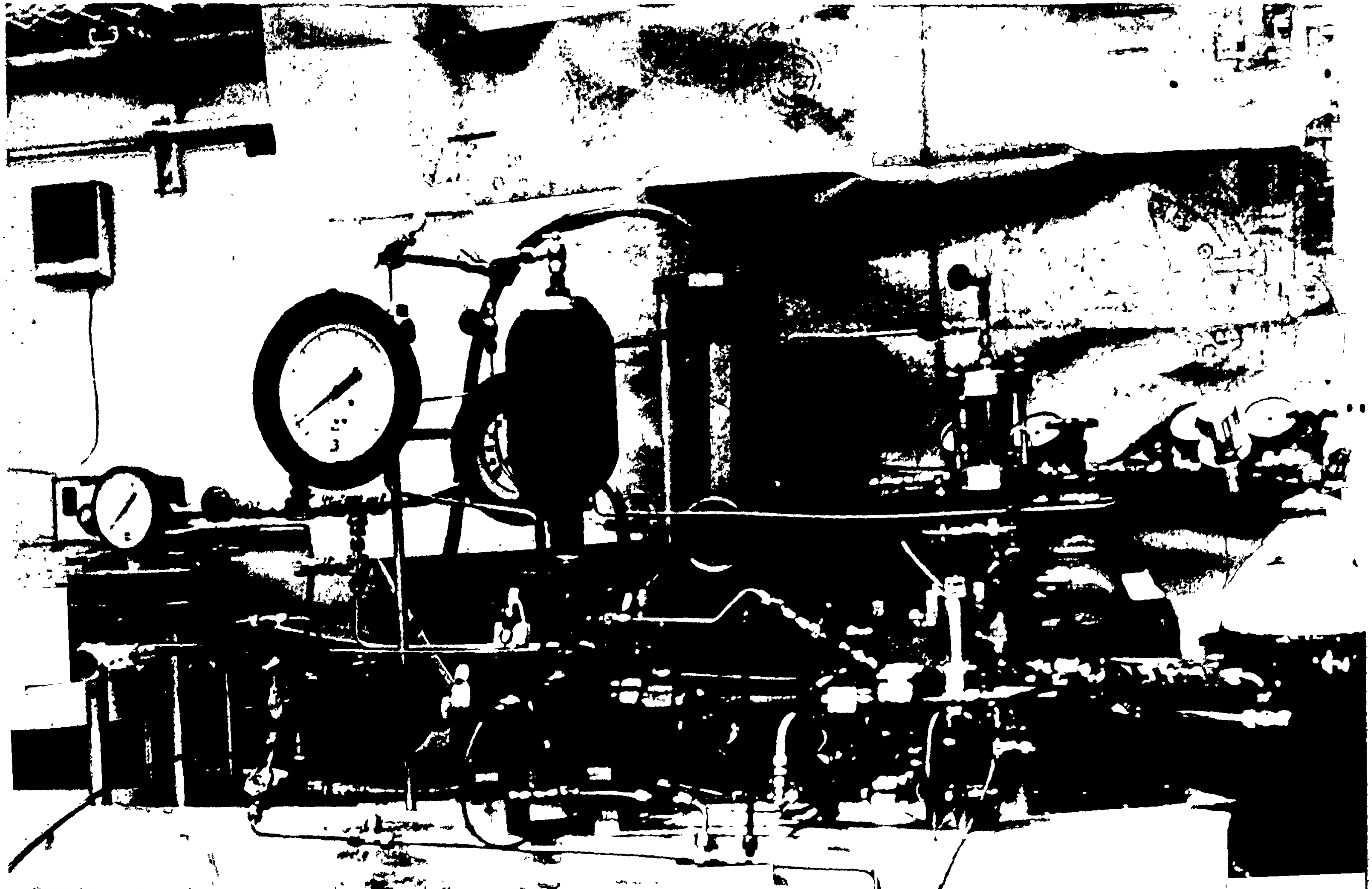


MCDONNELL DOUGLAS ASTRONAUTICS CO.
SYSTEM 5 ENGINE MODULE

FIG. 22

has exchanged a wear and detritus problem for a fatigue problem, which is thought to be more predictable. It does, however, limit the stroke, which in this instance is .188 inch. The lower, cold end of the displacer rod is guided by a lubricated bush, separated from the cylinder by a bellows to prevent ingress of "oil", which is also the fluid of the hydraulic power system, to the cylinder. Losses from thermal effects are inevitably relatively high in very small engines. The direct heat loss from the engine and fuel capsule to the surrounding body organs is minimized by multi-foil, zirconium oxide spaced, vacuum insulation. The insulation is arranged as far as possible for the foil layers to follow the natural isotherms. Losses through the insulation are now down to about 4 W, bringing the expected temperature elevation of adjacent body tissue to within 2°C of normal body temperature. Relatively short tests have been performed in animals with a 4°C elevation with apparent success. Examples of the constant effort to reduce parasitic losses are to be found in the displacer wall which has been reduced from .010 inch to .005 inch thickness saving approximately 6 W, and also in the many radiation baffles in the displacer. These baffles are also structural stabilizers. A thermal saving of another type is found in the use of a sodium-filled heat pipe which reduces the loss between the 50 W Plutonium 238 fuel source and the hot surface of the cylinder. The waste heat from the engine is rejected through a heat exchanger at the lower end of the cylinder. The hydraulic fluid used for power transmission to the pump actuator flows through the heat exchanger absorbing the waste heat. This heat is subsequently transferred to the blood through a wall of the blood pump.

Fig. 23 shows a typical bench set-up for developmental engines. Extensive in-vivo tests in animals and in-vitro testing have lead to several changes with the objectives of improving reliability, weight and vol-



FI ENGINE [illegible]ST F

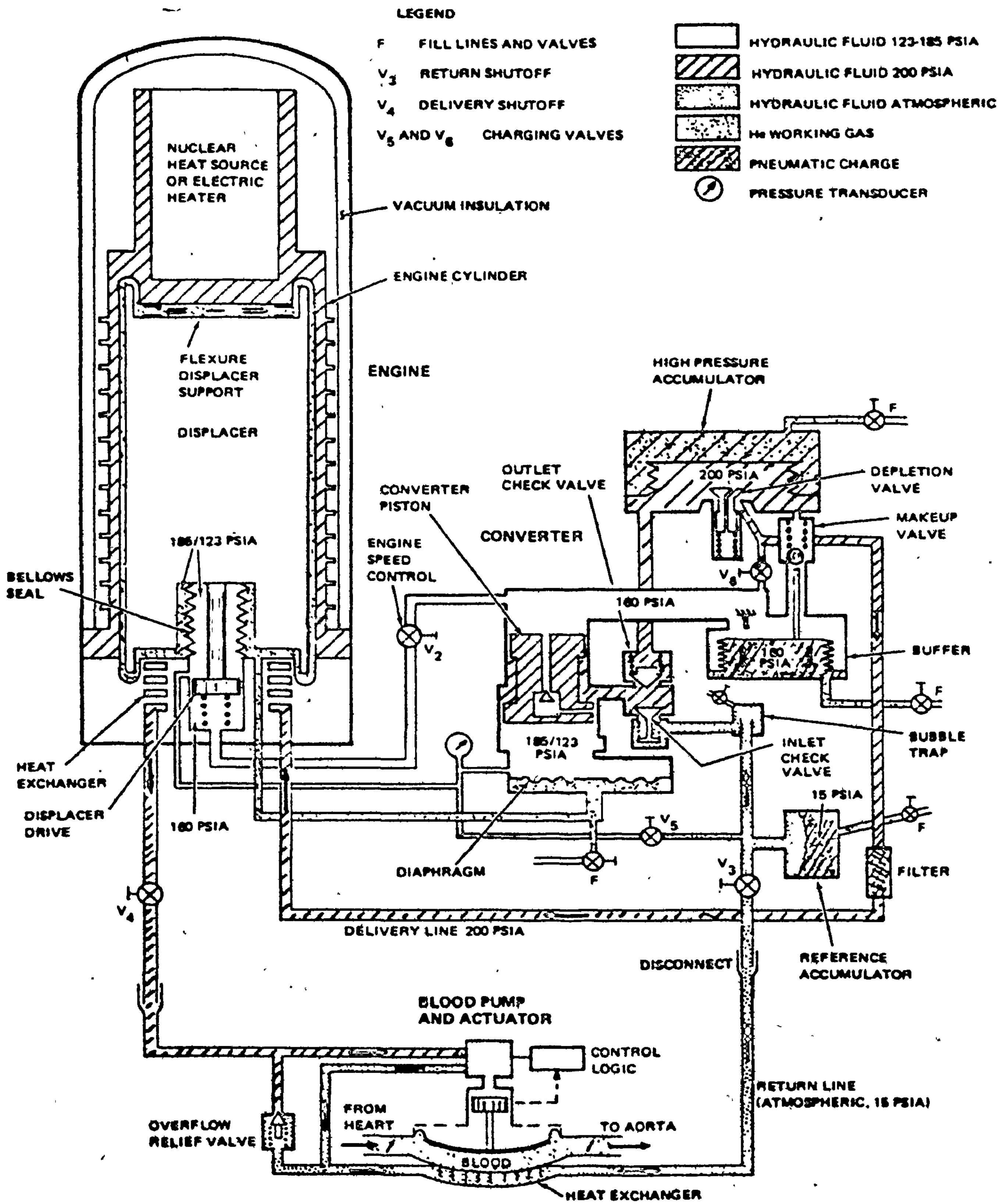
ume. Thus the pending System 6 implantable heart has about one half of the weight and volume of the System 5 heart. Figs. 24 and 25, respectively illustrate the System 6 schematic and the engine module. The most significant change in the engine is the reversion to purely reciprocating devices to control the displacer motion, eliminating the flywheel-crank mechanism. Substantial performance improvements are expected in both the production of power and in its application to the blood pump, resulting in reduction of the Plutonium 238 inventory from 50 W to 33 W. The net thermal efficiencies of the engine are from reference [52]:

Documented System 5 Implantable	Documented System 6 Research Engine	Pending System 6 Implantable
14.2%	15.8%	22.4%

Also of interest in the proposed System 6 engine is the use of a cesium-filled heat pipe to prevent over-temperature excursions of the heat source during long periods of low power demand. The improvements necessary to bring the System 6 engine to reality are seen by DWDL to be:

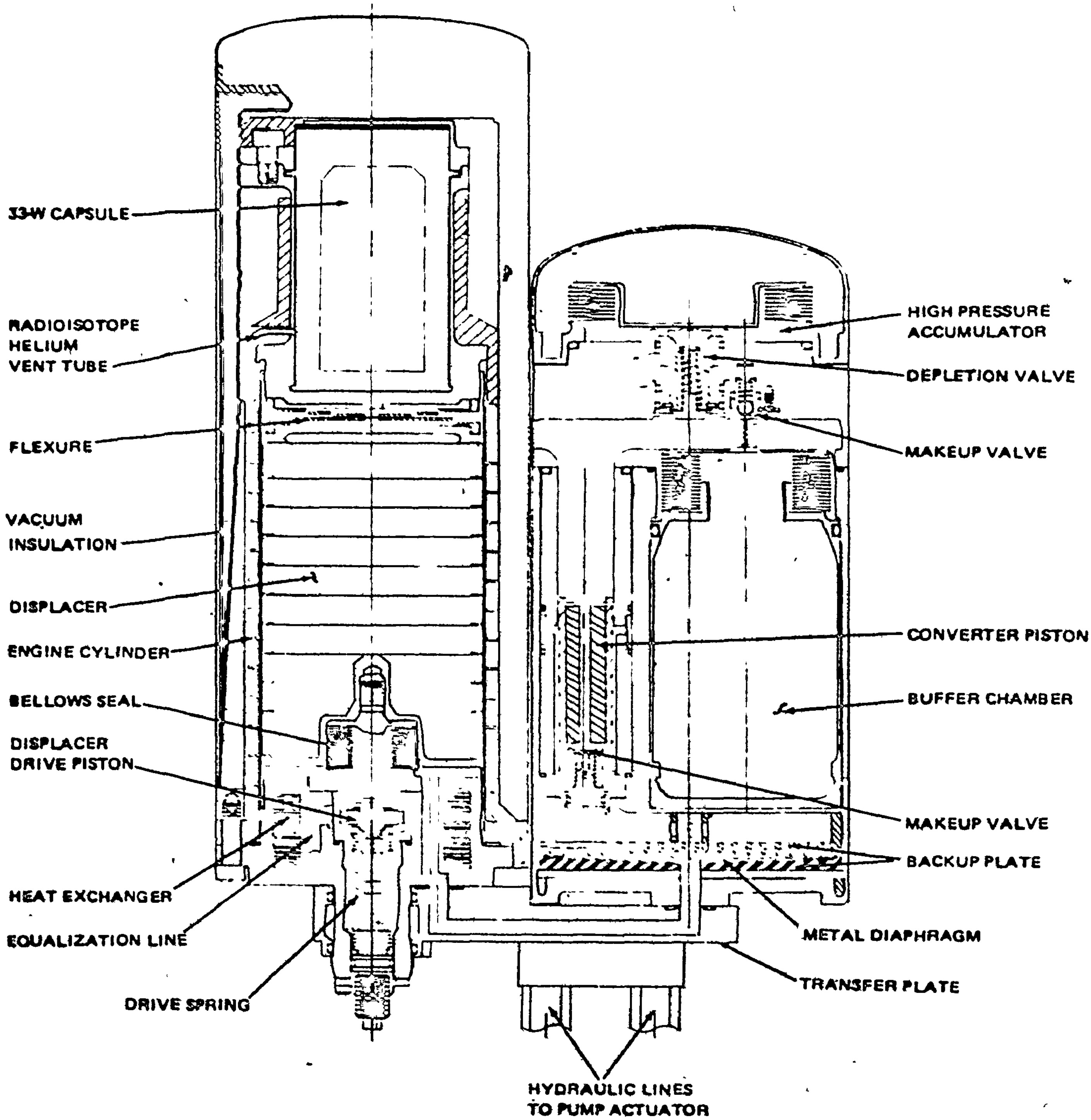
1. A higher efficiency regenerator.
2. A larger displacement and more efficient converter.
3. A free displacer drive for compactness, self-starting and power control.
4. An engine power controller operating on the ratio of diastole to systole time in the blood pump.
5. A smaller, more efficient vacuum insulation package, using the engine cylinder as part of the vacuum envelope.
6. Compact packaging of the engine module, biologically compatible in shape, tissue interface material and tissue interface temperature.

Another problem to be solved before long-term implants become feasible is that of working fluid containment. While rubber o-rings are a design convenience for meeting near term objectives, they do not provide absolute



MCDONNELL DOUGLAS ASTRONAUTICS CO.
SYSTEM 6 SCHEMATIC

FIG. 24



MCDONNELL DOUGLAS ASTRONAUTICS CO.
SYSTEM 6 ENGINE MODULE (FULL SCALE) FIG. 25

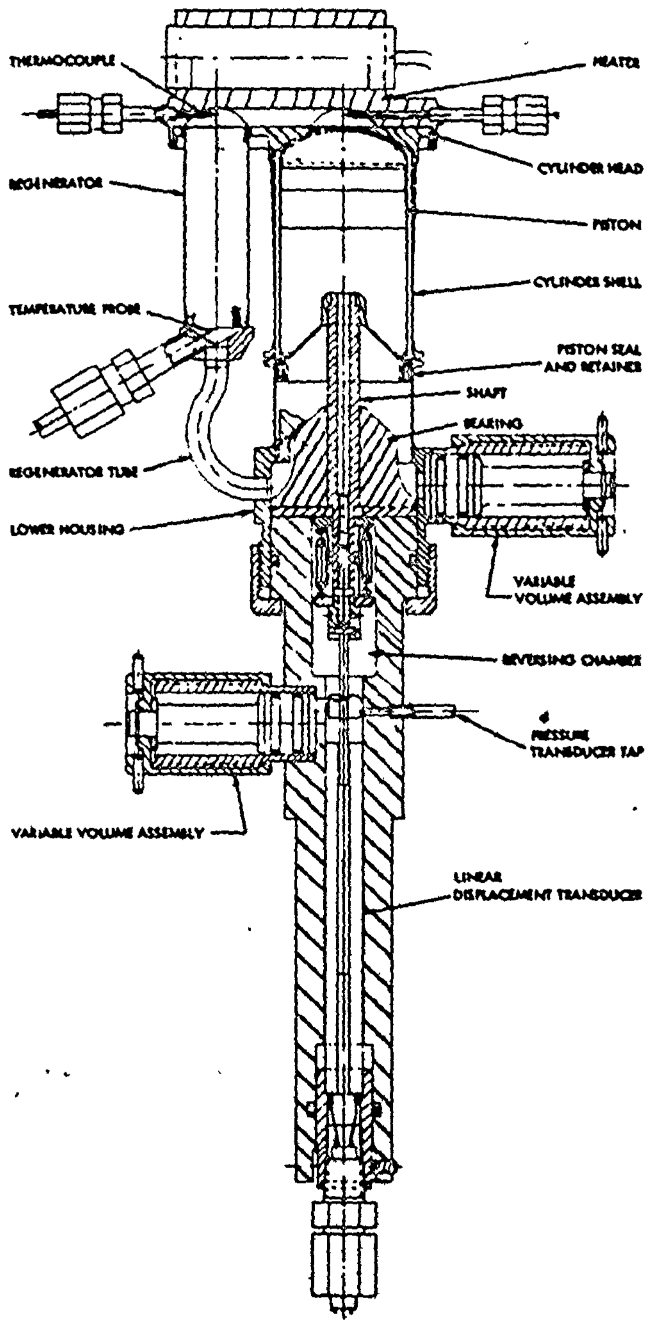
helium containment on a long-term basis. Redesign for sealing by welding is only practicable after solving all other development problems.

2. Aerojet Liquid Rocket Company

This company's engine [46,53] commenced with an externally generated, valved Thermocompressor (type (a) of Fig. 20) built into a system that is entirely pneumatic in its power application and control. As for other heart engines based on the Thermocompressor cycle, the working fluid is helium. The engine rate is about 1800 cycles/minute. Aerojet's work leads to a claim of 17.4% engine efficiency with a near term promise of 22% realizable. Although later publications give practically no information on engine details, an earlier publication [54] gives sectional views of the Mark II and Mark IV engines, reproduced in Figs. 26 and 27. The Mark IV engine retains the purely linear displacer control device of the test bench Mark II engine. The regenerator, however, has become coaxial with the cylinder and within it, although still stationary. Considerable axial space is saved by doubling the displacer reversing spring back within the displacer body. By using a very long guide bearing at the lower end of the displacer an upper end bearing between hot surfaces is eliminated. The radioisotope fuel capsule for this engine is now down to 33 W. Future development of the engine will concentrate on achieving adequate durability.

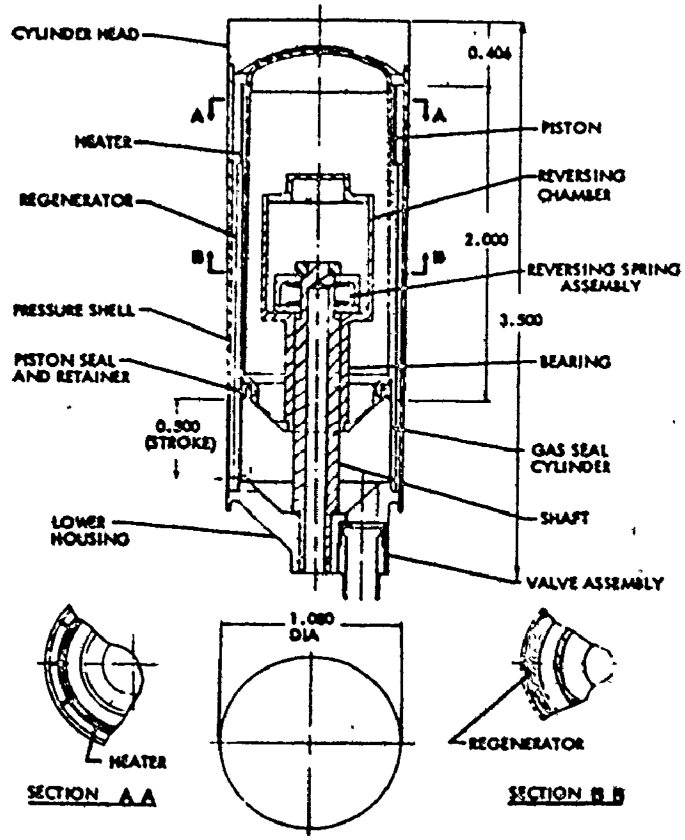
3. Philips

The many years of effort in the Stirling engine field by the Philips organization has lead to the joint development of an artificial heart engine by the Westinghouse Electric Corporation and the North American Philips Corporation. The present status of this is described in [55], from which Figs. 28 and 29 are taken. Compared with other efforts in the artificial heart field, the engine (also the power transmission and the blood pump) is notable for its complexity.



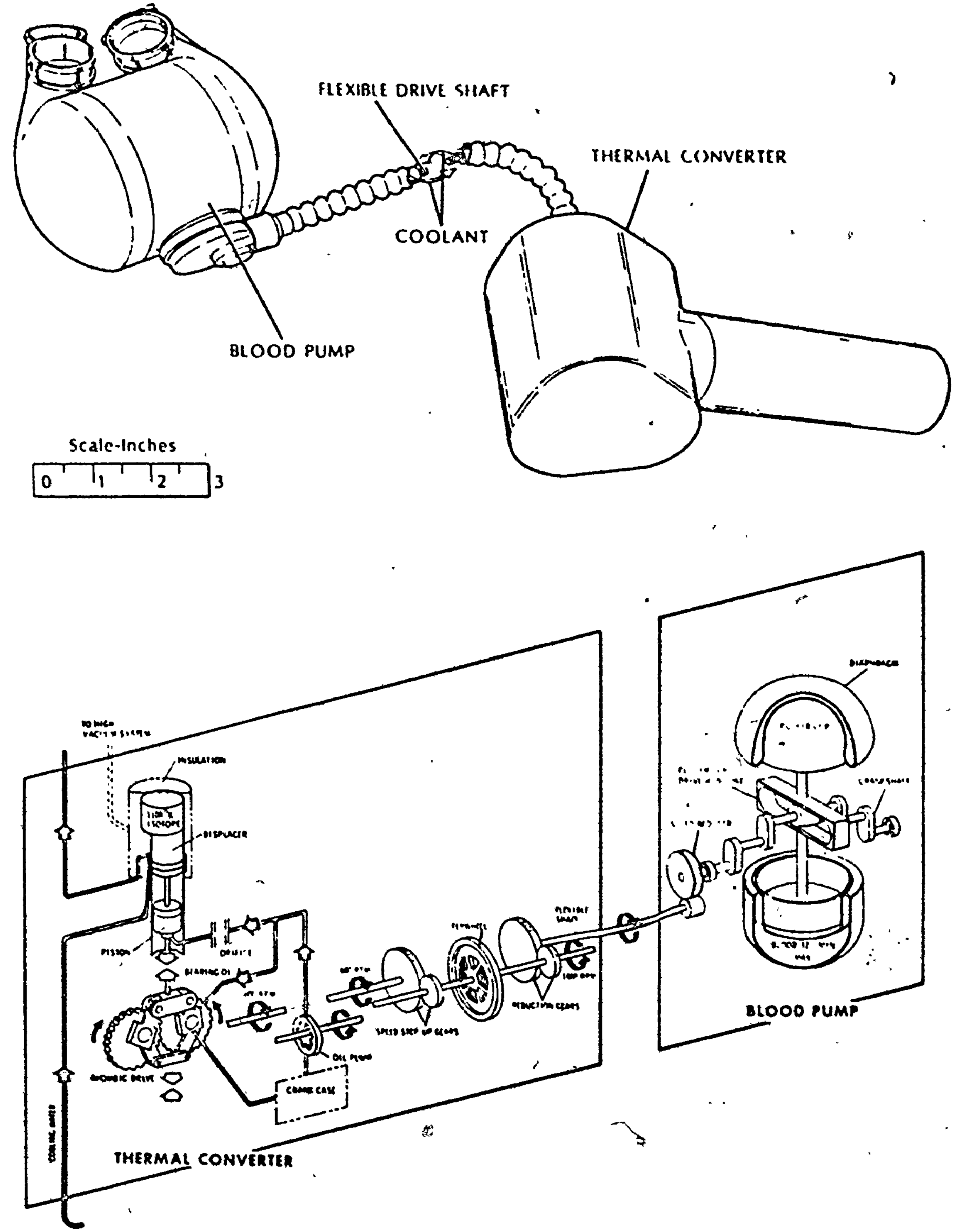
AEROJET-GENERAL CORP.
MARK II ENGINE DESIGN

FIG. 26



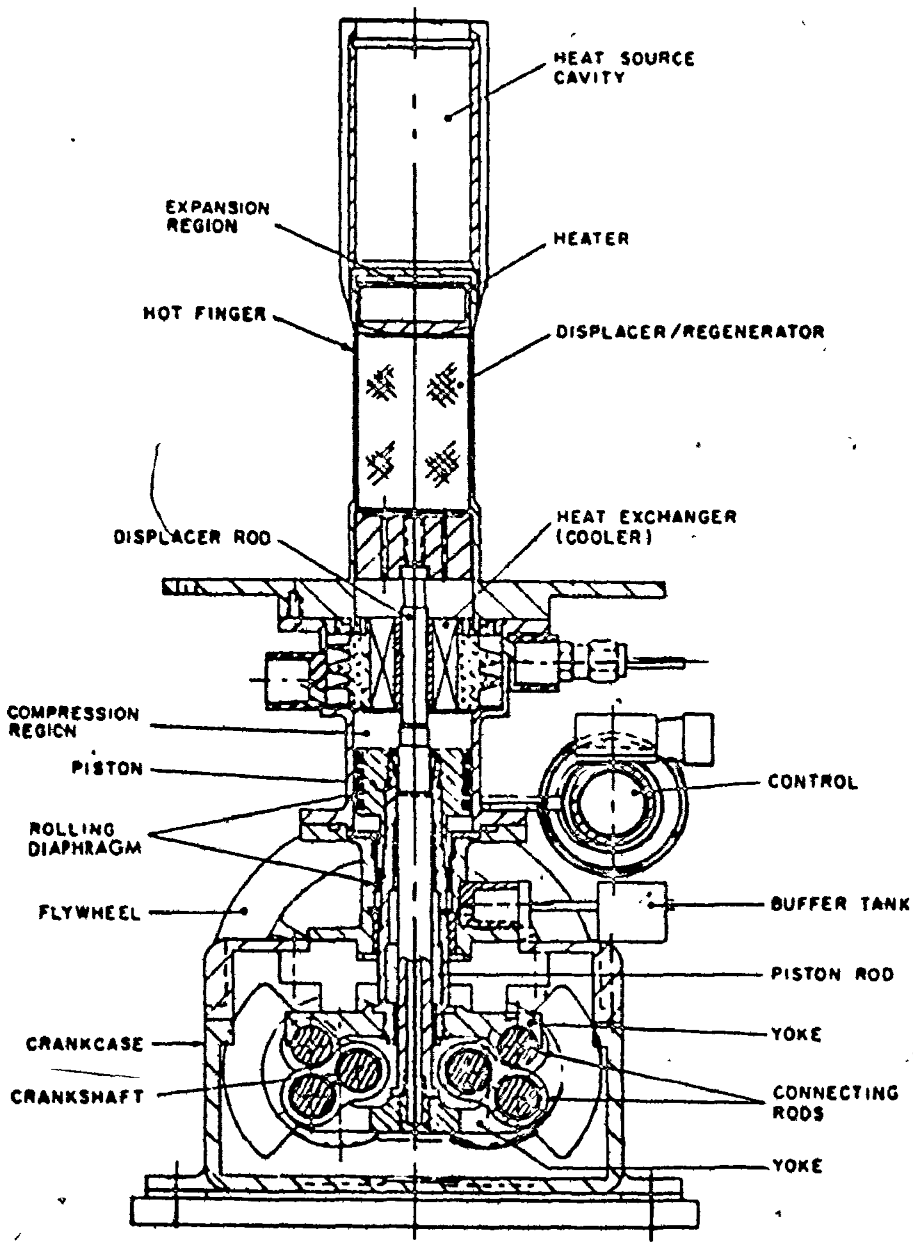
AEROJET-GENERAL CORP.
ADVANCED (MARK IV) ENGINE

FIG. 27



NORTH AMERICAN PHILIPS CORP.
SCHEMATIC OF BENCH MODEL SYSTEM

FIG. 28



NORTH AMERICAN PHILIPS CORP.
CROSS-SECTIONAL VIEW OF BENCH MODEL SYSTEM ENGINE

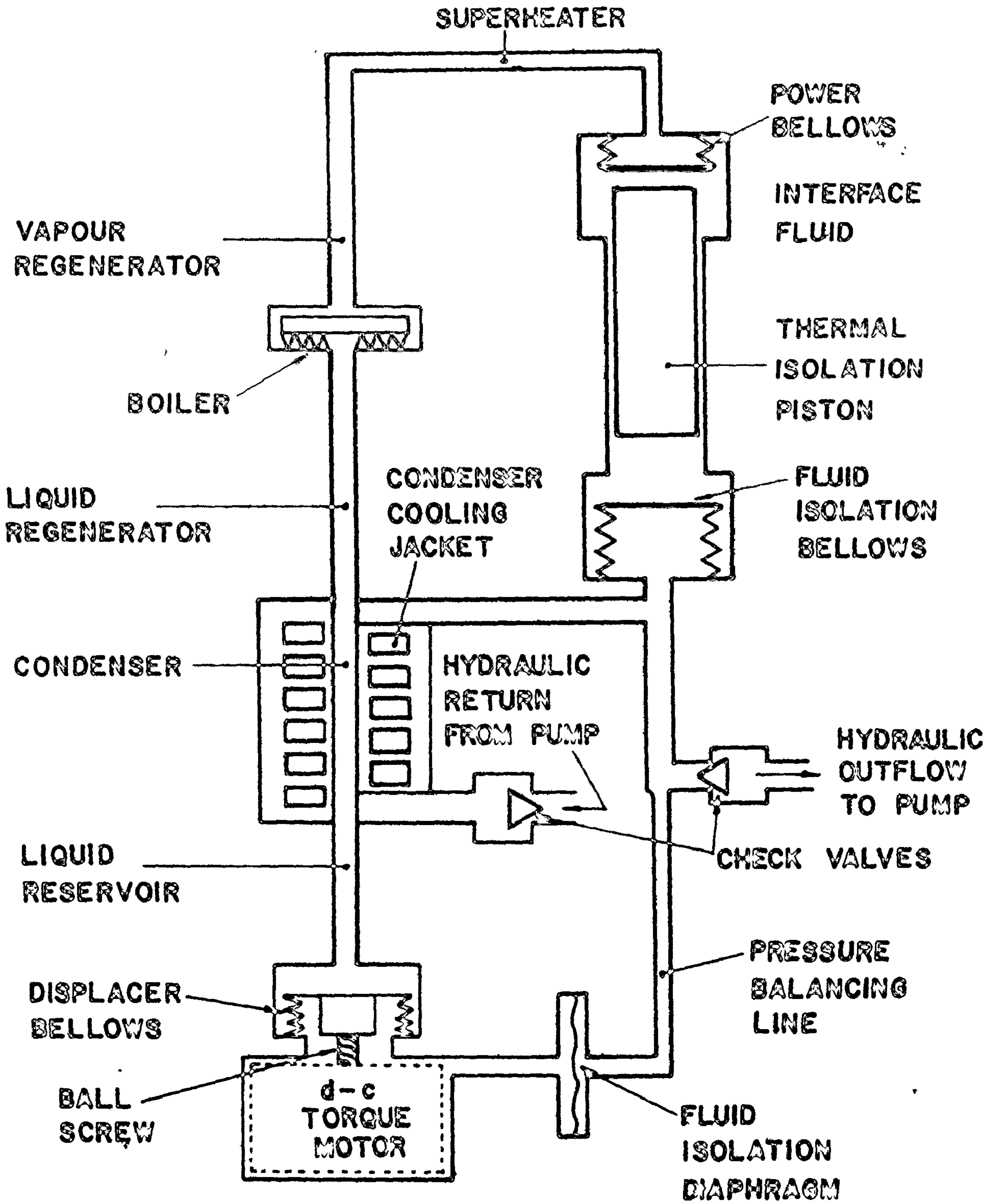
FIG. 29

The engine appears to be a miniature of the multi-horsepower engines developed by Philips in the past, complete with rhombic drive, power piston and displacer/regenerator, except that the latter combines the functions of displacer and regenerator instead of keeping these separate as in the larger Philips engines. The engine rotates at 600 RPM, this being stepped up to 1800 RPM by gears before the flywheel, after which it is reduced to 900 RPM before transmission through a flexible cable to the blood pump. This engine is the only one utilizing a mechanical transmission between engine module and pump module. "Rolling sock" seals are used to hermetically seal the crankcase from the cylinder spaces. These are, in principle, the same as for commercial multi-horsepower engines. Though so dissimilar from other engines mechanically, the thermal insulation problems are much the same. A major portion of the insulation, not shown in the figures, consists of 131 layers of .0005 inch thick molybdenum foil separated by zirconia spacers, wrapped around an inner cylinder formed from .002 inch thick molybdenum foil. This is enclosed in an outer cylinder also constructed of .002 inch thick material and operating under vacuum. The engine has been tested at similar average pressures to those of the DWDL and Aerojet engines, in this case 210 lb/in^2. Though designed for krypton, the engine was tested on argon, giving a representative efficiency of 11%. This is rather low in comparison with the DWDL and Aerojet engines and is evidently due to a number of component deviations from design, both plus and minus, giving a performance deficit of approximately 20%. The tests and performance deviations of this engine are discussed in the contractual report [56]. As a result of these bench tests it was recommended that the development of a heart system be undertaken and fabrication of an implantable version is now underway.

4. Thermo Electron Corporation

This company is developing an interesting engine of a type that is probably unknown to most engineers unacquainted with the artificial heart field. The engine, known as the Tidal Regenerator Engine (TRE), has elements of compression, expansion and regeneration, giving it some qualification for consideration in the "Stirling" field. On the other hand it also has strong elements of the Rankine concept in its cycle, utilizing liquid-vapour phase changes. The engine is relatively recent, following earlier work of this company with simple Rankine cycles. The engine and its development are described in the patent specification [23] of its inventor F.N. Huffman and references [24,57,58,59]. An appealing feature of the engine is its design for operation at blood pump (natural heart) frequency, as was one of the earlier Stirling engines, though this was abandoned due to large mass, volume and (presumably) heat losses.

Fig. 30 is a schematic of the TRE engine illustrating the principles of operation. It is seen that a small D.C. motor is used, operating a displacer bellows across which there is negligible pressure difference. This movement controls the engine cycle. The D.C. motor itself is controlled by an electronic logic module responsive to the venous return. The motor input power is very low, so that sufficient power to operate it may be generated at about 97% effective efficiency by a thermoelectric converter drawing its heat supply from the nuclear heat source. The converter also powers the logic module. It is seen that the fluid isolation bellows is equivalent to a power piston, and with the exceptions that there is a change of phase of the working fluid and that the displacer bellows is not mechanically coupled to the power system but is separately operated in response to body demands, the engine is similar to the Stirling engine



SCHEMATIC OF TIDAL REGENERATOR ENGINE

FIG. 30

When power to the torque motor causes the displacer bellows to rise the working fluid, water, level is raised (hence "tidal") from the condenser into the boiler where some is evaporated. The pressure, approximately constant throughout the loop, rises greatly and steam flows through the superheater round to the power bellows. The superheater and boiler receive heat from the nuclear energy source and the flow channels incorporate a regenerative filling for the betterment of the cycle efficiency. The high pressure on the power bellows is transferred through the interface fluid, thermal isolation piston and fluid isolation bellows to the hydraulic fluid which carries power to the blood pump module. When the blood pump has performed the systole function the displacer motor moves the displacer bellows downwards, lowering the tidal level and exposing the steam to the cold surface of the condenser. The condenser is cooled by the hydraulic fluid which is constrained to have one way flow through the condenser by means of the check valves. Thus the steam pressure falls and some of the superheat and latent heat is saved as the steam flow reverses. The reduced pressure, substantially the same as the hydraulic pressure of the blood pump actuator, allows diastole to occur and when this is complete the motor receives power to move the displacer bellows upwards to repeat the cycle.

The motor works best in its own fluid, which must be a liquid to maintain the principle of approximately zero pressure difference across the bellows, so that a fluid isolation diaphragm is used to isolate the motor fluid from the hydraulic power fluid. The blood pump frequency of engine operation permits simple hydraulic coupling between the engine and pump modules, a coupling efficiency better than 97% being claimed.

A set of typical engine design parameters is given:

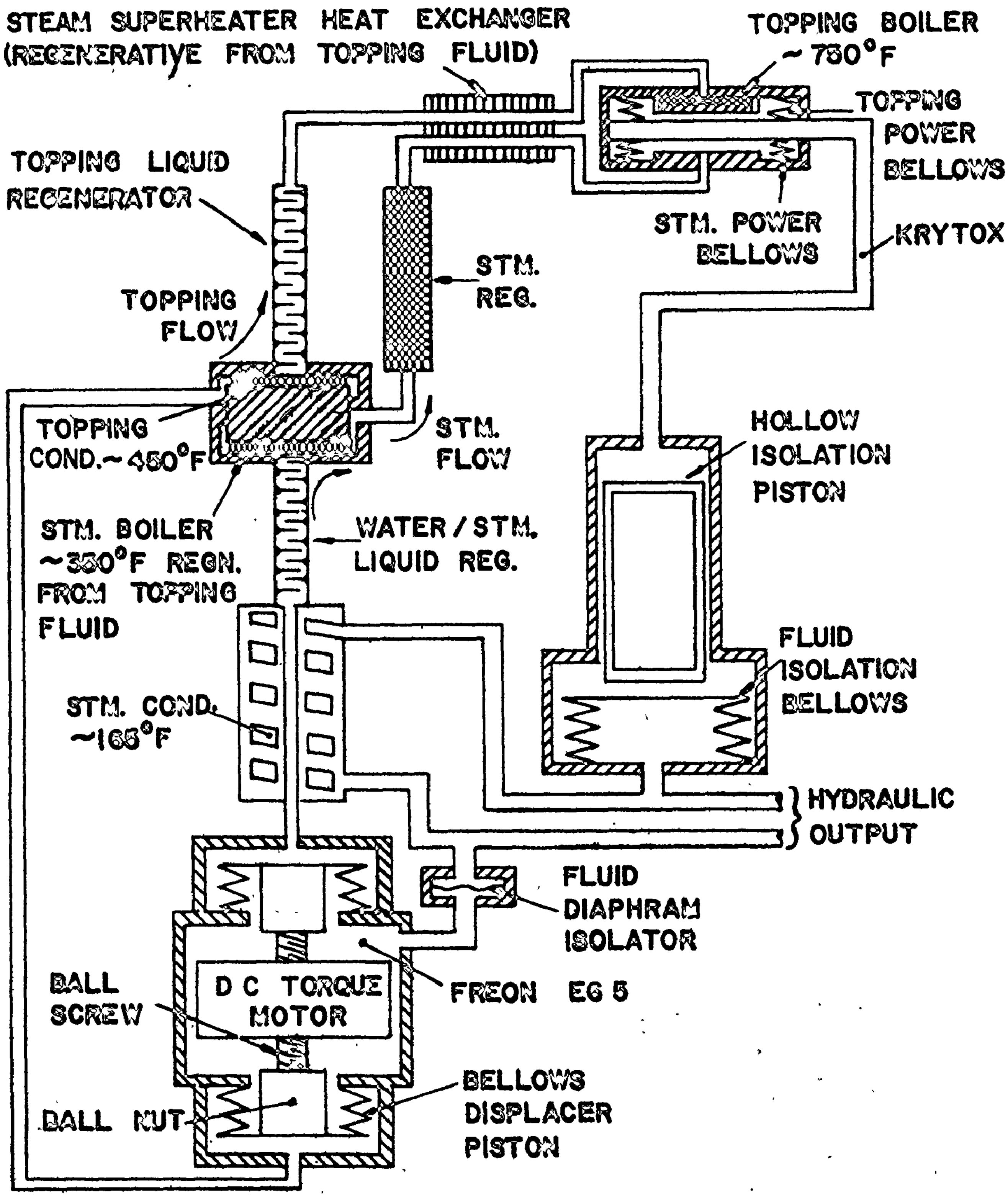
Working Fluid	Water/Steam
Peak Pressure	135 psia (9.3 bars)

Condensing Pressure	5 psia (0.34 bar)
Superheat Temperature	700°F (371°C)
Boiler Temperature	360°F (182°C)
Feed Water Rate	.001 in^3/cycle (.016 ml/cycle)
Output Stroke Volume	0.222^3in (3.64 ml)
Work Output	2.8 ft lb/cycle (3.8 joules/cycle)
Displacer Electric Power	0.16 joules/cycle
Displacer Stroke Volume	0.15 in^3 (0.25 ml)
Torque Motor Rotation	± 90°

In this cycle all of the internal void volume must be fully pressurized for each cycle, but the associated vapour does no work. Thus the ratio of internal void volume to the power piston stroke is an important parameter affecting the cycle efficiency. The practical thermal efficiency achievement for this cycle is approximately 10%. It is considered that continued development of the basic concept will raise the efficiency to about 14%, using water/steam as the working fluid. The fundamental optimization problem is a matter of achieving large heat transfer surfaces while minimizing void volumes and pressure drop losses of vapour flow.

This engine is similar to other very small heat engines in that realistic testing of separate components is difficult to do; also the separation of cause and effect is not easy. The boiler is an example of this, many designs being tried with mediocre results before finding a sound prototype [59]. These designs included ball matrices, monotubes, parallel tubes and combinations of these.

The importance of achieving the highest practicable thermal efficiency is evident in Thermo Electron's current work on the application of the binary cycle to the TRE, using the Emmet [60] process. This is shown schematically in Fig. 31. The fluids selected for this are Dowtherm A and



SCHEMATIC OF BINARY TIDAL
REGENERTOR ENGINE

FIG. 31

water. Theoretical predictions of cycle efficiency show values up to 24% with zero void volume in the engine, falling to about 20% for practical void volumes. Breadboard type experiments show efficiencies of over 13% for this engine. The engine had rather large void volumes, corresponding to a calculated efficiency of approximately 12-1/2%. In this light, therefore, the results are encouraging. Reduction of void volumes will be the main effort in raising the engine efficiency.

5. Messerschmitt-Bölkow-Blohm GmbH

This organization is relatively new to the artificial heart field, and their current work is outlined in [61]. From basic haemodynamic requirements and a system analysis [62] a Bush type Thermocompressor has been selected as the engine basis for both a left ventricular assist and a total replacement heart. Messerschmitt-Bölkow-Blohm's studies, supported by some component testing, indicate that a low frequency engine is feasible for the assist heart. The tests showed an engine efficiency of 12%. This, however, was based on fibre insulation. If the fibre insulation is replaced by their already proven vacuum foil insulation, the engine efficiency would rise to 16%. With some component improvements and an optimization of geometry and operating parameters the efficiency should improve to more than 20%.

CHAPTER 6

FURTHER PERSPECTIVES

Introduction

The purposes of this chapter and the following one are (a) to develop some further perspectives for the better understanding of the "Stirling" cycle, and (b) to compare the performances of those cycles that are feasible and if possible to select a cycle that appears the most suitable for the practical attainment of high thermal efficiency.

The interdependence of the performances of the components of a Stirling engine suggests that each factor considered in proposing either engine concepts or detail designs has to be taken as supporting, or detracting from, a mass of theoretical circumstantial evidence rather than being assertive in nature.

1. Fluid Mass Distribution

Consider a compression-regeneration-expansion engine in which at any moment its working fluid volume is comprised of three series connected volumes:

(a) A variable hot volume V_H consisting of a portion of the hot swept volume together with the "dead" hot volumes (this includes heater tubes, cleaners).

(b) A constant volume V_R, representing the regenerator void space, having an assumed uniform temperature gradient.

(c) A variable cold volume V_C consisting of a portion of the cold swept volume together with the dead cold volumes.

These volumes and their temperatures are represented in Fig. 32. The total mass of the working fluid is

$$W = w_H + w_R + w_C \tag{6.1}$$

$$= \frac{PV_H}{RT_H} + \int_0^{V_R} \frac{Pdv}{RT_H[1-(1-\frac{T_C}{T_H})\frac{v}{V_R}]} + \frac{PV_C}{RT_C} ,$$

$$= \frac{PV_R}{RT_C} \{\tau\frac{V_H}{V_R} - \frac{\tau}{1-\tau}\ln\tau + \frac{V_C}{V_R}\} ,$$

$$\text{or } P = \frac{WRT_C}{V_R\{\tau\frac{V_H}{V_R} - \frac{\tau}{(1-\tau)}\ln\tau + \frac{V_C}{V_R}\}} . \tag{6.2}$$

The fluid masses may be expressed non-dimensionally:

$$\frac{w_H}{W} = \frac{\tau\frac{V_H}{V_R}}{\{\tau\frac{V_H}{V_R} - \frac{\tau}{(1-\tau)}\ln\tau + \frac{V_C}{V_R}\}} \tag{6.3}$$

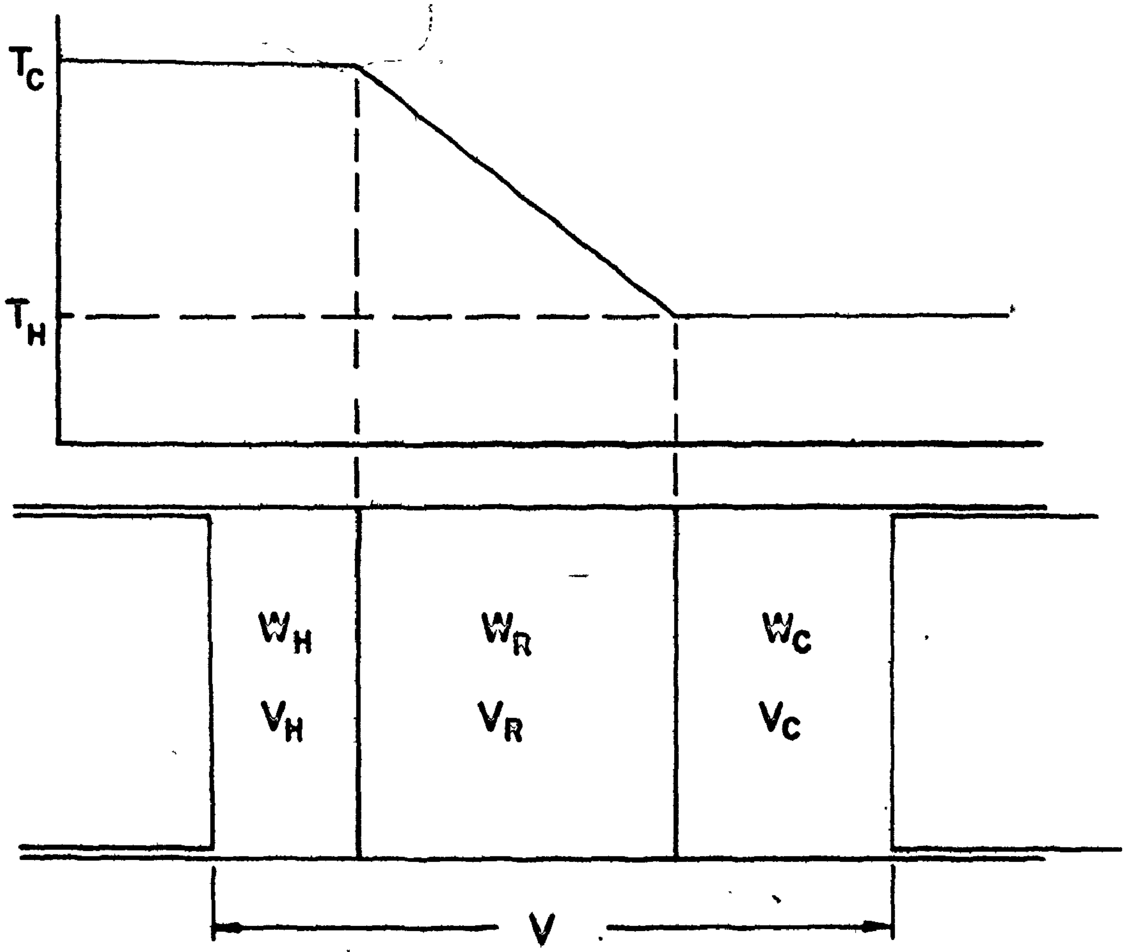
$$\frac{w_R}{W} = \frac{-\frac{\tau}{(1-\tau)}\ln\tau}{\{\tau\frac{V_H}{V_R} - \frac{\tau}{(1-\tau)}\ln\tau + \frac{V_C}{V_R}\}} \tag{6.4}$$

$$\frac{w_C}{W} = \frac{\frac{V_C}{V_R}}{\{\tau\frac{V_H}{V_R} - \frac{\tau}{(1-\tau)}\ln\tau + \frac{V_C}{V_R}\}} \tag{6.5}$$

For the assumptions of isothermal hot and cold spaces and neglecting pressure differences, these quantities represent the distribution of working fluid mass through the engine.

Schmidt Engine

The mass distribution will be considered for two cases, i.e. one in which the regenerator volume is relatively small, the other in which



FLUID MASS DISTRIBUTION FIG. 32

Table 1: Schmidt Engine Relative Mass Distribution for Small Regenerator

$\theta°$ =	0	20	40	60	80	100	120	140	160	180
w_H/W =	.0721	.0915	.1342	.1982	.2704	.3306	.3636	.3675	.3496	.3188
w_R/W =	.2481	.2808	.3144	.3408	.3507	.3397	.3127	.2789	.2464	.2914
w_C/W =	.6797	.6277	.5514	.4610	.3789	.3297	.3238	.3536	.4040	.4618

$\theta°$ =	200	220	240	260	280	300	320	340	360
w_H/W =	.2821	.2436	.2056	.1696	.1366	.1078	.0853	.0719	.0721
w_RW =	.1989	.1849	.1768	.1743	.1771	.1855	.1998	.2207	.2481
w_C/W =	.5190	.5715	.6176	.6561	.6863	.7067	.7149	.7074	.6797

Table 2: Schmidt Engine Relative Mass Distribution for Large Regenerator

$\theta°$ =	0	20	40	60	80	100	120	140	160	180
w_H/W =	.0905	.1052	.1326	.1699	.2097	.2427	.2619	.2659	.2573	.2402
w_R/W =	.3112	.3414	.3697	.3897	.3951	.3841	.3604	.3307	.3011	.2754
w_C/W =	.5983	.5534	.4976	.6404	.3952	.3732	.3777	.4035	.4417	.4844

$\theta°$ =	200	220	240	260	280	300	320	340	360
w_H/W =	.2183	.1942	.1699	.1465	.1252	.1071	.0939	.0875	.0905
w_R/W =	.2554	.2415	.2337	.2318	.2358	.2457	.2617	.2838	.3112
w_CW =	.5264	.5643	.5964	.6217	.6390	.6472	.6444	.6287	.5983

the regenerator volume is relatively large, namely twice that of the first case. In each case the engine is optimized in accordance with Walker's [19] curves* using a temperature ratio $\tau = 0.3$. Putting all volumes in terms of the regenerator volume V_R we have

	Rel. Small Regenerator	Rel. Large Regenerator
Swept volume, hot side	$2.0\ V_R$	$1.0\ V_R$
Dead volume, hot side	$0.5\ V_R$	$0.5\ V_R$
Swept volume, cold side	$1.5\ V_R$	$0.9\ V_R$
Dead volume, cold side	$0.5\ V_R$	$0.5\ V_R$
Phase angle, hot displacement leading cold displacement	$0.57\,\pi = 102.6°$	$0.53\,\pi = 95.4°$

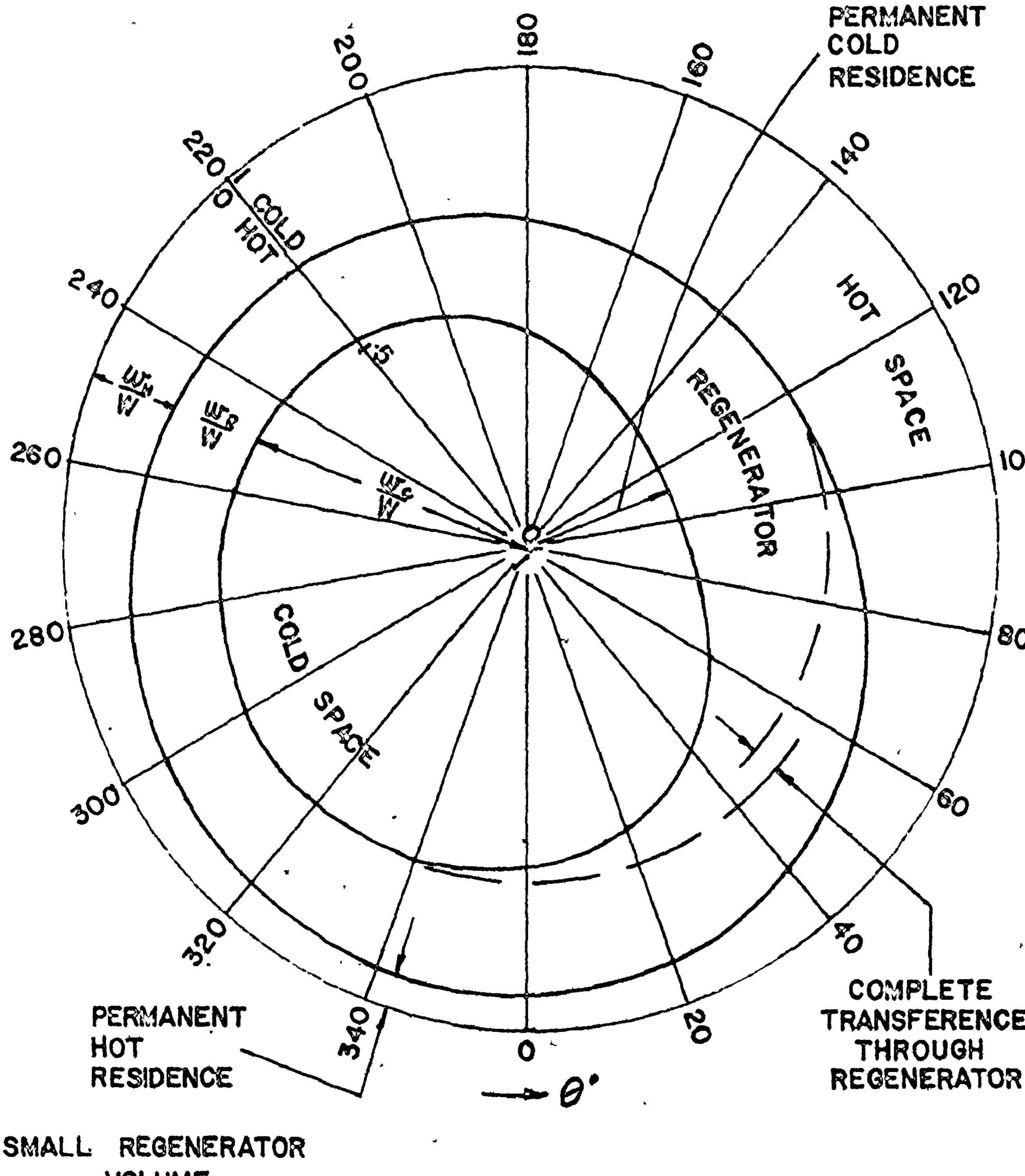
SHM motions for the Schmidt engine give:

	Rel. Small Regenerator	Rel. Large Regenerator
$\frac{V_H}{V_R}$	$1.50 - 1.0 \cos\theta$	$1.0 - 0.5 \cos\theta$
$\frac{V_C}{V_R}$	$1.25 - .75 \cos(\theta - 102.6°)$	$0.95 - 0.45 \cos(\theta - 95.4°)$

Inserting these in the expressions for mass distribution, $\frac{W_H}{W}$ and $\frac{W_C}{W}$, relative mass distributions result as shown in Tables 1 and 2. These results are shown in the polar plots of Figs. 33 and 34 for the small and large regenerator volumes respectively. The significance of these plots is:

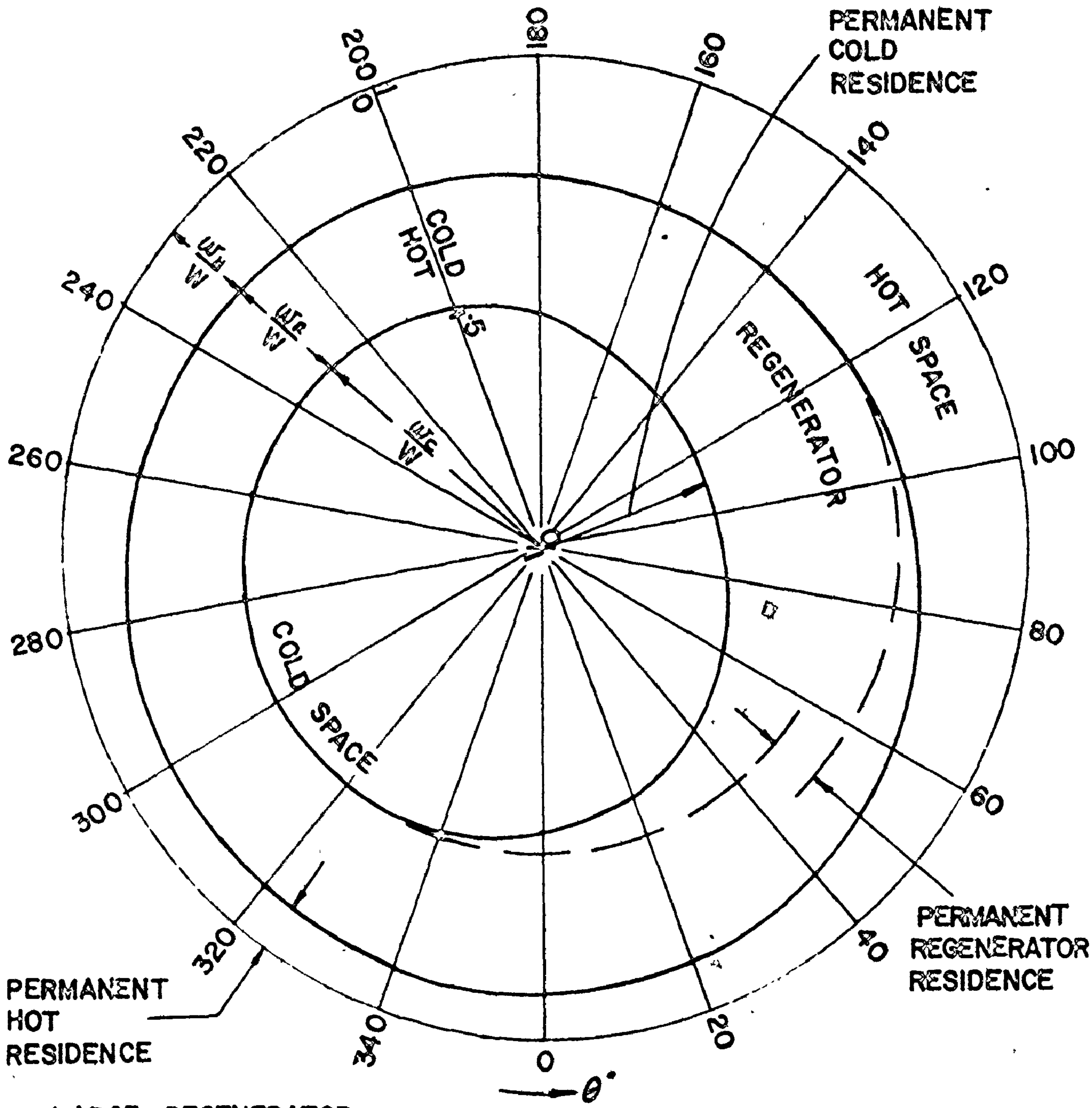
(a) Definite masses of fluid have either permanent hot space or permanent cold space residence (except for diffusion). This is inevitable since the volumes associated with the heat supply and rejection surfaces cannot be entirely absorbed by piston movements. These permanent resident masses experience pressure-temperature hysteresis cycles even when

* Walker's curves optimize the isothermal Schmidt engine, giving the phase angle and hot/cold swept volume ratio for given dead volume and temperature ratios.



SCHMIDT ENGINE RELATIVE MASS DISTRIBUTION OF WORKING FLUID

FIG. 33



LARGE REGENERATOR
VOLUME

SCHMIDT ENGINE RELATIVE MASS
DISTRIBUTION OF WORKING FLUID

FIG. 34

they are on the wrong side of the regenerator to take part in the reception or rejection of heat.

(b) For the small regenerator case, Fig. 33, only a relatively small fraction (.08 app.) of the total fluid charge completely traverses the regenerator and a substantial portion of this spends so long within the regenerator that it cannot contribute very much to the processes of isothermally taking in or rejecting heat. Bearing in mind that these are activities of thermodynamic equilibrium between the fluid and its bounding surfaces and that the fluid has traversed the full temperature difference across the regenerator with accompanying pressure losses, the "utilization" is quite low; this utilization may be recognized as the quantity $\frac{\text{power output}}{\text{wRT}}$ in other references to Stirling engine performance, e.g. Walker [26], Kirkley [64]. It may be seen how rapidly this ratio falls as the void volume increases.

(c) For the large regenerator the situation is even worse. None of the fluid charge completely traverses the regenerator to be utilized on both sides, but on the contrary there is now a fluid mass that is permanently resident in the regenerator. Also a substantial part of the fluid charge resides entirely on the cold side or traverses only a part of the regenerator when it is not in the cold space. Walker [19] gives mass flow rates and "cyclic trajectories" for particles of the working fluid for a reversed (i.e. cooling) isothermal Schmidt engine. These curves indicate that here too, net flow through the engine occurs for only a portion of the cycle time and that the working fluid is not well utilized. Better utilization of the working fluid can be achieved by (i) reducing dead volumes, (ii) by better coordination of the piston movements. These steps are certainly consonant with increasing the compression ratio which has been shown to have a strong probable influence on cycle efficiency. These achievements must therefore be recommended as engine design objectives if the Stirling cycle is selected. In the following chapter a proposal is made for at least partly advancing towards these objectives.

2. Mean Effective Pressure of Schmidt Engine

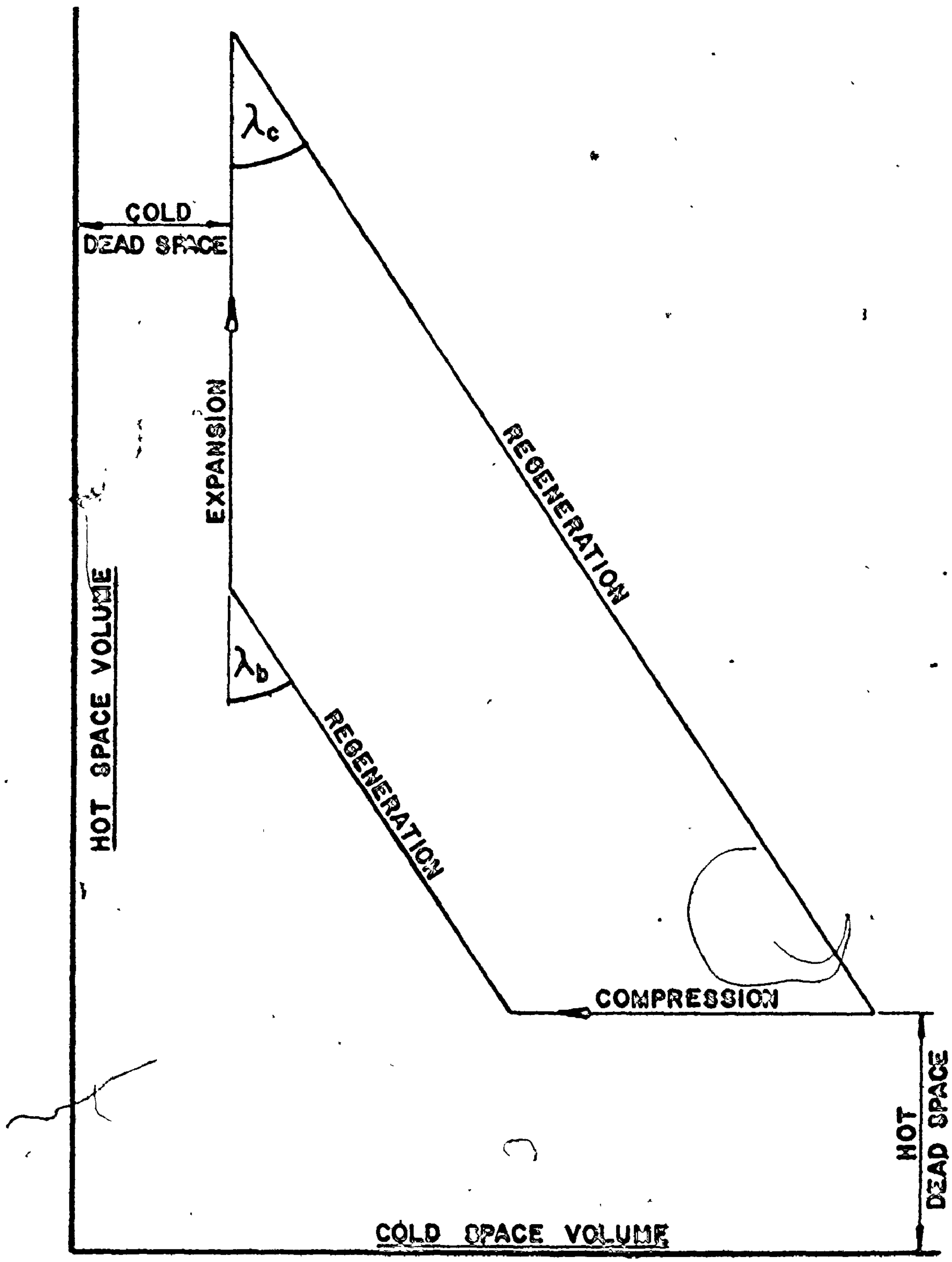
Referring to Fig. 20, the type of engine arrangement (a), (b) or (c) significantly influences the mean effective pressure. The Schmidt isothermal theory, developed for type (b) engines, has been extended by Kirkley [64] to types (a) and (c). Kirkley gives curves of "optimum power parameter" which are essentially measures of the mean effective pressure to maximum pressure ratio. For a representative $\tau = 0.3$ and a dead volume/expansion volume of 1.0, Kirkley's pressure ratios for the three engine ratios for the three engine types are approximately:

	type (a)	type (b)	type (c)
$\frac{MEP}{P_{max}}$	.15	.17	.20

Owing to marked advantages of the type (c) engine, i.e. compactness, minimization of sealing problems and relatively low temperature gradients I feel that this is the best of the three for the artificial heart. Fortunately, it also has the best mean effective pressure. It is noted that this arrangement, but using a displacer-regenerator, with a rhombic drive is the basis of the North American Philips development engine, Fig. 29.

3. Volume Relationships

The performance of Stirling engines depends much upon the relationships of the hot and cold volumes of the engine at any moment. To facilitate understanding of this I have devised a method of showing the hot space volume/cold space volume as a plot, Fig. 35. On such a plot, not only the ideal cycles, but also the contemporary cycles, Schmidt and rhombic drive (which I later analyze mechanically) may be shown for comparison.



VOLUME RELATIONSHIPS FOR CYCLES INTERMEDIATE BETWEEN STIRLING AND ERICSSON CYCLES

FIG. 35

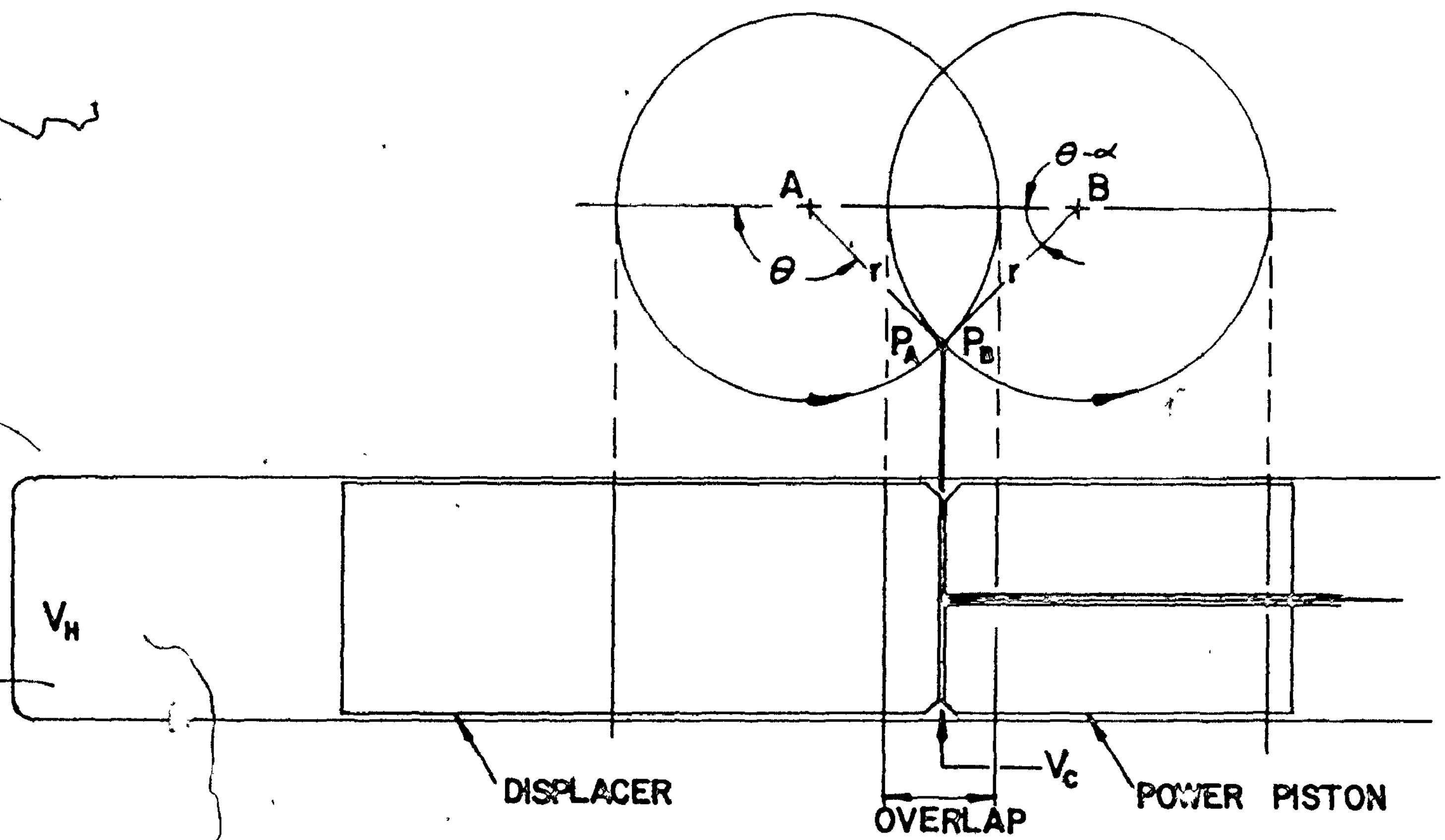
The true Stirling cycle is by definition one of constant volume during the regeneration processes, thus these lines are such that $\lambda_b = \lambda_c = 45°$, assuming equal scales for the coordinates. For the Ericsson cycle which has constant pressure regeneration, one volume of cold space is equivalent to $1/\tau$ volumes of hot space, thus for this cycle $\lambda_b = \lambda_c = \arctan \tau$. As pointed out earlier there is an infinite number of cycles other than these ~~two that~~ have Carnot efficiency. All of these cycles having linear hot-cold space changes may also be shown as straight-sided figures. In the final development of the piston movements of an engine it is conceivable that λ_b and λ_c are not equal for optimum results. This diagram has the further advantages of being easy to relate to mechanical design. Also, lines of constant presure (and therefore of constant local density) are straight lines inclined to the vertical axis at $\arctan \tau$. To compare their hot space-cold space relationships with those of the ideal cycle I have kinematically analyzed the Schmidt and Meijer (rhombic drive) engines.

(a) Schmidt Cycle

The hot and cold volume relationships are considered for a displacer type engine having the displacer and power piston in one cylinder, i.e. of type (c) in Fig. 20. In the Schmidt cycle the piston and displacer have harmonic motions so that the opposing faces of the displacer and power piston may be considered as projections of P_A, P_B, see Fig. 36. In this diagram P_A and P_B coincide, i.e. the piston and displacer are practically touching. It is important that the opposing faces nearly touch at their closest positions to obtain the maximum compression ratio. The overlap of the displacer and power piston travels must be a maximum. Assuming equal strokes of $2r$ and that the hot space leads the cold space by α, then

$$AB = 2r \sin\frac{\alpha}{2} \quad ,$$

and the overlap is $2r(1-\sin\frac{\alpha}{2})$. For $\alpha = 90°$ this becomes .586r or .293 x



ENGINE WITH HARMONIC MOTIONS (SCHMIDT)

FIG. 36

stroke. The minimum possible total cylinder volume is when P_B lies in AB and is then represented by $2r\sin\frac{\alpha}{2}$. The maximum possible volume is when P_B lies on AB produced, at the extreme right of P_B's excursion. This volume is represented by $2r + 2r\sin\frac{\alpha}{2}$. Thus defining the compression ratio as maximum volume/minimum volume,

$$\text{Maximum Compression Ratio} = \frac{2r + 2r\sin\frac{\alpha}{2}}{2r\sin\frac{\alpha}{2}}$$

$$= 1 + \operatorname{cosec}\frac{\alpha}{2}\,. \qquad 6.6$$

For a phase angle $\alpha = 90°$ this equals 2.41. This is a basic ratio that is greatly reduced by the large volumes of the heater, regenerator and cooler. At any shaft angle θ from the zero hot space displacement,

$$\text{Hot space displacement} = r\,(1-\cos\theta) \qquad 6.7$$

$$\text{Cold space displacement} = r\,(2\sin\frac{\alpha}{2} + \cos\theta - \cos[\theta-\alpha])\,. \qquad 6.8$$

If we multiply the cold space quantity by a factor $k = \frac{\text{piston area} - \text{rod area}}{\text{piston area}}$ then the multiplied cold space displacement represents the cold volume to the scale that the hot space displacement represents the hot volume. Selecting $k = .9$ and $\alpha = 90°$ we have the relative volume relationship shown in Table 3.

Table 3: Schmidt Engine Displacements

$\theta°$ =	0	20	40	60	80	100	120	140	160	180
Rel. V_H =	0	.0302	.1170	.2500	.4132	.5868	.7500	.8830	.9698	1.0000
Rel. V_C =	1.0864	.9053	.6919	.4717	.2714	.1151	.0217	.0024	.0596	.1864

$\theta°$	200	220	240	260	280	300	320	340	360
Rel. V_H =	.9698	.8830	.7500	.5868	.4132	.2500	.1170	.0302	0
Rel. V_C =	.3674	.5809	.8011	1.0014	1.1577	1.2511	1.2704	1.2132	1.0864

Equal strokes have been taken in this example to make the Schmidt volume relationships comparable with those of the rhombic drive engine, which, as in this example, has equal strokes for the displacer and power piston.

(b) The Rhombic Drive

This mechanism is shown in Figs. 37 and 38. The kinematics are essentially those of two engine mechanisms in which the lines of action of the connecting rod wrist pins (yoke pins) are offset substantially from the centre lines of the crankshafts. From the dimensions marked in Fig. 38 it may be shown that:

Hot space displacement from zero,

$$x = \sqrt{l^2 - h^2} - \sqrt{l^2 - h_o^2} + r\cos\lambda - r\cos(\theta - \beta) \quad , \qquad 6.9$$

where h_o is the value of h when $\theta = 0$, i.e. when the hot displacement is zero, which occurs when the crank radius and conrod are in line and overlapping, making an angle β with the horizontal line through the crankshaft axis. Making θ the only variable this becomes

$$x = \sqrt{l^2 - (d - r\sin[\theta - \beta])^2} - \sqrt{l^2 - (\frac{dl}{l-r})^2} + r\cos\beta - r\cos(\theta - \beta) \qquad 6.10$$

The maximum hot displacement occurs when the crank radius and the conrod are again in line, extended:

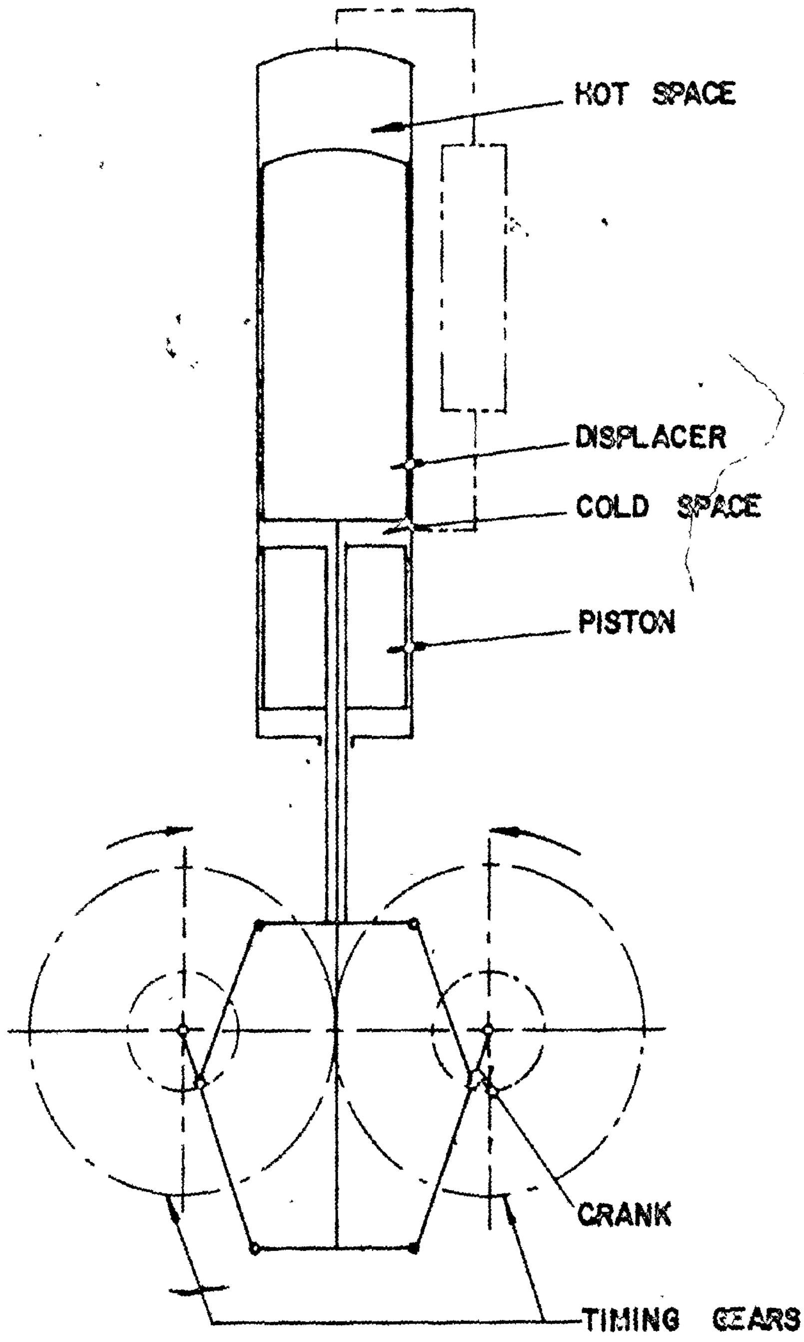
$$x_{max} \doteq \sqrt{(l + r)^2 - d^2} - \sqrt{(l - r)^2 - d^2} \qquad 6.11$$

The cold "space", i.e. the distance between the faces of the displacer and power piston (assuming that these nearly touch when closest and neglecting the clearance) is:

$$2[\sqrt{l^2 - (d - r)^2} - \sqrt{l^2 - (d - r\sin[\theta - \beta])^2}] \qquad 6.12$$

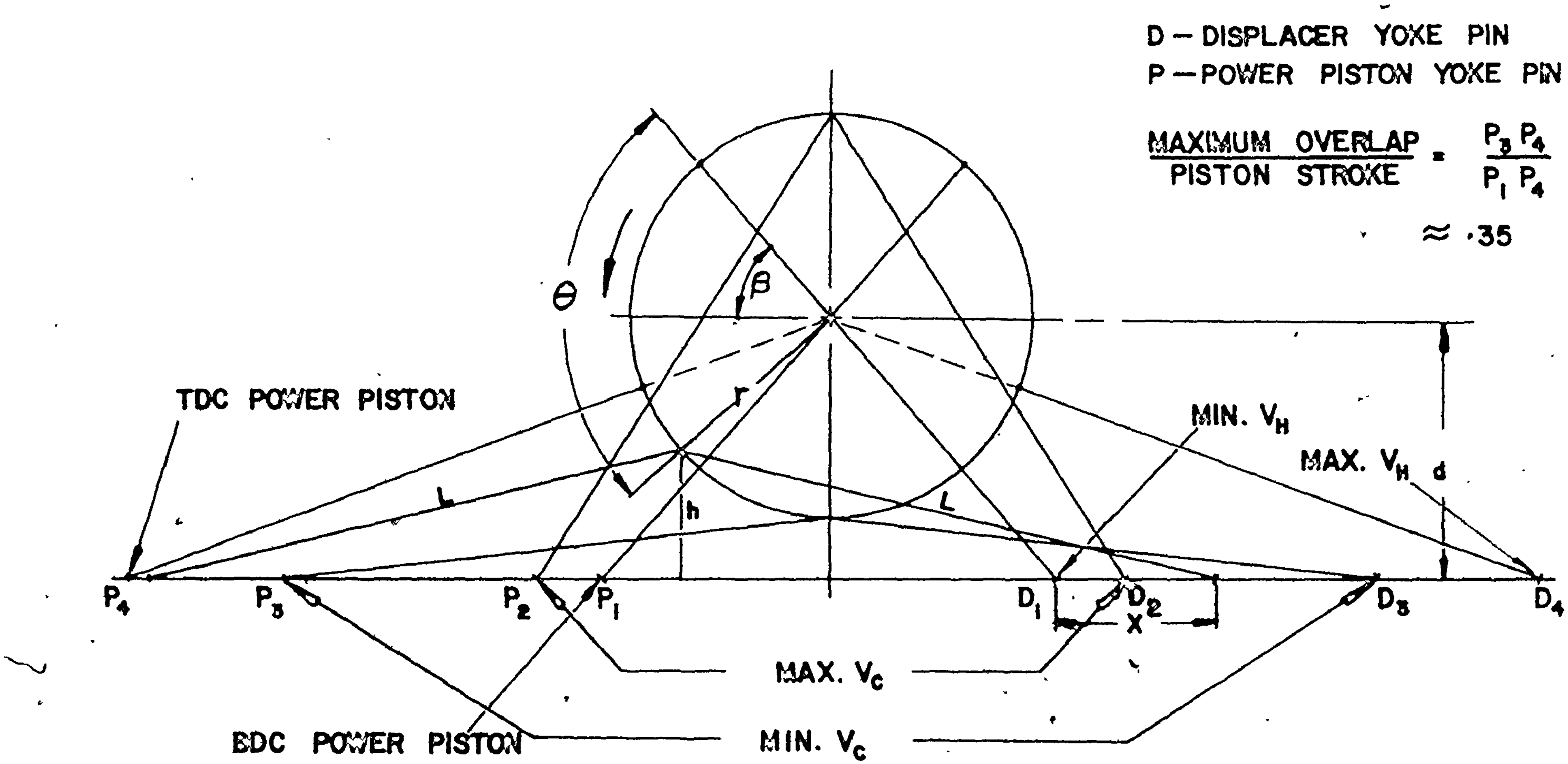
Applying a factor k to the cold "space" to allow for the displacer rod area the displacement on a volume basis (now comparable with the hot displacement as a volume displacement) is

$$2k[\sqrt{l^2 - (d - r)^2} - \sqrt{l^2 - (d - r\sin[\theta - \beta])^2}] \qquad 6.13$$



RHOMBIC DRIVE

FIG. 37



SALIENT POINTS OF RHOMBIC MECHANISM

(ONE HALF OF MECHANISM SHOWN)

FIG. 38

Dividing both quantities by the maximum hot displacement, the maximum relative hot displacement is unity and the engine, on this basis, may be compared with the Schmidt engine.

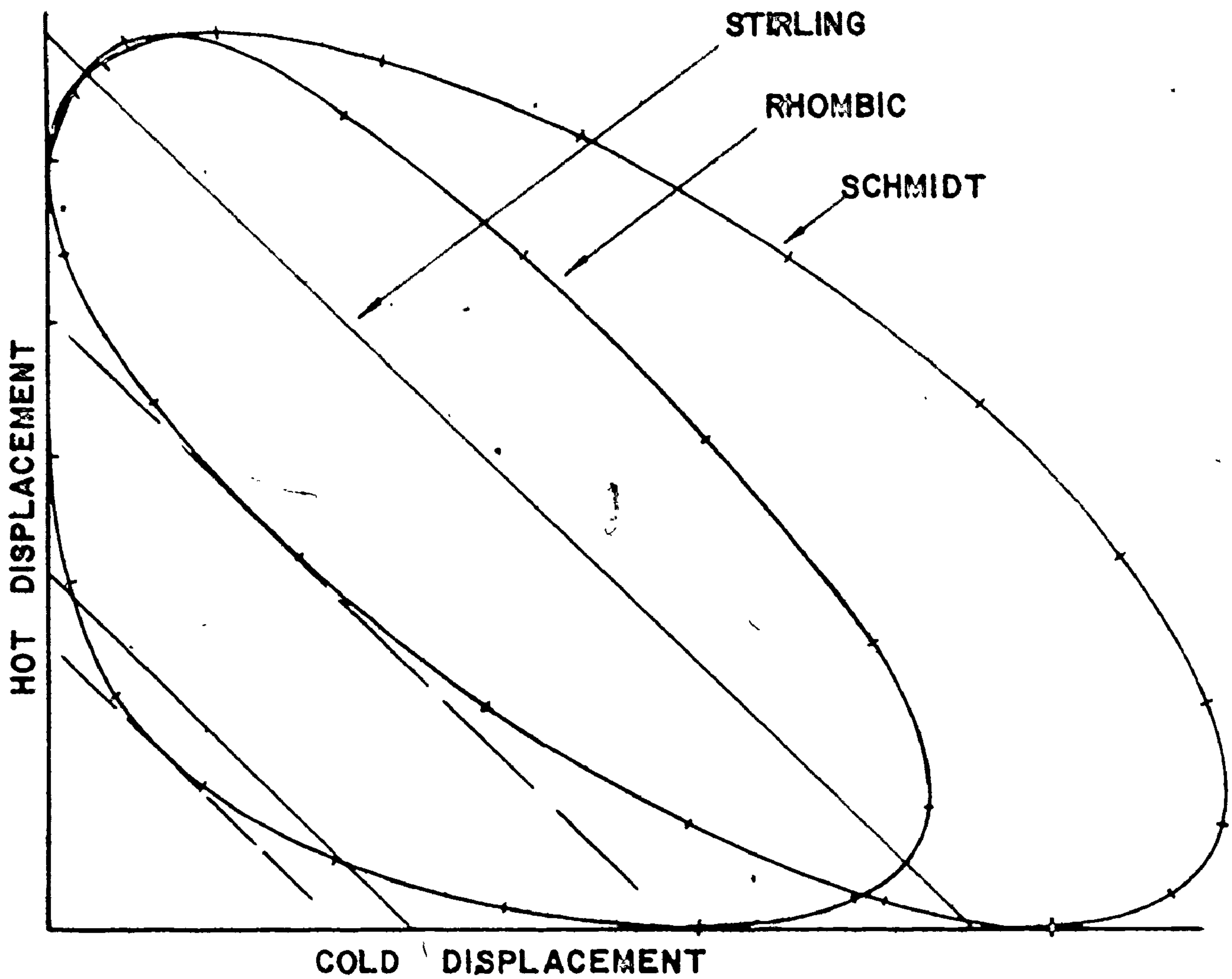
Taking k = 0.9 and using the proportions of Fig. 36 (r = 32.5 mm, d = 42 mm, l = 88 mm) we obtain the displacement relationship shown in Table 4.

Table 4: Rhombic Drive Displacements

$\theta°$	0	20	40	60	80	100	120	140	160	180
Rel. V_H =	0	.0217	.0772	.1584	.2619	.3857	.5259	.6740	.8153	.9291
Rel. V_C =	.7035	.4970	.3101	.1677	.0755	.0260	.0051	0	.0060	.0288

$\theta°$	200	220	240	260	280	300	320	340	360
Rel. V_H =	.9929	.9879	.9044	.7466	.5358	.3129	.1317	.0290	0
Rel. V_C =	.0813	.1774	.3239	.5137	.7199	.8897	.9543	.8789	.7035

This displacement relationship is plotted in Fig. 39, together with that of the Schmidt engine having the same piston and displacer strokes. The true Stirling diagram is also shown for comparison. On these diagrams the marks represent equal intervals of shaft rotation. Since the rhombic and Schmidt diagrams are derived from the motions of rotating shafts and defined linkages, suitable "starting" points may be located so that their time-displacement relationships may be progressively compared. If the minimum total volumes, i.e. the points of the beginning of expansion, are taken as the starting points, these are located as the tangent points with lines at 45° to the axes on the left side of the diagrams. The Stirling diagram, however carries no timing marks having been simply defined in terms of volumetric relationships only and reminding us that there would be some freedom, when developing kinematics for this cycle, to optimize the fluid flow velocities in the engine.



VOLUME DISPLACEMENT
RELATIONSHIPS

FIG. 39

Divergencies of these diagrams from the ideal straight lines of the Stirling diagram are quite marked, also the rhombic drive diverges less than the Schmidt. From both diagrams we see that significant heating of the fluid charge occurs during volumetric compression and significant cooling occurs during volumetric expansion, these having inevitable adverse effects on the mean effective pressure ratio, the work ratio and the efficiency.

4. Stirling Cycle Design Flexibility

Recalling the expression for cycle efficiency, with allowance for regenerator efficiency,

$$\eta = \frac{(1-\tau)}{1 + \frac{(1-\tau)(1-\eta_R)}{(\gamma-1)\ln r}} \qquad 3.28$$

it is seen that for given τ and γ the cycle efficiency depends upon $\frac{(1-\eta_R)}{\ln r}$. Thus over a range of η_R values the cycle efficiency may be maintained if suitable r values are used. Assuming $\tau = 0.3$, $\gamma = 1.667$ for example, the following compression ratios r would maintain the cycle efficiencies η for the regenerator efficiencies shown.

Cycle eff. η	$\eta_R = .98$	$\eta_R = .95$	$\eta_R = .90$
.66	1.41	2.38	5.65
.62	1.18	1.50	2.26

In comparing regenerators having the same matrix construction and equal flow velocities the required surface area is proportional to the thermal load, i.e. to η_R, and inversely proportional to the mean temperature difference, i.e. to $(1-\eta_R)$. Thus the surface areas, and therefore the void volumes, of these regenerators, are approximately proportional to $\frac{\eta_R}{(1-\eta_R)}$. Optimized regenerator designs may not have equal flow velocities of course, but they would have void volumes that increase very rapidly as η_R approaches unity. Alternatively the reduction of regenerator void volume that occurs as η_R is reduced from very high values to lower values may significantly assist in increasing the compression ratio! For example, if a Schmidt engine has a regenerative void volume of a similar magnitude to the expansion volume, its regenerator efficiency being .98, then reducing this efficiency to .95 would decrease the void volume to approximately 41% of the original, assisting in raising the compression ratio.

Summary

(a) Diagrams have been prepared showing the effect of a regenerator void volume and of other dead spaces for a Schmidt cycle. The utilization of the working charge is reduced due to various degrees of retention of the working fluid in the regenerator matrix. Large regenerators can completely retain a portion of the charge so that it never leaves the regenerator matrix. The concept of utilization of the working charge that these diagrams give is akin to the "power parameters", i.e. $\frac{\text{output power}}{\text{pressure x swept volume}}$ or $\frac{\text{output power}}{\text{wRT}}$ calculated by other workers.

(b) Using Kirkley's results the mean effective pressure ratios of engine types (a), (b) and (c) of Fig. 20 were estimated and type (c) found to be the best.

(c) I have developed diagrams for showing the coordination of the piston movements of Schmidt and rhombic drive engines in terms of the hot and cold space variations. These diagrams reveal substantial discrepancies when compared with those of the Stirling cycle.

CHAPTER 7

CONCLUSION

This chapter reviews the results of the preceding analyses and endeavours to point out a direction in which Stirling engine development might be tried out with the objective of better efficiencies, especially with regard to very small engines, amongst which that for the artificial heart has a commanding interest. In comparison with most heat engines, the efficiency of these very small engines is, perhaps expectedly, below that of most modern heat engines of large size, i.e. of multihorsepower. Nonetheless the achieved efficiencies are already high enough for immediate application of the engine to its intended purpose, even though far below the theoretical potential for the ideal cycle; that is there is much room for improvement. Together with a large potential demand for the artificial heart, without forseeable end, it therefore appears technically justified to continue developing the engines towards higher efficiencies. It is through higher engine efficiencies that a smaller envelope, lower weight, reduced radioactive material inventory or extended "recharging" periods will be largely achieved.

Although it would be surprising to see no further efficiency improvement from current developmental engines that have now had several years of development, it would be more surprising to obtain a further substantial efficiency gain from them.

1. Cycle Evaluation

(a) Adiabatic Cycles

The ideal regenerative Otto cycle was examined in Chapter 4. It was seen (Fig. 16) how rapidly efficiency falls with increasing compression ratio. In a practical cycle there must always be some losses however, resulting in zero efficiency for the compression ratio of unity (zero output) as opposed to maximum efficiency for the ideal engine. Therefore the actual efficiency-compression ratio curve is expected to be peaked.

Another adiabatic cycle that perhaps represents many "Stirling" engines more closely than the regenerative Otto cycle is the Schmidt adiabatic cycle. It differs from the regenerative Otto mainly in that operations are performed in two sinusoidally varying volumes and their interconnecting heater, regenerator and cooler. Walker and Khan [34], proceeding from Finkelstein's work, have plotted the ideal cycle performance, from which some representative figures of interest are given in Appendix I. These figures reveal an efficiency trend very similar to that of the ideal regenerative Otto cycle, the best efficiency being associated with very large dead volumes, that is with compression ratios apparently approaching unity. Allowing for some losses in a practical engine, a peak in the actual efficiency-compression ratio curve is expected.

At the same time I have shown that large dead space volumes are associated with relatively large fluid mass residence in the dead spaces and in the regenerator of the engine with low utilization of the working fluid.

It is concluded that:

(i) Practical adiabatic engines must have optimum compression ratios significantly greater than unity, at which ratio it would be impossible to operate.

(ii) Large dead space volumes favour the ideal engine but adversely effect the efficiency of practical engines in terms of utilization of the fluid mass.

(iii) Also, from Fig. 17 and from Walker and Khan's work [34], the mean effective pressure of adiabatic engines is lower than that of isothermal engines.

(b) Isothermal Cycles

From Chapters 3 and 4 the isothermal cycles qualifying for consideration are the Stirling and the Thermocompressor. Calculated parameters are given for these cycles for temperature ratio $\tau = 0.3$, $\gamma = 1.667$ and perfect regeneration:

	Ideal Stirling			Thermocompressor
	r = 1.25	1.8	3.0	Optimum
1. Output/total reversing heat	.047	.123	.230	.035
2. Work ratio	.538	.538	.538	.059
3. Mean pressure/max. pressure	.625	.514	.385	.152

Each cycle has the potential Carnot efficiency 0.7. Although these parameters are for ideal engines they are expected to be applicable to practical engines when viewed as approximate figures of merit. As an engine is developed to higher efficiencies it will more nearly approach the ideal, tending to validate the merit principle upon which the engine cycle was selected.

The ideal Stirling cycle efficiency is independent of compression ratio, but the effect of even small losses must inevitably reduce the efficiency to zero as the compression ratio approaches unity, as shown for the adiabatic cycles. Unlike the adiabatic cycles, however, the efficiency should improve continuously as the compression ratio rises if the losses are due to regenerator imperfections (Fig. 13). The adverse effect of parasitic heat losses on engine efficiency is expected to increase as engine dimensions increase for an engine of given power, conceivably causing a peak in the efficiency-compression ratio curve. The engine dimensions would increase due to falling mean effective pressure as the compression ratio increases as indicated by the third parameter in the above table.

For the isothermal cycles therefore, it is concluded that:

(i) The Stirling engine appears much more promising than the Thermocompressor.

(ii) Also, the compression ratio is likely to be an important factor in developing a high efficiency Stirling engine.

To be consistent with these conclusions a programme for developing an engine of maximum efficiency would commence with an engine constructed to be able to follow the true Stirling cycle and to be adjustable in its compression ratio, permitting exploration and exploitation of this parameter. For this concept, features are suggested for incorporation in an initial engine:

1. Piston and displacer motions would be cam controlled, facilitating the predetermination of mechanical relationships. While primarily intended to ensure engine spatial relationships to the true Stirling cycle (other cycles for comparison and "adjustments" to the Stirling in course of development) cam control gives more independent control over the relative timing of each element of the cycle for the better balancing of heat transference and pumping losses. In later engine development, having found optimum cam profiles, if desired, suitable linkages might be developed to approximate the motions. It is appreciated that this could imply a considerable development programme in itself.

2. The initial engine would be designed for a rather higher power output

i.e. a larger engine having greater internal volumes and not so extremely light in construction as that for the artificial heart requirement. This would reduce distortion of the engine's performance due to the connection of a mass of instrumentation and would permit component manufacture at a relatively comfortable level of technology. It is emphasized that the main purposes of the prototype engine tests are to verify, or otherwise, that trueness to the ideal Stirling cycle is important and to determine how critical the compression ratio really is with respect to efficiency.

3. Cam based mechanisms for controlling the piston and displacer motions will incur mechanical losses. Conceivably quite a wide range of cam contours might be tried, possibly with considerable variation of the obliquity of the cam thrust on the follower and therefore a variation of the frictional losses. The cam mechanism should therefore be designed so that the frictional power lost can be measured and allowed for. This would prevent the distortion, or even apparent reversal of the internal effects on engine performance when the cams are changed. Calorimetric methods appear suitable for this, based on temperature rise of the mechanism as frictional heat is added. To facilitate this, the cam mechanism would be thermally insulated from the engine cylinders.

4. It has been shown that the coaxial displacer-piston engine, i.e. of the configuration depicted by (c) of Fig. 20, is slightly superior to other engine configurations. The closed end form of this engine cylinder is suitable for a solid nuclear isotope fuel capsule, and indeed the closed cylinder is used in all current developmental work. This configuration also minimizes the dead spaces which control the ultimate compression ratio possible. Further it is the most compact, yet permits relatively low temperature gradients in the engine cylinder and dis-

placer. It would therefore be adopted, with the modification of using a displacer-regenerator rather than the external regenerator. The prototype engine would be electrically heated, in lieu of a nuclear source.

5. To promote isothermal conditions in each working volume of the engine the heat transfer surfaces should be large. This may be achieved by using contoured rather than flat surfaces to bound the ends of the respective volumes. To a greater or lesser extent this may be seen in the current engine designs shown in Chapter 5, but they do not persuade one that the maximum effort has been made in this direction.

[The flexure used to locate the hot end of the displacer of the DWDL engine, Fig. 25, is interesting in that it provides two "extra" surfaces for heat transfer, their effectiveness depending upon the absorbtivity and emissivity of the relevant surfaces, also on the cylinder end surface temperature.]

For the type of engine proposed there may be a possibility of approximately doubling the heat removal surface at the cold end (in addition to any gains from contouring) by making the displacer rod a good thermal conductor, probably using a larger rod sectional area than otherwise necessary, in good thermal contact with the lower surface of the displacer-regenerator. This implies the need to remove the heat further down the rod; also a mass penalty would probably be incurred. In principle the additional mass could be dynamically balanced to prevent vibrations from this source. In practice, any mass balancing would be preferably integrated with the linkage controlling the motions of the pistons. A rod having nearly perfect conductivity would require a relatively very large mass of material, i.e. ~~the~~ practical rod must be optimized and will be dependent upon the total performance of all engine components. It is not therefore possible to predict the precise thermodynamic worth of the enhanced heat dissipation at this stage and the idea must stand as a desirable feature for the advancement of engine efficiency.

APPENDIX I

Schmidt Adiabatic Cycle

Data taken from reference [34]:

Engine parameters for temperature ratio $\tau = 0.3$

k = Cold swept volume/hot swept volume = 1.0

α = Phase lead of hot swept volume = 90°

x = Dead volume/hot swept volume

Performance:

x =	0.25	0.5	1.0	1.5
Work output $\frac{P}{wRT}$ =	.37	.26	.15	.10
Cycle efficiency η =	.54	.58	.62	.63

APPENDIX II

Performance of a Current Artificial Heart Engine

Data for DWDL documented System 6 research engine (Fig. 25) and the pending System 6 implantable engine from reference [52]:

	Documented System 6 Research Engine	Pending System 6 Implantable Engine
Peak cycle temperature	430°C	500°C
Heat sink temperature	39.5°C	39.5°C
Carnot efficiency	.555	.596
Gross pneumatic output	6.1 W	8.9 W
Less displacer pneumatic drive power	0.9 W	1.5 W
Net engine pneumatic output	5.2 W	7.4 W
Gross efficiency	.185	.270
Net efficiency	.158	.224
Parasitic engine heat losses	14.5 W	11.9 W
Regenerator reheat losses	5.1 W	5.9 W
Net efficiency relative to Carnot	.28	.38

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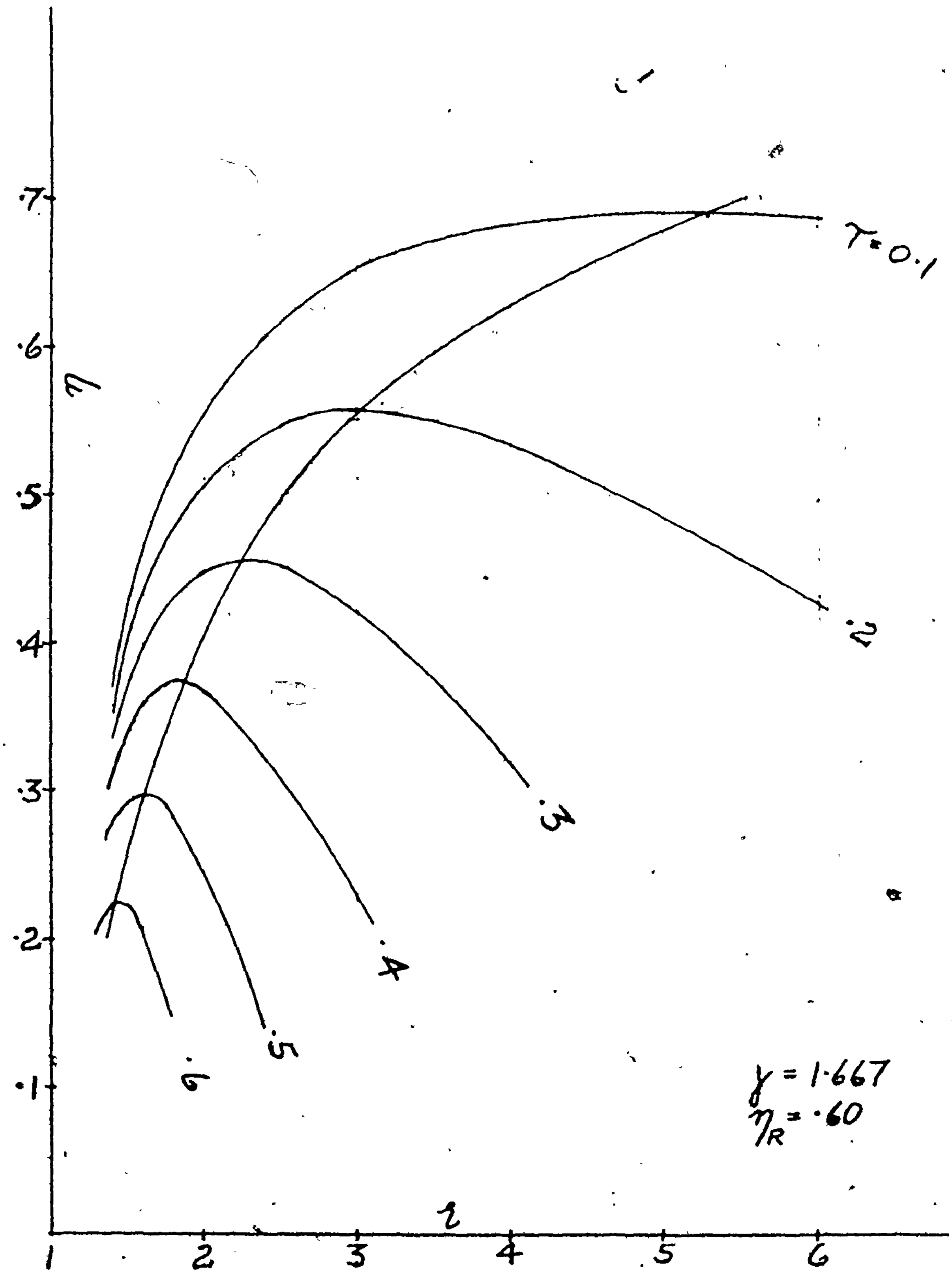

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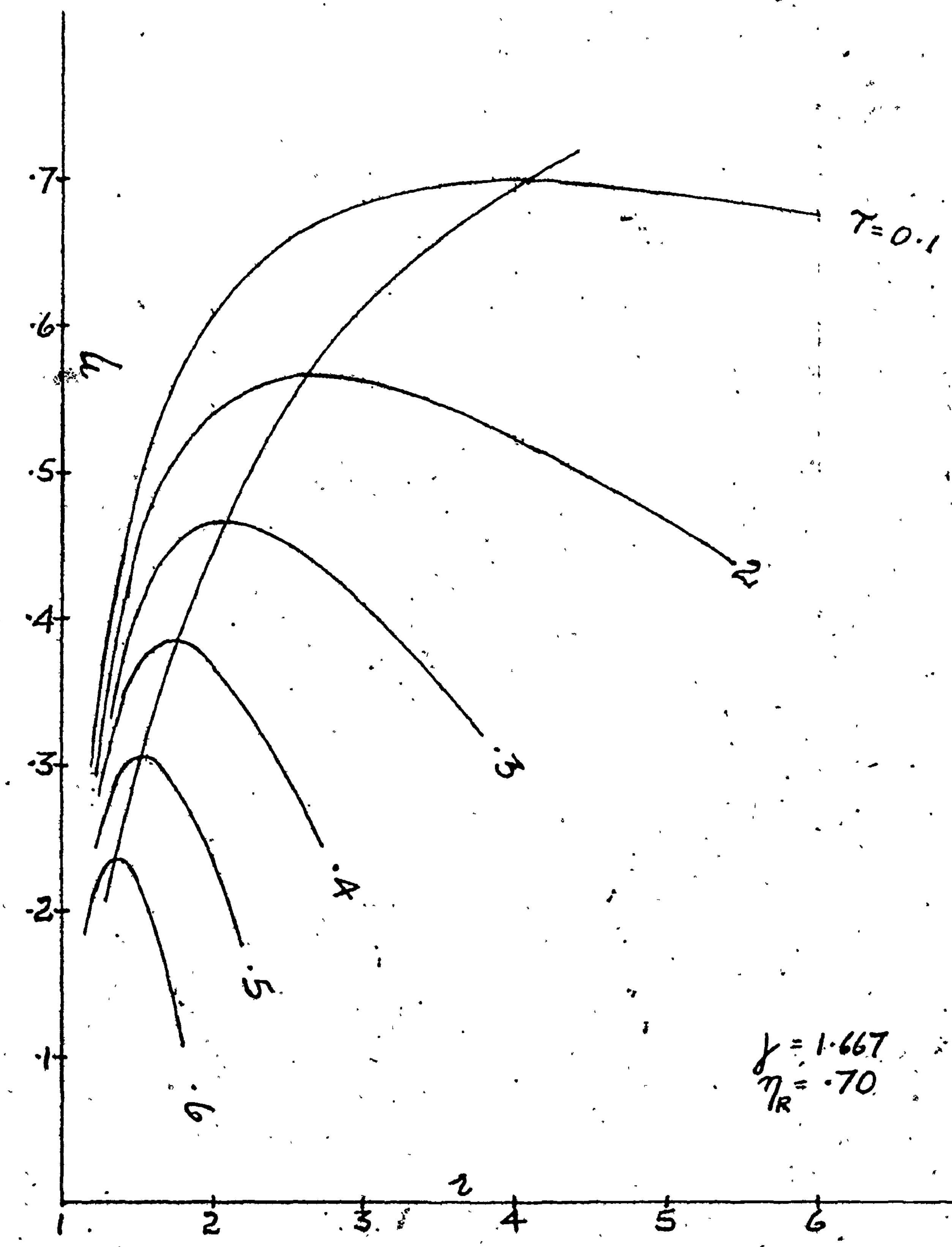

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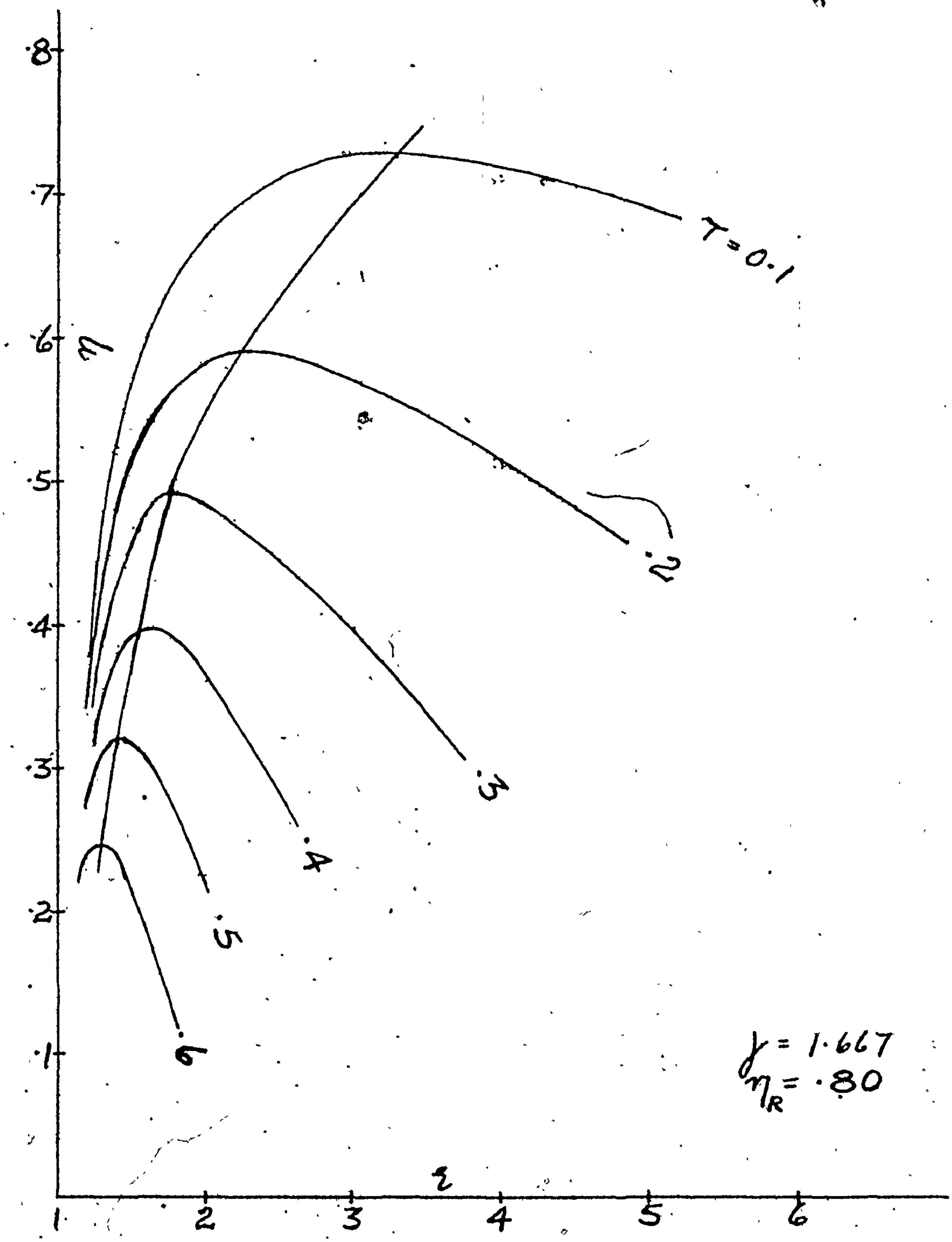

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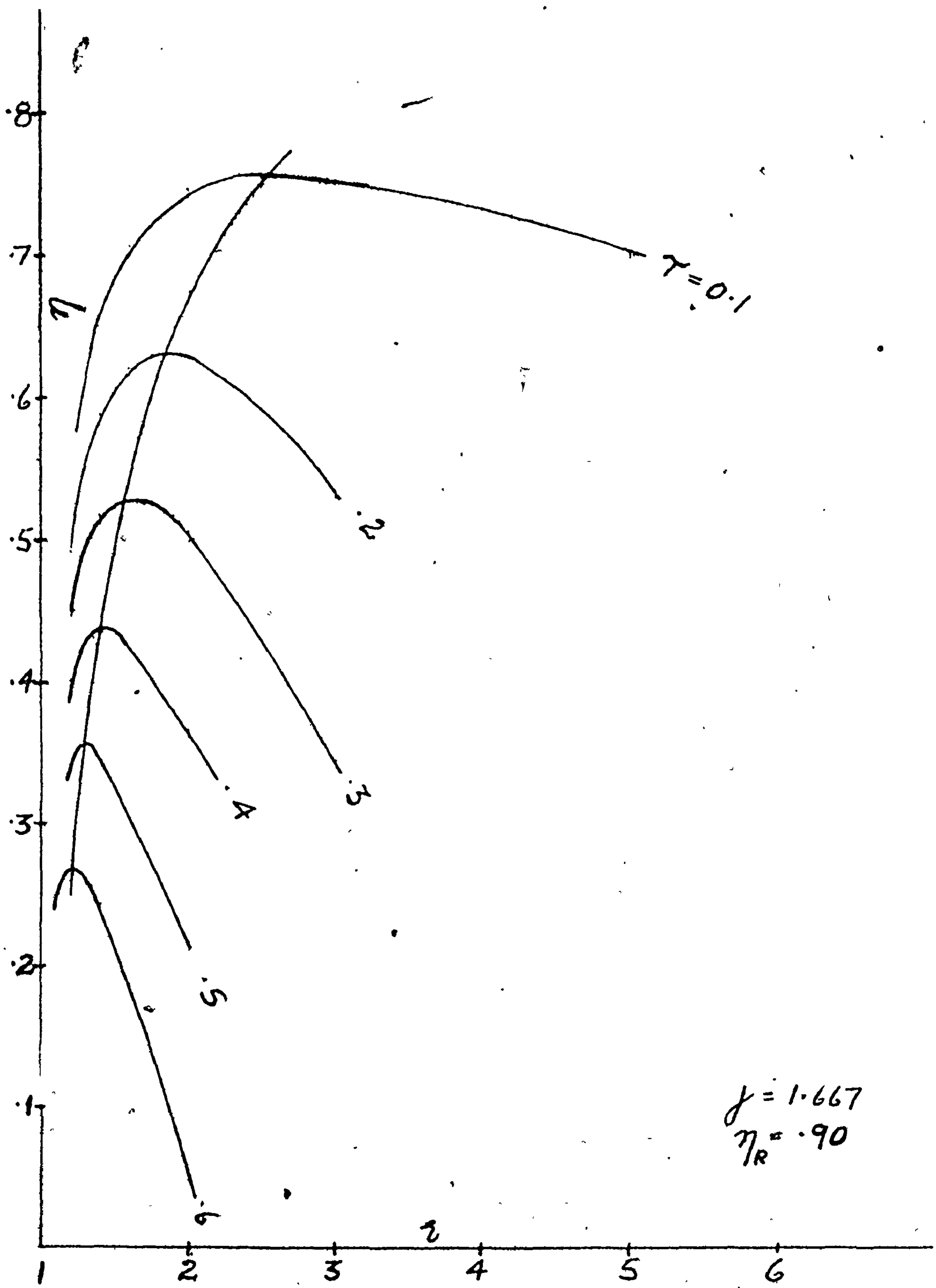

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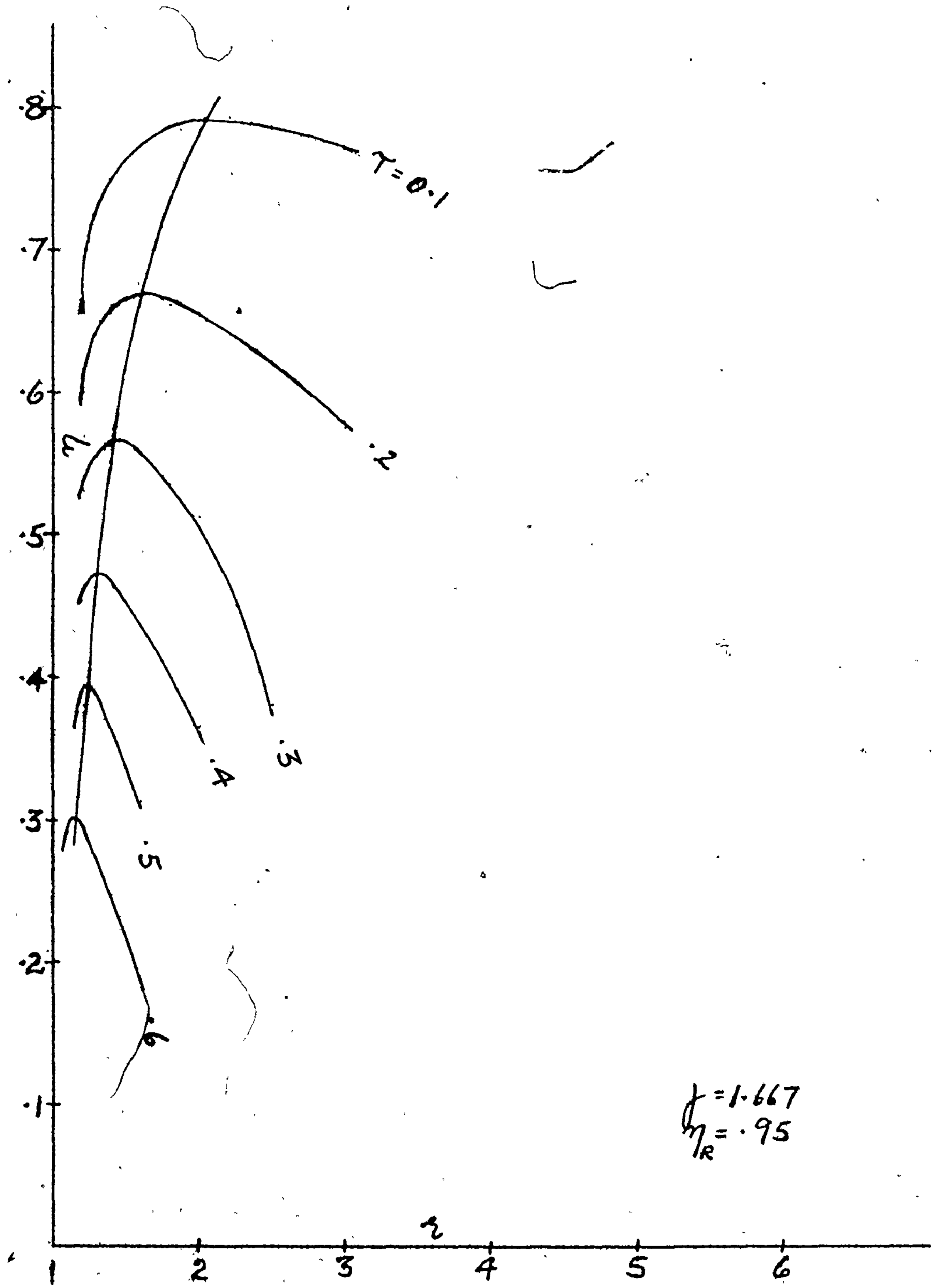

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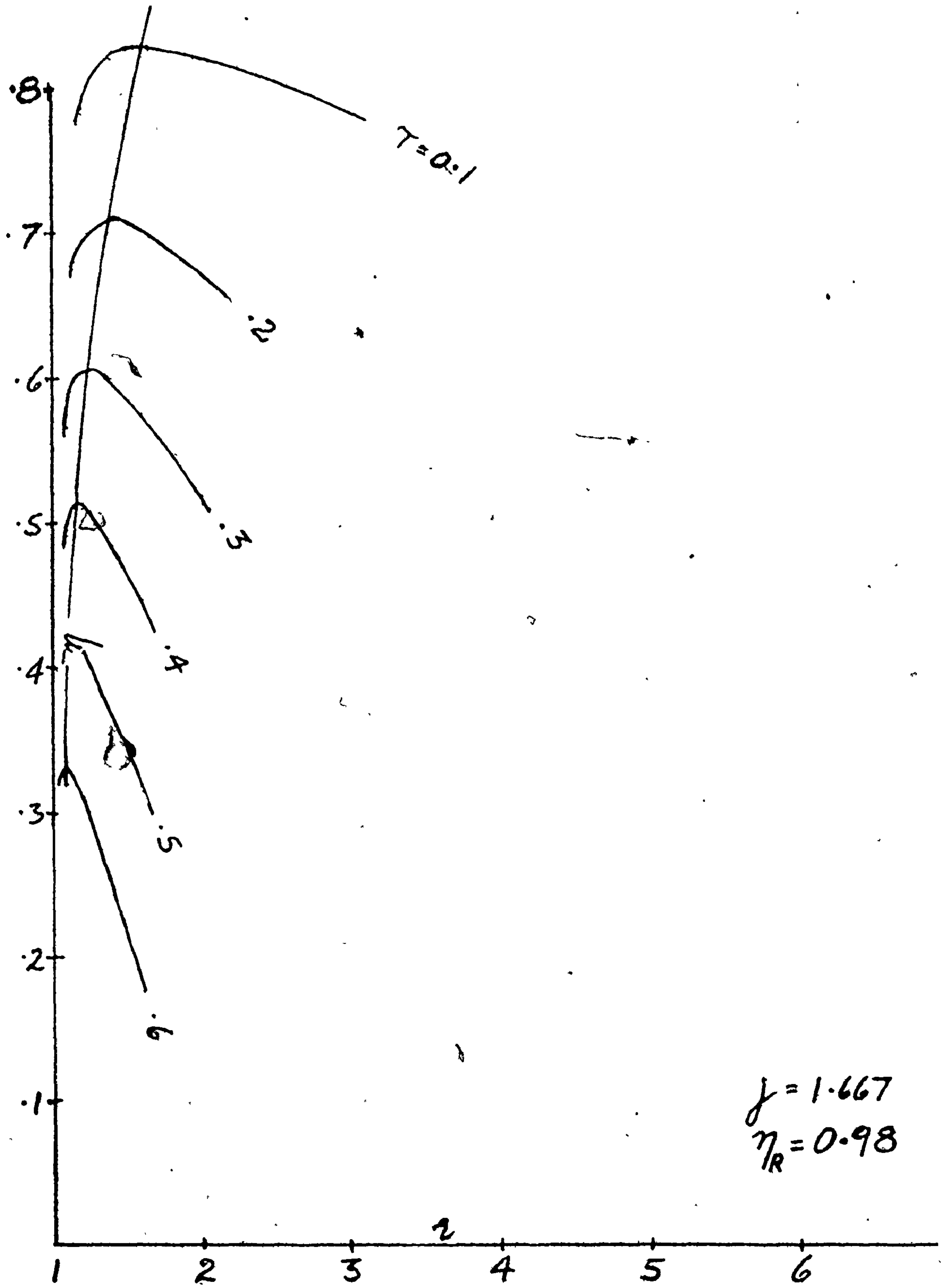

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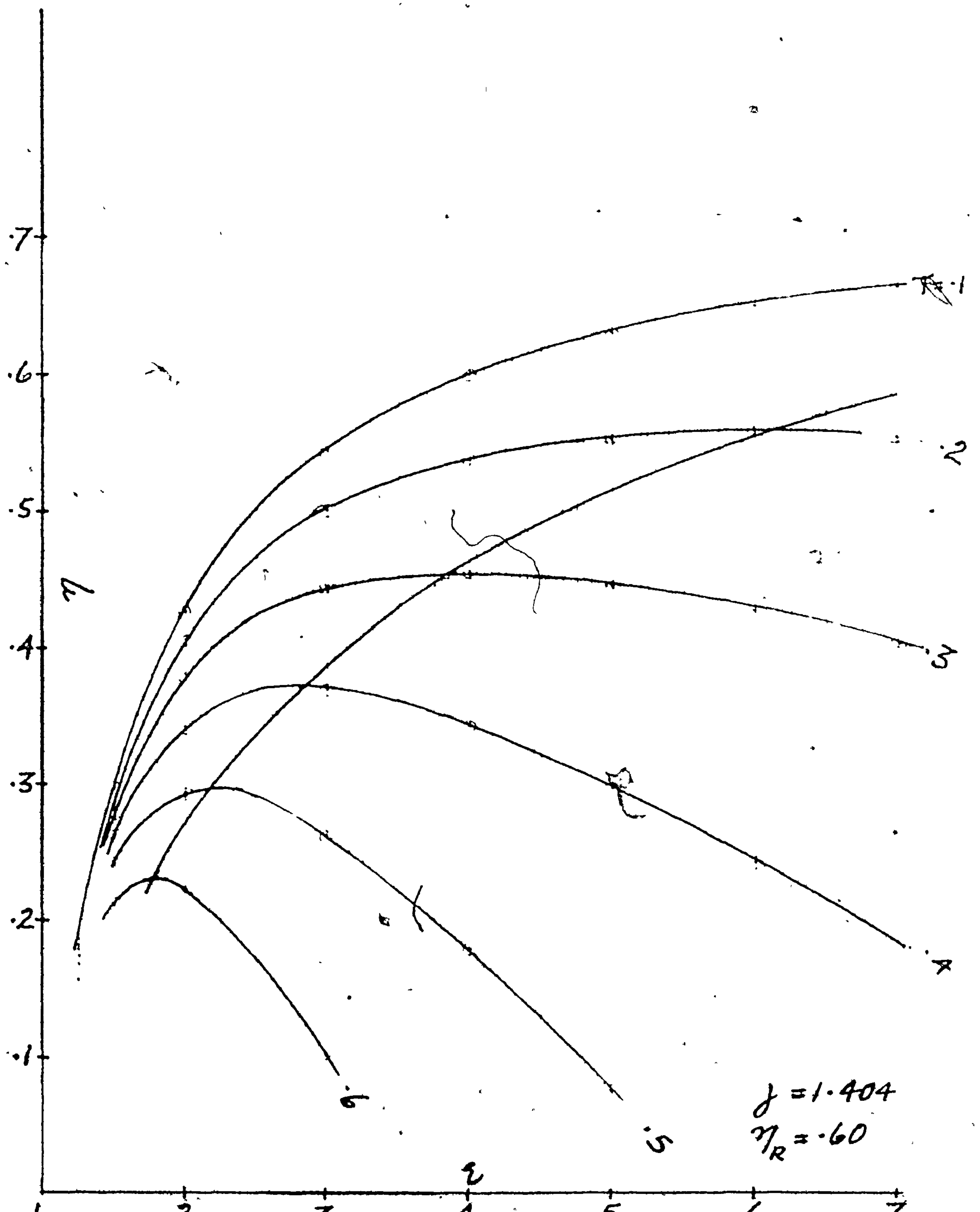

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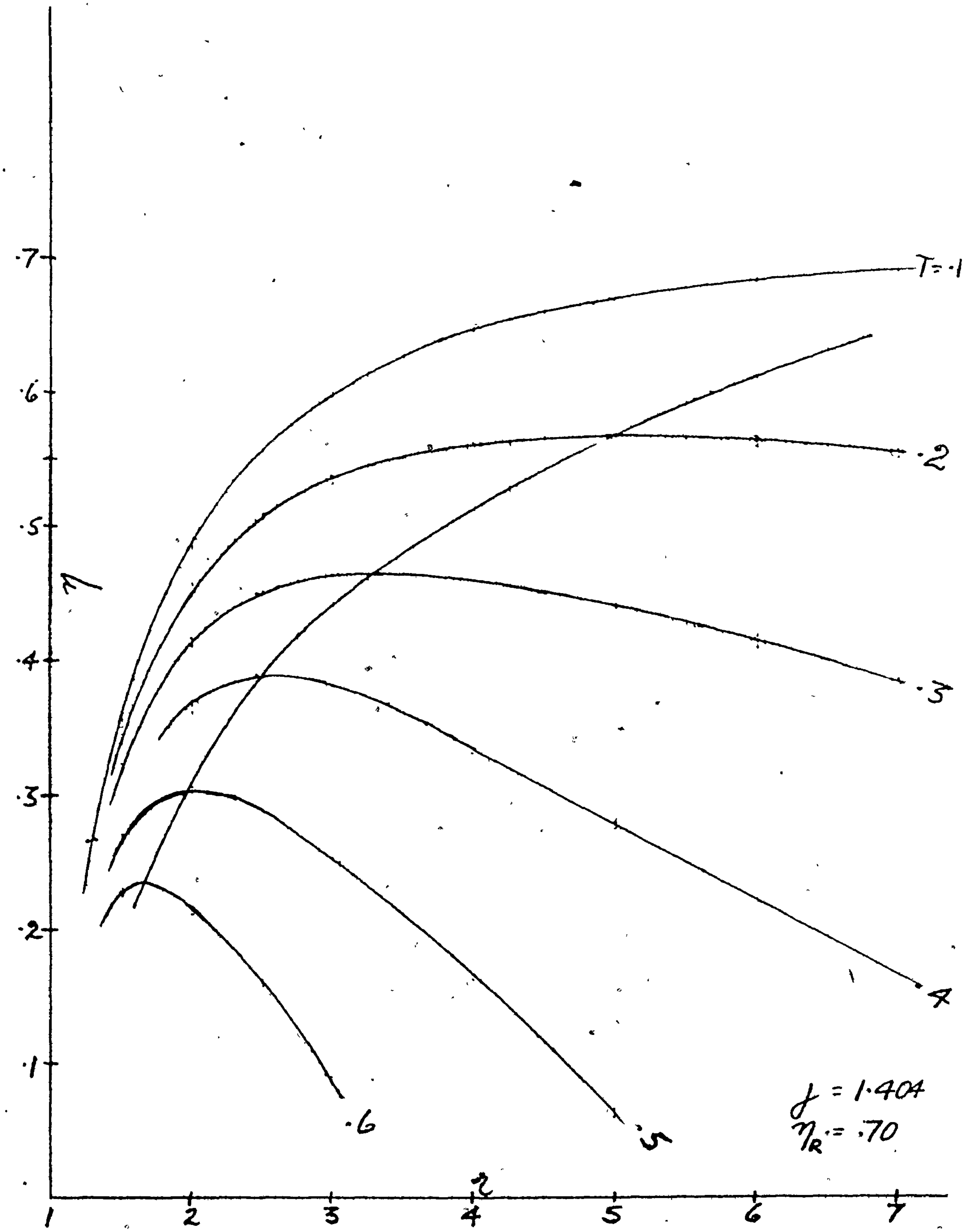

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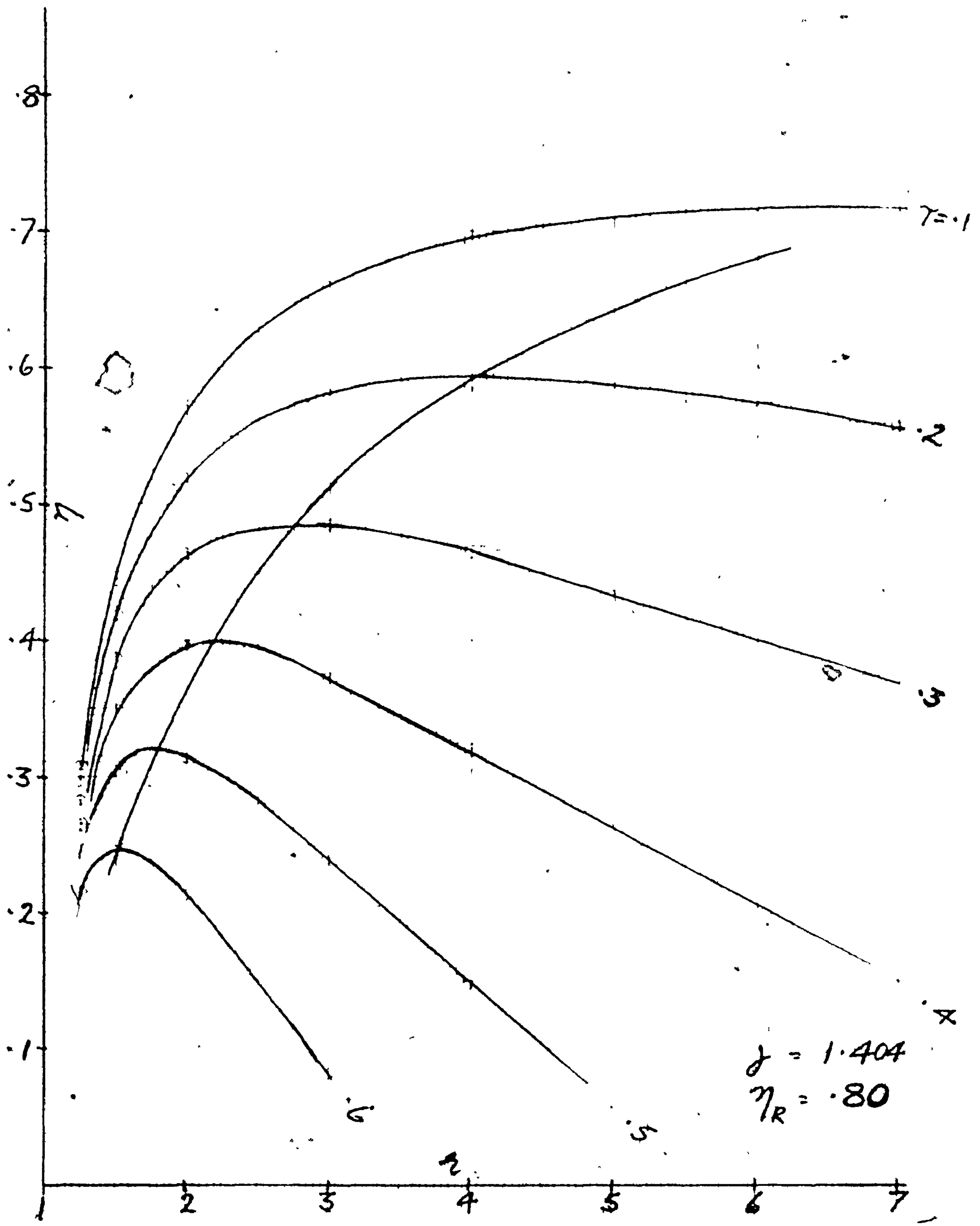

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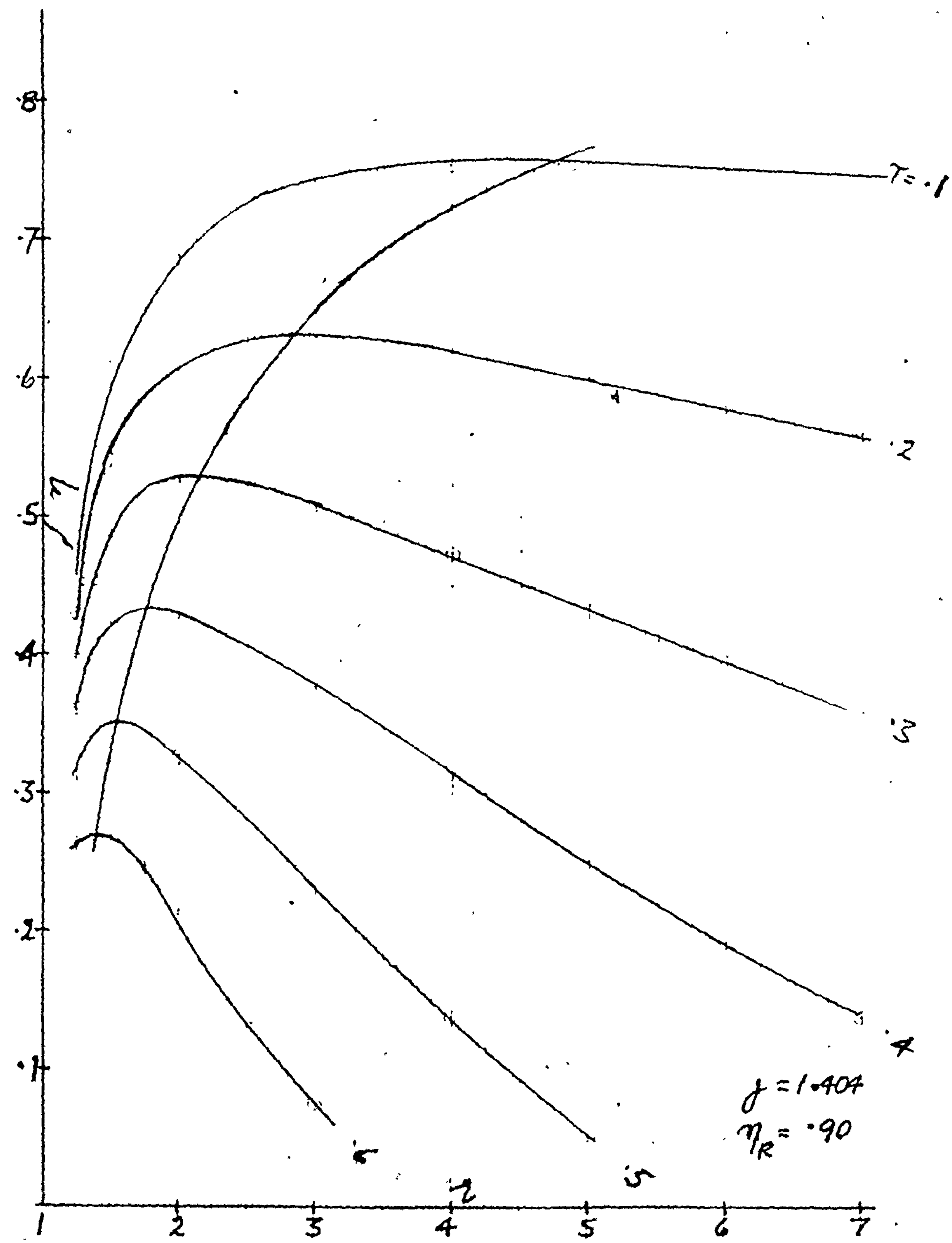

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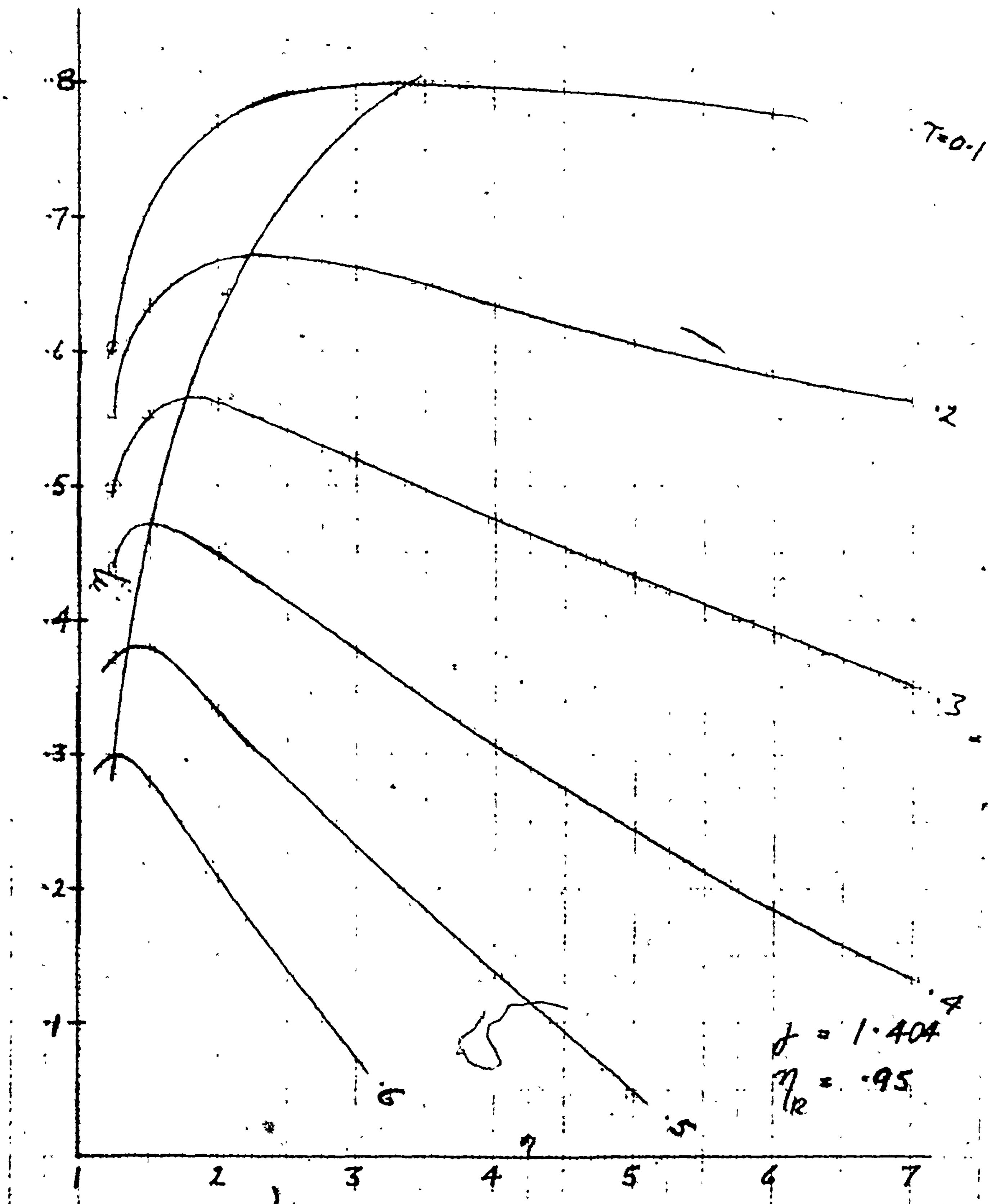
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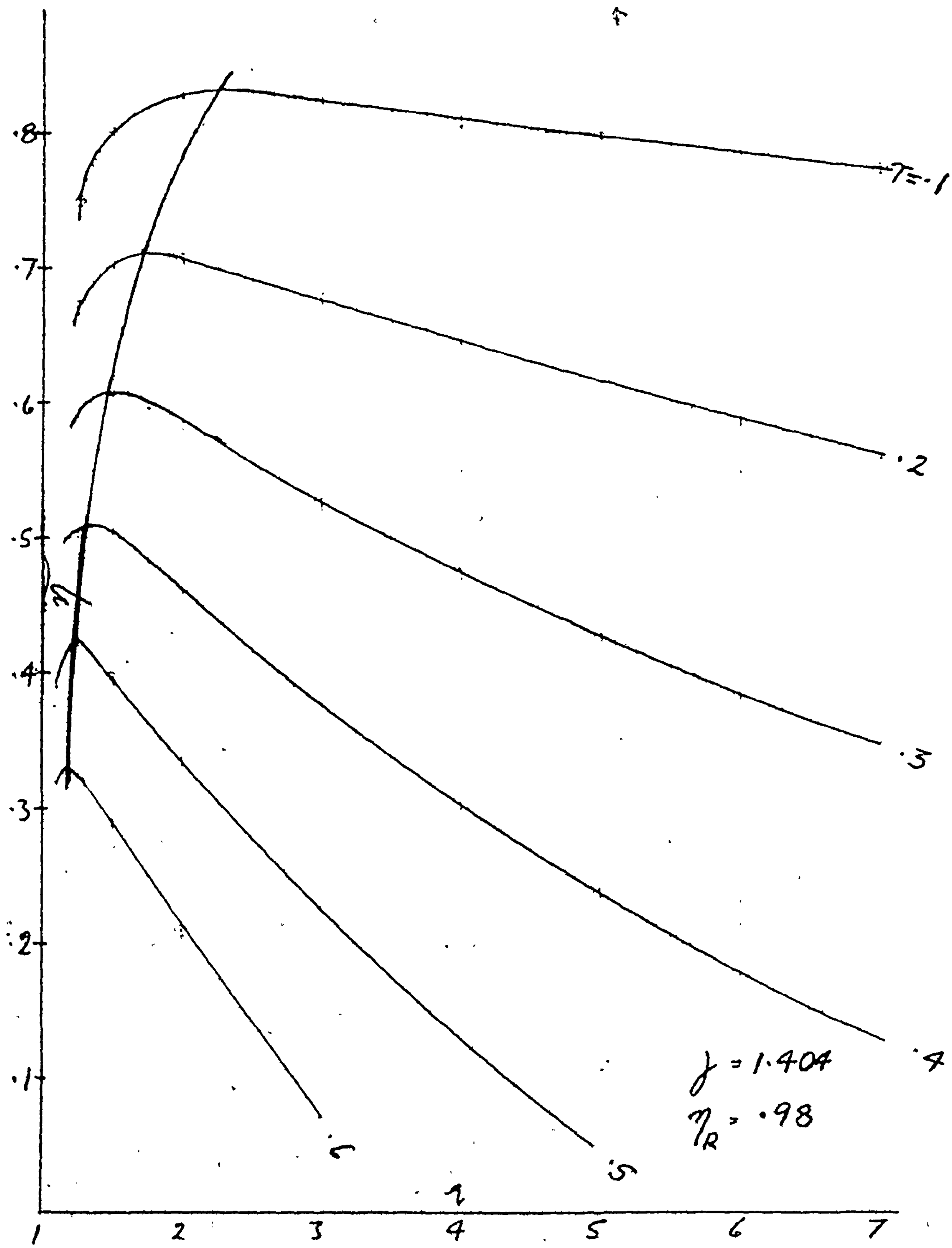

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