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16. Abstract

A historical review of Stirling engine development in the United States is given for the period 1960 to 1970. Most of the Stirling engine hardware activity during this period resulted from General Motors developmental efforts on an Army Ground Power Unit (GPU). Discussions of the problems encountered and the solutions attempted during this developmental period are given. A compilation of available engine and component test experience is given which lists types of tests, durations, conditions, and failures. A discussion of the technical problems remaining to be solved for Stirling engine automotive application is also given as well as a list of potential suppliers and/or developers for critical engine components and materials.

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# FOREWORD

The contract effort reported herein was supported by the Alternative Automotive Power Systems Division of the Environmental Protection Agency through EPA contract number 4-E8-00595. The Alternative Automotive Power Systems Division of EPA has since become a part of the Energy Research and Development Administration, and the program management responsibilities for the Stirling engine effort now lie within the Division of Transportation Energy Conservation of ERDA. ERDA has assigned project management responsibility for the Stirling engine effort to the NASA Lewis Research Center in Cleveland, Ohio. As a part of that responsibility, NASA-Lewis is publishing this report for ERDA.



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## INTRODUCTION

This report is submitted in fulfillment of EPA contract number 4-E8-00595, calling for a historical review of Stirling engine development in the United States from 1960 through 1970. The author was in charge of the Stirling engine project at General Motors Research Laboratories, Warren, Michigan, as an assistant department head from project inception in 1958 through 1967, and as a department head in 1968 until project termination in 1970. The author was also acquainted with the Stirling work at the Cleveland Diesel Engine Division (1958 - 1962), Electro-Motive Division (1962-1968) and Allison Division (1958 - 1964). There was no significant Stirling hardware activity outside of General Motors in U.S., until after 1970; this report will, therefore, concentrate on the results of the GM program exclusively.

The contract calls for the following information:

1. Discussion of problem areas encountered and solutions attempted during this development period.

2. Compilation of available engine and component test experience listing types of tests, duration, conditions and failures.

3. Discussion of technical problems remaining to be solved for Stirling engine automotive application and potential solutions to these problems.

4. List potential suppliers and/or developers for critical engine components and materials.

# I THE STIRLING ENGINE

Like all heat engines, the Stirling produces power by compressing the working substance at low temperature and expanding it at high temperature. During each cycle, heat is supplied at high temperature and rejected at low temperature. The difference between these two heat quantities represents the work of the cycle. The Stirling is different from the familiar gasoline or diesel engine in two respects; first, it uses the same working fluid over and over again, and second, the heating and cooling takes place by heat transfer through metal walls. In other words, the Stirling operates on a closed cycle and combustion takes place externally, as in the Rankine cycle powerplant. The Stirling differs from the Rankine in that the working fluid does not change phase; it is always a gas and is usually hydrogen or helium. All demonstrated Stirling engines are reciprocating piston devices which operate without valve gear. Two pistons per cylinder are required for single cylinder engines; engines of 3 to 8 cylinders may employ one double-acting piston per cylinder.

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

STIRLING ENGINE NOMENCLATURE

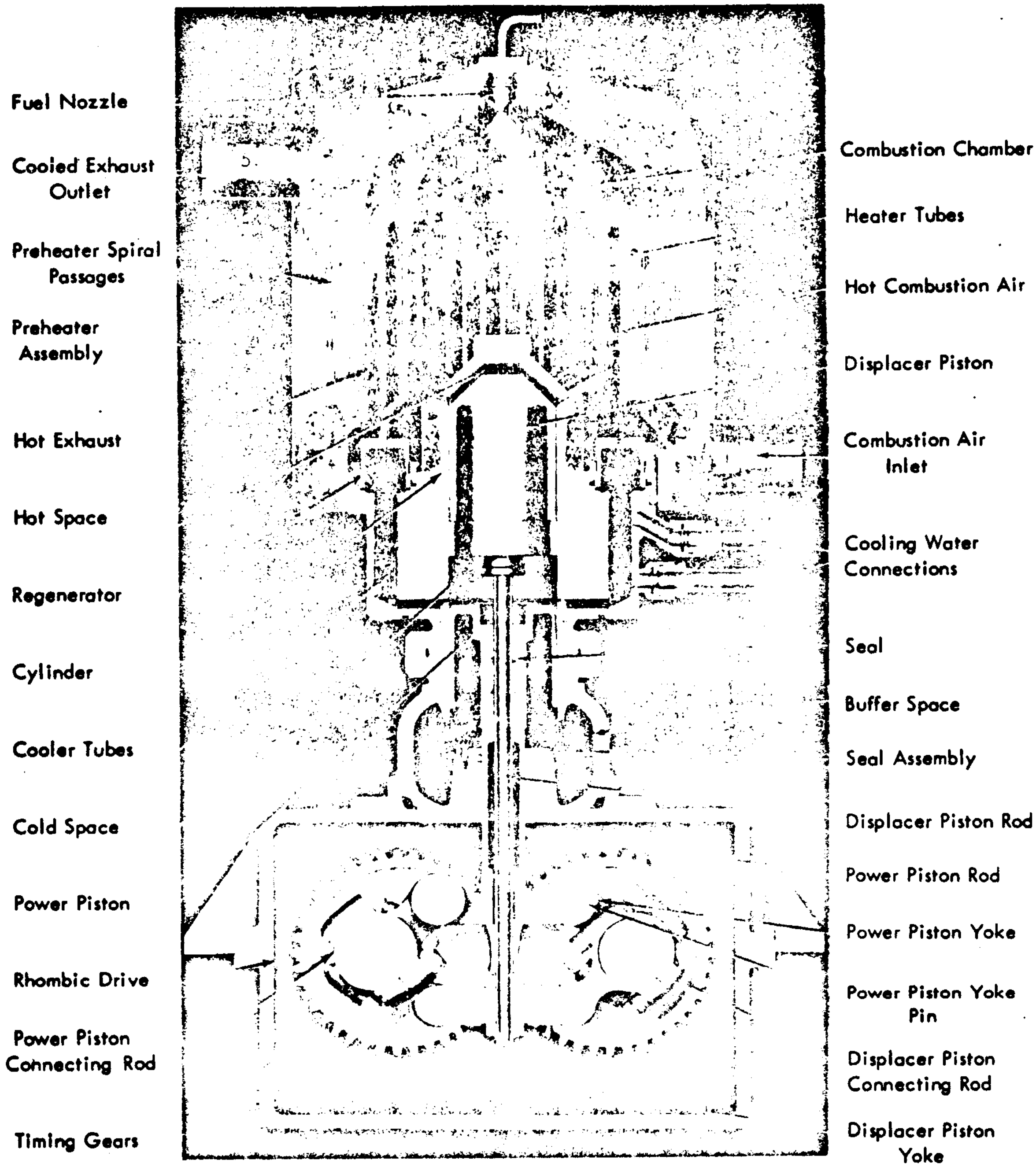


**Figure I**

## II GENERAL MOTORS AND THE STIRLING ENGINE -- BACKGROUND

Interest in the Stirling cycle at General Motors Research Laboratories* was initiated by several published papers on "air engines" in the Philips Technical Review in 1946 and 1947. As a result of a simplified analysis (Zeuner--Technical Thermodynamics--1905) an internal GMR report, written by the author in May 1948, concluded that the efficiency and specific power claimed by Philips were possible. An historical review of air engines was made, and a speech on air engines was delivered to GMR management in 1950 during which a large Rider pumping engine, borrowed from the collection of the Henry Ford Museum, operated on the auditorium stage. A smaller Henrici engine was also loaned to GMR and an unsuccessful attempt was made to install a regenerator and larger heater.

A General Motors representative in Europe was requested by GMR management in 1950 to investigate the possibilities of a working agreement between GM and N.V. Philips, Eindhoven, The Netherlands. The contact was made but Philips believed the time was not right and turned down the offer. GM's main interest in the Stirling engine in the late 40's was for marine applications, particularly for submarine propulsion. The Cleveland Diesel Engine Division had supplied the majority of submarine engines during World War II.

In 1948, the Philips' Stirling engine was still operating on air as the working fluid with an overall brake thermal efficiency of about 15 percent. By 1952 work on the engines at Philips was nearly stopped. Gradually, a new group of engineers and physicists interested in cryogenics began to replace the old group. They concentrated on achieving a low temperature device instead of an engine. Hydrogen replaced air to reduce pumping losses, improved heat exchangers and regenerators were constructed and a more accurate cycle analysis was developed. Finally, a new drive mechanism with twin crankshafts called the rhombic drive, solved the balancing and phasing problem. By 1954 the new group succeeded in making liquid air with the Stirling cycle machine. With encouragement from Philips' management, they made a new start on the more difficult task -- development of a commercial Stirling hot gas engine.

In 1957 GM learned that Philips was interested in a licensing agreement and in October, Mr. A.F. Underwood visited Philips Laboratory and was given some technical data. These data showed Philips had

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* Mr. Arthur F. Underwood, head of Department ME-5 in 1948, was the person responsible for starting the Stirling project in GM, both in 1948 and in 1957. Mr. Underwood became Manager of the Laboratories in 1958 and retired in 1969.

achieved a brake thermal efficiency of over 30 per cent, while bulk and weight were approaching the diesel engine. On November 20, 1958, a licensing agreement was signed between General Motors Corporation and the two Dutch affiliates, N.V. Philips Gloeilampenfabrieken of Eindhoven and North American Philips of New York. The agreements provided for a ten-year information exchange and had provision for mutual licensing of patents related to Stirling engines. General Motors was to pay N.V. Philips a total of $850,000, in annual installments, running until 1974, and nominal minimum royalties annually until 1968. General Motors was to pay North American Philips minimum annual royalties on a schedule of increasing amounts which would total $360,000 after the last payment in 1968.

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

In 1958 GMR received considerable encouragement from the U.S. Army Engineer Laboratories at Ft. Belvoir, Virginia, to develop a Stirling outboard motor as well as small generator sets which would be nearly silent. Presumably they would have been purchased in large quantities to replace existing internal combustion power units. The near certainty of an Army hardware contract* for a Stirling powerplant was one of the major considerations which influenced GM corporate officials to sign the GM-Philips agreement in 1958.

For commercial applications of the Stirling engine for marine propulsion, locomotives and generator sets, <u>bulk</u> and <u>weight</u> were not of primary importance. Also, the level of engine efficiency attained by Philips in 1958 was high enough to permit GM to put their efforts on other immediate problems. Consequently, during the first 5 years of R & D activity, GM Research concentrated in these areas: sealing of the pistons, sealing of the piston rods to prevent loss of gas as well as ingress of lubricant, reduction of gear noise, improvement

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* It should be pointed out that all contracts with the Army at Ft. Belvoir were strictly hardware type -- not for R. and D. This was because the GM-Philips contract prohibited sharing of "know-how" and design details outside of GM. This restriction, however, made it difficult to obtain further support from other branches of the Armed Forces and government agencies.

in combustion and burner-nozzle designs, durability of the preheater, engine speed governing, reduction of regenerator cost, endurance testing, and refinement of the cycle analysis. In addition, numerous studies and demonstrations were made of thermal energy storage systems combined with the Stirling engine in an effort to interest the U.S. Navy. In the last 5 years GM concentrated more effort in other areas, including: cooler and heater heat transfer, rolling seal quality control, lower cost preheaters, swash plate drive bearing studies, reduction of engine volume, stress analysis of heater cylinders, vehicle applications and exhaust emissions, governor refinements, controls reliability and reduction of friction.

## III PROBLEM AREAS AND SOLUTIONS

This report will cover major problem areas as well as some lesser ones, the methods used to solve them, and the results. Engine and component test experience at GMR, Allison and EMD will be included in the discussion of the problem areas where appropriate. A general summary of engine test data and projected performance of new designs will be reported also. Throughout the report a recurring discussion of the Army Ground Power Units will be noted, since this project constituted the major effort at GMR from 1960 through 1966.

The following topics or problem areas were arranged in an approximate chronological order, i.e. each succeeding topic (project) was initiated at a later time period. For example, "Piston Sealing" was started in 1959 while the "Four Cylinder Coach Engine" was started in 1968. Projects starting at about the same time were placed in an arbitrary order.

## PROBLEM AREAS & SOLUTIONS
## PISTON SEALING

Piston sealing was early recognized by Philips as a severe problem which required a unique solution, different from conventional internal combustion engine practice; it was more analogous to piston sealing in commercial air compressors which demanded sealing with minimum lubrication. Most air compressors used carbon multi-piece rings having tolerably high wear rates. It was soon discovered that these wear rates in a closed cycle engine would result in unacceptable contamination of the heat exchangers. Before 1952 Philips' various experimental air engines used from 3 to 5 rather conventional iron piston rings on the power pistons, while the displacer pistons required only a close clearance wear band. Leakage of oil past the piston rod seals provided enough lubrication to run for test purposes. The U.S. Navy tested a 1/4 hp Philips air engine for 1015 hours in 1949.* It is the author's recollection that the original set of rings were still functioning at the end; but the heat exchangers required several cleanings during the test because of oil contamination.

From about 1955 to 1961 Philips developed the so-called "close clearance" seals on both the power piston and the displacer. These consisted of a tin-lead alloy band with circumferential grooves and treated with moly disulphide. In some cases the $MoS_2$ was dispersed within the alloy itself. The piston band was machined slightly oversize, then shrunk in a dry ice bath for initial fitting in the cylinder. The piston was then "honed" into the cylinder liner by "motoring" the crank-piston assembly for several hours. The procedure proved to be more of an art than a science and as a consequence was not seriously considered by GMR after 1960. When done acceptably, the resulting engine demonstrated slightly higher mechanical efficiency than for conventional piston rings and with good sealing. It is believed that the system might have been acceptable for marine engines (cold water) but the results generally proved to be too erratic and the procedure too time consuming. When it was necessary to remove a piston, the procedure usually had to be repeated.

The first engine run at GM Research was a Model 30-15 single cylinder Dutch engine capable of about 30 hp at 1500 rpm. It was equipped with close clearance seals. The engine was operated for less

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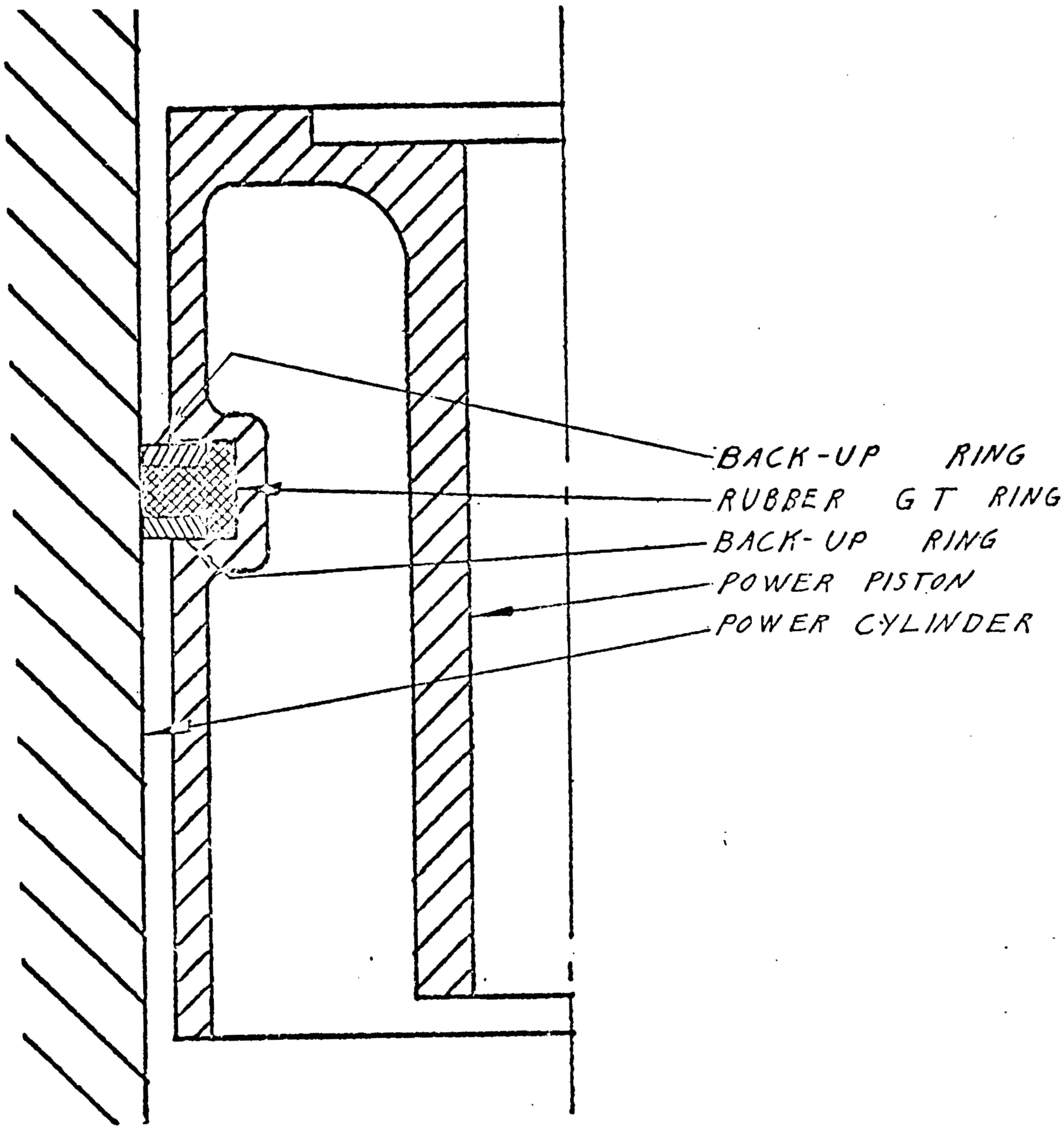
*U.S. Naval Exp. Station Rpt. # NS-623-305, 1 Feb., 1951 than sixty hours in 1960 when the displacer piston internal heat shields failed. The resultant overheating damaged the clearance seals on both pistons. At that point, a commercial Palmetto GT* elastomer ring was installed on the piston. The material was Buna-N. The ring cross section is "T" shaped with the top of the "T" forming the inside diameter, as seen in the accompanying sketch. Also shown are phenolic anti-rolling shields, or back-up rings, on either side of the section which seals against the cylinder wall.

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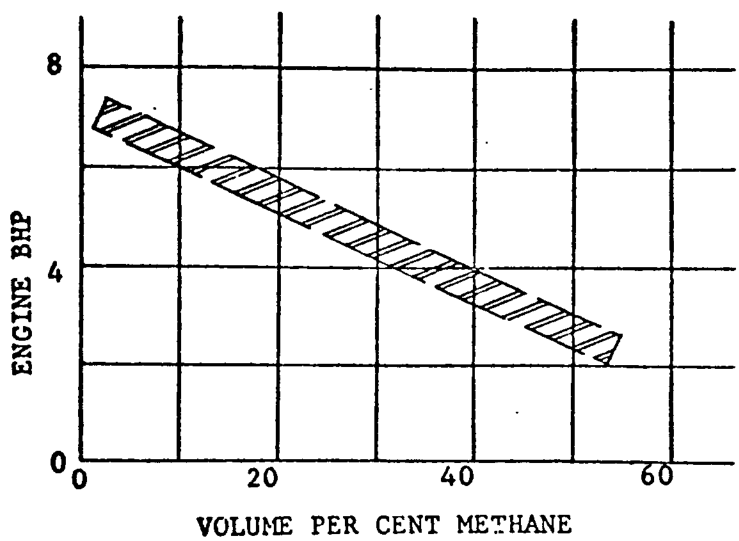
* GT means Greene Tweed & Company, North Wales, Pa.

The GT rings had already been successfully tested as substitutes for conventional "O" rings on the piston rods of the smaller 10 hp engine running at GM Research.

These same rings were used on all the GMR engines as piston seals until 1963. They offered the best compromise in sealing ability and wear resistance. They required minimum lubrication which was supplied by leakage past the piston rods. These rings enabled GMR to get on with other jobs and particularly the development of the first Army Ground Power Unit from 1960-1962. On the other hand, the friction appeared to be higher than the best clearance seal; also in some cases excessive lubricant gradually became thermally cracked resulting in generation of methane, which reduced power drastically. The effect was much more significant than ever had been noted from plugging of heat exchangers by solid residue. Proof of this relationship, shown on the graph, was obtained from a series of tests in cooperation with the Chemistry Department. Samples of gas were withdrawn as the engine output deteriorated. Correlation was shown between observed power losses, calculated gas losses, and the degree of contamination revealed by Chemistry's chromatographic analyses.




It should be explained that the reason oil leakage did not produce more problems than indicated was because all dynamometer engines were running with what was called an "open cycle" hydrogen system, i.e., filling from bottled hydrogen and dumping to atmosphere, which tended to purge the engine of methane contamination. None of the engines had complete governing systems nor hydrogen compressors which would have permitted recovery of dumped gas --- a truly "closed" system.

In 1962 GMR was becoming dissatisfied with the GT rings while Philips was still trying to improve the close clearance seals at Eindhoven. The Cleveland Diesel experimental engine, built in Holland, was also running with close clearance seals. At the Allison Division the goal was to develop a long life engine for satellite power. Convinced that neither sealing method would give sufficient life, they tested various commercial and experimental filled-Teflon materials for piston rings.

In 1962 GM Research made a study of the effect of leakage on power loss, and some limits on piston-cylinder radial clearance. Curves of leakage rate vs. equivalent radial clearance were produced which correlated data accumulated by Philips, GM Research and Allison, over a three year period, on elastomer rings, close clearance seals, metal piston rings and numerous Teflon formulations. The leakage data were obtained by unidirectional pressure gradient measurements in test fixtures or in engines used as test fixtures. It was discovered that the three research groups had been studying piston sealing systems with leakages ranging over seven orders of magnitude. The survey indicated the one order of magnitude which was critical for acceptable engine performance and thus enabled the investigators to concentrate on sealing systems in this range. An equivalent radial clearance of less than about 400 microinches was found essential for acceptable engine performance. The GT ring leakage was equivalent to less than a 10 microinch clearance. This was better than needed; but it then became the standard for comparing all other sealing methods.

Early in 1963 serious testing of new ring materials got under way at GM Research. These were essentially screening tests and included Buna-N running on polyurethane coated cylinder walls, fiberglass filled Teflon, and Rulon which is a proprietary Teflon-based substance,* in addition to several combinations of these with various clearances and configurations. Nine of ten seals meriting consideration were then tested in dynamometer engines. Typical test results are shown in the curves which compare four types against the base line GT ring of Buna-N. At that time the Rulon clearance seal had operated 112 hours with no loss in

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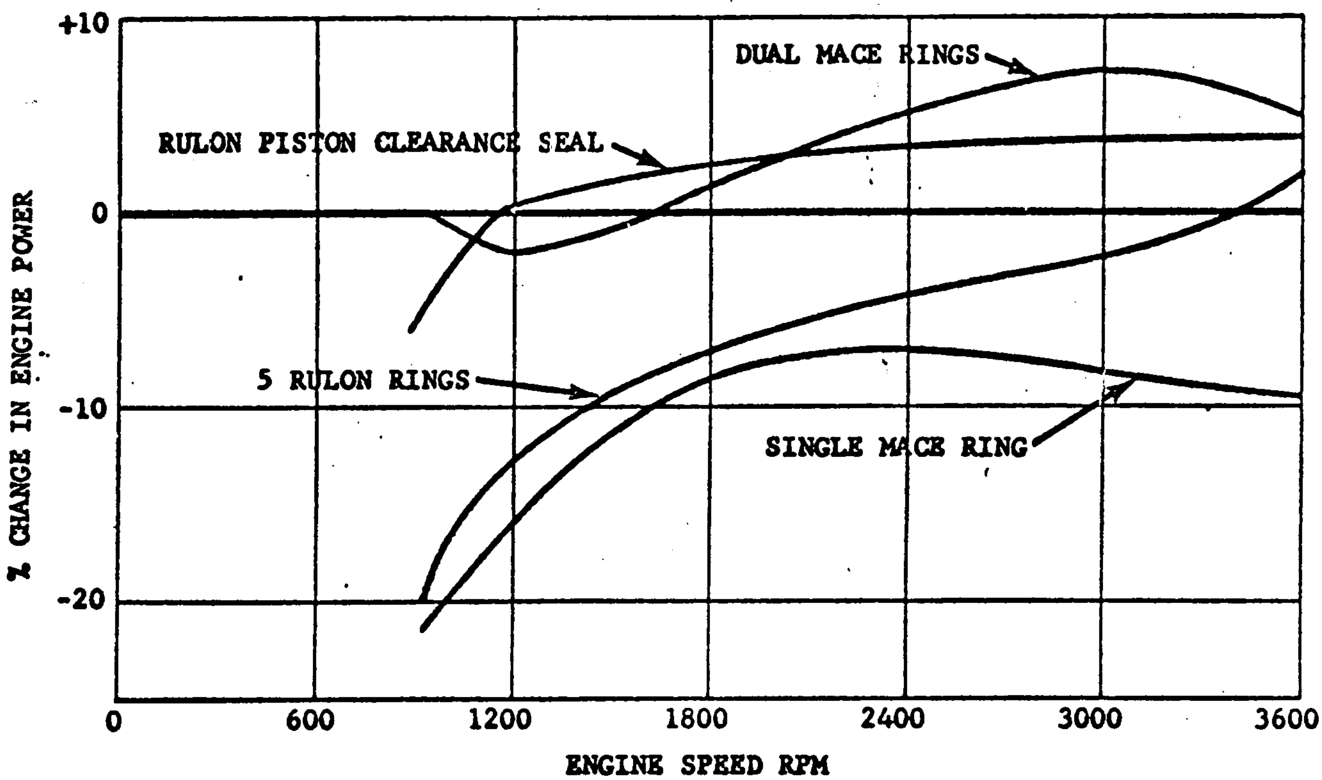
* All Rulon referred to in this report is Rulon LD, a fluorocarbon resin sold by Dixon Corporation of Bristol, Rhode Island.

performance. The one factor which tended to cloud the picture at that period was the slight oil leakage. Until the new materials and ring designs could be tested in a totally oil free atmosphere, there was always some question about the validity of the comparison. It was the great hope in 1963 that the new rolling or "sock" seals for the piston rods would soon be ready for engine installation. These were

PISTON SEALS

BASE LINE PERFORMANCE IS WITH A BUNA-N GT RING RUNNING ON A CAST IRON CYLINDER



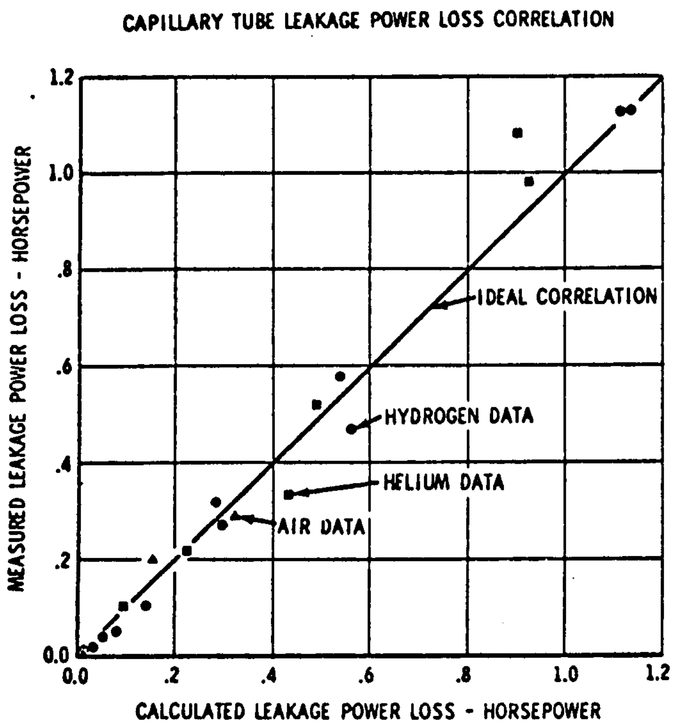
"absolute" seals which would permit realistic piston ring evaluation. Philips was optimistic that the rolling seal could soon be developed for the piston seal itself, but this was never realized by 1970. Detailed discussion of rod seals is found in the next section.

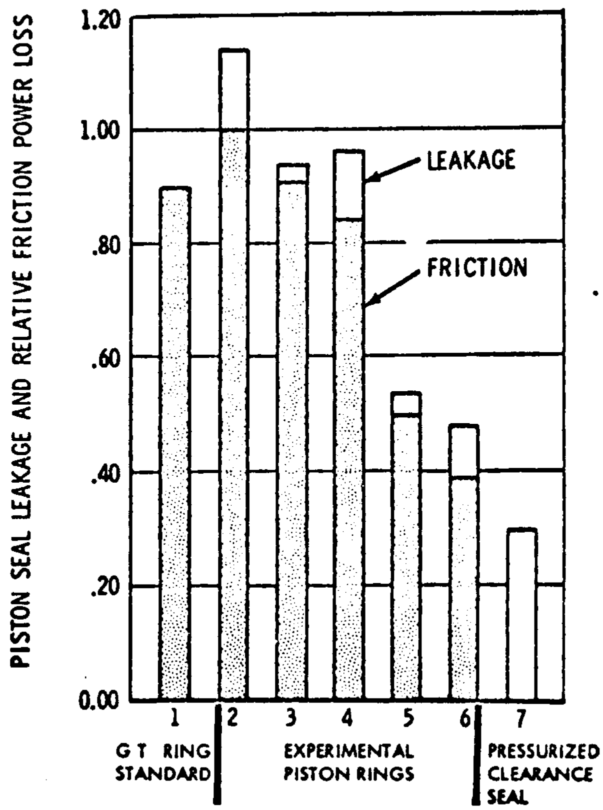
In 1964 the Army Ground Power Unit designated GPU 2-1 completed its 500 hour test program at ERDL, Ft. Belvoir. Design studies were also completed for the new GPU 3, which was delivered in 1966. The target objective for the GPU packages was 500 hours operation between major overhauls. The only engine component which prevented this attainment was the GT piston seal which had to be replaced at 250 hour intervals on the GPU 2-1. This result forced even stronger consideration of the Rulon type rings for all future engines.

The 9 rings which had been engine tested in 1963 (reported above) were re-tested in a special seal test fixture in 1964, which permitted a more basic study of all sealing effects. It was noted that fixture power differences were the same as in the engine, showing that the fixture was a suitable analog of an engine for developing seals. Three fundamental properties of Stirling engine piston seals were studied: Power loss due to leakage, power loss due to friction and "pumping characteristic." The latter is the tendency of a seal to leak more in one direction than another which causes a higher mean pressure in either the working space or buffer space. (See pg.3 Figure I) This phenomenon occurs in Stirling engines because the direction of the pressure gradient across the piston reverses during the cycle, similar to a double acting steam engine piston.

The seal test fixture was motored by a dynamometer which permitted measurement of the total seal friction power, the seal leakage power, and the test fixture friction losses. In order to separate the seal power losses, an analytical investigation was made of the relation between seal leakage and pumping power loss associated with that leakage. Then a method was developed to measure seal leakage as it was actually operating in the test fixture or an engine. The method was checked by inserting known leaks in a piston sealed with a GT ring (zero leakage) and then comparing the calculated and observed power differences from the fixture dynamometer. The power loss changes correlated within about 10% as can be seen in the graph.

A manual was then written containing step-by-step procedures and the necessary calculation charts for application to any Stirling engine, as an aid to other GM Divisions. The differences in total power loss could now be separated into an absolute leakage power and a quantitative difference in friction power between seals. Testing of the seals previously screened in the engine showed the greatest potential gain in performance would be realized by reductions in seal friction.




The bar chart compares relative friction and leakage power losses for 6 experimental seals against a standard GT ring. The GT ring has zero leakage loss while seal number 7 has essentially zero friction (piston does not touch cylinder wall) to provide a base line for the relative friction rating of the other seals. Seal number 6 was a balanced pressure, Rulon piston ring. Its total power consumption was 0.42 horsepower less than that of the GT ring, and it had run 250 hours without deterioration in the test fixture.

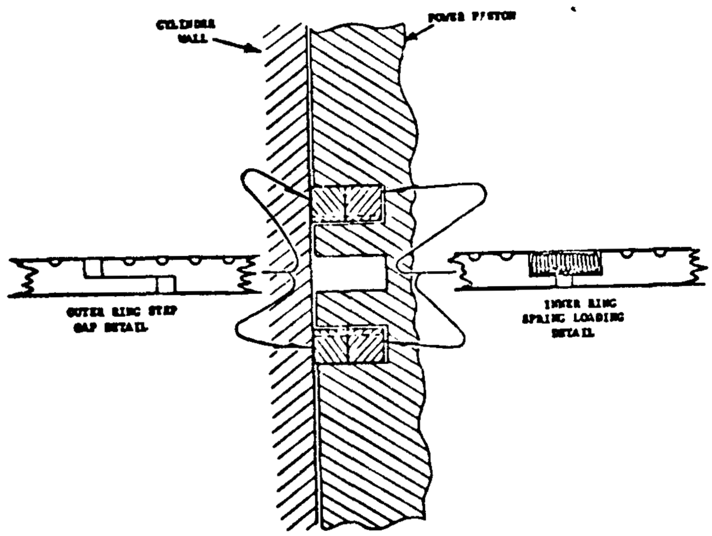
In 1965 further intensive development of the Rulon type rings took place. Seven ring configurations were tested in a special test rig for a total of 5607 hours during the year. For the first time an evaluation of wear rates could be made. The best rings averaged about .002 inches per 1000 hours and no sign of engine degradation due to wear particles was noted. While low ring leakage was vital, even more important was stabilization and control of leakage rate to maintain a nearly constant ratio of mean pressure above and below the power piston, i.e. the working space and buffer space.

The table shows 7 rings tested in 1965 and some of the results. Number 9, the "inward opposed pumping rings" showed great promise, with 8 rings having been tested to insure a life of at least 500 hours -- the goal of the GPU 3. A sketch of this sealing arrangement is shown. It consisted of two Rulon rings designed to leak toward each other at a much higher rate than they leak away from each other.

| SEAL TYPE NUMBER AND DESCRIPTION | TEST CONDITIONS AND DATA | | STATUS AT END OF 1965 |
|---|---|---|---|
| | a. Wear Rate*<br>b. Cylinder Material<br>c. Test Speed-rpm | a. No. of Samples<br>b. Total Test Hrs.<br>c. Longest single test-hrs.<br>d. Power Loss, HP** | |
| 5 - Two outward opposed, pressure-balanced, split rings with back side O-ring | a. .0112<br>b. Chromium<br>c. 3600 | a. 7<br>b. 501<br>c. 286<br>d. .35 less | Testing stopped because needed external pressure control valve |
| 6 - Two inward opposed, pressure-balanced, split rings with backside O-ring. Middle damper volume | a. .0071<br>b. Chromium<br>c. 3600 | a. 2<br>b. 446<br>c. 286<br>d. .20 more | Testing stopped because of pressure instability |
| 7 - Two inward opposed, pressure-balanced, split rings without backside O-ring. Middle damper volume. | a. .0048<br>b. Chromium<br>c. 3600 | a. 3<br>b. 562<br>c. 478<br>d. Equal | Testing stopped because of pressure instability |
| 8 - Two inward opposed pumping rings with continuous metal back-up rings | | a. 2<br>b. 5<br>c. 3 | Testing stopped because of temperature instability |
| 9 - Two inward opposed pumping rings with spring-loaded, split back-up ring. Middle damper volume | a. .0010<br>b. Meehanite cast iron<br>c. 3600 | a. 8<br>b. 3273<br>c. 587<br>d. Equal | Development continuing |
| 10 - Floating clearance seal | | a. 1<br>b. 73 | Development continuing |
| 11 - Two outward opposed, pumping rings with spring-loaded, split back-up ring. Eight .020" wide x .030" long grooves in cylinder wall. | a. .0008<br>b. Meehanite cast iron<br>c. 3000 | a. 2<br>b. 747<br>c. 557<br>d. .40 less | Development continuing |

* Average radial wear of ring in inches per 500 hours operation.

** Expressed with respect to power loss of Buna N piston seal which was standard seal at start of program.




A small volume is provided between the seals in the form of a groove in the piston. The seals act to maintain the pressure in this space equal to the maximum pressure of both the buffer space and working space -- these spaces are thus stabilized at the same peak cyclic pressure.

On the piston ring test machines, the following variables were investigated: Pumping ring seal width, pressure balancing, seal groove clearance, seal material, cylinder surface roughness, cylinder material, wear-in procedure, the effect of wear particles and the effect of oil. Chrome plate, pearlitic cast iron, Meehanite cast iron, and 310 stainless were tested as cylinder materials. Meehanite gave the lowest seal wear rates.

Control of oil in the engines as well as the piston ring test fixtures was still dependent on the GT type rod seal and scraper device. Marginal improvement in life and oil control had been realized since 1964.

In 1966 two of the opposed pumping rings had operated for nearly 1000 hours in engines and one reached 1670 hours on a ring test fixture. With the conclusion of the GPU 3 program in 1966, the piston ring test fixtures were converted from GT oil seals to rolling type piston rod seals so as to eliminate all oil leakage to the Rulon piston rings.

Further testing of cast iron cylinder liner material took place with 150 hour screening tests of 7 liner materials; unfortunately the results were inconclusive because of scatter of the data. The best combinations gave wear rates of about .001 inch per 1000 hours; the average wear rate was about .0015 per 1000 hours. Work continued on the floating clearance piston ring seal (seal #10 of pg14 ) This ring is loosly attached to the piston so it may center itself, regardless of piston eccentricity. Preliminary results from two tests of 380 and 851 hours indicated slightly lower friction power and an order of magnitude less wear. Unfortunately, its pumping characteristics were erratic; and because of other priorities, no further work was done at GMR on this seal design.

In 1967 GMR operated the GPU3-3, an improved copy of the package sold to the Army, for 1800 hours. Of 4 opposed pumping rings required for the engine, two were operating satisfactorily after 1800 hours and two others after 1434 hours. In addition, 4500 hours of ring fixture testing, under "dry" conditions (rolling seals on the piston rods), confirmed last year's data, namely an average wear rate of .0015 inches per 1000 hours.

In 1968 it became evident that future multi-cylinder Stirling engines might depart from the traditional two piston-displacer type toward the double-acting piston type with interconnected cylinders. This was because of the greater emphasis on reducing engine volume and weight, particularly for vehicle applications. The double acting engine piston operates under more severe temperature conditions than a displacer engine. In addition, the vehicle type engines require higher mean pressures than the GPU engines to be competitive with internal combustion engines.

In addition to environmental temperature, the two criteria to be considered for judging severity of piston ring operating conditions are (1) the PV factor, or mean product of rubbing velocity and pressure difference over the ring, and (2) the maximum pressure differential. The pressure differential tends to extrude the rings into the clearance space, while the PV factor influences the wear rate. In 1968 Philips constructed a special ring testing machine called the "Hot Compressor" to simulate temperature, pressure and velocity in a double acting engine. GM Research also started testing rings at much higher pressure. Extrusion became evident after only a few hundred hours of operation.

The following table presents piston ring operating conditions as seen by GMR. The 4 cylinder engine was the new model 4L23 double-acting, 4 cylinder, inline engine in the design stage at GMR in 1968. It was completed but never run before the program was canceled in February, 1970.

PISTON RING OPERATING CONDITIONS

| | PV (psi)(ft/min) | Pressure Difference (psi) |
|---|---|---|
| GPU-3 Engine | 115,000 | 650 |
| GMR Test Rig | 110,000 | 760 |
| Philips Hot Compressor | 156,000 | 1500 |
| Four-Cylinder Engine | 214,000 | 1235 |

A test program was started in March, 1968, and completed in January, 1969, to determine the maximum cold space temperature which the piston rod sealing system of the GPU3 engine could tolerate for a minimum of 500 hours between overhauls. This program accumulated 3634 hours of dynamometer operation on one engine and established the maximum cold space temperature of 275°F. Water temperature out was 160°F.*1

Piston ring wear based on 3583 hours of operation at the higher temperature ranged from .0015 to .0024 inches/1000 hours, which was only slightly greater than at the usual 125° water temperature.

The general conclusion from all testing through 1969 of Rulon balanced pressure type piston rings was a predicted life of 10,000 to 12,000 hours under running conditions encountered with the GPU series of engines -- namely, a pressure difference of about 750 psi, PV of about 110,000 and a maximum cold space temperature of 275°F.*2

Late in 1969, a piston ring test fixture was converted to allow testing at conditions approximating those for the new 4L23 engine. In 1711 hours of testing, the rings were operated at gradually increasing PV values until reaching 156,000 (same as Dutch tests -- see table above.) Temperatures were held at 240°F, the estimated value expected for the 4L23 ring area. Wear rate was 3 to 4 times that encountered under GPU conditions; rings might be expected to last about 3000 hours, which was considered adequate for initial engine operation. Tests continued until the end of the program but nothing significant was reported.

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*1 Coolers on the GPU engines were too short; with an optimized design, the space -to-water $\Delta T$ could be less than 50°, or a water temperature of about 225°F. This would reduce vehicle radiator size.

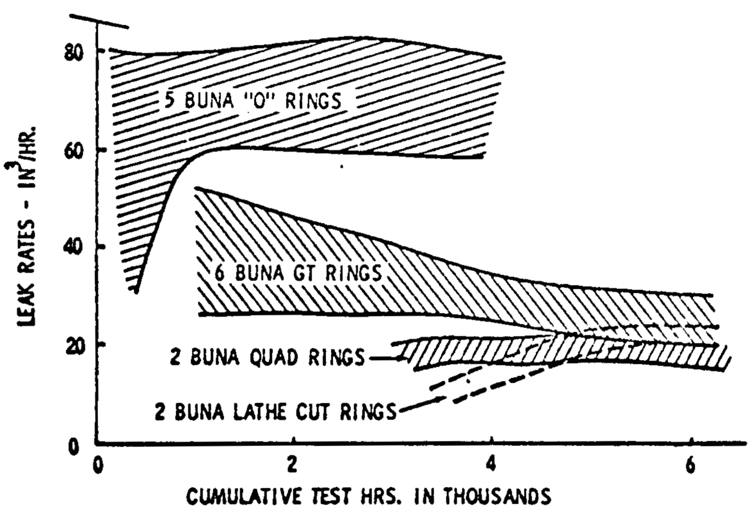
*2 One weakness of the ring program was uncovered in 1968. The local Rulon sales representative switched jobs to another company, also making a proprietary Teflon-filled material. He approached the GMR Purchasing Department, unknown to the Mechanical Development Department, and sold them on buying his product instead of Rulon. It was used for rings with the belief it was Rulon -- wear rates went up an order of magnitude! When the trouble was finally traced, Rulon was again purchased exclusively. Unfortunately, the <u>exact formulation</u> of either product was unknown to GMR at the end of the program.

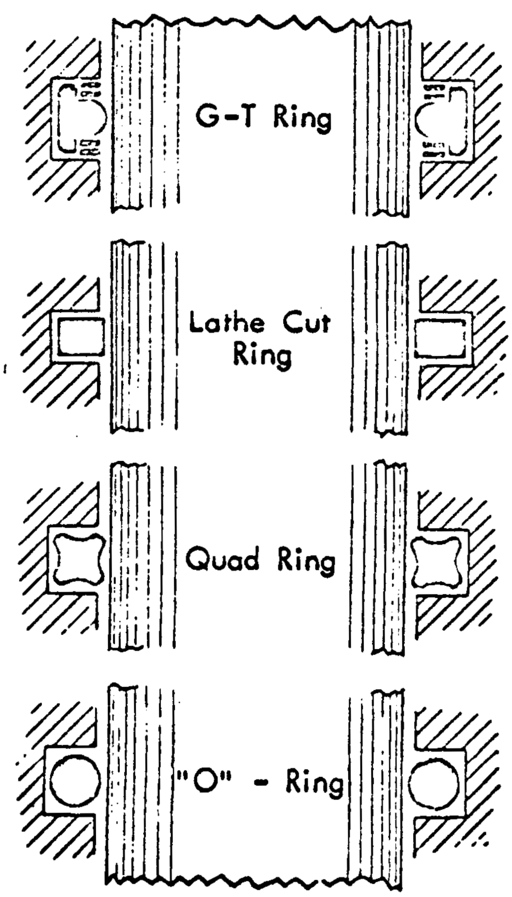
PISTON ROD SEALING

A. SLIDING SEALS

In the first year of the program, 1959, GM Research constructed 6 piston rod seal test fixtures. Each consisted of a crankcase and single crank assembly driven directly by 2 HP induction motor at 3590 rpm. The connecting rod drove a 1" diameter reciprocating shaft with a stroke of about 1 3/8 inch to simulate the piston rod of the small 8-10 HP engines being developed at GMR, Allison Division and at Philips. The shaft was sealed at two locations with pressure gas admitted between the seal assemblies. Leakage was measured out of the top seal assembly only, by means of water displacement.

In 1960 GMR tested 54 reciprocating shaft seals of 15 different kinds against helium at 1500 psi. One commercial seal, a GT ring (discussed in detail on page 8) proved superior to the original "O" ring seal. These were used by GMR, with slight modification, as seals for both the power piston rod and the smaller displacer rod throughout the entire engine program to 1970, except for the few engines equipped with the rolling or "sock" seals. Results of leak tests on four kinds of commercial seals are shown in the graph.



The one major change in sliding seal technology took place in 1963. The quality of the commercial Buna-N GT seals was so erratic that a request was made to the seal section of the department (specialists in lip seal technology, "O" rings, and elastomer compounding) to compression mold some copies of the GT rings, using their own Buna-N formulation (nitrile) of about 78 durometer. Molding was done so as to eliminate all flash, which was found detrimental in the commercial seals. Before this, the displacer rod seal life averaged 20 - 30 hours (higher temperature than piston rod seal); the "new" GT ring operated over 190 hours. Beginning in 1963 all GT type sliding rod seals were made by General Motors.

To assist in preventing oil from the crankcase passing through the GT ring seal, an oil "wiper" was installed just below the GT ring. It was a commercial sharp edge ring device made of polyurethane and known by the trade name of Disogrin. Each was selectively picked after examining the sharp edge under a magnifying glass. Numerous other wiper crossections had been tried with little success. The life of the Disogrin type was considered marginal because of temperature effects; however, by the end of the program they would operate successfully as long as the GT rings. No metal wiper material was ever used.

Another device also helped to prevent oil from passing above the buffer space region. It was a simple metal "sponge" made of Met-Net (discussed in regenerator section) about 1/8 inch thick and placed in the bottom of the buffer space to absorb oil. Whenever inspected it was always found to contain some oil; but no quantitative test of its effectiveness was ever made. Straight mineral oil was always used in the crankcase of the GPU engines. One experiment was made with a synthetic oil (Ucon) which resulted in much greater solid residue than ever noted with mineral oil.

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

By 1965 several rod seals had operated successfully over 400 hours and one seal was not leaking after 987 hours. They were so temperature sensitive, however, that a special water cooled buffer space was added to the GPU-3 to extend their life.

In 1966 two more seals achieved over 900 hours successfully. On the GPU2-1 the power piston rod seal was replaced at 581 hours while the displacer rod seal ran 646 hours without leaking. Little basic study of these seals was made after 1966.

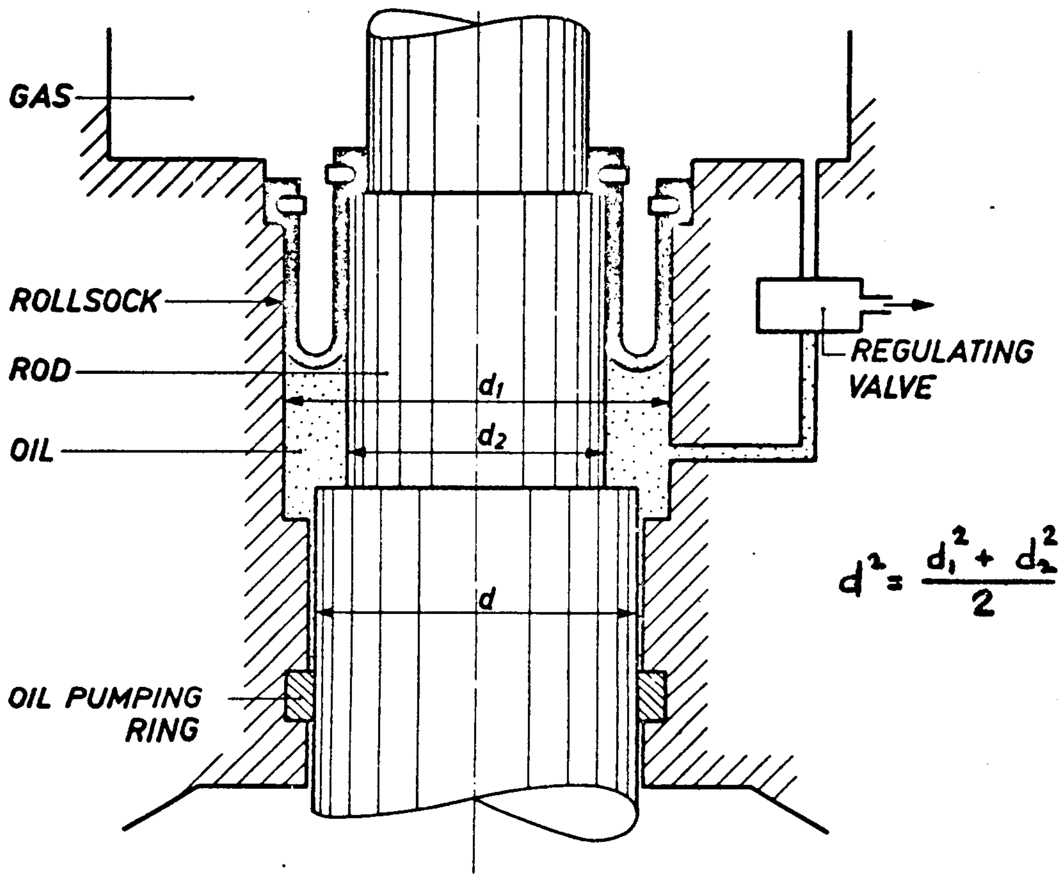
In 1967 the GPU3-3 durability test was completed at GMR. A power piston rod sliding seal failed after 861 hours in the seal mount, but not in the elastomer element. The displacer rod seal was removed for inspection after 1172 hours, at which time it was not leaking.

Leakage rate of hydrogen through the power piston rod seal was measured in the early 60's on the seal test machines. It amounted to about 60 cu in/hr (STP); the smaller diameter displacer seal was never measured but was assumed proportional to the diameter. Total leak rate for both seals was estimated at 100 cu in/hr. This was considered quite acceptable and would permit a 500 hour run with a storage reservoir of reasonable size. Later in the 1960's, leakage rate was measured for the GPU engine packages, which necessarily included the sum of seal leakage, plumbing leaks and hydrogen diffusion through the high temperature heater tubes. Seven tests made in 1969 on the GPU3-3 showed leakage rates ranging from 71 to 191 cu in/hr, for an overall average of 156 cu in/hr, including one stop. Static leakage was not usually included since it was the standard practice to shut off the main reservoir valve after stopping. The maximum gas lost then amounted to the hydrogen remaining in the engine at elevated pressure---about 45 cubic inches, or about 765 cubic inches at STP conditions. Therefore, each shutdown was equivalent to about 7 hours operation under dynamic conditions. The contributions of plumbing leaks and hydrogen diffusion were never determined. Using the equation and constants for hydrogen diffusion which Philips believed valid, yielded a value of about 150 cu in/hr for the GPU conditions---higher than the rod seals alone for some tests!

It was the conclusion that dynamic seal leakage of hydrogen was under "control", and would enable a commercial engine to be manufactured. Static leakage control might require a special expandible "O" ring device for clamping on the piston rod when it is stationary.

PISTON ROD SEALING
B. ROLLING SEALS

The rolling or "sock" seal was invented by Philips in 1960. Initial operation on test machines and some experimental cryogenerators was so encouraging that Philips expected it would finally solve the sealing problem on both the rods and pistons very soon. Unfortunately, the initial problem with quality control of the elastomer did not go away; also as temperature rose the life of the best seals dropped drastically. GM Research was first informed of the work in 1961 and began to convert some of their seal test machines in 1962 from sliding seals to rolling seals. A sketch of the seal is shown.




In 1963 GMR was studying polymer materials, EMD was studying mechanical design of the pumping devices and controls required, while Philips was returning to a more fundamental approach with analytical studies. According to their theory, failure was primarily due to creep rather than tensile load. Allison Division did not require seals, since their space engine incorporated a pressurized crankcase which enclosed twin alternators.

GM Research tested eleven polyurethane seals made by Philips for 2235 hours; the longest single life was 549 hours. Temperatures ranged from 90°F to 125°F on the oil side. GMR materials tested were of three general classes of oil resistant polymers; nitrile (Hycar and Buna-N), polyurethane (Vibrathane and Genthane), and polybutadiene (Adflex). Vibrathane was the most successful, one seal achieving 372 hours.

Work also began in 1963 on the hydrodynamic metal "pumping ring" which was used on all test machines to maintain the high oil pressure required to back up the rolling diaphragm. An oil pressure 50-90 psi less than the working gas pressure was believed to be optimum. Considerable time was spent developing automatic controls to maintain this differential pressure, particularly during start up and shut down.

By 1964 Philips had determined that seal life decreased one order of magnitude for every 45°F increase in operating temperature. Most testing had been done at around 100°F; GMR believed a life of at least 10,000 hours at 200°F would be required for commercial success--and this became the goal.

All six GMR sliding seal test machines had been converted to rolling seal testing by 1964. The Polymer Department at GMR was responsible for molding the seals and the Mechanical Development Department for conducting the tests. It soon became evident that because of quality control problems, the Polymer Department could not keep up with the demand. Furthermore, the first 6600 hours of rolling seal testing showed it would be impossible to assign any level of significance to seal material or design detail changes. Finally, a quality control program was started. A detailed written procedure covered seal manufacture and a program to produce and test 6 batches of duplicate seals was instituted. Molded seals were screened initially by Polymers and turned over to Mechanical Development for further inspection and static leak testing. Seals passing these steps were installed in the test machines and operated at controlled conditions and a temperature of 100°F., until they either failed or ran for 300 hours. A 300 hour seal was defined as a "good" seal. Results of this new program for 1964 were encouraging. Before this, only 7% of the seals which were believed to be good seals reached 300 hours. "Non-valid" failures were caused by machine breakdown, safety controls failure, or by seal installation errors.

Polymers Department submitted eleven seals for high temperature testing at 200°F. Eight were of a refined polyurethane, two of epichlorohydrin and one of Neoprene. Five seals operated less than one hour and the longest, of polyurethane, lasted 46 hours.

At the "normal" 100°F temperature level, a new thin-walled seal operated for 2376 hours; the longest life of any seal at the end of 1964 was 3596 hours. Total test machine time was 24,263 hours.

In 1965 test machine time was 43,163 hours, with about 100 seal tests conducted. Of the polyurethane tests at 100°F, 15 operated over 1000 hours. Average life of twelve that failed was 3074 hours and minimum life was 1386 hours. One ran 5849 hours and another 3589 hours. These compared favorably with Philips results as to total cycles. Philips had several tests showing 8000 - 13,000 hours life at 43% of GMR cylic rate.

High temperature seal material tested included Viton A, caroxylic nitrile, silicone, propylene oxide, Hypalon, Neoprene, and epi-chlorohydrin. Only propylene oxide and Viton A appeared promising. This also checked Philips conclusions. The oxide elastomer operated 524 hours at 130°F and 417 hours at 150°F; however, it was not considered for further testing because of lack of availability in the future. Viton A had proved very difficult to mold. One seal ran 13.5 hours at 180°F.

The quality control program begun in 1964 was completed in 1965. The last two batch tests achieved 100% to 300 hours, for an overall average of 81% of valid tests reaching 300. This was some progress; but the high temperature program was severly limited as to quality seals. Five new test machines were added. Little time was available to improve the metal "pumping rings" used on the test machines to pressurize the seals with back-up oil. They consisted of a simple sharp edged ring usually of white bearing metal, with spring "fingers" around the perimeter to hold the upper edge against the piston rod -- and consequently to allow oil on the rod to pass in one direction only, toward the rolling seal chamber. In 1965 a new aluminum ring was tested and was still operating after 1734 hours.

In 1966 performance of low temperature seals was nearly the same with 8 Vibrathane seals averaging 3033 hours' life; the minimum was 2116 hours and the maximum 4840 hours. Twenty-five seals operated over 1000 hours.

Results of high temperature testing showed 2 Viton seals operated 3408 and 2255 hours at 165°F, while another seal made by AC-Flint Division of GM was still running after 1930 hours at about 220°F. However, the test machine on which it was installed operated at 1750 rpm to more closely duplicate Philips testing of Viton seals. Nevertheless, the Viton results were dramatically better than in 1965. However, the quality of other seals being received from the Polymers Dept. had not improved since the previous year.

It was decided that instead of further searching for new materials, the search would concentrate on finding a commercial source of high quality seals made of polyurethane and Viton only. EMD also emphasized their need for larger diameter quality seals for their new test machine and 4 cylinder, 400 hp engine. It was estimated that at least 300 quality seals would be needed in the Corporate program each year, which was beyond the capacity of the Polymer group at GMR.

Outside sources contacted were E.F. Houghton Co., Chicago Rawhide Mfg. Co., Vernay Laboratories, and Diaphragm Industries. Within the Corporation, AC-Flint and the Mechanical Development Seal Group were asked to help. By the end of the year, Vernay Laboratories of Yellow Springs, Ohio was the only outside source seriously interested, and they actually delivered 8 seals made from a GMR compression mold. From their experience, they expressed the desire to transfer-mold the seals instead, in the future.

The AC-Flint Laboratory had consistantly been a high quality supplier of seals, having developed their own materials and molding techniques--unfortunately, it was a one man operation -- which illustrates the importance of the "art" of elastomer manufacturing, requiring understanding, diligence and concentrated interest.

The department seal group suggested use of liquid polymers instead of milled gums, to permit easier cleaning. They also suggested centrifugal molding and a simple mold was built. The casting of liquid polyurethane (both Adiprene and Vibrathane) was done by the Naugatuck Chemical Division of U.S. Rubber, after discussions with them. The mold spins at several thousand rpm and is surrounded by an induction heating coil to assure initial setting of the polymer. Final curing is done in an oven. Seven of their seals passed the 300 hour screening test by the end of 1966. While this appeared encouraging, by 1968 it was concluded that this process made the urethane more temperature sensitive, and therefore it was abandoned for the more conventional millable gums.

In March, 1967, all development work on rolling seals and elastomer materials was transferred to the Seal Group of the Mechanical Development Department. The Seal Group had been gradually building up sufficient man power and facilities, including a new polymer laboratory, to assume full responsibility and thus allow the Stirling Group to concentrate on other aspects of the program.

For the first time, a dynamometer engine was assembled with rolling seals on both piston rods and incorporating a semi-automatic control system.

Results of low temperature seal testing were similar to the previous year with 8 representative polyurethane seals averaging 2966 hours. One high temperature Viton seal ran 3200 hours at 240°F. One pumping ring was still running after 16,500 hours.

The most discouraging aspect of the rolling seal program by 1967 was the poor correlation between seal life, failure type and the conditions under which the seal operated -- this resulted from an analysis of hundreds of seals at GMR as well as at Philips. It was attributed to the wide array of random operating conditions and the difficulty of obtaining groups of seals which were uniform in composition, homogeneity and wall thickness.

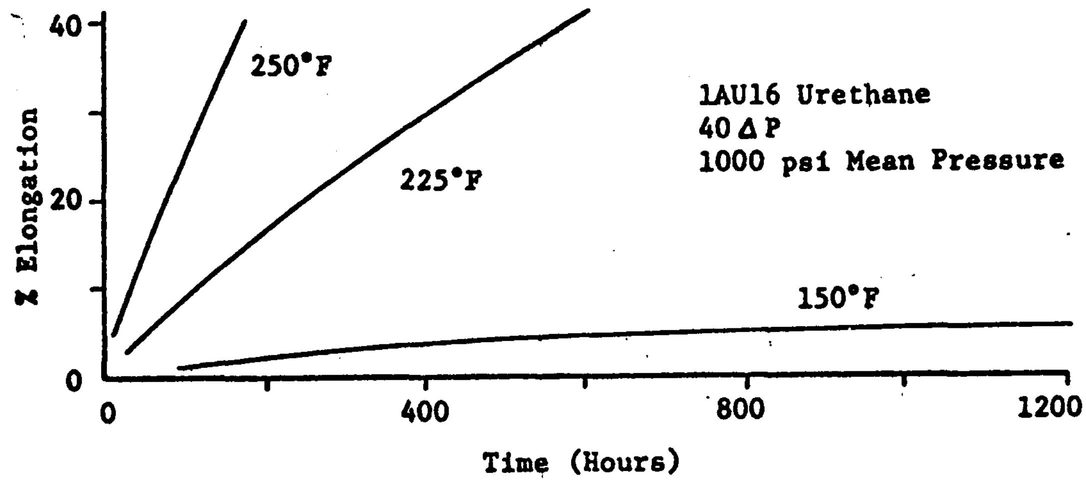
The rolling seal program had made some definite progress by 1968 under the control of the Seal Group. Approximately 49,000 hours were accumulated during the year with a total of 31 low temperature seal failures. In the previous year, 84 seals failed in the same number of test hours. Creep rate (elongation) of all seals tested were measured every 200 hours. It was established that the percent elongation was dependent on temperature and pressure differential across the seal and was linear with time; it was independent of the absolute pressure level.

Based on these conclusions, a new four station axial seal tester was constructed, driven by a swash plate mechanism. The seals were pressurized with a gas on one side and had no oil support on the opposite side, thus greatly simplifying the mechanical details and control system. While it did not simulate an engine with rolling seals, it facilitated testing of elastomer materials at a much greater rate. Seals were tested for 10 million cycles, examined for elongation and defects, and based on the findings, accepted or rejected for further testing.

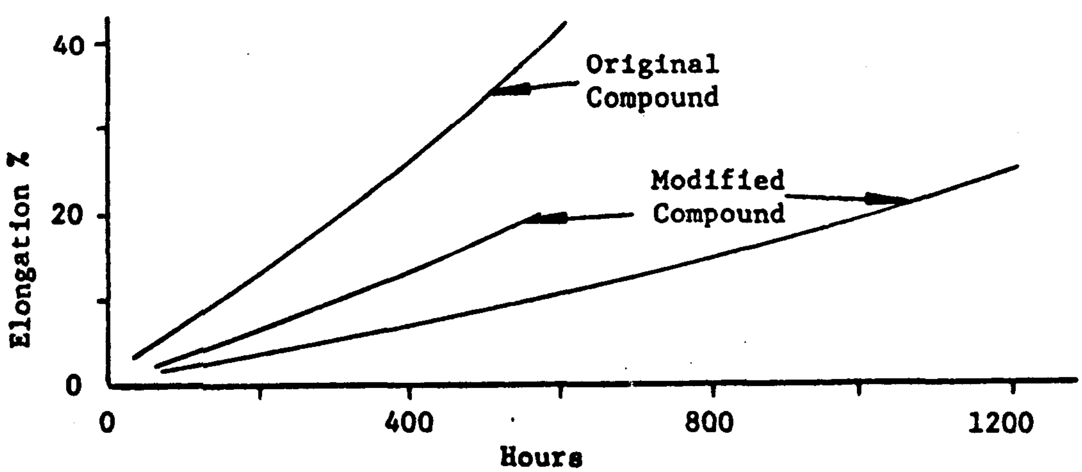
A new compression mold design was introduced which was self-aligning, which filtered the prep before it entered the mold cavity, and which utilized vacuum to scavenge the cavity. It was decided to concentrate on polyurethane for high temperature as well as for low temperature operation by improvements in the polymer and processing.

In 1969, the last full year of the program, the operating life of polyurethane seals at 200°F had increased from 200 to 1400 hours. Only 4 seals ran over 180°F in 1968 while 59 seals were tested in excess of 180°F in 1969.

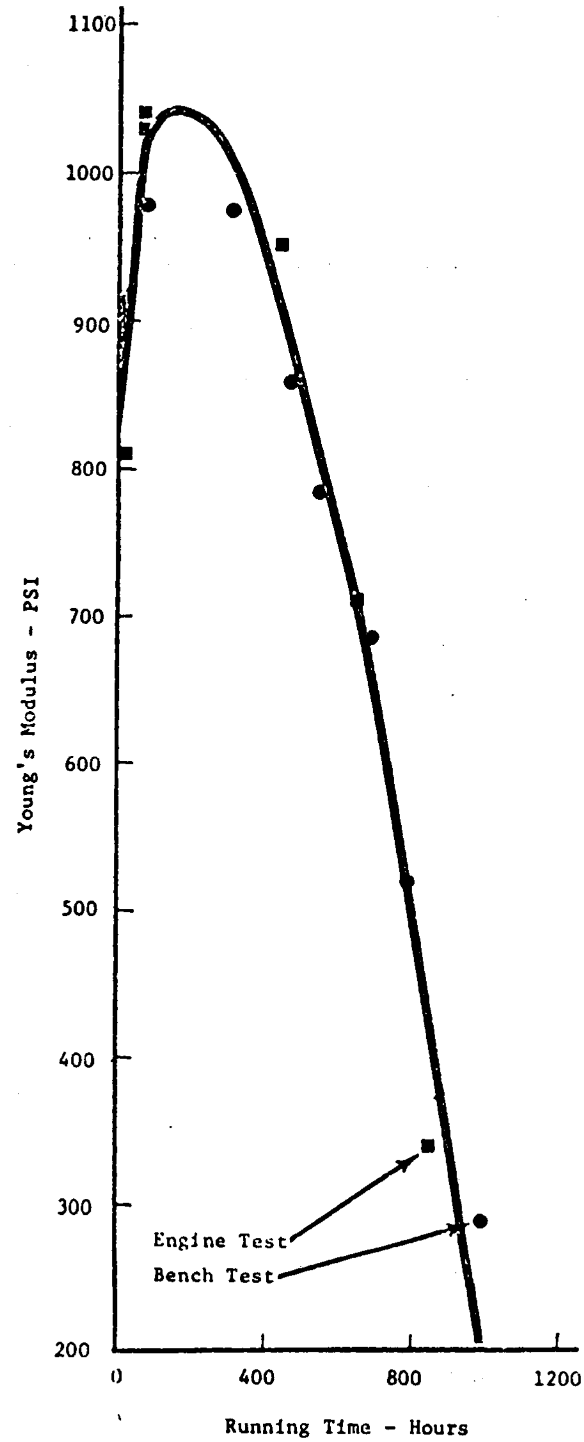
Two non-destructive measurements were devised to describe the state of polyurethane seals. The first was Young's Modulus, determined by stretching the seals 10% on a commercial Instron machine. The second was seal elongation. Initial values of Young's Modulus for a given formulation were used to check seal manufacturing reproducibility. Monitoring both measurements during subsequent seal testing gave an indication of the rate of degradation. The graph shows the change in seal elongation of a particular urethane compound at 3 different temperatures.



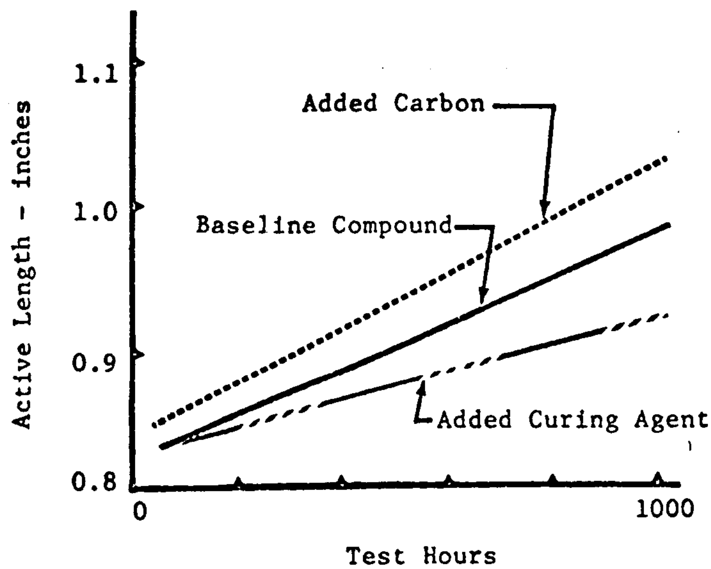

Success in modifying urethane compounds to improve seal durability is shown in the curves below. Modifications included "moca" curing agents and a granular type of carbon black.



Before the conclusion of the program in February, 1970, it was discovered that the primary limitations to the life of high temperature urethane seals was not pressure stresses, but degradation of the compound caused by exposure to hot oil which reduced the modulus. It was found that the useful life of a seal could be approximated by a simple oil soaking test, plotting the change of Young's Modulus over a time period, as seen in the curves at the right.




Increasing the amount of carbon black and curing agent above the level found in commercial baseline urethane compounds was investigated. Increased carbon raised both brittleness and creep rate; but increased curing agent was beneficial, as can be seen in the curves below.




Comparing urethane seals of 3 different wall thicknesses, it was concluded at the end of a 1400 hour test that creep rate varied directly with the average axial stress and was independent of bending stresses. Failure in all 3 occurred when the creep reached 135% of original seal length. The last significant test of urethane rolling seals, from February, 1969, to February, 1970, operated 2654 test hours. The purpose was to determine the maximum gas temperature in the seal area for a 500-hour life -- the value was near 238°F.

Although good Viton seals were far superior to urethane seals at temperatures over 200°F, Viton was still plagued by early failures which were not understood. A few seals were made of a silicone-Viton composite formulation which showed good flex life and good flaw resistance, but no life testing was done.

The hydrodynamic pumping ring was always considered a vital accessory for the rolling seal because it eliminated the requirement for a separate high pressure oil pump. In 1970, quality control was improved by machining test rings using externally pressurized bearing spindles for precision boring. Rings were produced which successfully functioned over 5000 hours in a life test machine at 1000 psi back pressure and an oil temperature exceeding 180°F.

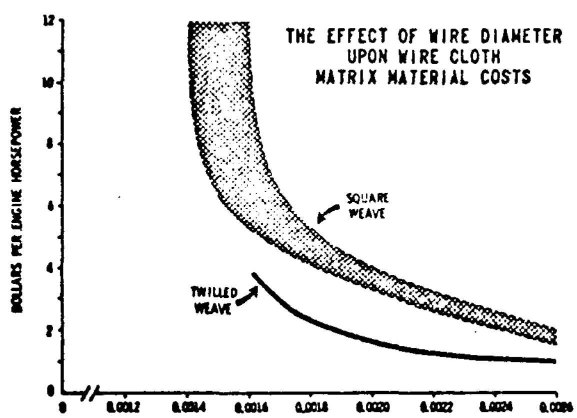
In conclusion, it appeared in 1970 that while much progress had been made in the development of high temperature rolling seals since 1963, the goal of 10,000 hours at 200°F was still far away. For temperatures under 150°F, polyurethene was unmatched and "good" seals were averaging about 3000 hours at 3600 rpm -- 648 million cycles -- several orders of magnitude better than were being realized by commercial "bellows" type reciprocating seals.

## REGENERATOR COST STUDIES

In 1961 Philips, GM Research and Allison Division measured flow losses in regenerators, which was a comparatively simple procedure. Determination of regenerator heat capacity is much more difficult and at that time, only Philips had built the complex apparatus required for such measurements. GM decided not to duplicate Philips' heat capacity machine because of cost and time required. In 1962 at a conference of General Motors' and Philips' engineers it was confirmed that some revision of the Philips regenerator analysis procedure was required. Real flow losses of the latest regenerators were found to be 50 - 100 per cent greater than those predicted by the original design procedures of 1958. Similarly, it was noted that measured heat capacities were from 70 to 85 percent of the predicted values. Corrections were agreed upon by all the participating groups. Meetings of this type were common throughout the eleven years of GM activity, and GM Research Laboratories assumed the responsibility for coordinating the analysis efforts.

In 1963 both Allison Division and the Electro-Motive Division (Cleveland Diesel was dissolved in 1962 and the Stirling project and personnel transferred to EMD) singled out the regenerator as the most serious expense item -- an order of magnitude too high for commercial engines. At that time regenerators were made exclusively of stacked layers of 200 mesh, .0015 inch diameter stainless steel screen or "cloth." It was determined that fabrication or weaving of the screen was the principal cost factor; the wire cost was about 30 percent of the total. GM Research immediately began a detailed study of regenerator costs, and a survey of potential substitutes for wire screen. Some materials examined included "Feltmetal," various foamed metals, electro-plated screens, photoetched thin sheet, sprayed porous metal sheet, polymer and metal powder matrix, aluminum oxide "whiskers," and single parallel lay wire screens. What was lacking at that time was the amount of performance loss which could be tolerated in exchange for reduction in cost.

In 1964 additional studies of wire cloth were made. Costs ranged from \$11.70 per engine horsepower for the best square weave to a low of only \$1.00 per horsepower for twilled weave material. Curves are shown of the range of wire screen costs. It was estimated that regenerators built with the \$1.00 material would result in a sacrifice of between 4 and 5 percentage points in best efficiency and a slight reduction in maximum power.



A new punching method for standard wire cloth regenerators was devised which resulted in a 50 percent savings in manufacturing cost at GMR and which had the potential for automatic production and assembly of regenerators.

Studies continued of other matrix materials. The most encouraging was a foam metal (nickel) being developed at that time by the Metallurgical Dept. of GMR for filters, later termed "Met-Net." Costs were estimated to range from $0.15 to $0.30 per horsepower. Philips determined from samples tested that fluid friction losses were greater than for wire cloth while heat transfer characteristics were satisfactory.

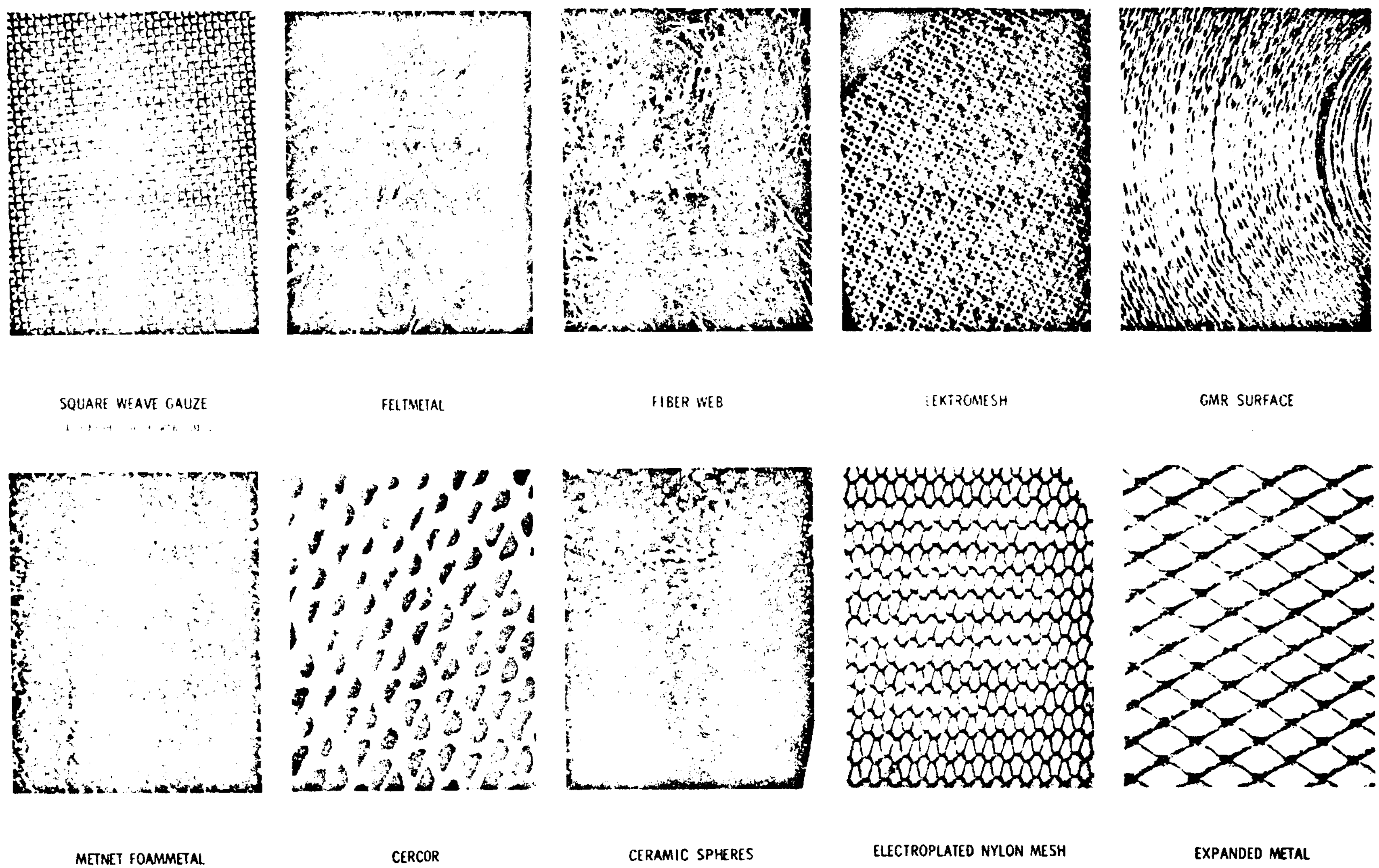
Considerable effort was made by the Electrochemistry Department of GMR to develop a very fine 60% porosity electroplated as well as a photo-etched screen matrix, but with little success. Their estimated costs ranged between $0.80 to $0.90 per horsepower.

Regenerator calculations were performed with the aid of a computer program originally developed for gas turbine regenerator studies. The purpose was to determine the upper limit on longitudinal thermal conductivity for the new regenerator matrix materials. It was found that for certain conditions, the limitation must be based on <u>direct</u> conduction heat transfer losses, rather than on the influence of conductivity upon regenerator <u>effectiveness</u>.

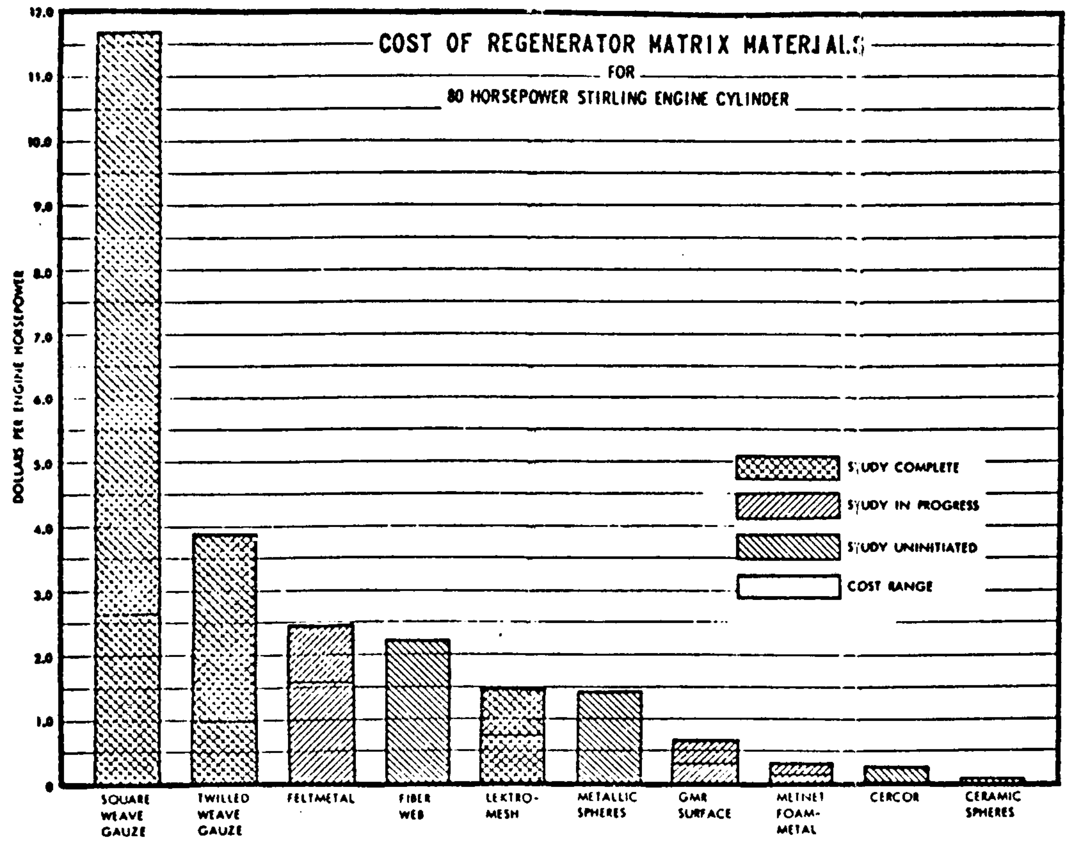
By the end of 1965 studies of alternate materials were complete on Lektromesh (C.O. Jelliff Mfg. Co.), and Feltmetal (Huyck Metals Co.), while studies continued on GMR Met-Net and a thin folded sheet material originally designed for gas turbine regenerators, called GMR Surface. Still to be examined were Fiber Web by Brunswick Corp., metallic spheres, Cercor by Corning Glass, and ceramic spheres. Eight regenerators were constructed of Met-Net and 6 of GMR Surface. They were flow tested and then sent to Philips for heat transfer evaluation.

In 1966 the GMR Met-Net manufacturing process was modified to increase the surface to volume ratio about 40 percent. In 1967, the first engines were tested with Met-Net regenerators. After a 100-hour run there was no loss of structural integrity in the matrix. Engine efficiency was 89% and the power 91% of that obtained with standard wire cloth regenerators. Since the engine was not optimized for Met-Net nor was the regenerator filling factor optimum, it was believed that the efficiency and power would improve in future designs.

A magnified view of 10 matrix materials is shown and results of the cost comparison is illustrated on the bar chart.



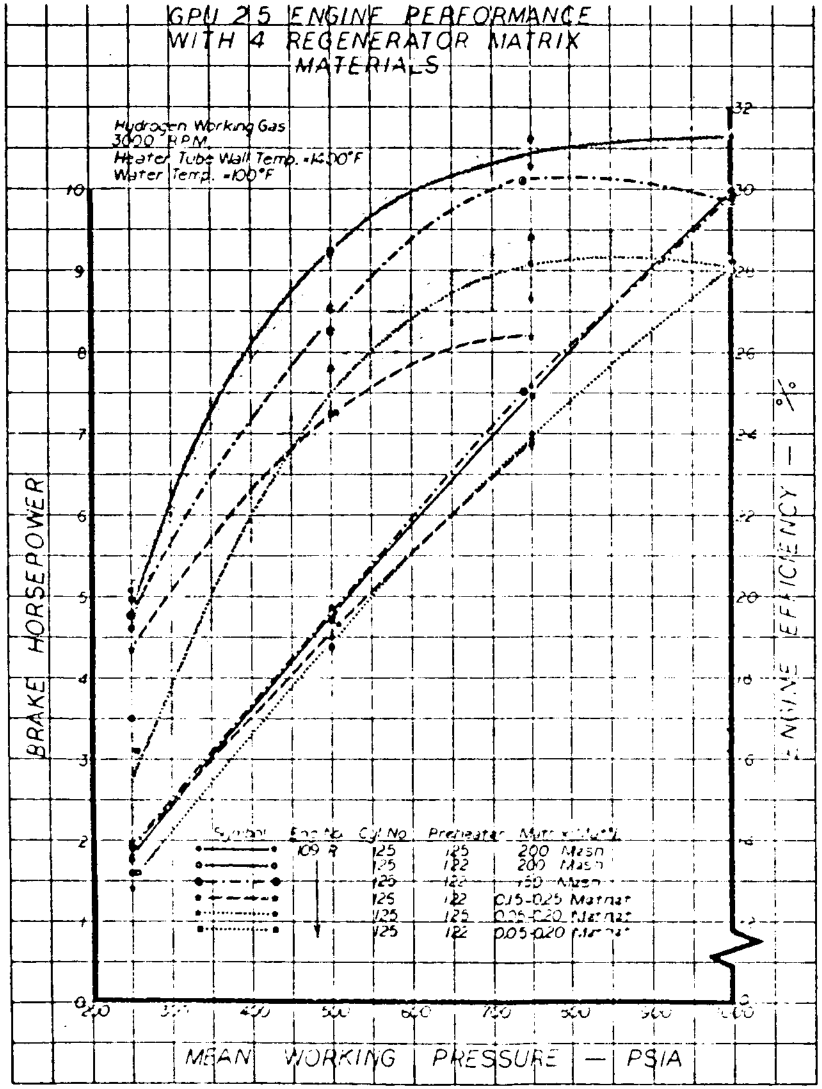

STIRLING REGENERATOR CORE MATERIALS SHOWN MAGNIFIED 15 TIMES

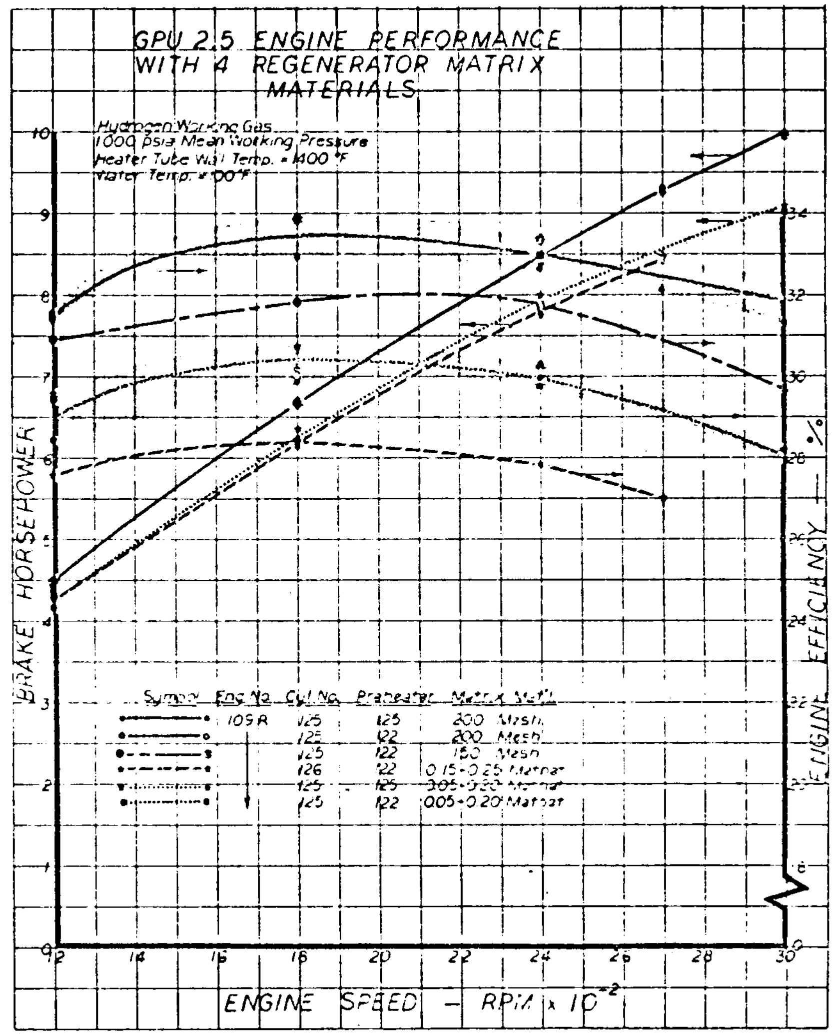



The original Met-Net was made by electroless nickel plating on a polymer sponge material, followed by burn-out of the sponge. The sheets produced were about 1/8 inch thick and had filling factors ranging from 0.05 to 0.15. By experiments, it was shown that higher filling factors were needed to reduce engine dead space. This was achieved by compressing the Met-Net. Numerous combinations were tested; the one producing the least degradation of power and efficiency (see paragraph above) had an initial filling factor of 0.05 and a final filling factor of 0.20. Its effective longitudinal thermal conductivity as measured by Philips was 0.017 watts/°C cm. This is approximately three times that of 200 mesh wire cloth. Comparison of engine performance with 4 regenerator matrix materials is shown on the next page.

By August, 1968, the dynamometer test engine reached 1200 hours, operating with Met-Net regenerators. There was no indication of fatigue failure nor structural change in the matrix nor was there any measurable change in engine efficiency over the entire run.

In 1969 the same engine had operated over 4000 hours on a severe schedule to determine maximum hot space temperature for the piston rod sealing system. At the end, fuel consumption was within 2.6% of the initial fuel rate, and visual inspection verified that the regenerators were in good condition. This marked the successful end of a 6 year program. An enlarged picture of one of the regenerators after 4101 hours is seen on page 96.


GPU 2.5 ENGINE PERFORMANCE
WITH 4 REGENERATOR MATRIX
MATERIALS
Hydrogen Working Gas
3000 RPM
Heater Tube Wall Temp. = 1400°F
Water Temp. = 100°F
Symbol Eng No. Cyl No. Preheater Matrix Mat'l
109 R 125 125 200 Mesh
125 122 200 Mesh
125 122 150 Mesh
125 122 0.15-0.25 Matnat
125 125 0.05-0.20 Matnat
125 122 0.05-0.20 Matnat
BRAKE HORSEPOWER
ENGINE EFFICIENCY — %
MEAN WORKING PRESSURE — PSIA


GPU 2.5 ENGINE PERFORMANCE
WITH 4 REGENERATOR MATRIX
MATERIALS
Hydrogen Working Gas
1000 psia Mean Working Pressure
Heater Tube Wall Temp. = 1400°F
Water Temp. = 100°F
Symbol Eng No. Cyl No. Preheater Matrix Mat'l
109 R 125 125 200 Mesh
125 122 200 Mesh
125 122 150 Mesh
126 122 0.15-0.25 Matnat
125 125 0.05-0.20 Matnat
125 122 0.05-0.20 Matnat
BRAKE HORSEPOWER
ENGINE EFFICIENCY — %
ENGINE SPEED — RPM x 10^-2

## EXTERNAL COMBUSTION SYSTEM

## AIR PREHEATER STUDIES AND HEATER-CYLINDER HEAT TRANSFER

The air preheater is the heat exchanger in the external combustion circuit for recovering surplus heat from the exhaust gases and transferring the heat to the incoming combustion air.

Testing of the first Ground Power Units in 1963 pointed to the combustion air preheaters as one of the more unreliable parts of the package, and an expensive one. At that time it was a sheet metal assembly consisting of spiral passages (rectangular crossection) conducting air from the outside inwardly, and alternate passages conducting hot exhaust gas from an inside header to the outside. The stainless steel sheet parts were brazed together; the spiral configuration was supposed to permit thermal expansion and contraction without breaking the joints. In 1963, one GPU preheater failed after 614 hours from an accumulation of warpages and cracks which allowed cross leakage to become excessive.

In 1964 a series of preheater failures and engine heater-cylinder* failures occurred in the 360 hp 4-cylinder Philips-built engine which had been sold by the Electro-Motive Division to the U. S. Navy Engineering Laboratory. Pre-heater and heater tube parts were found to have melted. Also, the 80 hp single cylinder engine at EMD failed several combustion chambers from overheating; and two heater assemblies failed at GMR. A careful study of these failures led to the following conclusions:

Examination of the thermal equilibrium conditions which must be satisfied by the heat flows and temperatures in the external combustion system showed that the inherent regenerative nature of the system allowed it to establish temperature levels that would automatically correct for low heat transfer coefficients into the engine heater. This self-correcting action simply raised the temperature level of the combustion air, combustion products, and

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* The engine heater-cylinder assembly consists of the stainless steel displacer cylinder, the circular tube cage brazed into the cylinder head and the regenerator cups attached to alternate tubes by brazing. It will usually be referred to as the "heater," as distinct from the air preheater.

exhaust gases until sufficient temperature difference prevailed between the engine heater tube walls and the combustion gases to force transfer of the amount of heat required by the engine. Also, due to the high effectiveness of the preheater, the temperature level of the entire internal circuit could be raised significantly with hardly noticeable changes in the exterior temperatures or the fuel rate. The conclusion drawn from knowledge of this system characteristic was that an insufficiency of heat transfer capability in the engine heater, either from low unit area heat transfer coefficients or from low total surface area, would not result in a notable reduction in engine output; but rather it would result in an increase in combustion system temperature and overheating of the burner, heater, or preheater. It also became evident that the effectiveness of the preheater had to be matched to the heat transfer capability of the engine heater and to the temperature limitations of the burner, heater, and preheater. The system efficiency could not be improved simply by increasing the preheater effectiveness without regard to the matching of all components.

Additional tests were made in 1965 of heat transfer to the heater-cylinder. It included three procedures, (1) the insertion of baffles in the burner cage to direct the combustion gases to three passes across the heater tubes, (2) packing insulation on sections of the heater tubes to reduce heat transfer area, and (3) altering the air and fuel flow to cover all operating conditions. Twenty shielded thermocouples were installed to measure burner inlet and heater exhaust gas temperatures. During the investigation, a preheater burned out in the same manner as those in the 360 hp Navy engine.

These general conclusions were drawn from results of the tests:

1. The insulation on the heater tubes had little or no influence on the engine performance. This can be interpreted to mean that either a uniform heat flux pattern was not necessary for good engine performance or that the normal heat flux pattern for this burner cage was not greatly changed by the insulation.

2. While the apparent heat flux per unit area was increased three-fold there was very little relative change in the burner exhaust temperature. This indicated that the heat transfer coefficient between the combustion gas and the tube surface was much higher than had been anticipated.

3. Temperatures measured with the minimum heat transfer surfaces exceeded 2000°F on the inner exhaust header of the preheater. If some carbon or other combustion products had accumulated on the preheater it was believed that combustion could be initiated with the resultant burnout and melting of stainless steel parts.

In 1966, a heater tube failed when a chunk of metal (N-155) blew out of the tube wall. With help of the Metallurgical Department, the cause was found to be extreme oxidation due to presence of copper on the tube. The copper source was found to be a commercial anti-seize compound recommended for assembly of high temperature parts -- in this case, the preheater top to the core section. It is brought to attention here because of the subtle nature of this kind of failure and the possibilities of its repetition in the future.

In 1966 a special heat transfer apparatus was designed to measure accurately the net heat flow into the heater end to obtain an overall heat balance. The air preheater was located separate from the main engine in order to place additional thermocouples with radiation shielding in the hot air and exhaust gas regions. This system in conjunction with an engine was chosen after numerous studies had shown that other kinds of heat transfer apparatus, including a closed high presssure helium loop, would require longer to construct and still not duplicate real engine conditions.

The preheater for the GPU-3 engine had been designed by Harrison Radiator Division of GM. Instead of spiral passages formed by sheet stock as in the Philips design, they selected stainless steel tubes for carrying the exhaust gases in an outward spiral (arc of about 180°); air was blown over the tubes while confined within spiral sheet passages in close proximity to the tube bundles. This preheater design was plagued with early loss of effectiveness in some models, while one, after 900 hours, suffered gross breakage of tubes; others exhibited severe oxidation. In 1967, an "improved" Harrison spiral tube type was placed on test for 500 hours. During that period the exhaust temperature rose from 420° to 850°F, indicating a serious loss of effectiveness. All spiral types were finally abandoned for more conventional shell and tube designs.

In 1967, Harrison and GMR began a program to study lower cost preheater designs. It was shown that sheet material was preferred to tubes and that folded stock or "accordion" types were the most promising with plate-fin types the second choice.

In 1966, Philips was beginning to study rotary ceramic preheaters similar to those being used in regenerative gas turbines. In 1967, independent of Philips, EMD tested a rotary Circor model on their 80 hp engine. It proved too effective, resulting in damaged burner parts from overheating -- illustrating the necessity of careful matching of the external combustion system components. It was concluded by Philips and GM that the rotary type was best suited to multi-cylinder engines.

The new heat transfer apparatus designed in 1966 was fabricated and assembled in 1967 and testing was started. Before this was available, it was only possible to measure brake fuel consumption and power as well as heat rejection from the cooler;* but not heat input to the heater except by calculation. By the end of 1968, GMR had determined the first heat balance, the "true" engine thermal efficiency and the apparent heat transfer film coefficient for the outside of the heater tubes. A sample heat balance is shown in Table 2 of Reference #6.

Approximately 96% of the heat which enters the engine was accounted for. The unaccounted loss was undoubtedly the result of thermometry errors in measuring the exhaust gas from the engine heater, which is in the range of 1500 -- 1700°F., and to a lesser extent, errors in measuring the heated inlet air in the range of 900° to 1100°F.

The "true" engine brake thermal efficiency based on net heat to the heater was equal to 33 percent. This is the value required for comparing computed engine efficiency vs. measured and is also useful for evaluating Stirling engines running from thermal energy storage, isotopic or nuclear reactor heat sources, or some other "closed" heat source such as lithium metal combustion. Reference #6 discusses these topics in detail.

The primary purpose of the heat transfer experiments was to obtain data on the apparent heat transfer coefficient on the outside of the heater tubes. The correlation of these data on the basis of conventional Nusselt and Reynolds Numbers is shown in the graph. The fact that the Stirling data and the McAdams data are different is a reflection of the fact that there are significant differences in the conditions under which the data were determined. The McAdams data are for pure convection heat flow from a bank of relatively widely spaced tubes to gases passing over the tubes. In the Stirling engine, the heat transfer is from reacting combustion gases to relatively closely spaced tubes under circumstances where radiation, dissociation, molecular diffusion, and other boundary layer effects may influence the heat transfer.

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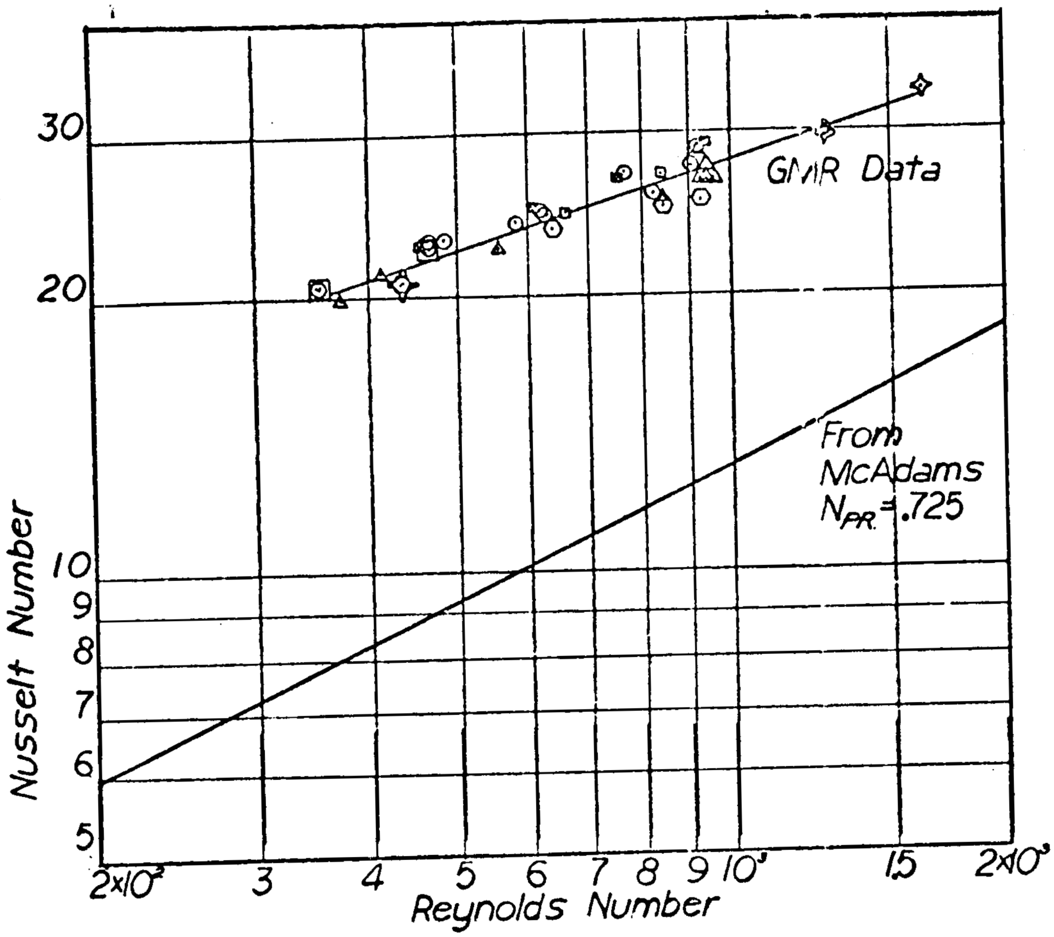
* Cooler heat flow had been measured before, but not with the accuracy achieved with the new heat transfer rig. Buffer and oil heat rejection had not been measured before.




In 1968 the preheater cost reduction program was beginning to show some results with hardware under test. A cost comparison based on core material alone, without considering assembly and manufacturing costs, is shown in the table. The price for .01 thick stainless steel sheet for design III and IV was assumed at $0.30 per pound. Tubing costs were based on lot sizes of 1000 feet.

PREHEATER CORE MATERIAL COST COMPARISON

| | I | II | III | IV |
|---|---|---|---|---|
| | Spiral Tube | Straight Tube | Plate Fin | Accordion |
| Heat Exchanger Surface per BHP - sq.ft./hp. | 2.1 | 1.65 | 3.0 | 1.2 |
| Core Weight per BHP - lb./hp. | .672 | .53 | 1.20 | .48 |
| Core Material Cost - $/lb. | 34.40 | 34.40 | 2.00 | .80 |
| Core Material Cost per BHP - $/hp. | 23.10 | 18.30 | 2.40 | .38 |

The data clearly indicate the advantages of using basic cores constructed of sheet materials. The plate fin and accordion cores were designed so that manufacturing methods which would be used in quantity production would retain the basic price advantage of the core material.

Heat exchanger surface area of number III is highest because the extended surface of the fins is only about 60% as effective as the primary surface; all the surface on the accordion design is prime surface.

Pictures of 4 different preheaters which were operated in 1968 are included.

1) One spiral tube design was still running after 1612 hours; the last 200 hours were with natural gas fuel; the first 1412 hours on diesel fuel resulted in signs of severe corrosion from sulphur in the fuel.

2) Seven straight tube preheaters were built, only as an interim solution, while Harrison was working on the newer sheet metal designs. One failed at 1264 hours from poor brazing; all others were running with one at 750 hours.

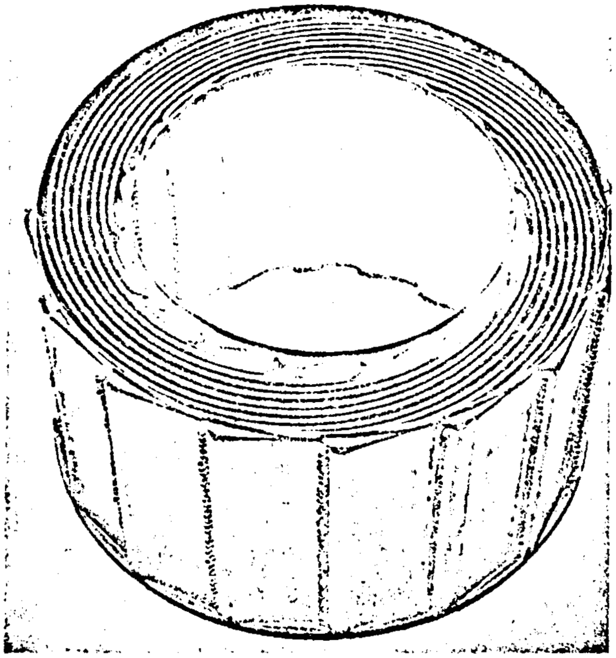
3) One plate fin type was built. It operated for 209 hours before breaking apart at a header joint.

4) The accordion unit appeared most attractive for single cylinder engines. Unfortunately, the first one failed after 25 hours at a braze joint; it was to be repaired for further testing. However, records indicate that it was not operated again before the program was stopped.

At the conclusion of the Stirling program, one straight tube preheater had operated 2475 hours without failure. This is in contrast to the period 1960 - '61 when 3 preheaters failed in less than 10 hours and the best one survived less than 300 hours.

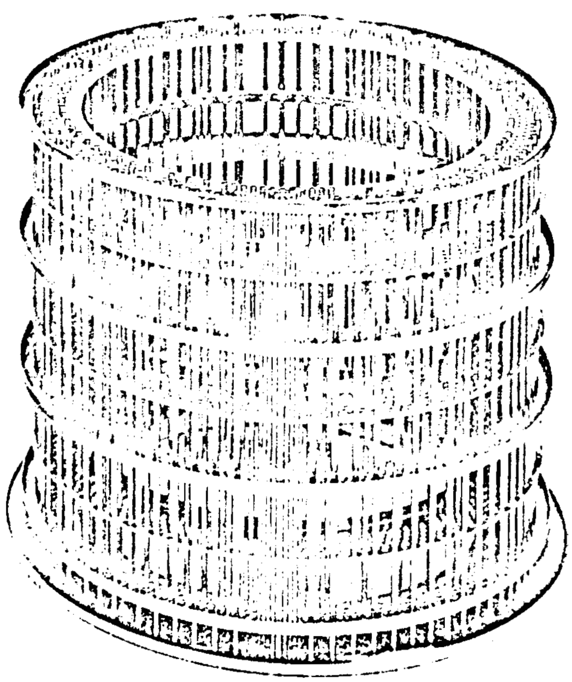
At the conclusion of the program, one heater-cylinder assembly had operated for 4803 hours and was still usable. In 1960 - '61 two heaters failed in less than 80 hours.

A general summary of engine component durability is shown in a separate section of the report.



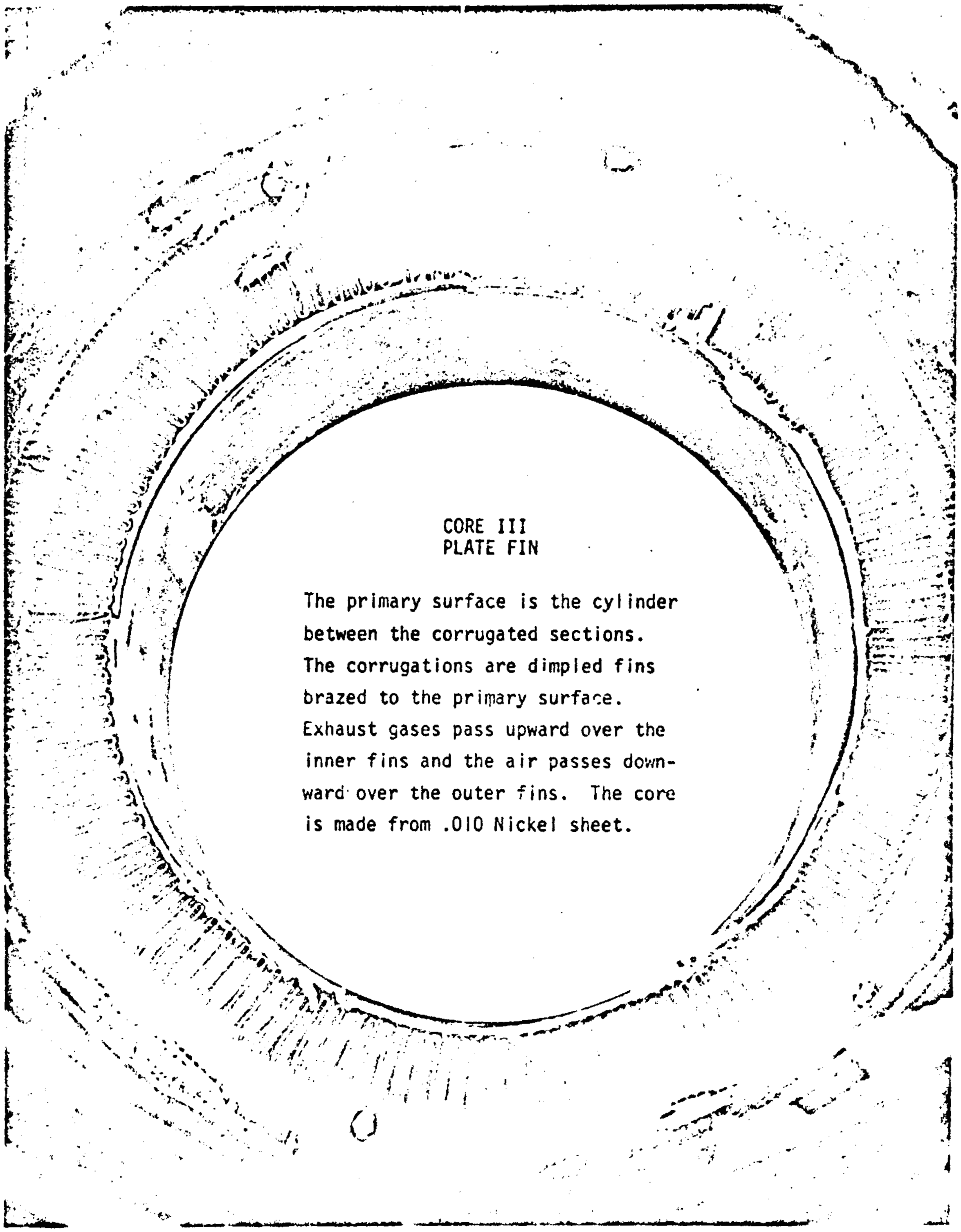
CORE I
SPIRAL TUBE

The primary surface is made of .125 O.D. x .008 wall thickness 310 stainless steel. Exhaust gases enter the tubes at the inner header and pass outward to the outside header. Vertical plates between the tubes form spiral paths on the outside of the tubes. The air enters at the bottom outside and flows inward through these paths and goes to the combustion chamber upward through a header at the inside diameter.



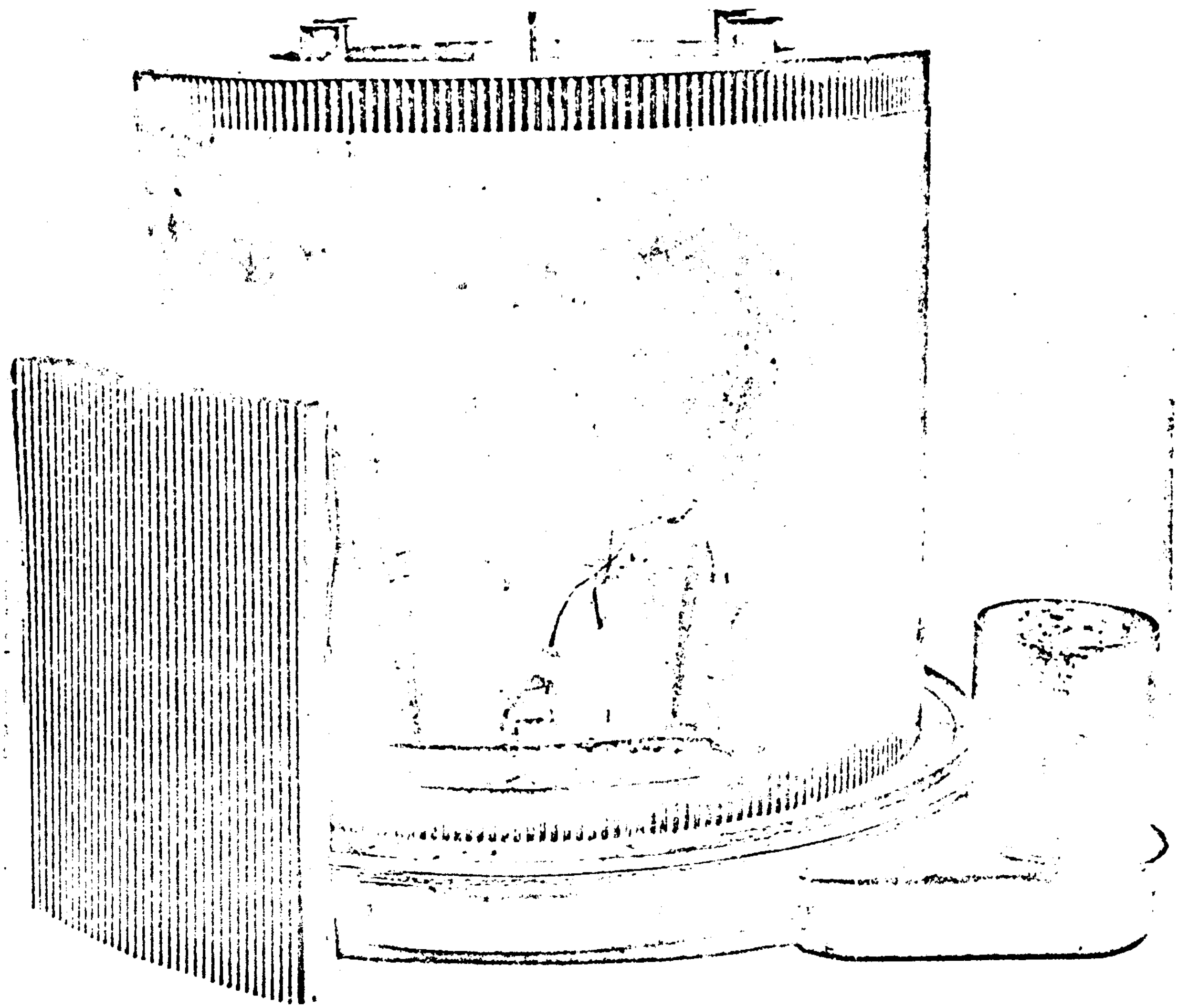
CORE II
GMR STRAIGHT TUBE

The primary surface is .125 O.D. x .008 wall thickness 310 stainless steel tube. Exhaust gases pass upward through the tubes and the incoming air makes six passes back and forth across the tubes as it moves downward through the shell.



CORE III
PLATE FIN

The primary surface is the cylinder between the corrugated sections. The corrugations are dimpled fins brazed to the primary surface. Exhaust gases pass upward over the inner fins and the air passes downward over the outer fins. The core is made from .010 Nickel sheet.



CORE IV
ACCORDION

The corrugated section is made from .010 310 stainless steel sheet and it is the primary surface. Passages on the inside and outside are formed by the accordion and inner and outer cylindrical surfaces. Incoming air enters the outer passages through the opening at the bottom, passes upward and radially outward to a header. Exhaust gases similarly pass downward through the inside passages to a collector and out the tube at the lower right.

## EXTERNAL COMBUSTION SYSTEM
## BURNERS, NOZZLES & EXHAUST EMISSIONS

The Philips built engines operated at GMR in 1959 and 1960 were equipped with vaporizor type burners and air pressure atomizing nozzles. Two fuels were required for start-up, a gaseous fuel such as propane or butane to preheat the "target" plate on which the liquid fuel was sprayed, and a distillate such as No. 1 or No. 2 diesel which was the normal running fuel.

The department at GMR which had specialized in gas turbines and their burner systems was requested to assist in the development of a single-fuel gas turbine type burner for the GPU program. By the end of 1961, such a burner was successfully operating on the GPU. A distillate fuel was sprayed from an air-atomizing, water-cooled nozzle. The igniter plug was a separate assembly, but in the next model the plug was combined with the nozzle assembly. This improved starting reliability. The combustion chamber or burner "can" was designed similar to a gas turbine burner with typical slots and holes for admission of primary and secondary air.

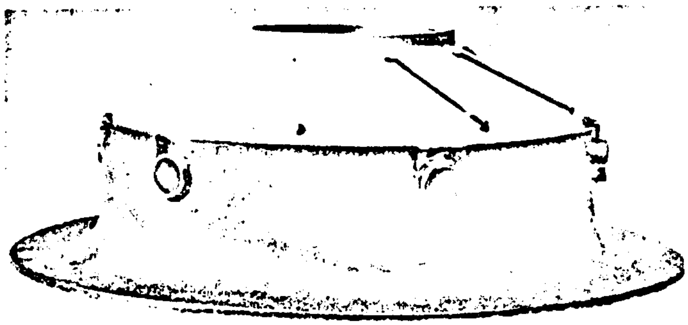
In 1960, six vaporizer burner "cans" failed in less than 30 hours; in 1962 one of the new burners failed in 180 hours while 11 others were operating successfully, up to 275 hours.

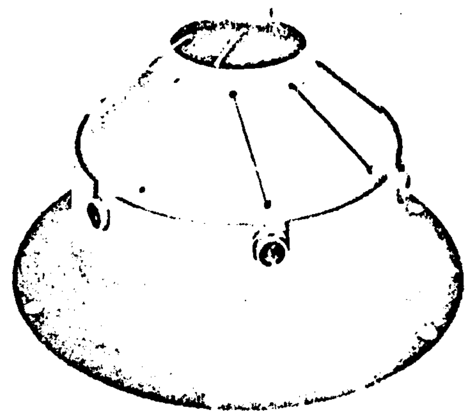
In 1962, the first automatic fuel controller, based on a high temperature (1200°F) bi-metallic rod-type thermostat immersed in the heater tube hydrogen circuit, was successfully tested. Also accomplished was a new air-fuel controller which eliminated operator attention during starting, warm-up, and acceleration after cranking.

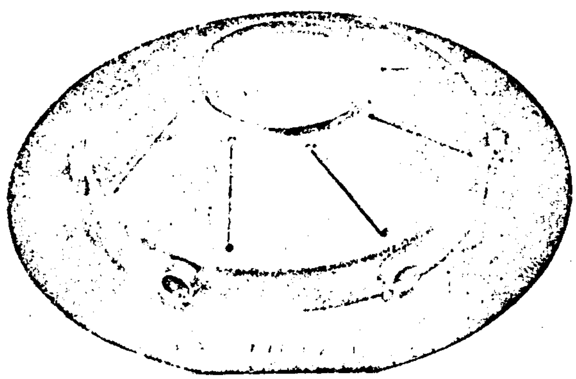
A study was started to determine the reasons for non-uniform heater temperature distribution--variations as high as 180-200°F in some cases. The main cause was found to be variations in the rate of heat supply; i.e. the burner, rather than variations in the rate at which the engine takes away heat from the heater. A rotating fuel nozzle, driven by an electric motor, was constructed and tested. Temperature differences were reduced by as much as 70-80°F. Next, a study of combustion air leaks into the burner-heater-nozzle system was made -- these can cause deflections of fuel spray as well as having a cooling effect on some portions of the heater tube cage. The most critical fit was found between the fuel nozzle and the burner.

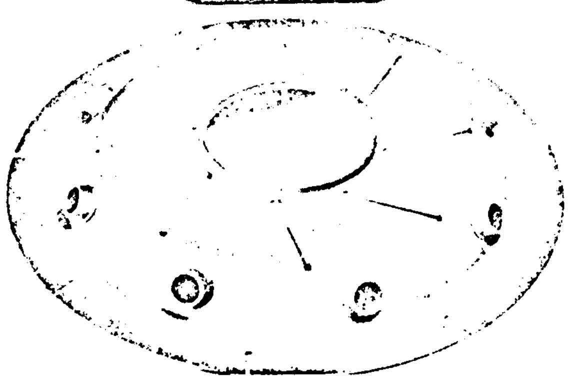
In 1964, at the conclusion of the 500-hour test of the GPU-2 by the Army at Ft. Belvoir, the combustion chamber was found to be still usable, but was distorted to such an extent that its life was considered marginal to 500 hours.

In 1967, the durability of burner parts as tested in the GPU-3 series exceeded 900 hours. Success was achieved in reducing the height of the GPU-3 burner in preparation for the 4L23 engine program. The pictures show the progress in cutting in half the height of the standard burner of 2.57 inches to 1.28 inches. The lowest design of 0.63 inches was never tested.









**Above:** Standard GPU Burner 2.57 in. high

**Top Left:** 1.76 in. high

**Center Left:** 1.28 in. high

**Lower Left:** .63 in. high

The engine in the Stir-Lec hybrid car represented the best achievement in automatic start-up and operation of any combustion system which had been developed prior to 1969. In its final version, the engine required 45 seconds from key-on to full load running.

In 1969, the combustion system in the GPU endurance test program achieved 2475 hours without failure.

In 1967, a GPU combustion system was modified to operate on natural gas. The project was initiated by a request from the Institute for Gas Technology in Chicago. It had operated about 500 hours by 1968, mostly for the purpose of developing and improving governor systems and the hydrogen compressor.

Emissions from a GPU engine were first measured in 1967 as a result of the rising interest in vehicle applications. Results will not be presented here since the program was thoroughly covered in reference #4. This reference also discusses burner construction.

In 1968, Philips began tests of exhaust gas recirculation as a means for reducing oxides of nitrogen. In 1971 and 1972 Wayne State University performed similar tests on a GPU engine, donated to them by GMR, and reported in references # 8 and 9.

In 1970, the Stir-Lec II car was tested for closed cycle, true mass emission values, for the warmed-up condition, in accordance with the Federal 7 cycle test. The engine combustion system was not modified for the test in any way. Following the chassis dynamometer driving test, the Stirling engine was operated at full load for 26 minutes to return the batteries to their original state of charge. Approximately 58 percent of the total emissions, shown in the Table, were emitted during the recharging period. The inertia weight of the dynamometer was set at 3000 pounds. A road load setting of 3.5 horsepower at 50 mph was used. This setting was the result of matching motor current and voltage on the road at 30 and 40 mph to current and voltage obtained at equal speeds on the dynamometer. Because of the low range of exhaust pollutant concentrations expected, the long path Beckman Model 315L carbon monoxide and nitric oxide NDIRA's, the Beckman Model 315 carbon dioxide NDIRA and the Beckman Model 108 flame ionization detector were used for this test. Chart recorders were provided for all instruments. A GM Proving Ground fuel meter was connected immediately upstream of the burner fuel nozzle. No. 1 heating oil was burned throughout the emissions test.

EMISSION RESULTS

| | |
|---|---|
| Unburned hydrocarbons | 0.0045 gm/mile |
| Carbon monoxide | 0.57 gm/mile |
| Oxides of nitrogen (as $NO_2$) | 3.94 gm/mile |
| Air/fuel ration (range) | 26:1 - 30:1 |
| Fuel economy | 15.6 mpg |

It should be added that the "transmission efficiency" from engine to rear axle was estimated at between 40 and 45 percent, because of losses through the generator, battery, speed control system, motor and motor speed reducer. With a Stirling engine driving through a mechanical transmission, emission values would be about half of those reported above.

## GOVERNOR AND HYDROGEN SYSTEM

The torque of a Stirling engine can be controlled by several methods. In most cases, it has been done by modulation of the working gas pressure. Other methods have included "dead space" control, by-pass control, parasitic control, and temperature control; the latter is perhaps the least effective and evidences the slowest response. During the 11 year activity at General Motors, the temperature and torque were always independently controlled, and this appears to be the best solution.

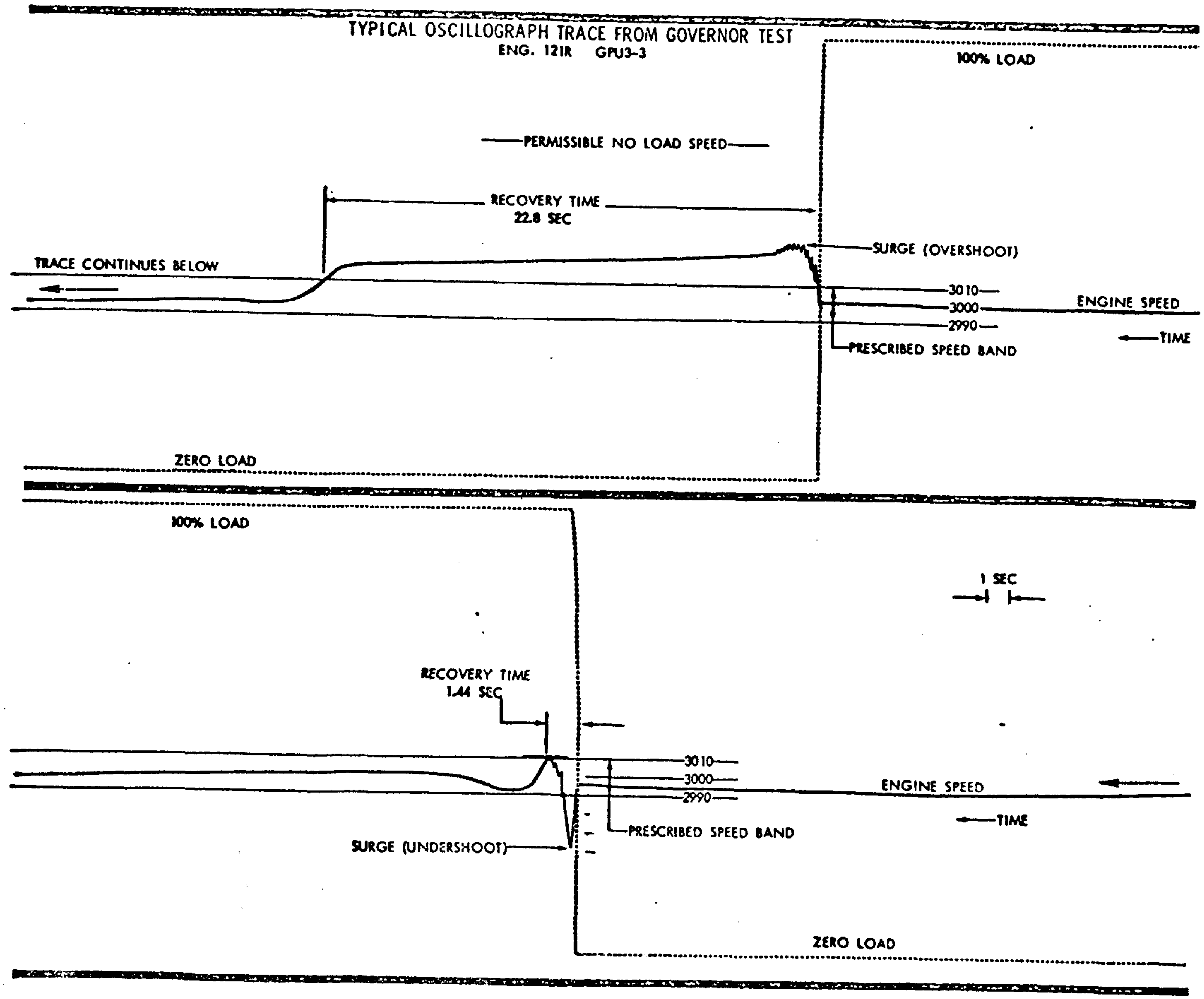
The GPU series of engines required control of the speed to specified limits. Stability was to be maintained at ± 10 rpm at 3600 rpm; regulation was not to exceed a droop of 90 rpm; surge limit was not to exceed 216 rpm and recovery time for 100 percent load change was not to exceed 6 seconds. Discussion of the results of work at GMR is found in references 3, 5 and 6, while a general discussion of governing is found in several Philips' publications by R. J. Meijer. The high speed recording oscillograph was a vital tool for the governor development program. A typical record is shown on the next page.

By 1967, the GPU-3 governor system was capable of holding the stability at ± 5 rpm, the droop did not exceed 10 rpm, the surge limit for sudden increase in load was met, and the recovery from surge for 100% sudden decrease was met in two seconds.

From the standpoint of reliability, however, the entire speed governing system was a constant source of trouble until nearly the end of the program. The hydrogen compressor was perhaps the major problem in the beginning. It was incorporated into the base of the crankcase of the GPU-3 as an extension of the displacer piston rod in the form of a hydraulic plunger. Hydraulic pressure activated a diaphragm compressor which eliminated the need of a sliding or rotating seal. This made servicing more difficult. The hydraulic plunger required precision machining and was subject to binding. In retrospect, it would have been better to mount an experimental compressor outside of the engine and drive it from a gear, with a break-away coupling, or from a belt. On the other hand, an outside compressor requires a good seal to prevent hydrogen leakage.

Another item which often stopped endurance tests was failure of the small (½" dia.) hydrogen check valves and main control valve -- usually the seats were pounded out or distorted sufficiently to leak. The hydrogen control valve was actuated by hydraulic pressure delivered by the speed sensing governor which was mounted on the crankcase and gear driven.

TYPICAL OSCILLOGRAPH TRACE FROM GOVERNOR TEST
ENG. 121R GPU3-3



In 1964, the governor system had 5 separate valving units and 10 adjusting screws; by the end of 1965, it had 2 valving units and one adjusting screw.

Results of endurance testing of the GPU3-3 at GMR in 1967 showed that the hydrogen compressor had failed twice in the 1537 hour run and the governor hydrogen valve had failed 4 times. A summary of the various failures which ended service-free periods is included on the next page, extracted from a 1967 report.

The hydrogen explosion mentioned was the only incident of its kind in the 11 years of activity at GMR. No such accident occurred at either Philips or EMD, so far as the author is aware.

The GPU3-2 which had been delivered to the Army completed its 500-hour test in 1967. The main item which caused shut-downs was the governor hydrogen valve, which failed 4 times, just as in the GMR test reported above.

In 1969, the GPU3-3 at GMR was operated on a more rigorous 500-hour test, equaling a military qualification test. In order to meet military requirements, a "Certified Parts List" for the package was established so that all parts were like the engineering drawings. This defined exactly what was being tested and prevented casual substitution of components which would have caused the test to lose significance. At the conclusion, the maximum overhaul life was extended slightly but under more rigorous conditions to 560 hours from the previous 553 hours.

The longest run with no service was extended to 525 hours from the previous high of 196 hours. The longest run without stopping was extended from 159 hours in 1967 to 235 hours. In all, the engine was stopped 4 times, all caused by building safety interlocks and in no way connected with the GPU operation. The limit of 560 hours was caused by the hydrogen compressor--a small valve assembly failed to function properly. However, the hydrogen check valves and governor control valve were in excellent condition.

An alternate system for compressing hydrogen was investigated briefly in 1961. It was based on electrolytic generation of hydrogen and diffusion through palladium tubes. Pressures to 1150 psi were maintained inside the tubes; but the concept was abandoned when piston rod seals were found to seal hydrogen better than expected.

## 1967 GPU 3-3 DURABILITY TEST

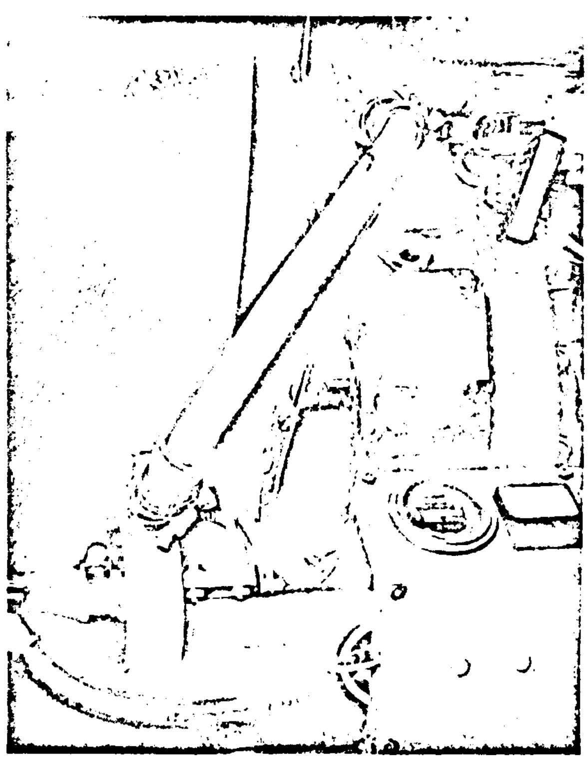
| | |
|---|---|
| Longest run without service | 196 hrs. (one normal working month) |
| Longest run without stopping | 159 hrs. (6.6 days) |
| Longest run without removing cylinder | 277 hours |

### Failures Ending Service-Free Periods

| | |
|---|---|
| Fuel Pump Drive | 5 |
| Governor Hydrogen Valve | 4 |
| Nozzle Air Pump | 3 |
| Accessory Belts | 2 |
| Hydrogen Compressor | 2 |
| Alternator Drive | 2 |
| Cooler Tube Leak | 1 |
| Piston Rod Seal | 1 |
| Hydrogen Explosion | 1 |
| Unknown | 1 |

Some comments on the failures noted above are appropriate. First, as noted above the governor hydrogen valve and the hydrogen compressor have been constantly improved during the test period. The hydrogen compressor is currently operating 335 hours since its last modification, and the governor hydrogen valve has operated 607 hours since its last service.

The fuel pump drive failures, the alternator drive failures, and the nozzle air pump failures are probably related to torsional vibrations introduced into the accessory drive by the non-uniform angular motion of the single-cylinder GPU engine.



The hydrogen explosion occurred during unattended operation, and is thought to have been caused by a static electric spark jumping from an ungrounded pulley to the engine frame. It caused relatively little damage, and the engine eventually stopped because the crankcase front cover was distorted, preventing operation of the water pump which in turn actuated a water pressure safety switch and stopped the unit. The damaged cover can be seen in the photograph.

## FRICTION STUDIES

Motoring an engine is the procedure for determining friction power of an internal combustion engine. It is not a true picture since bearing loads are different from an engine in which combustion is taking place. In order to determine friction power of a Stirling engine, motoring in the IC engine sense is not possible since driving a Stirling engine in the normal rotation results in a refrigerator and absorbs power of the same magnitude as the engine power. Driving it in the reverse direction results in a heat pump which absorbs power. To properly motor a Stirling displacer type engine, it is necessary to replace the displacer piston with a dummy piston of equal weight but having large clearance so as to produce negligible pumping effect. Then by charging the engine to the normal pressure and motoring, the bearings are loaded in a manner similar to, but not identical with, the normal engine; but, in any event, a close approximation to the true mechanical efficiency can be found by the procedure.

In 1961 GMR was trying to determine by careful motoring tests why the Philips built engine required only 62 percent of the power required by the GMR built engine having nearly identical dimensions.* The one main difference was a floating bushing between the connecting rods and crankshaft of the Philips model. In discussions with the bearing experts in the department, it was brought out that the load reversals in a Stirling, quite unlike most IC engines, tend to center the bushing, resulting in greater total oil film thickness and reduced friction. Bearing friction calculations were made for the two cases, and the results are shown on the curves. The measured curves include wrist pin and vertical rod bearing losses which were not included in the calculations. The floating bushing is certainly useful, but the practical application requires a split crankshaft, as was employed by Philips. This drastically increases cost. The only other solution is some kind of cemented bushing, which, to the author's knowledge, has never been successfully applied.

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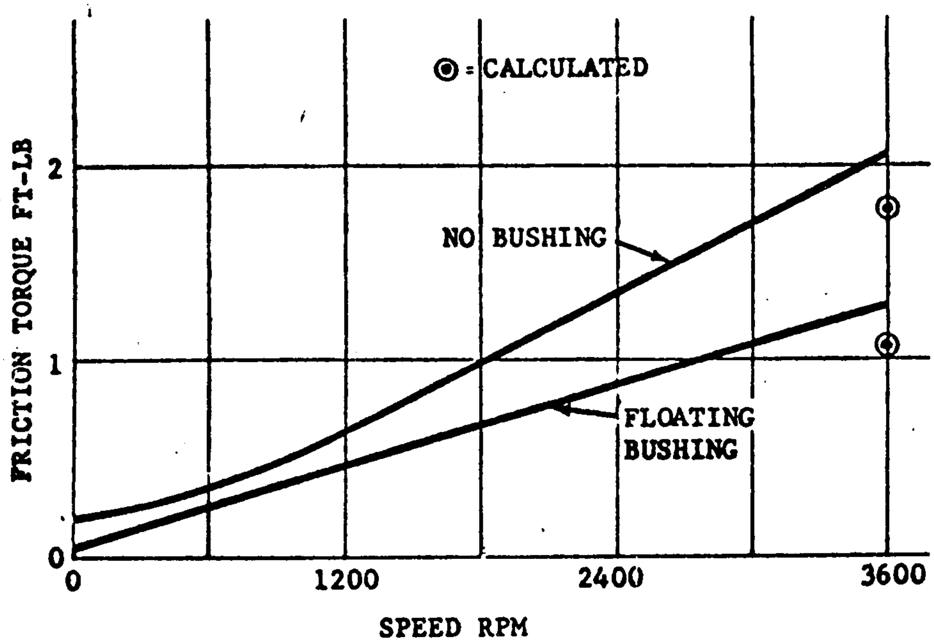
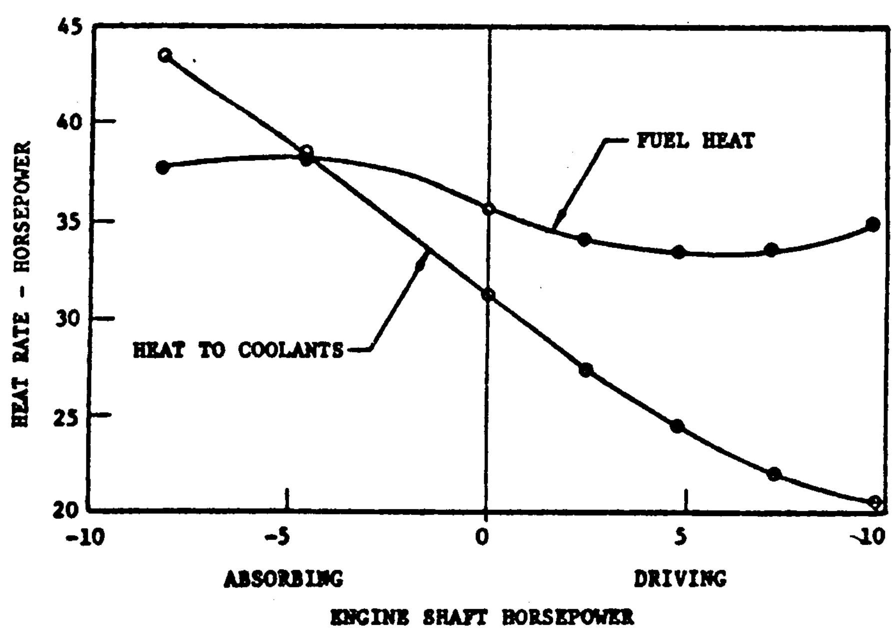
* The dynamometer used for this work was equipped with hydrostatic air bearings on the cradle -- a special arrangement for accurate measurement of small torques.




In 1968 some tests were made of the ability of the Stirling to quickly absorb power -- normal motoring as was pointed out, absorbs power, but not until the heater temperature has dropped below ambient. By installing an extra large by-pass valve between the working space and the buffer space (larger than is used for normal engine governing), it was discovered that the engine could absorb power nearly equal to its full power output -- at the instant the valve was opened wide. This feature makes the engine attractive for vehicle braking. The curves below show the effect on the fuel rate and heat to coolant for the two conditions.



## NOISE AND VIBRATION

Stirling engines are "quiet" engines for the following reasons:

1) Intake of air and exhaust of combustion products are steady flow processes instead of periodic as in the IC engine. Consequently, there is little "coupling" to the atmosphere to produce air borne vibrations. In addition, the air preheater serves to muffle combustion noises.

2) Cylinder pressure changes are very smooth in comparison to the sharp changes in IC cylinders. Therefore the cylinder walls are not excited into vibration.

3) No valve gear is present.

4) No fuel injection system is present.

5) The drive mechanism can be designed to be in complete balance, as in the rhombic drive.

On the other hand, Stirling engines are not "silent" engines. Since they have bearing systems with finite clearances, gears for proper timing in the rhombic drive, and require accessories such as blower, fuel pump, nozzle air pump, water pump, generator, etc., there are numerous sources of noise which can be objectionable. Then, too, one source of vibration can excite other parts to vibrate, as for example if the engine is improperly balanced, or if torsional vibration is severe enough to excite the fly wheels or other driven parts.

Early in the program with the first Army GPU, it was obvious there was enough noise from the complete package to prevent achieving the Army goal of "inaudible at 100 feet."

A thorough investigation of the noise sources was made by separate dynamometer tests of the GPU accessory package and of the engine itself. Some noise sources were found in the accessories, but most of the sounds arose as reverberations excited by vibrations from the engine. Surveys of the engine noises revealed a significant "rap" once per revolution. For some time, it was believed to be the result of sudden movements of the crankshaft within the main bearing clearance spaces as the loads changed during each cycle.

Beginning with the GPU, the noise study was expanded to include a Dutch 10 hp engine, the 30-15 Dutch engine on which structure-borne and air-borne recordings had been made in a Navy test in 1960, a 4 cylinder, 300 hp Dutch engine on which Philips had made a sound tape, and finally the Allison PD46 space power engine. All engines showed the characteristic rap, with the 4 cylinder engine having 4 per revolution.

By 1963 it was believed that the rap was coming from the helical timing gears rather then from the bearings. A one-third octave band analysis of the GPU was made after vibration isolation mounts were installed and stiffer couplings used between the engine and generator. Two pure tones were prominent, one at the frequency of the fan blades and the other was believed to be the timing gears. A study was begun of the effects of different gear designs including nylon teeth, aluminum spur gears, extra fine steel teeth, and a set of herringbone with the two sides offset half a tooth. A flywheel resonance at near the 3600 rpm operating speed was also discovered; it was cured by building new ones of a laminated construction.

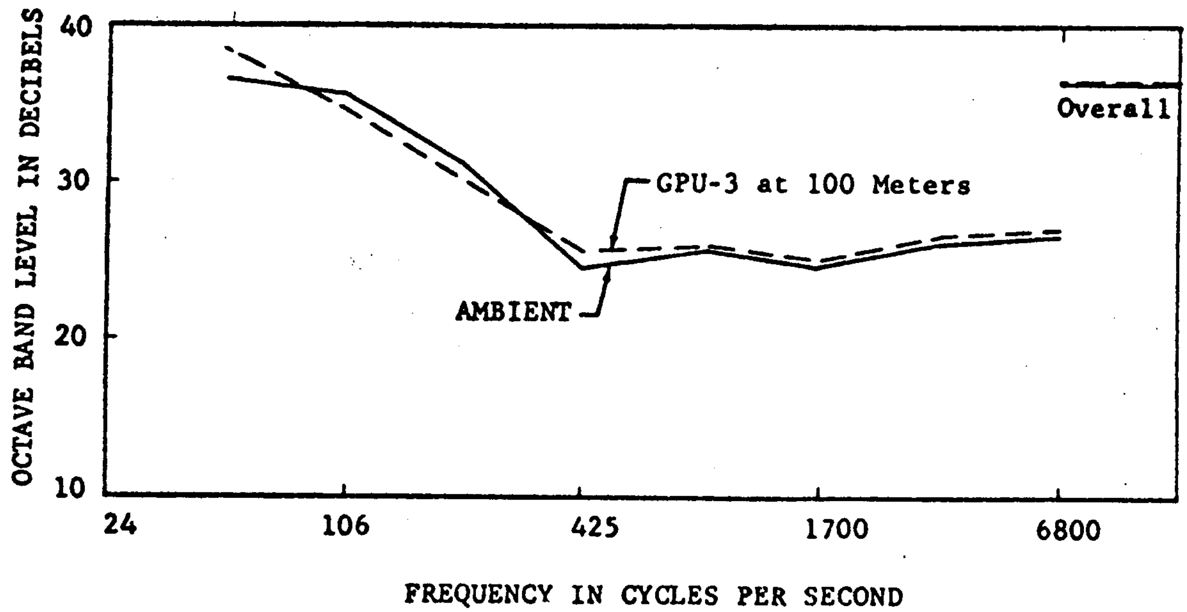
In 1964, a more extensive analysis of noise was made in the anachoic chamber at the GM Proving Ground with the help of their noise and vibration staff specialists. These tests confirmed the gears and fan as the predominent sources, with the nozzle air pump and combustion air blower as important if the others were eliminated. Eight different gear sets were run in an engine; from these, only two looked promising, a pair with nylon teeth and a steel set with very fine teeth. A relatively simple method was developed for mechanically bonding the nylon rim to an aluminum hub. A test fixture was built to load the teeth as in the engine; after 782 hours at elevated temperature, no failure occurred in the teeth or hub bond.

A single crankshaft engine was designed and built in 1964 and tested in 1965, and a special rhombic drive engine was tested in 1965. The latter had a heavy-duty crosshead which enabled it to operate as a conventional rhombic drive either with timing gears or without timing gears; also as a single crank half-rhombic drive. None of the variations was acceptable. The single crank engine was certainly quieter than the standard rhombic when all accessories were prevented from shaking; but because of the time required to find the optimum balancing and mounting and because accessories and instrumentation were subject to periodic breakage from the severe vibration, the project was stopped.

From a study of flywheel inertia distribution between the loaded and unloaded crankshafts, it was found that 25% of the <u>total</u> inertia placed on the unloaded shaft was sufficient to reduce gear tooth noise to a minimum. A further test was made of torque effects on gear noise by installing an AC generator with a $\sin^2$ load curve and comparing to a constant load DC generator. Changes were noted but nothing of consequence could be correlated with the nature of the load.

A cast iron crankcase was compared to the standard aluminum GPU crankcase for the first time in 1965. A reduction of overall sound level of 6 db (10 feet) at 2400 rpm was significant, but the extra weight prohibited using cast iron.

In 1966, the GPU-3 met the new Army requirement to be inaudible at 100 meters in a quiet environment. Results of the test are shown in the curves.



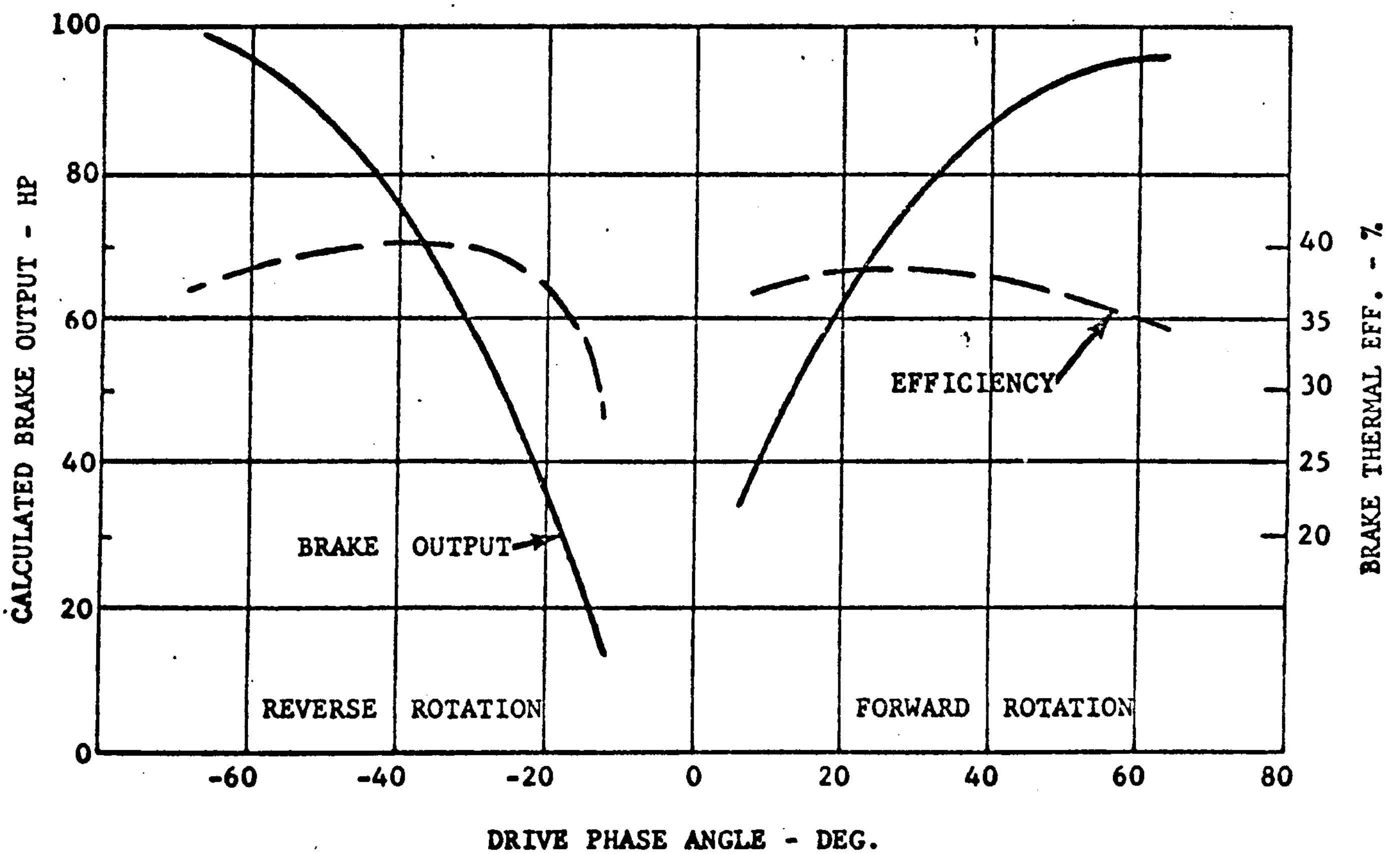

The only noise and vibration study of any significance after 1966 involved a gear resonance study on the Stir-Lec II hybrid-electric car, in 1969. A change in flywheel inertia corrected the problem.

## CYCLE ANALYSIS

The basic Stirling cycle analysis was developed by Philips during the late 1950's and was constantly improved during the 1960's until it was possible to optimize engine designs by 1967. The basic calculation procedure, loss analyses, and computer programs are highly confidential and proprietary to NV Philips.

Early contributions from GM consisted of special programs, for example, to study applications of thermal storage and torpedos. In 1963, GMR programmed the Philips' calculation prodecure for the IBM 7094 computer. Subroutines were developed which generated pressure, drive dynamics, drive forces, torques, and bearing loads as a function of crank angle; special drives were studied for Electro-Motive Division. Some interesting results for the case of variable phase-angle drive and reversible drive mechanism are shown on the graph below. EMD built a 4 cylinder 400 hp variable phase-angle engine and tested it in 1965.

FORWARD AND REVERSE PERFORMANCE CHARACTERISTICS
CALCULATED FOR A VARIABLE PHASE ANGLE STIRLING ENGINE



At GMR in 1965, the computer program was brought up-to-date and re-written in Fortran IV. Some effort went into reducing computer time per engine calculation -- the lowest time achieved was 0.3 seconds. Analyses were made of 2 Stirling proposals outside of GM, the Stratos-Dinnen engine and a liquid-vapor engine proposed by Dr. G. Walker of I.I.T. The first was found to be incapable of producing the power hoped for; the second was found not to be a Stirling cycle at all and would not be able to achieve the efficiency predicted for it.

In 1966 GMR discovered a basic error in the Philips procedure for predicting cooler performance. Philips used ideal cooler heat flux to establish cold end temperatures from the assumed cooler tube wall temperatures. The actual cooler heat flow was measured to be 40% higher than the ideal. With this and some other minor corrections the cold space temperature calculated was in close agreement with those measured. Using the new calculation procedure, the design of the GPU type engine coolers was re-examined. They were found to be too short and should be lengthened by at least 50 percent to improve both power, efficiency, and reduce cold space gas temperature.

An improvement in kinematic analysis was made at GMR. Philips' analysis was based on sinusoidal piston motion; for the rhombic drive, so called "higher harmonic" corrections were introduced which are semi-empirical, but not accurate for the new (1966) EMD variable phase drive. The Philips analysis was therefore reprogrammed so that any drive system could be analyzed without simplification errors.

Some typical comparisons between actual test data and calculated performance, as was analyzed in 1966, are given in the Table on the next page.

Numerous studies were made for Electro-Motive Division in 1967 of their new "W" engine having one double-acting power piston and two separated displacer pistons. Cold end duct flow losses were predicted to be rather severe--amounting to 10 percent of the engine gross power. Additional thermodynamic studies were made to generate optimum engines, using Met-Net regenerators, to follow the "W" design.

From 1967 to March, 1970, studies were made for the government (classified contracts) of a small, light weight, "V" engine of 2 hp, at 5000 rpm; and a 9 hp, 4 cylinder swash plate engine. Both were to operate from non-combustion heat sources.

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

## COMPARISON OF CALCULATED STIRLING ENGINE PERFORMANCE WITH EXPERIMENTAL DATA

Independent Variables: Mean Working Pressure; Temperature of Cooling Water; Speed (RPM); Temperature of Heater (T. Htr.) °F

Dependent Variables: Brake Horsepower (BHP); Brake Thermal Efficiency - % (Eff); Heat Rejected to Cooling Water - B/Min (Ht.Wtr.); Temperature of Gas in Engine Cooler - °F (T. Clr.)

Mean Working Pressure - 1000 psi
Cooling Water Temperature - 125° F

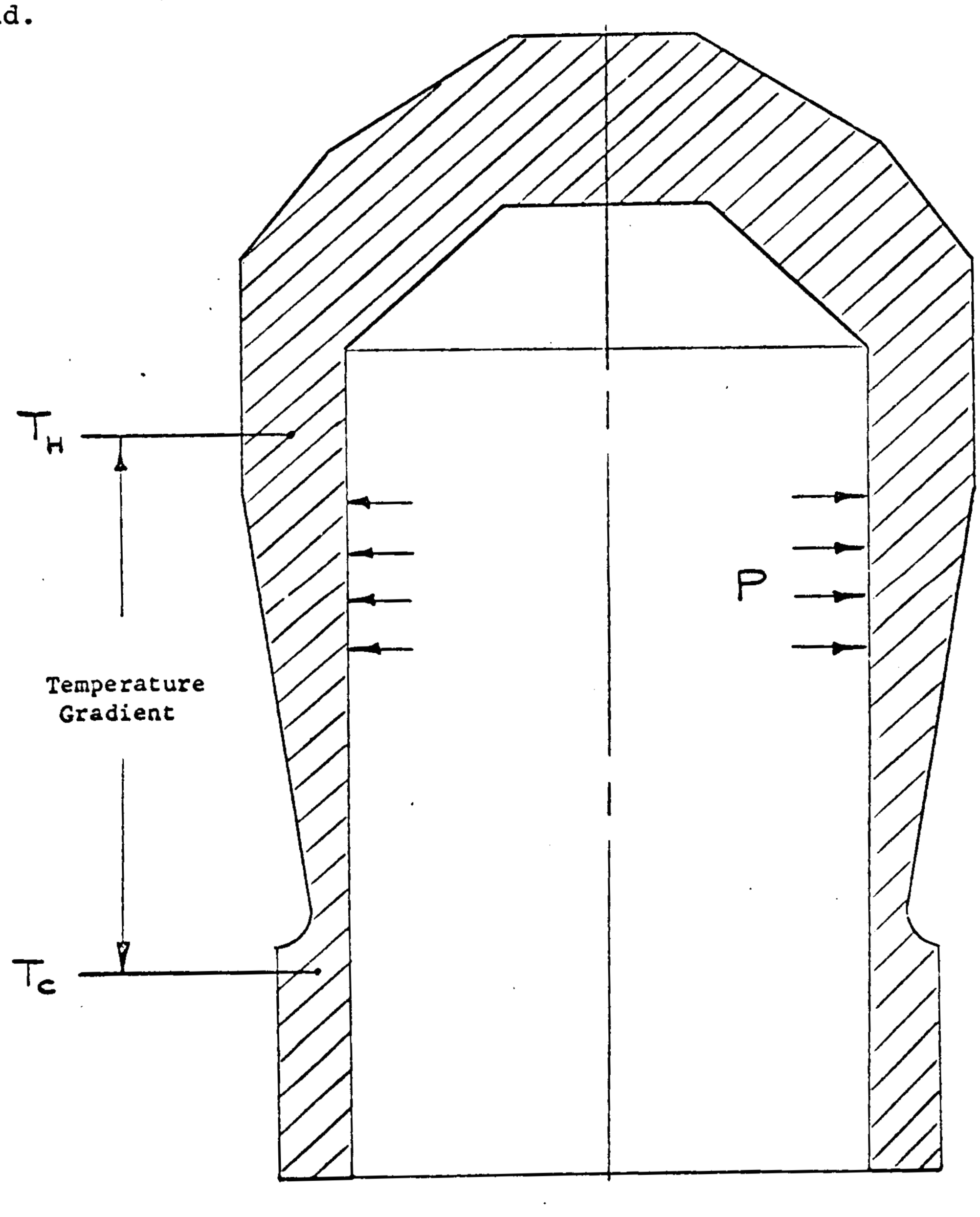
| | Meas. | Calc. | Accuracy | Meas. | Calc. | Accuracy | Meas. | Calc. | Accuracy |
|---|---|---|---|---|---|---|---|---|---|
| RPM | 2780 | 2800 | | 2400 | 2400 | | 1810 | 1800 | |
| T. Htr. | 1278 | 1279 | | 1296 | 1282 | | 1265 | 1286 | |
| BHP | 8.97 | 9.53 | 94.2% | 8.17 | 8.52 | 95.9% | 6.25 | 6.64 | 94.1% |
| Eff. | 26.6 | 27.9 | 91.7% | 27.3 | 28.7 | 95.2% | 27.7 | 29.2 | 94.7% |
| Ht. Wtr. | 792 | 742 | 93.7% | 691 | 637 | 92.2% | 504 | 489 | 97.0% |
| T. Clr. | 247 | 237 | | 237 | 230 | | 222 | 221 | |

Mean Working Pressure - 800 psi
Cooling Water Temperature - 125° F

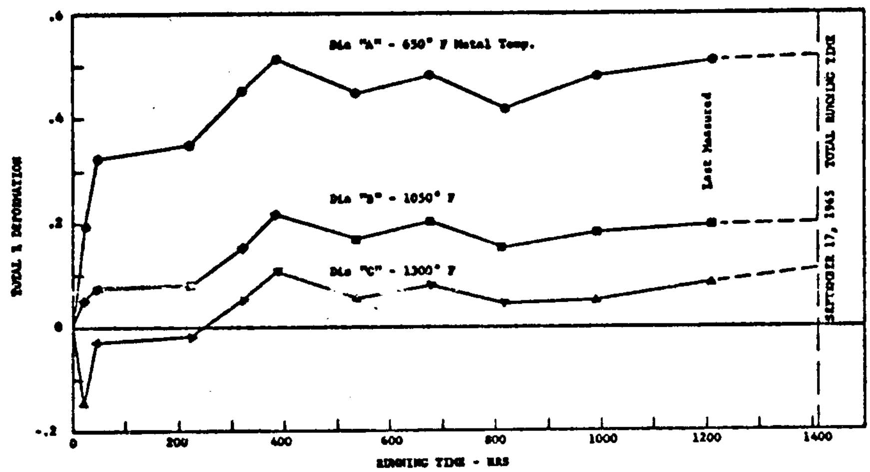
| | Meas. | Calc. | Accuracy | Meas. | Calc. | Accuracy | Meas. | Calc. | Accuracy |
|---|---|---|---|---|---|---|---|---|---|
| RPM | 2775 | 2800 | | 2395 | 2400 | | 1810 | 1800 | |
| T. Htr. | 1285 | 1283 | | 1279 | 1285 | | 1270 | 1289 | |
| BHP | 7.15 | 7.68 | 93.1% | 6.57 | 6.86 | 95.7% | 5.13 | 5.34 | 96.1% |
| Eff. | 26.2 | 27.6 | 94.8% | 27.3 | 28.4 | 96.1% | 27.3 | 28.7 | 95.2% |
| Ht. Wtr. | 645 | 598 | 92.7% | 562 | 516 | 91.8% | 441 | 400 | 90.6% |
| T. Clr. | 234 | 228 | | 226 | 222 | | 214 | 216 | |

DISPLACER (HEATER) CYLINDER STRESS STUDIES

The Stirling engine displacer cylinder, of the type used from 1957 through 1969 by Philips and GM, is a pressurized vessel whose walls act as a thermal barrier between the hot and cold regions of the engine. A typical cross section is shown below. Although the cylinder loading is simple in nature (internal pressure and an axially varying wall temperature), it produces a complex tri-axial stress state resulting from tangential (hoop), radial, longitudinal (axial) and radial shear stresses. Knowledge of these stresses, in conjunction with the resulting cylinder wall deformation, was required for analyzing the designs at that time as well as proposed future designs. In the GPU engines, the temperature at the top of the cylinder was about 1350°F while the bottom, only 3 inches away, was near 200°F. Internal pressure at full load cycled between 700 and 1400 psi. Before the more general application of finite element analysis in the middle 60's, attempts had been made at Philips and GM to solve the governing differential equations written for an infinitesimal element of the cylinder wall; an exact mathematical solution was not found.



In 1965 the cylinder for the forthcoming GPU-3 was re-designed for a larger bore so as to produce more power for higher ambient conditions not encountered with GPU-2 testing. Because of strong doubts about the new cylinder's strength and lack of means for calculating the stresses, a test run was made of the experimental cylinder. It was intended to last only 500 hours, but actually ran 1400 hours at 125% of GPU-3 rated load, with measurements of bore deflection being made every 100 hours. As can be seen in the curves of percent deformation, for the first 400 hours the distortion was rapid. Afterwards the creep rate stabilized, and the total change in cylinder diameter remained in the 0.4 to 0.5 percent region.




The type of step-wise yielding exhibited by the displacer cylinder has been referred to in the literature as "cyclic-strain induced creep" or "cylinder ratcheting." The stabilized creep rate shown in the curves may indicate a type of stress relief by yielding which reduces the possibility of failure by creep over a long period of time.

Improvements which were incorporated into the GPU-3 displacer-cylinder assembly and combustion system components were: Elimination of controls from the top of the heater by installation of all accesses to the hydrogen circuit at the bottom of the cooler where the mounting is more rigid; use of unit cooler-regenerator assemblies to reduce the number of parts to be handled in assembly; improvement of design details so that the entire assembly can be brazed in one furnace operation instead of the two previously required, and with a reduced number of brazing fixtures; installation of the optimum cylinder material for best piston ring life; better flow passages for combustion gases to utilize more of the heater tube surface; revision of the burner-cylinder-preheater complex to allow freer movement of the combustion chamber under thermal expansion forces while creating less opportunity for hot gas leaks in the combustion chamber zone, and also permitting proper mating of the parts with less strict manufacturing tolerances; and arrangement of the cylinder exterior insulation for quicker assembly and a cleaner appearance.

In 1967 a finite element analysis was made and a computer program was made available for stress and deformation studies of the displacer cylinder. The analytical results were compared to known exact solutions to simpler cases with good success. Also, an elaborate test program was proposed for further checking the calculation procedure, but it was never done because of lack of manpower.

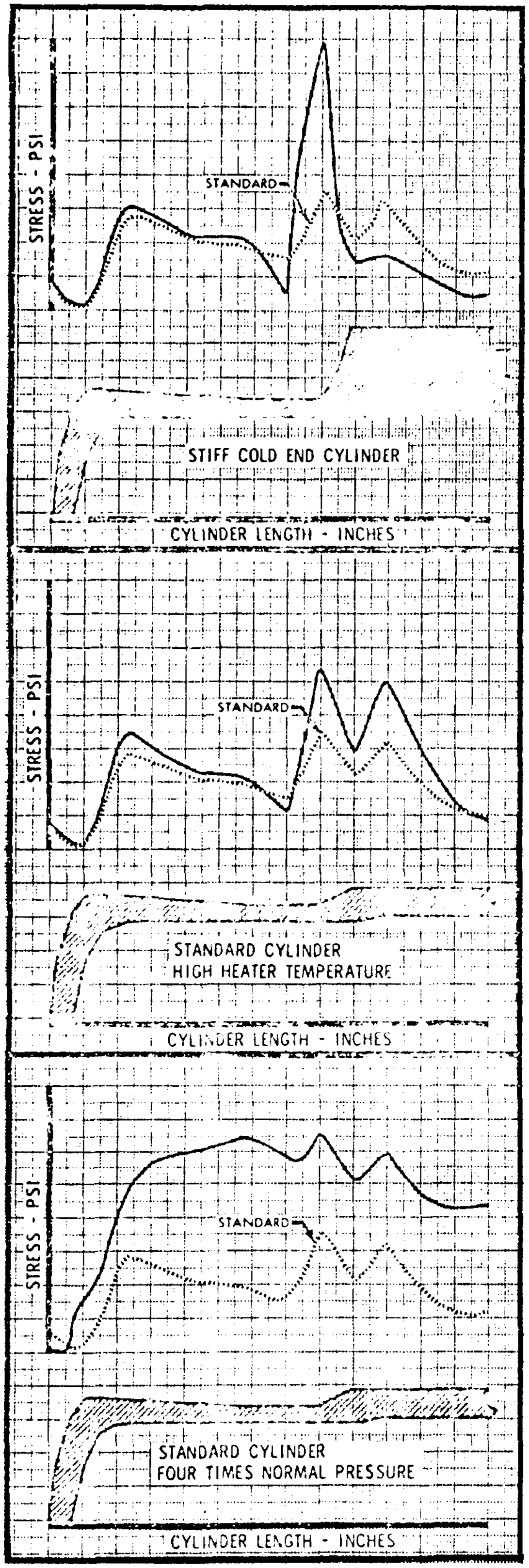
A brief study, in cooperation with Pontiac Motor, was made in 1967 of displacer cylinder costs. Pontiac was studying the potential of the Stirling in a small car because of its low emissions. The final heater design included simplified cylinder and regenerator cups, simplified heater tubes and extensive use of cast iron. An outside supplier of investment castings provided a quotation for the cylinder and regenerator cups of 347 stainless steel, accurate to 0.005 inch/inch. The costs varied from $400 per engine in quantities of 50, to $83 per engine in quantities of one million engines.

In 1968, the finite element analysis of displacer cylinders was expanded to examine the effects of arbitrary design changes on stress levels. Six of these studies are summarized on the next two pages. In each case the stress in a standard cylinder under full load operating conditions was compared to the stress with the special condition under study. The cylinder was the EMD, 80hp model. Full load internal mean pressure was 1500 psi; hot end temperature was 1200°F and cold end temperature was 200°F.

The relative stresses are plotted with a cross section of one-half of the cylinder to orient the stress pattern. The water jacket surrounds the cylinder to the right of the offset (I.D.). The plotted stress is the Von Mises equivalent stress on the inside of the cylinder wall. The calculation is based upon an adiabatic temperature profile which was computed for each case using actual wall areas, and the temperature dependent properties of 316 stainless steel.

A somewhat puzzling but gratifying aspect of the displacer cylinder hardware at Philips and GM was the fact that not one cylinder ever ruptured -- it was hard to explain because GM and Philips believed the stress in the hot section was near the stress rupture limit. Twenty-four cylinders were built at GMR, about 10 were built by EMD and at least 50 built at Philips over a 15-year period. All GMR cylinders were made of 310 stainless and the tubes were originally 310 steel; but eventually all tubes were made of Multimet (N-155).* The accompanying figure for the GPU-3 cylinder shows the maximum hot stress (regardless of pressure) to be about <u>3 times</u> the 1000-hour stress-rupture valve for 310 stainless! Two GPU cylinders operated over 4000 hours and two over 2500 hours without failure.

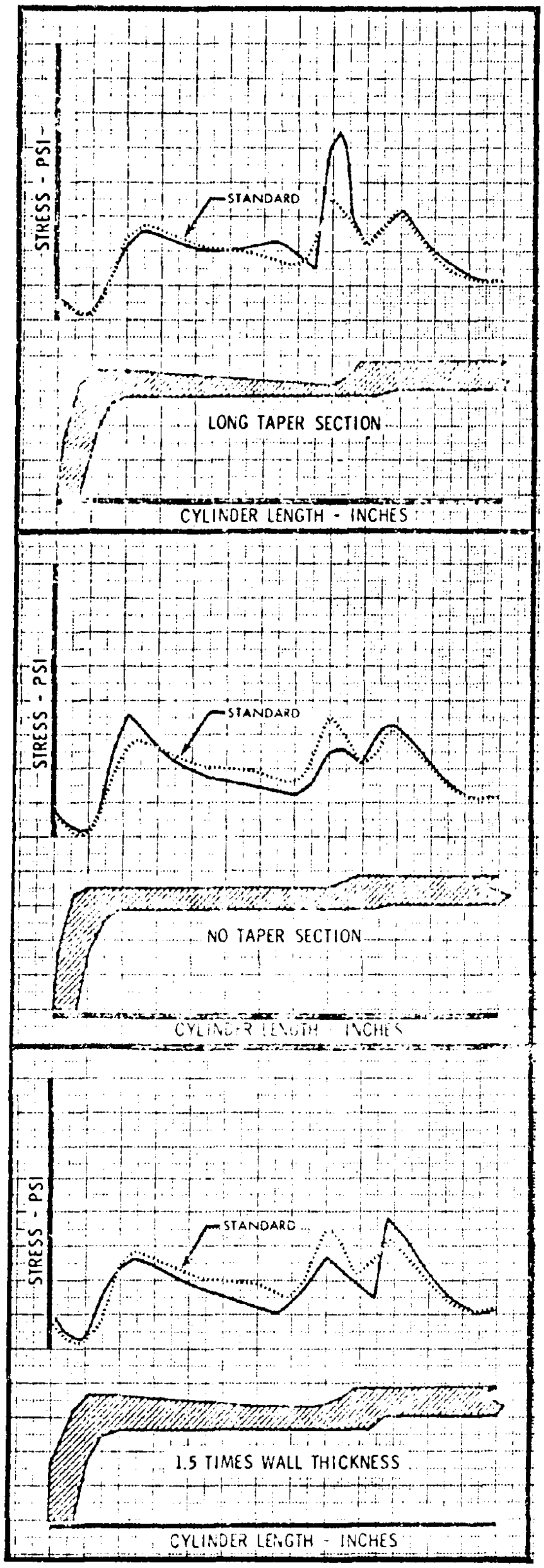
* N-155, a nickel-cobalt alloy, was selected because experience had shown that tubing of this material was practically 100% reliable---free of inclusions and other defects---and was manufactured according to ASM Specification. The consequence of failure of one tube in a heater assembly after final brazing was the scraping of the entire unit, which was the most expensive part of the engine.




Of the design changes studied, the biggest influence on stress results from making the cold end of the cylinder thicker and stiffer. This reduces the flexibility of the cylinder and imposes high bending stresses in the region where deformations are normally found in test engine cylinders.

The next largest effect comes from increasing the temperature gradient. In the example, this was accomplished by raising the temperature of the hot end of the cylinder to 1400°F.

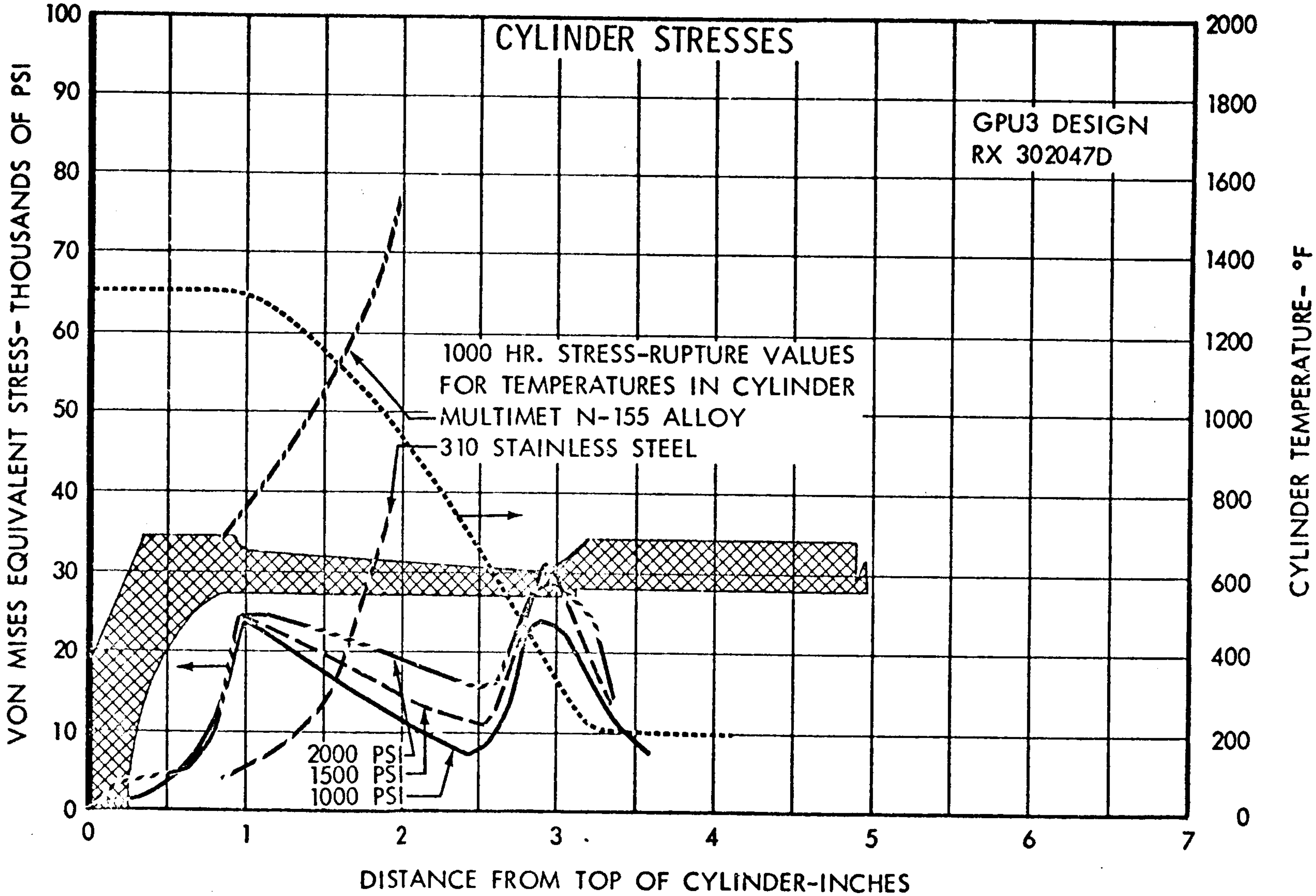
Increasing the internal pressure raises the general stress level as might be expected, but produces no sharp stress concentrations or large variations in the pattern. It is also noted that the stress level is less than doubled for a four-fold increase in pressure.




The shape of the cylinder wall is frequently under scrutiny when manufacturing costs are being considered, and these three examples would indicate that the wall contour is not too critical. Here the taper section has been extended so that the uniform thickness part of the wall is eliminated. This causes a stress increase at the cold end as a result of the wall thickness being smaller at that point. This long taper section was actually the original design of the 8015 cylinder, but it was changed to the current standard shape when large deformations were noted at the point where the high stress is indicated in the graph.

A uniform wall is easier to make and has the effect of increasing the stress somewhat at the hot end where the material has the lowest creep strength.

Multiplying the wall thickness by 1.5 uniformly reduces the stresses as might be expected.

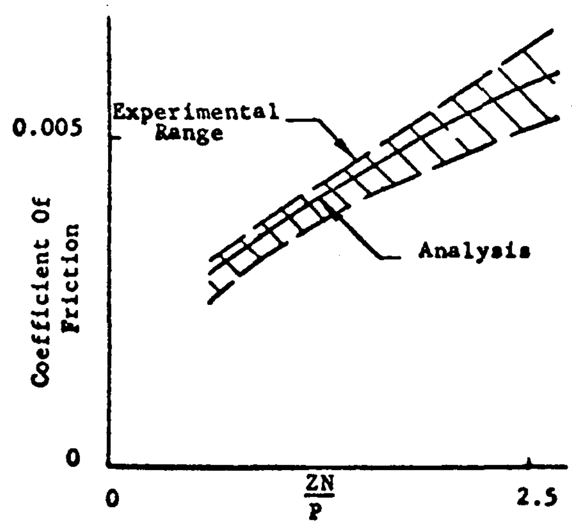

CYLINDER STRESSES
GPU3 DESIGN
RX 302047D
1000 HR. STRESS-RUPTURE VALUES
FOR TEMPERATURES IN CYLINDER
MULTIMET N-155 ALLOY
310 STAINLESS STEEL
2000 PSI
1500 PSI
1000 PSI
VON MISES EQUIVALENT STRESS- THOUSANDS OF PSI
CYLINDER TEMPERATURE- °F
DISTANCE FROM TOP OF CYLINDER-INCHES
0
10
20
30
40
50
60
70
80
90
100
200
400
600
800
1000
1200
1400
1600
1800
2000
1
2
3
4
5
6
7

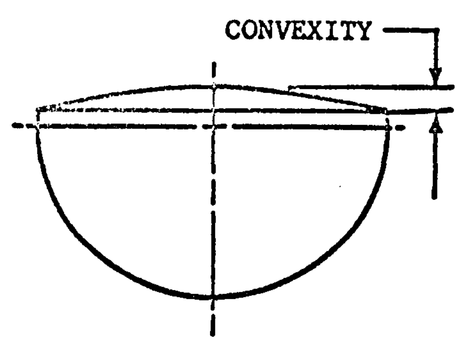
COMPACT ENGINE STUDIES
SWASH PLATE DRIVE AND DOUBLE ACTING PISTONS

Reference #6 discusses compact engines in general and some of the swash plate drive work at GMR. Reference #7 goes into the history of swash plate drives and covers the analysis and test experience at GMR in greater detail.

The swash plate drive concept for Stirling engines was initiated by the Philips Laboratory in the 1940's; it was abandoned for unknown reasons. It was revived by GMR in 1965 for the purpose of obtaining U.S. Navy interest in advanced torpedo drives. The bearing group of the Mechanical Development Department was asked to analyze the bearing and swash plate friction losses. This was accomplished in 1966 for a 6 cylinder 500hp engine design. The analysis showed that the total power consumed by all the slider bearings, cross head bearings, main bearings, and thrust bearings was under 15 horsepower; also the bearing loads, temperature rise, and film thicknesses were all within standard design limits. The only question remaining was the effect of the relatively high surface speed of the slider bearings which bear on the swash plate.

In 1967 it was established that the agreement between measured and predicted slider bearing characteristics was excellent; this can be noted in the curves. The view of the slider and amount of convexity is also shown.




Slider convexity was varied from 0.0001 in. to 0.0036 in. It was established that a convexity less than 0.00025 in. would result in increased friction. An upper limit on the amount of convexity was not found within the range investigated.

In 1967 great interest in compact Stirling engines was expressed by EMD, who began construction of a double acting version of a displacer engine called the "W" model; also by Pontiac, Oldsmobile, Truck and Coach, and Engineering Staff -- the last 4 were stimulated by possibilities of low emissions and low noise.

Engineering Staff considered the Stirling for a series of golf-cart like vehicles for shopper's cars. A full scale plastic model of a 25hp swash plate Stirling engine with accessories, transmission and vehicle drive line was constructed for study. Oldsmobile and GMR cooperated in a study of a 250hp Stirling swash plate engine in a Toronado. The Truck and Coach Division requested information on a 150 hp Stirling engine for a new bus for New York City. It was alleged that city officials gave management assurances they would pay up to $15,000 more per bus for a replacement for the diesel engine. The Stirling's advantages over the diesel were less noise, less smoke, less odor, lower emission of hydrocarbons and $NO_x$, and, essentially, no oil consumption; also a thermal energy storage heat source had been considered. GMR undertook a short study of matching a Stirling engine to a vehicle -- not a hybrid such as the Stir-Lec.

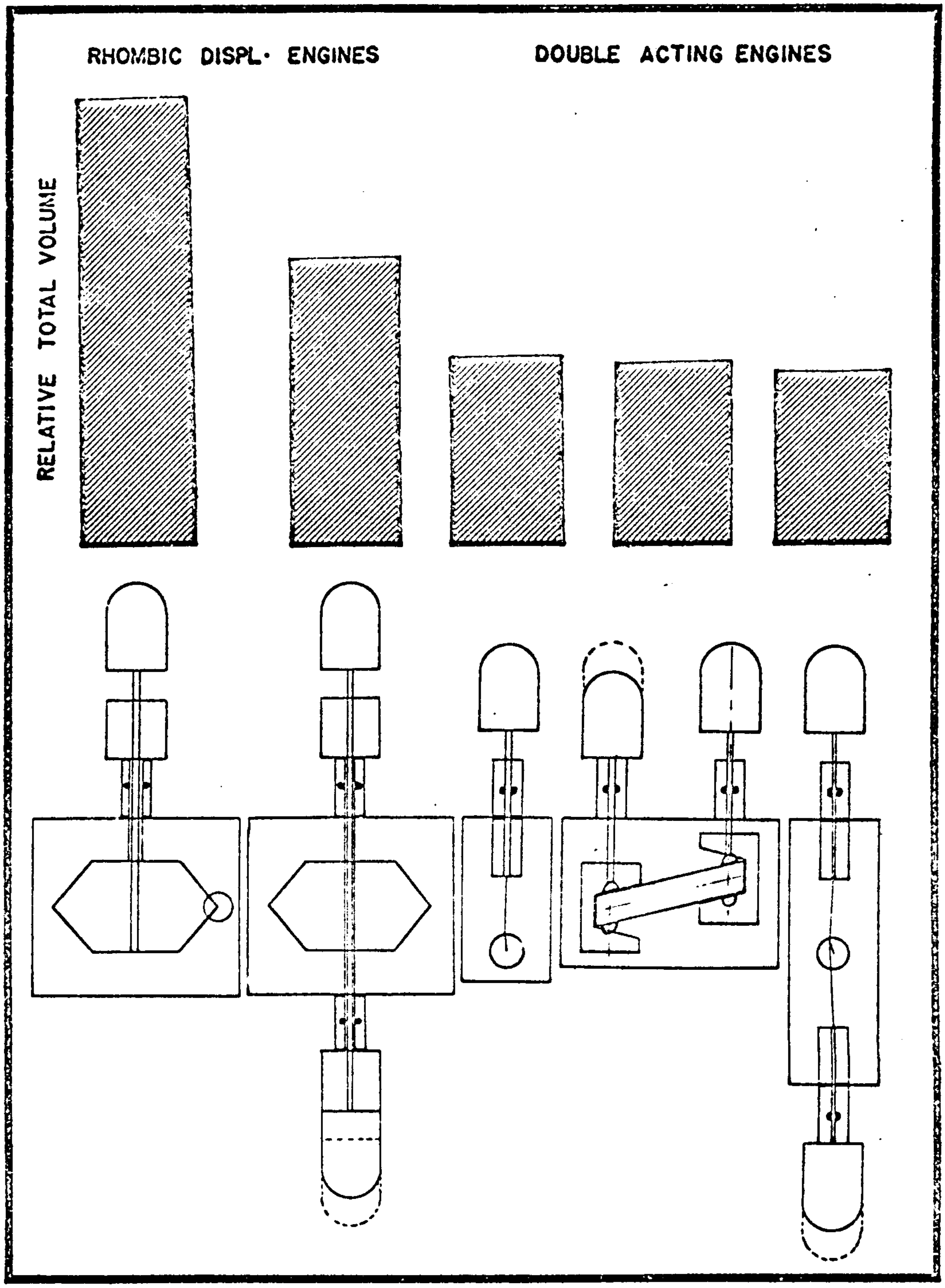
In 1968, the compact engine study was expanded to include five fundamental arrangements of multi-cylinder engines. The preliminary survey indicated that double-acting swash plate, in-line, and opposed-piston ("Boxer" in European parlance) engines would be about the same size, and that double-acting engines would be near one-half the size of either rhombic in-line or rhombic Boxer engines.

A more detailed design study of the three small double-acting configurations was then undertaken. The study involved the virtually complete design of each engine, and was arbitrarily based upon four-cylinder engines of 120 horsepower total output.

After some work was done, it was seen that the single-crank Boxer engine would be about the same overall volume as the single-crank in-line engine; and in the size selected, the Boxer introduced balancing and other mechanical complexities. The Boxer was, therefore, set aside and a Scotch Yoke in-line study carried through in its place.

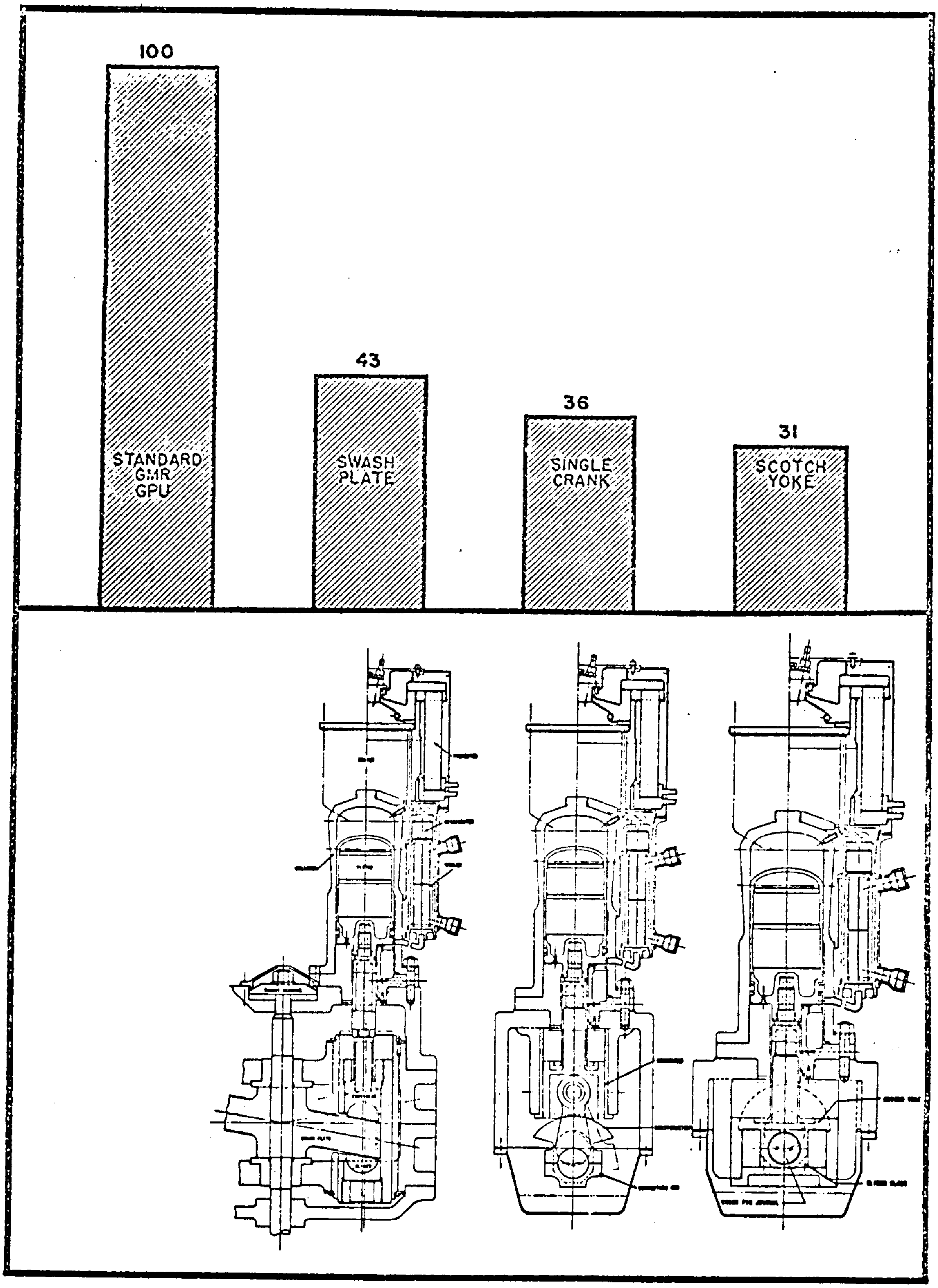
The results of both parts of the compact engine study are shown on the following pages. The size of the GMR standard engine is added as a final basis for comparison.

From the results of the study, it can be concluded that double acting engines are more compact than rhombic drive engines; but that the swash plate drive engine is not necessarily the most compact of the double acting designs, particularly if "packing-crate" volumes are compared. This is quite different than a comparison for a torpedo drive engine, for example. It must be remembered, however, that the study was made in 1968 and may not hold true today.




Preliminary Size Survey Results

**Left to Right: 1. Rhombic In-Line; 2. Rhombic Boxer; 3. Single Crank In-Line; 4. Swash Plate; 5. Single Crank Boxer.**




Final Size Survey Results

Relative Total Volumes of Equal Horsepower Double-Acting Engines

COMPACT ENGINE STUDIES

FOUR CYLINDER COACH ENGINE

A four-cylinder double-acting engine design was started in 1968 for the purpose of eventually demonstrating an advanced Stirling engine of about 150 horsepower. In the beginning it was by no means certain as to what kind of demonstration -- in fact, GMR management first decided it should be only a research engine; later it was to be installed in a boat, but early in 1969 they switched to a passenger coach. It was to be a part of GMC Truck & Coach Division's new demonstration program which included gas turbines as well as new diesel engine powerplants.

The engine became known as the 4L23 because of the piston displacement of 23 cubic inches. A single crankshaft was used, with crossheads, and only one piston per cylinder was needed. The crankcase incorporated balancing shafts to eliminate the inherent pitching moment. The initial full load design point was 2000 rpm and 1500 psi mean working pressure of hydrogen. Helium had been considered but the penalty was considered too great. In order to provide for future progress on the engine "hot parts" and combustion system, the crankcase, crankshaft and bearings were designed to accept 3000 psi mean pressure. The 4L23 was GMR's first computer designed (optimized) engine.

Some other features included a hollow sealed piston dome in place of a piston having a small bleed hole, which was standard with all engines built before. This required a careful compromise between sufficient internal ribbing to prevent buckling and excess weight. The final design was tested with external pressure and found to be satisfactory for the standard 1500 psi; but not "double" pressure operation. Eliminating the bleed hole reduced the total inventory of hydrogen, which otherwise would have been required to fill each piston at maximum working pressure. This feature reduced the danger associated with hydrogen -- it was not a serious problem, but in discussions with management, the question of safety was often asked.* (See Appendix B)

The 4L23 was the first engine to be optimized for use with Met-Net regenerator material. Twenty-four assemblies were built and a special procedure for flame-spray bonding the edges of the Met-Net discs was established.

The connecting rod and crosshead were fatigue tested in the GMR hydraulic fatigue test machine for 10 million cycles at full simulated load without failure.

* It should be remembered that this occurred before the "hydrogen economy" proposals became so popular, after which the fear of hydrogen was considerably mitigated.

The 4L23 was equipped with rolling seals. Only one seal per cylinder was required instead of two in the rhombic drive designs; but the pressure was higher than for the GPU and the cyclic pressure variation was computed to be 2 to 1 compared with the 1.2 to 1 in the GPU engines. A test apparatus simulating these conditions was built and used to design the control system and seal support hydraulic circuit.

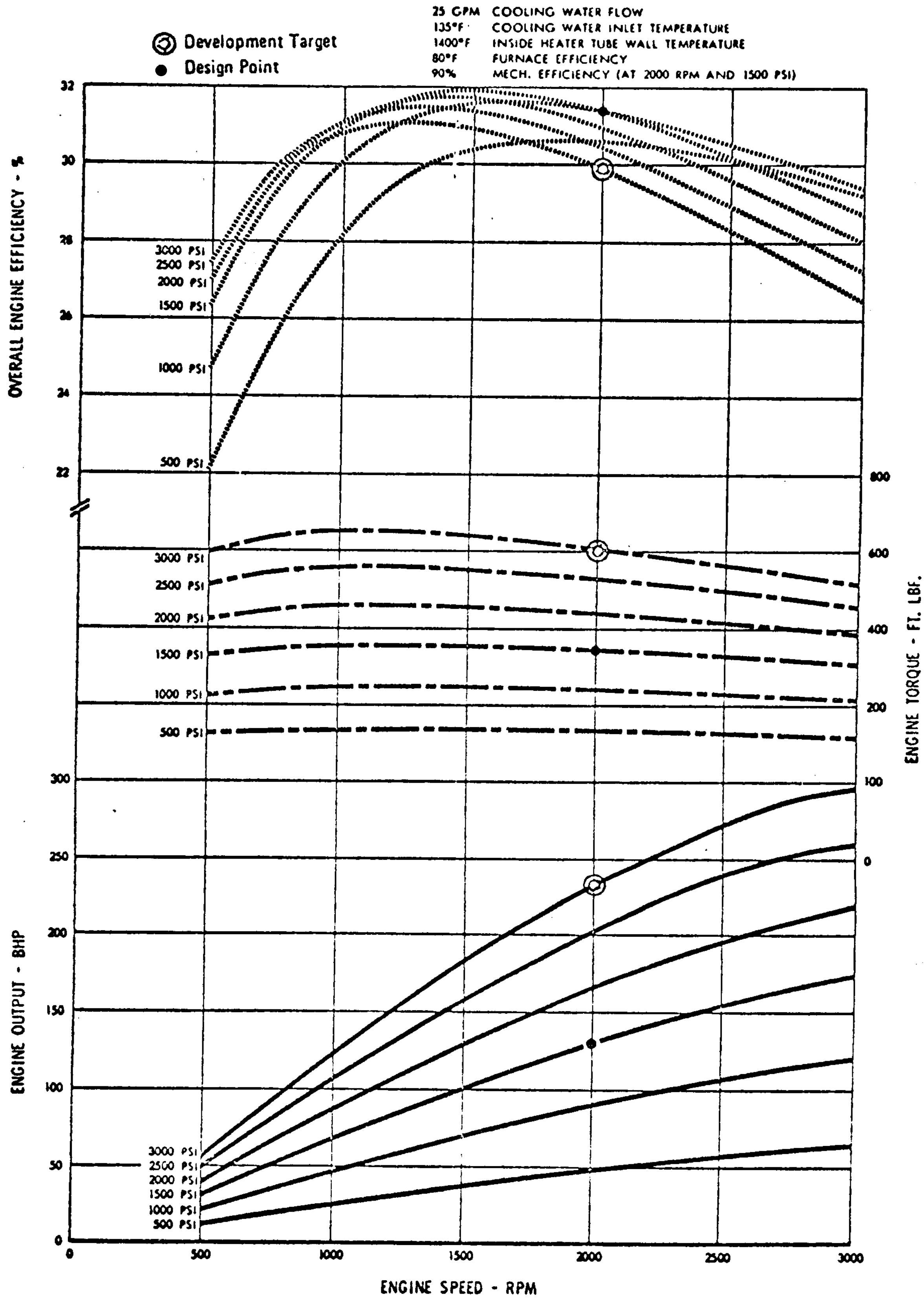
The next page gives the calculated performance of the 4L23. This is followed by a table comparing the new Stirling with the 4-53N and 6-71N GM diesel engines.

In 1969, initial studies of the use of stainless steel and super alloys, specified for the first test engine, showed that about half of the cost of these critical materials could be eliminated by substitution of other materials--all metals; no ceramic materials were considered.

Approximately 95 percent of the basic engine parts were on hand early in 1970, and the engine was being motored on the dynamometer as a part of the balancing procedure when the program was halted on February 27, 1970. Target date for start of the coach installation had been May 1, 1970.

## COMPACT STIRLING RESEARCH ENGINE
## MODEL 4L23

**Hydrogen Working Fluid at Various Mean Working Pressures**



| | | GMR Stirling | DDAD Diesel | DDAD Diesel |
|---|---|---|---|---|
| Model | | 4L23 | 4-53N | 6-71N |
| Bore | in. | 4.00 | 3.875 | 4.25 |
| Stroke | in. | 1.83 | 4.50 | 5.0 |
| Cylinders | | 4 | 4 | 6 |
| Displacement | cu. in. | 92 | 212 | 426 |
| Heater Temperature | °F | 1400 | | |
| Water Temperature | °F | 135 | | |
| Working Gas | | $H_2$ | | |
| Length | in. | 52.75 | 40 | 56 |
| Width | in. | 23.75 | 29 | 33 |
| Height | in. | 39.25 | 34 | 44 |
| Volume | cu. ft. | 28.50 | 22.8 | 47 |
| Weight | lb. | 1600 | 1190 | 1960 |
| Working Gas Pressure | | 1500 | | |
| Rated BHP/RPM | | 130/2100* | 130/2800 | |
| Min. BSFC/RPM | | .425/1400* | .40/2300 | |
| Working Gas Pressure | | 3000 | | |
| Rated BHP/RPM | | 225/2100* | | 218/2100 |
| Min. BSFC/RPM | | .415/1500* | | .37/1800 |

*Net values after combustion blower and water pump power have been deducted.

## ENGINE PERFORMANCE

Two tables of engine data are shown on the following pages, the first presenting test data on ten engines operated by GMR and EMD, the second presenting design data on seven engines. The only "design" engine actually operated was the VEE-1, in early 1969. It is described in reference #6. The Philips boxer engine was about 90% completed before being set aside for work on swash-plate designs. The torpedo powerplant illustrated the extreme possibility (1968) of a Stirling engine designed for limited life (5 hours). Heat was to be supplied from combustion of molten lithium and a Freon gas. Combustion products are liquids which occupy the same volume as the reactants. In the list of 10 test engines, the EMD variable phase drive V-8 engine is missing. However, it used the 1-S1050 cylinder assembly; and only 4 cylinders of the engine were actually tested. Numerous bearing and vibration problems were encountered. Several EMD engines are shown in reference #3, including the 4 cylinder Navy engine. The 3015 engine was installed in the Calvair car in 1964 and operated from thermal energy storage. It is described briefly in the Appendix in the Search report from GMR.

Following the two tables are two sets of curves for the "best" GPU-3 engine, tested in late 1969. Both maximum power and efficiency were improved over the results shown for the GPU-3 in the previous table, which were based on 1966 tests.

## STIRLING TEST DATA
(Revised October 9, 1968)

| | 10-36 Engine | GPU 2 Engine | GPU 3 Engine | 3015 (Calvair) | 4S1210 Navy 4 Cyl. | EMD 1-S1050 | EMD 2W17A | GPU 2 Package | GPU 3 Package | Hybrid Package |
|---|---|---|---|---|---|---|---|---|---|---|
| Max. HP | 7.47 | 7.3 | 11.2 | 40 | 380 | 75 | 138 | 3 KW | 3 KW | 4 KW |
| @ RPM | 3600 | 3600 | 3600 | 2500 | 1500 | 1500 | 1800 | 3600 | 3000 | |
| Max. Eff. | 26.3 | 28.03 | 26.5 | 39 | 35 | 28 | 28.4HHV | | | |
| @ RPM | 1800 | 2400 | 1900 | 1400 | 750 | 1200 | 900 | | | |
| Heater Wall (°F) | 1400 | 1400 | 1400 | 1270 | 1202 | 1270 | 1100 | | | |
| Water in (°F) | 75 | 126 | 100 | 70 | 90 | 100 | 100 | | | |
| Working Gas | $H_2$ | $H_2$ | $H_2$ | $H_2$ | $H_2$ | $H_2$ | $H_2$ | $H_2$ | $H_2$ | |
| F.L. Press (psi) | 1000 | 1000 | 1000 | 1560 | 1500 | 1436 | 1100 | 1000 | 1000 | |
| No. Cyl. | 1 | 1 | 1 | 1 | 4 | 1 | 2 | | | |
| Bore (in) | 2.362 | 2.375 | 2.75 | 3.47 | 5.70 | 5.70 | 6.50 | | | |
| Stroke (in) | 1.238 | 1.238 | 1.238 | 2.37 | 2.90 | 2.90 | 3.20 | | | |
| Displ. ($in^3$) | 5.44 | 5.47 | 7.32 | 22.3 | 296 | 74.0 | 212.0 | | | |
| $HP/In^3$ | 1.37 | 1.33 | 1.53 | 1.79 | 1.28 | 1.01 | 0.65 | | | |
| Wt. (lbm) | 127.4* | 90* | 127* | 550* | 5000 | 2300** | 3800** | 382 | 350.0 | $AL_{391/448}CI$ |
| L (in) | 14 | 12 | 14 | 17.5 | 74 | 36 | 36.25 | 35 | 38.25 | |
| W (in) | 14 | 16.5 | 15.5 | 17 | 40 | 27.5 | 62.25 | 24 | 24.75 | |
| H (in) | 28 | 26.5 | 28 | 37.25 | 76 | 65 | 84.63 | 32 | 28.6 | |
| Vol. ($Ft^3$) | 3.17 | 3.04 | 3.51 | 6.42 | 130.0 | 37.2 | 111.0 | 15.6 | 15.7 | |
| $HP/Ft^3$ | 2.36 | 2.40 | 3.19 | 6.25 | 2.92 | 2.02 | 1.24 | | | |
| #/HP | 17.1 | 12.3 | 11.30 | 13.75 | 13.20 | 30.60** | 27.50** | | | |
| $\#/Ft^3$ | 40.2 | 29.6 | 36.2 | 86.0 | 38.5 | 61.90** | 34.20** | 24.5 | 22.3 | |
| Preheater Wt. (lbm) | | 21 | 21 | 64 | | 97 | 194 | | | |
| BMEP (psi) | 151 | 147 | 168 | 284 | 339 | 268 | 143 | | | |

* Bare engine with preheater.
** Without flywheel.

## STIRLING DESIGN DATA
(Revised October 9, 1968)

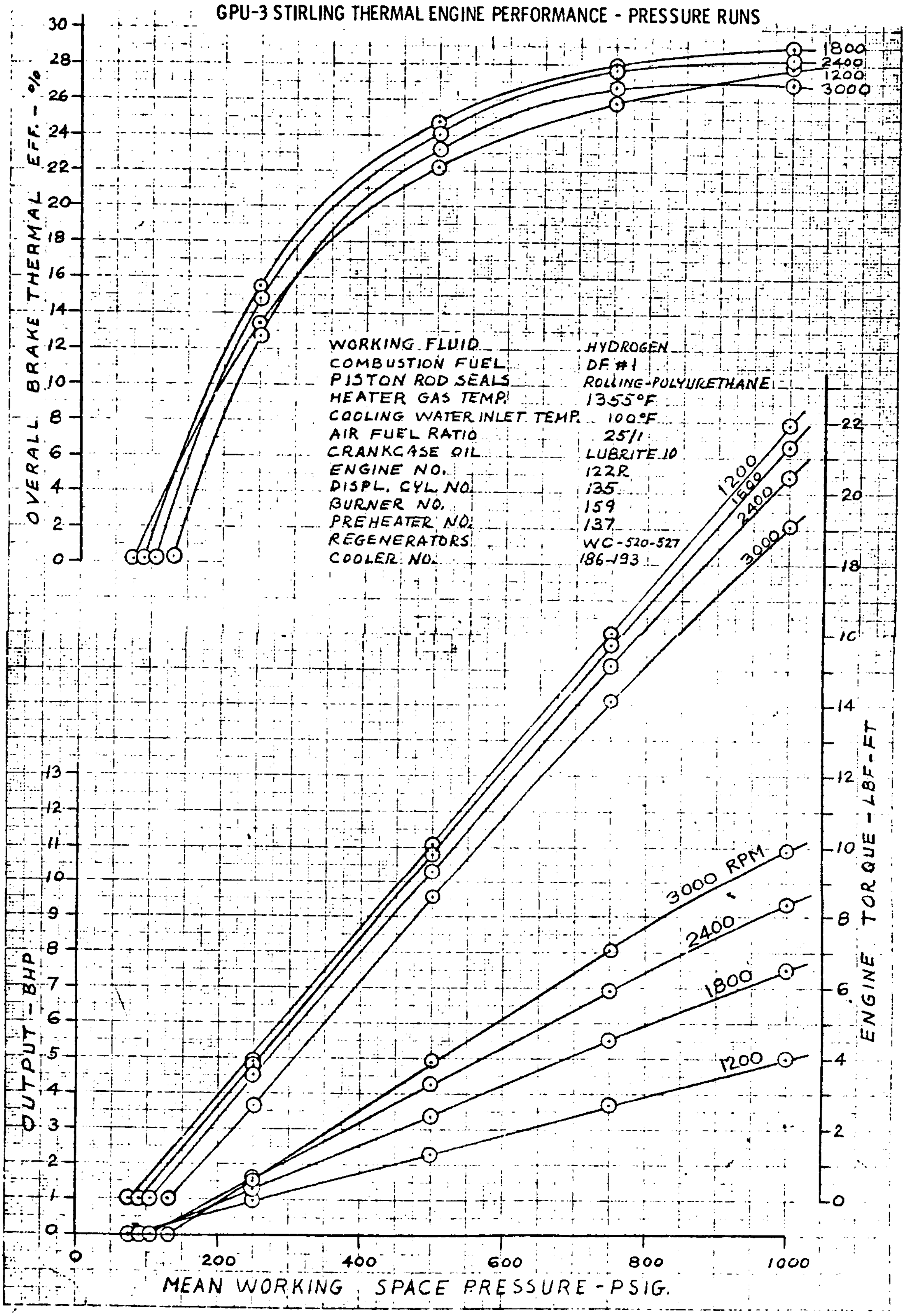
| | Philips Boxer | Philips In-Line | GMR In-Line | GMR In-Line | GMR VEE 1 | Swash Plate | Torpedo*** |
|---|---|---|---|---|---|---|---|
| Max HP | 120 | 200 | 148 ✱ | 129 | 2 | 22 | 690 (Gross) |
| @ RPM | 3000 | 3000 | 2000 | 2000 | 5000 | 2400 | 3000 |
| Max. Eff. | (36.5) | | 33 | 30 | 28.2 | 28.5 | 52 (No comb. loss) |
| @ RPM | 3000 | | 1500 | 1000 | 3200 | 2400 | 1200 |
| Heater Wall (°F) | 1292 | 1292 | 1400 | 1400 | 1300 | 1264 | 1500 |
| Water in (°F) | 104 | 104 | 125 | 125 | 170 | 150 | 60 |
| Working Gas | He | He | $H_2$ | He | $H_2$ | He | $H_2$ |
| F.L. Press. (psi) | 1720 | 3140 | 1500 | 1500 | 1500 | 1500 | 3500 |
| No. Cyl. | 4 | 4 | 4 | 4 | 1 | 4 | 5 |
| Bore (in) | 3.26 | 3.26 | 4.00 | 4.00 | 1.18 | 1.57 | 3.40 |
| Stroke (in) | 1.97 | 1.97 | 1.83 | 1.83 | 1.26 | 1.57 | 2.30 |
| Total Displ. ($in^3$) | 66.0 | 66.0 | 92 | 92 | 1.38 | 12.1 | 104.0 |
| $HP/In^3$ | 1.82 | 3.04 | 1.61 | 1.40 | 1.45 | 1.82 | 6.64 |
| Piston Speed (max rpm) ft/min | 984 | – | 610 | – | 1050 | 629 | 1150 |
| Wt. (lbm) | 850 | 880 | 1000 | 1000 | 24.6** | 200 | 600 |
| L (in) | 59 | 44.5 | 41 | 41 | 10 | 30 | 21 (Inc. gear) |
| W (in) | 39.7 | 17.3 | 18 | 18 | 10.5 dia. | 12 dia. | 19 dia. |
| H (in) | 17.3 | 37.9 | 34 | 34 | | | |
| Vol. ($Ft^3$) | 23.5* | 16.9* | 14.5* | 14.5* | .50 | 1.96 | 3.44 |
| $HP/Ft^3$ | 5.1 | 11.82 | 10.2 | 8.9 | 4.00 | 11.21 | 201.0 |
| #/HP | 7.10 | 4.40 | 6.75 | 7.75 | 12.30 | 9.10 | .87 |
| $\#/Ft^3$ | 36.2 | 52.1 | 69.0 | 69.0 | 49.2 | 102 | 174 |
| Preheater Wt. (lbm) | 150 | | | | | | |
| BMEP (psi) | 240 | 400 | 318 | 277 | 115 | 299 | 815 |

* Volumes include accessories.

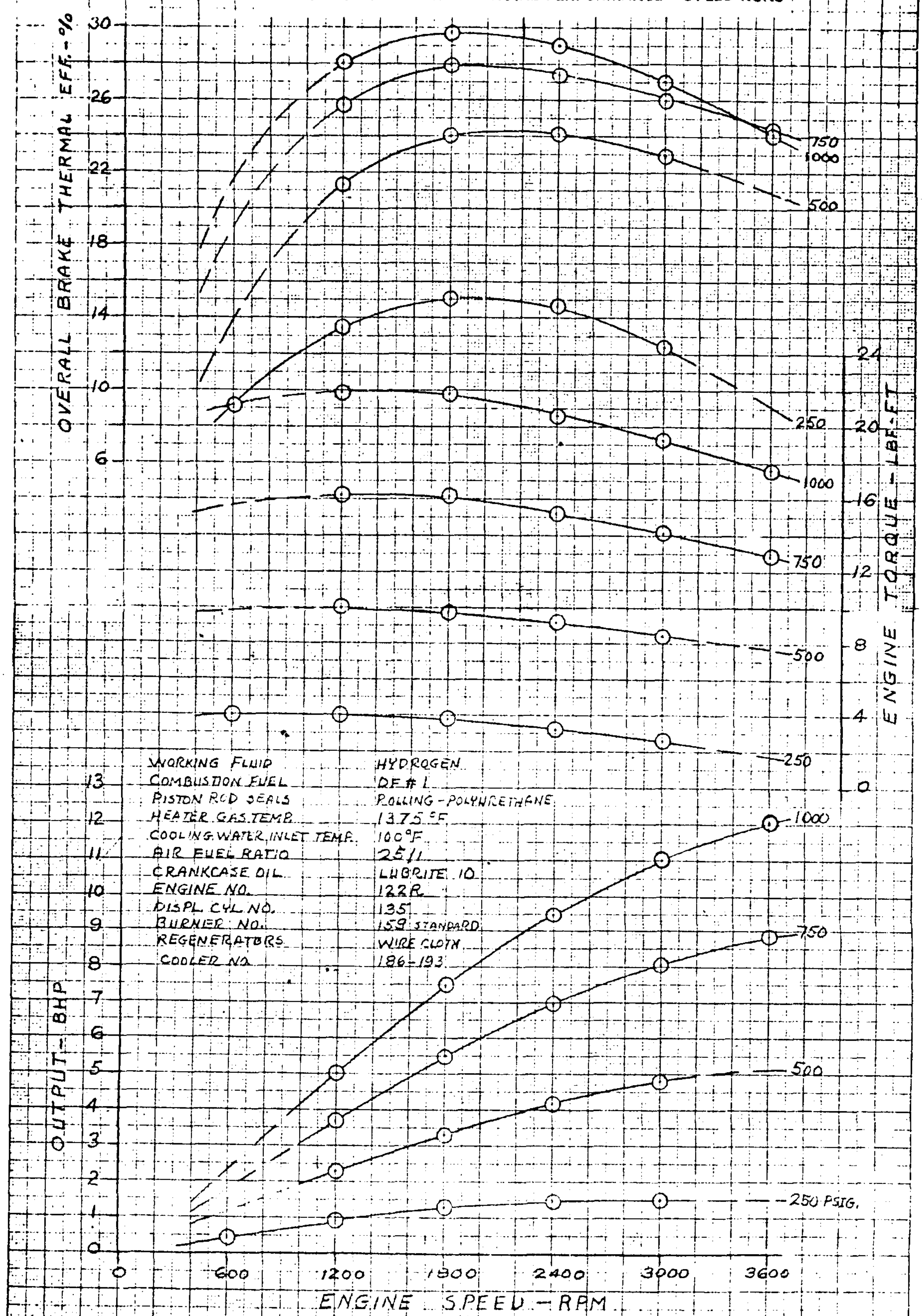
** With ring for isotope heat but less isotope capsules.

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

✱ 200 BHP, 3000 rpm, 1600 psi


GPU-3 STIRLING THERMAL ENGINE PERFORMANCE - PRESSURE RUNS
OVERALL BRAKE THERMAL EFF. - %
1800
2400
1200
3000
WORKING FLUID HYDROGEN
COMBUSTION FUEL DF #1
PISTON ROD SEALS ROLLING-POLYURETHANE
HEATER GAS TEMP. 1355°F
COOLING WATER INLET TEMP. 100°F
AIR FUEL RATIO 25/1
CRANKCASE OIL LUBRITE 10
ENGINE NO. 122R
DISPL. CYL. NO. 135
BURNER NO. 159
PREHEATER NO. 137
REGENERATORS WC-520-527
COOLER NO. 186-193
3000 RPM
OUTPUT - BHP
ENGINE TORQUE - LBF-FT
MEAN WORKING SPACE PRESSURE - PSIG.

## GPU-3 STIRLING THERMAL ENGINE PERFORMANCE - SPEED RUNS






Reproduced from best available copy.

## SUMMARY OF COMPONENT DURABILITY

A running time summary extracted from a 1970 report is shown on the next page. Following this are nine tables which document the tota running time in excess of 1000 hours accumulated by more than 326 critical parts of the experimental 10 hp Stirling engines operated by GMR between 1959 and 1970. Records of each part were kept on "history" cards.

Certain parts such as buffers, fuel nozzles, and pistons were not included either because they never failed, or failed for reasons which did not reflect on their basic design merit. It must also be noted that most of the parts listed were still in use or usable and would have continued to accumulate running time had the project not ended. It is therefore impossible to project ultimate lifetimes for any parts except the piston rings which appear to have predictable wear rates. Piston rod seals were certainly critical parts but were not included because history cards were not kept on each piece; because their lives rarely exceeded 1000 hours; and because their lives were not predictable. However, a photograph is included of two seals which attained 2469 hours of operation and were still usable. It does seem justifiable to conclude that all other major components have useful lives in excess of 2000 hours.

Full load design conditions for the 10 hp engine were 3000rpm, 1000 psi mean working pressure, 1300°F heater gas temperature and 125°F coolant temperature. Only a limited amount of running was done at genuine full load conditions. The GPU engines could not be loaded to capacity due to the limitations of the generator, while the two dynamometer engines which accumulated a large part of the time, operated at either lower speed or lower heater temperature. For some parts, this distinction would be important while for others it would be insignificant. Assessing the differences was probably not feasible anyway due to the variation in test conditions and the statistically small sample. Some indication of the type of operation experienced by each part is given in the "Remarks" column.

Final condition was noted mainly to distinguish between parts which had actually failed and those which were still usable. Except for a few parts, no more exact description was attempted beyond "good" if a part was still serviceable.

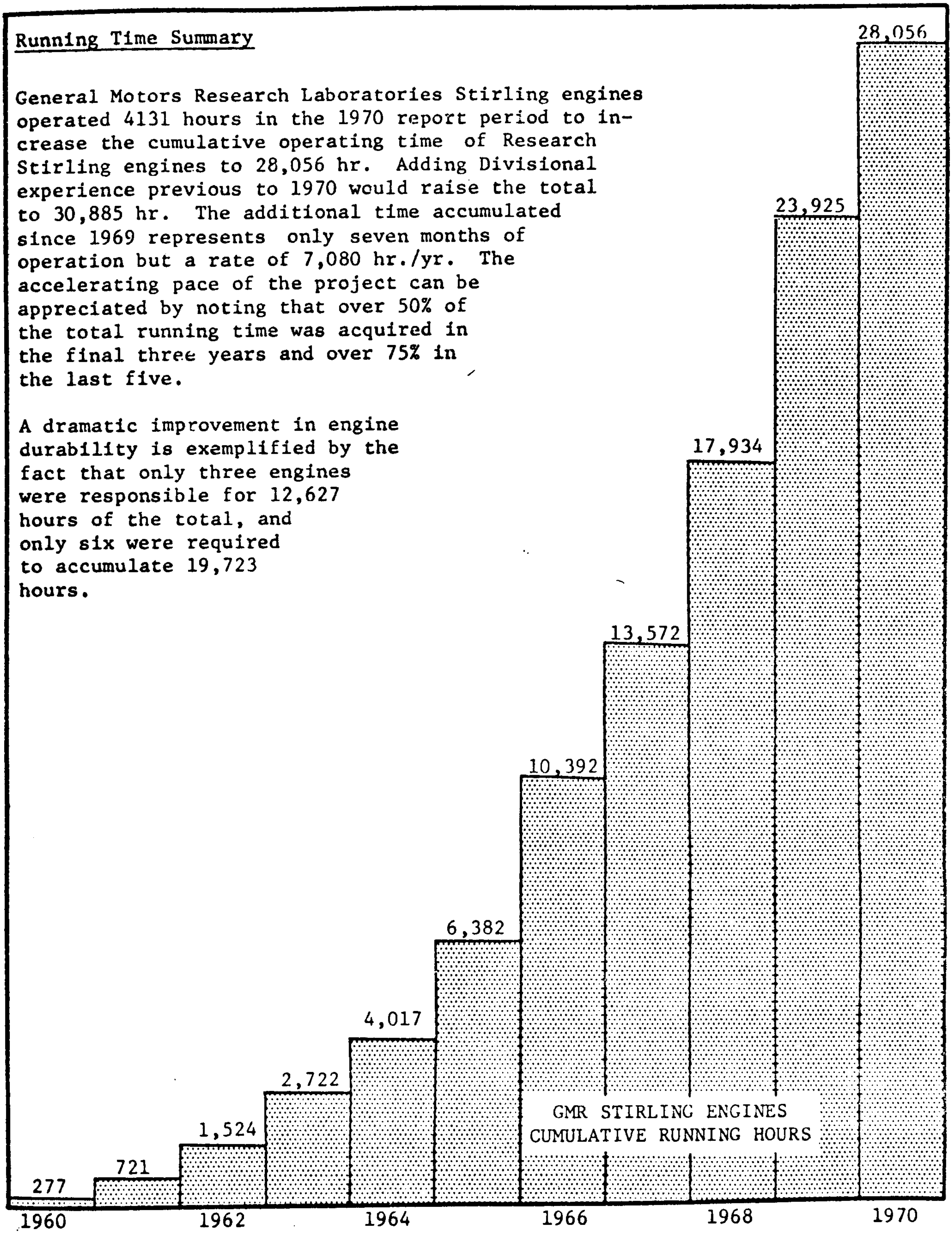
Photographs of some parts are included after the tables.

Durability runs of 500 hours or longer included the GPU-2 in 1964 and GPU-3 in 1967 at Ft. Belvoir; also 3 runs at GMR of which two

## Running Time Summary

General Motors Research Laboratories Stirling engines operated 4131 hours in the 1970 report period to increase the cumulative operating time of Research Stirling engines to 28,056 hr. Adding Divisional experience previous to 1970 would raise the total to 30,885 hr. The additional time accumulated since 1969 represents only seven months of operation but a rate of 7,080 hr./yr. The accelerating pace of the project can be appreciated by noting that over 50% of the total running time was acquired in the final three years and over 75% in the last five.

A dramatic improvement in engine durability is exemplified by the fact that only three engines were responsible for 12,627 hours of the total, and only six were required to accumulate 19,723 hours.



in 1969 were complete GPU packages operated on a rigorous schedule approximating a Military Standard Test, while the third run was a dynamometer test in 1967. The latter was remarkable in that it achieved a record 503.5 hours out of 504 hours of elapsed time -- only two shut-downs occured, the result of faulty building safety interlocks. As a matter of fact, by 1969 the reliability of the Stirling test engines was becoming proverbial -- in the numerous runs of a hundred hours or more the engine had always shown better continuity than had the building facilities. Electrical failures were the most common, occurring in the dynamometers, instrumentation, laboratory switch gear (often accidental shut-downs by electricians), and Detroit Edison stations due to lightning.

TABLE I

BURNERS

| History No. | Hours | Part No. | Remarks and Final Condition |
|---|---|---|---|
| 152 | 2475 | RX-305193-C | Ran in eng. 121R (GPU 3-3) entire time, never at full load. Final condition: Good (see Fig. 1) |
| 134 | 2277 | RX-365334-C | Ran in several engines at all loads. Final condition uncertain but believed to be Good. |
| 150 | 2191 | RX-312111-C | One piece design, ran in eng. 117R entire time at less than full load. Final condition: No Good due to accidental overtemperature (see Fig. 2) |
| 153 | 1597 | RX-305191-D | Dome only; ran in eng. 116R entire time, mostly at 2400 rpm full load. Final condition: Unsatisfactory; distorted, eroded, oxidized. |
| 140 | 1535 | RX-265334-C | Ran in several engines at all loads. Final condition: No Good; distorted, cracked when restruck in die. Skirt ran 4487 hours and was in Good condition. |
| 157 | 1385 | RX-312111-C | One piece design, ran in engine 117R entire time at less than full load. Final condition: Fair; bottom of burner distorted (see Fig. 2). |
| 139 | 1266+ | RX-265334-C | Total history uncertain, dome ran on engine 116R all of recorded time at 2400 rpm full load. Final condition: No Good; distorted. |

Two piece burners were generally installed as sets with the same history number. Once installed, however, it was usually impossible to see the number on the lower portion. In use, no deterioration of the lower or support section was ever noted.

Most common problems were closing of the air inlet gaps through distortion, and high temperature oxidation.

TABLE II

COOLER TUBE ASSEMBLIES

| History No. | Hours | Part No. | Remarks and Final Condition |
|---|---|---|---|
| 112-119 | 4063 | RX-303746-C | Special set made for use with MetNet regenerators; after 4008 hours, #112 was found to be leaking hydrogen into water and was replaced; ran much of the time at 2400 rpm full load. Final condition: Apparently Good (see Fig. 3) |
| 165-171,173 | 2472 | RX-303746-C | Ran in engine 121R (GPU 3-3) at less than full load entire time; one unit leaked after 2.5 hours and was replaced. Final condition: Good. |
| 157,158,161 163,164 | 2230 | RX-303746-C | Ran in engine 117R entire time at less than full load; three leakers replaced at 1385 hours. Final condition believed to be Good. |
| 159,160,163 | 1385 | | |
| 97-104 | 2173 | RX-302886-C | Ran in engine 117R most of time at less than full load. Final condition: Good. |
| 65,66 | 2063 | RX-302886-C | Ran in engine 115R (GPU 3-3) most of the time at less than full load; remainder of set developed leaks at 963 hours and 1127 hours. Final condition: Leaked. |
| 19-26 | 1081 | RX-284830-C | Ran in prototype GPU-3 heater at full load. Final condition believed to be Good. |

Cooler tube assemblies did not wear out in the usual sense. The only mode of failure was the development of leaks which allowed working gas to enter the coolant. The cause of leaks was never firmly established, but it appeared that attack by the brazing material reduced tube wall thickness to a dangerous degree on some occasions and chlorine ion attack of the stainless steel grain boundaries may have been responsible for other failures.

TABLE III

CRANKCASE AND OIL PAN ASSEMBLIES

| History No. | Hours | Part No. | Remarks and Final Condition |
|---|---|---|---|
| 101R | 112(Engine)<br>15,117(Fixture) | RX-146554-D<br>RX-146667-D | First GMR adaptation of Philips 10-30 design; after initial use as an engine was converted to piston seal fixture; same parts believed to have been in assembly entire time. Final condition: Good. |
| 102R | 1,237(Engine)<br>11,412(Fixture) | RX-146554-D<br>RX-146667-D | Essentially same design as 101R; was converted to rod seal test rig; same parts believed to have been in assembly entire time. Final condition: Good. |
| 117R | 4,746 | RX-302432-R<br>RX-302568-E | First rolling seal engine; GPU-3 design; ran at less than full load due to dynamometer capacity; most parts in assembly entire time. Final condition: Good. |
| 111R | 4,177 | RX-284737-R<br>RX-284738-E | GPU-2 design, cast iron; used for general dynamometer tests; one set of parts ran 3,900 hours before rebuild as rolling seal engine. Final condition: Good. |
| 116R | 3,704 | RX-302432-R<br>RX-302568-E | Same design as 117R but used sliding rod seals; ran most of time at 2400 rpm full load; same parts ran entire time. Final condition: Good. |
| 121R | 2,475 | RX-302432-R<br>RX-304492-E | Most recent GPU-3 design; was used in GPU 3-3 entire time and never ran at full load due to limited generator capacity; same parts throughout. Final condition: Good. |
| 115R | 2,328 | RX-302432-R<br>RX-302568-E | Original GPU-3 design; was used in GPU 3-3 and never ran at full load due to limited generator capacity; most parts ran entire time. Final condition: Scrap-as a result of loose screw. |
| 109R | 2,293 | RX-284737-R<br>RX-284738-E | GPU-2 design; ran on dynamometer for 906 hours; also used as a test rig (GPU 2-3) to evaluate ball speed increaser for 1387 hours; total history of parts uncertain. Final condition: Good. |
| 103R | 1,084 | RX-147925-R<br>RX-147926-E | GPU-1 design; ran on dynamometer for evaluation of timing gear designs; many parts changed. Final condition: Cracked when power yoke failed. |
| 108R | 334(Engine)<br>1,038(Fixture) | RX-284737-R<br>RX-284738-E | GPU -2 design; ran as an engine for 334 hours and then converted to a piston rod seal test fixture; parts history uncertain. Final condition: Good. |
| 104 | 167(Engine)<br>4,768(Fixture) | RX-147925-R<br>RX-147926-E | Same as 103R; ran in GPU 1-1 entire time as an engine, then converted to gear test fixture with crankshafts only. Final condition: Good. |

All crankcases were aluminum unless otherwise noted. No failures ever occurred solely due to lack of strength but there was some evidence of insufficient rigidity.

## TABLE IV

## CRANKSHAFTS

| History No. | Hours | Part No. | Remarks and Final Condition |
|---|---|---|---|
| 115 | 4746 | RX-302590-D<br>RX-302591-D | Ran in eng. 117R entire time; never at full load. Final condition: Good (see Fig. 4). |
| 107 | 4323 | RX-147763-C<br>RX-147764-C | Ran in engines 108R and 111R at all loads. Final condition: Good. |
| 114 | 3703 | RX-302590-D<br>RX-302591-D | Ran in engine 116R entire time, mostly at 2400 rpm full load. Final condition: Good. |
| 111 | 2712 | RX-302590-D<br>RX-302591-D | Ran in engine 112R (GPU 3-1) until compressor failure, then ran in 115R (GPU 3-3) until crankcase was destroyed; never ran at full load. Final condition: Inspected and OK. |
| 118 | 2475 | RX-302590-D<br>RX-304703-D | Ran in engine 121R (GPU 3-3) entire time, never at full load. Final condition: Good. |
| 109 | 2293 | RX-282060-C<br>RX-282061-C | Ran in engine 109R entire time; probably little full load operation. Final condition: Good. |
| 101 | 1046 | RX-147763-C<br>RX-147764-C | For first GPU-1 engine, were reworked twice; left hand shaft broke due to a manufacturing defect. Right hand shaft: OK. |
| 110 | 1000 | RX-282060-C<br>RX-282061-C | Ran in crankcase 108R as a test fixture entire time; not highly loaded. Final condition: Good. |

Two additional sets of cranks exist which were never numbered. They were used in engines 101R and 102R and, subsequently, in the test fixture utilizing the same crankcases. Since these were unique parts, they undoubtedly ran the same length of time as the crankcases (i.e. 15,229 hours and 12,649 hours, respectively) but no detailed histories exist for the parts. Only one shaft ever failed (#101).

TABLE V

GEARS - TIMING

| History No. | Hours | Part No. | Remarks and Final Condition |
|---|---|---|---|
| 158,159 | 4746 | RX-303020-C<br>RX-303021-C | Fine pitch steel; ran in engine 117R entire time at less than full load. Final condition: Good. |
| 160,161 | 3703 | RX-303020-C<br>RX-303021-C | Fine pitch steel; ran in engine 116R entire time, mostly at 2400 rpm full load. Final condition: Good. |
| 156,157 | 2712 | RX-302681-C<br>RX-302682-C | Nylon teeth on aluminum hub; ran in engines 112R and 115R (GPU 3-1 and 3-3), never at full load; survived one compressor seizure in 112R but were destroyed when massive failure occurred in crank mechanism of 115R. |
| 166,167 | 2475 | RX-302681-C<br>RX-302682-C | Nylon teeth on aluminum hub; ran in engine 121R (GPU 3-3) entire time, never at full load; same as 156,157. Final condition: Good (see Fig. 5). |

Two additional sets of gears exist for which there are incomplete histories. They were used in engines 101R and 102R and, subsequently, in the test fixtures utilizing the same crankcases. Since there were only two such sets of gears, they undoubtedly ran the same length of time as the crankcases (i.e. 15,229 hours and 12,649 hours, respectively).

TABLE VI

HEATER AND DISPLACER CYLINDERS

| History No. | Hours | Part No. | Remarks and Final Condition |
|---|---|---|---|
| 130 | 4803 | RX-302188-E | Ran on engines 115R and 121R (GPU 3-3) entire time, never at full load. Final condition: Good (see Fig. 6). |
| 134 | 4064 | RX-302188-E | Ran on engine 116R for 2640 hours mostly at 2400 rpm full load, remainder on 117R at less than full load. Final condition: Braze joint leak due to accidental overtemperature. |
| 132 | 2892 | RX-302188-E | Ran on engine 117R for 2230 hours at less than full load. Final condition: apparently Good although was overheated to 1700°F on one occasion. |
| 135 | 2490 | RX-302188-E | Ran on engine 117R for 2092 hours at less than full load. Final condition: Good. |
| 125 | 1775+ | RX-302188-E | Ran on several engines; used as a standard for thermodynamic tests at all loads. Final condition: Good. |
| 123 | 1419 | RX-284880-D | First GPU-3 prototype; ran on engines 108R and 111R mostly at full load for durability. Final condition: Good. |

No cylinder failures ever occurred although the theoretical analysis suggested that the design was marginal at best. The failures which did occur were due to poor tubing quality at first, intergrannular attack later, and poor brazing technique.

TABLE VII

PREHEATERS

| History No. | Hours | Part No. | Remarks and Final Condition |
|---|---|---|---|
| 130 | 4410 | RX-311711-D | GMR straight tube design; 5 rows of .008 in. wall, 310 SS tubes with internal fin; ran on engine 117R at less than full load. Final condition: Good, sectioned for study (see Fig. 7) |
| 133 | 3084 | RX-311711-D | Same construction as #130; ran on engine 116R almost entire time, mostly at 2400 rpm full load. Final condition: Good. |
| 134 | 2475 | RX-314187-D | GMR straight tube design; 6 rows of .012 in. wall, 310 SS tubes with no internal fin; ran on engine 121R (GPU 3-3) entire time at less than full load. Final condition: Good. |
| 125 | 1662 | 7533-1 | Harrison-built spiral tube design; .006 in. wall 316 SS tubes; ran on engine 115R (GPU 3-3) for 1427 hours at less than full load, then 153 hours on engine 118R (GAS 1-1) on natural gas. Final condition: Badly corroded but no leakage. |
| 128 | 1274 | RX-305550 | GMR straight tube design; 5 rows of .008 in. wall 310 SS tubes with no internal fin; ran on several engines at varying loads. Final condition: Brazed joint on exhaust side failed. |
| 117 | 1160 | NE-1275-0-3 | Harrison-built spiral tube design; .006 in. wall 304 SS tubes; ran on engines 109R and 110R at less than full load. Final condition: Uncertain. |
| 115 | 1074 | NE-1275-0-4 | Harrison-built spiral tube design; .006 in. wall 304 SS tubes; ran on engines 108R and 111R at various loads. Final condition: Bad Leakage, cut up for study. |
| 118 | 1049 | 7480-1 | Harrison-built spiral tube design; .006 in. wall 304 SS tubes extensively modified; ran on engines 110R and 111R at various loads. Final condition: Poor but usable. |

Failures were generally due to corrosion of the tubing on early units and thermally induced cracks on later ones. Plugging was encountered on some units.

TABLE VIII

REGENERATORS

| History No. | Hours | Part No. | Remarks and Final Condition |
|---|---|---|---|
| I-VIII | 4101 | RX-314665-A | Special MetNet material, 5% density squeezed to 20%; ran in engine 116R for 3640 hours mostly at 2400 rpm full load, then in 117R at less than full load (see Figs. 8 and 10). |
| 404-411 | 2173 | RX-302887-C | Integral cooler-regenerator design, 308 layers, ran most of time in engine 117R at less than full load. Final condition: Good, were removed to permit new engine build-up. |
| 348,349,350, 354 | 2064 | RX-302887-C | Integral cooler-regenerator design, 308 layers, ran in GPU engines at less than full load; half of set removed at lesser unning time due to leak in coolers. Final Condition: Good. |
| 510,511,513-516,518,519 | 2063 | RX-303750-B | Separate regenerator design, 320 layers, ran in engine 121R (GPU 3-3) entire time at less than full load. Final condition: Good. |
| 477-484 | 1385 | RX-311785-B | Separate regenerator design, 320 layers, ran in engine 117R entire time at less than full load. Final condition: Partially Plugged. |
| 286-293 | 1081 | RX-284827-C | Integral cooler-regenerator design, 308 layers; ran in prototype GPU-3 heater (#123) entire time, mostly at full load. Final condition: Good, but two destroyed in removal. |

## TABLE IX

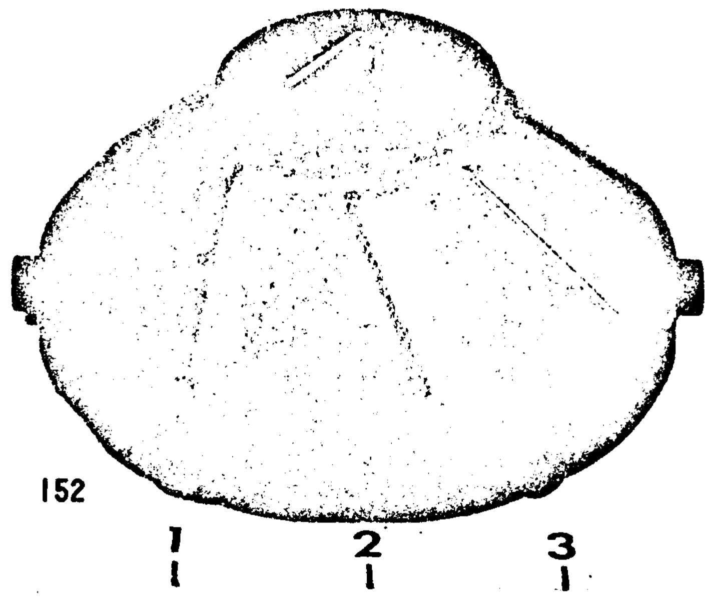
## RINGS - PISTON

| History No. | Hours | Part No. | Remarks and Final Condition |
|---|---|---|---|
| 111,112 | 4374 | RX-302639-A<br>RX-302638-A | Lap-gap design, ran most of time in engine 116R at 2400 rpm full load and higher than normal temperature; projected life based on average wear rate in 116R = 12,500 hours. Final condition: Good (see Fig. 11). |
| 133,134 | 2475 | RX-302639-A<br>RX-302638-A | Lap-gap design, ran entire time in engine 121R (GPU 3-3) at less than full load; projected life based on average wear rate = 11,000 hours. Final condition: Good. |
| None Assigned | 2153 | SK-83167 | Special 3-piece plain gap design, ran in engine 117R entire time at less than full load; projected life based on average wear rate = 8,140 hours. Final condition: No longer gave predictable performance. |
| 115,116 | 1641 | RX-302639-A<br>RX-302638-A | Lap-gap design, ran entire time in engine 117R at less than full load; projected life based on average wear rate = 17,700 hours. Final condition: Damaged by metal particles. |

Only power piston rings have been listed since displacer rings typically exhibit much lower wear rates. In addition, the wear rate for the top ring only was used for calculation of projected life because it wears the fastest under the conditions which prevailed during these tests. The projected life is based upon loss of 30% of the original radial dimension of the sealing ring. All of the actual sealing rings were Rulon LD material.

Typical mode of failure was erratic pumping, often for no apparent reason. Distortion of the lap joint was cause for replacement although the sealing may not have been affected. It proved very difficult to photograph the sealing surface, but in Figure 11 it can be seen that there were no scratches, nicks or other gross defects. A slight distortion of the lap-gap is visible on the upper ring.

BURNER DOME - EXTERIOR VIEW
2475 HOURS OF OPERATION




BURNER DOME - INVERTED
2475 HOURS OF OPERATION

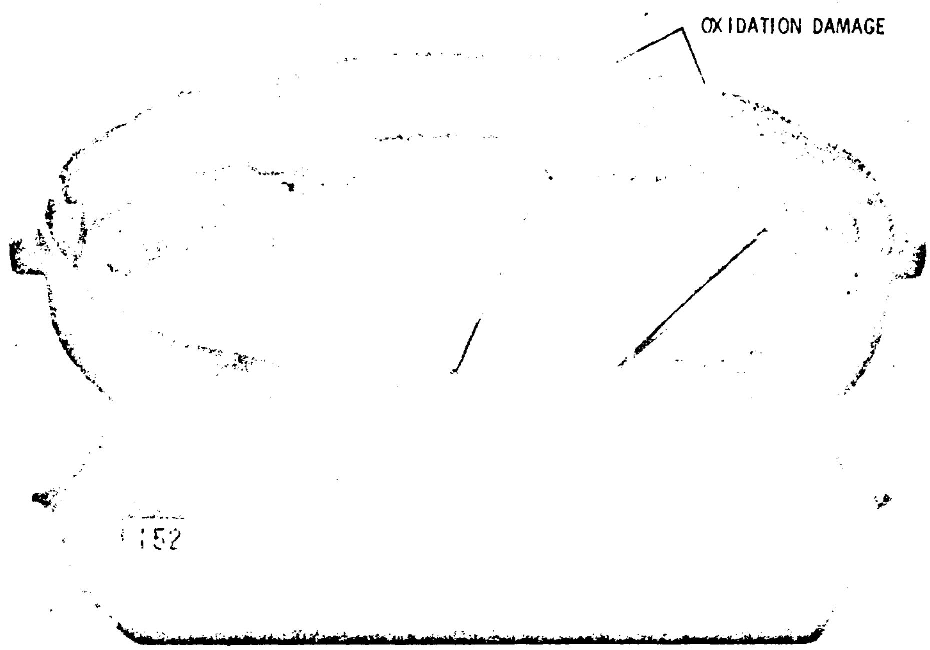



Figure 1

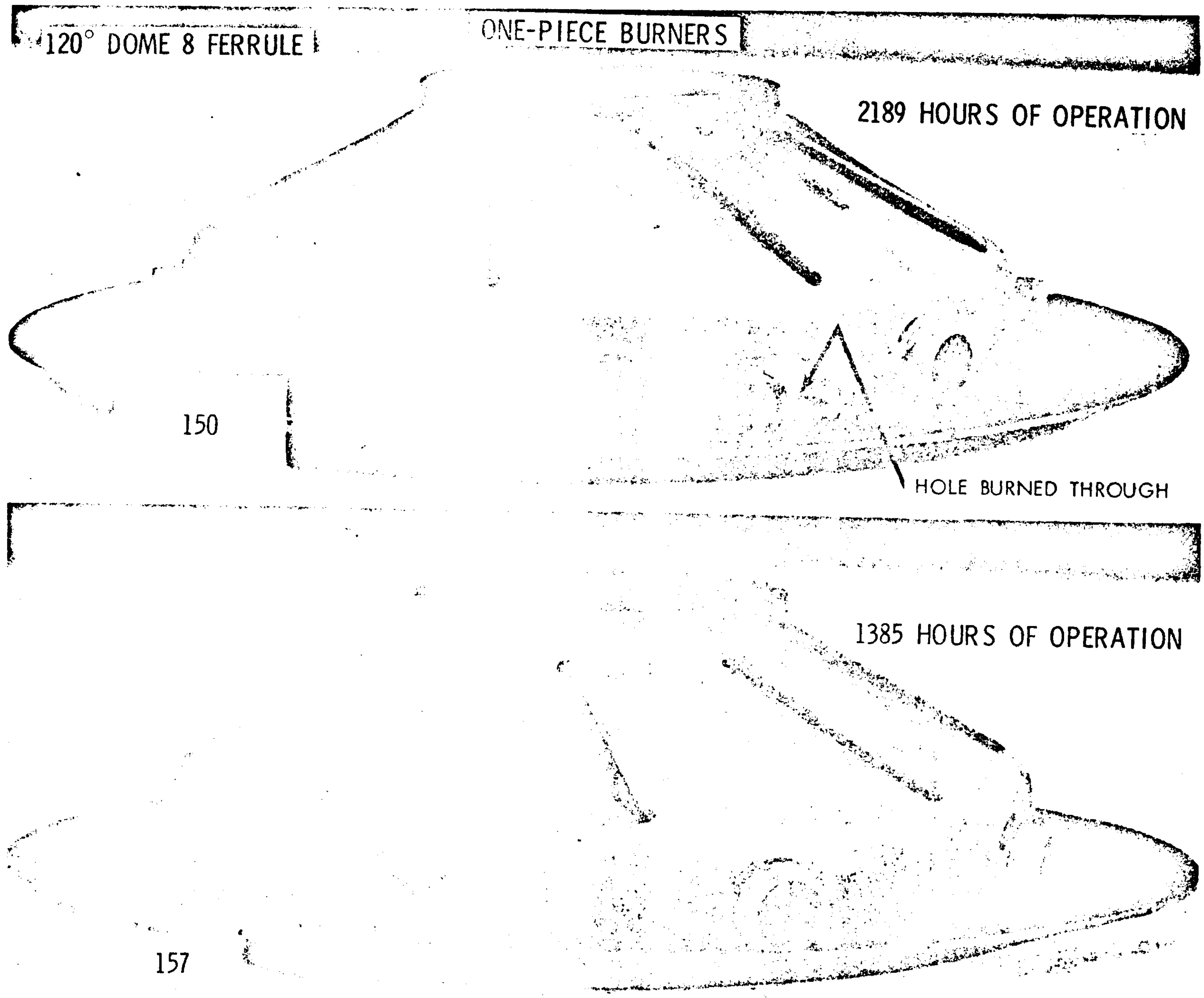
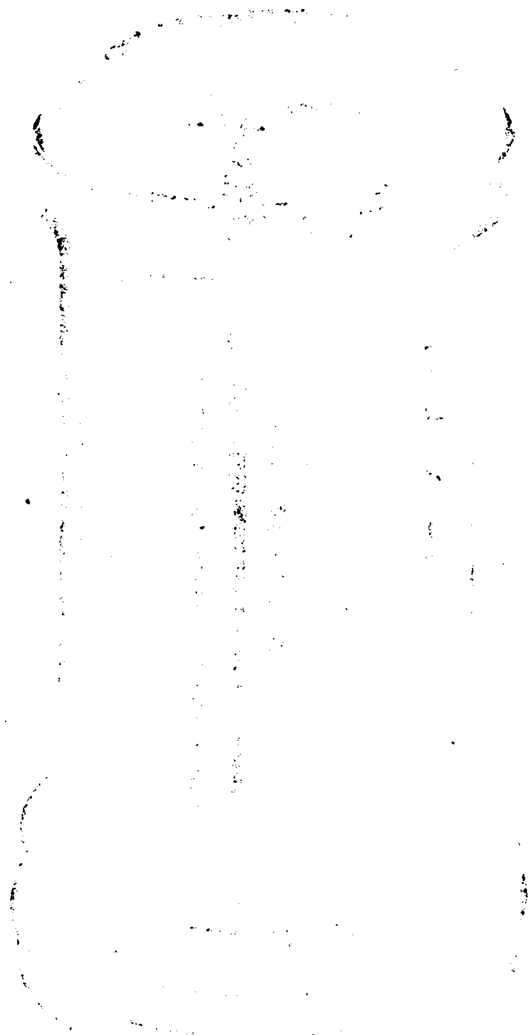
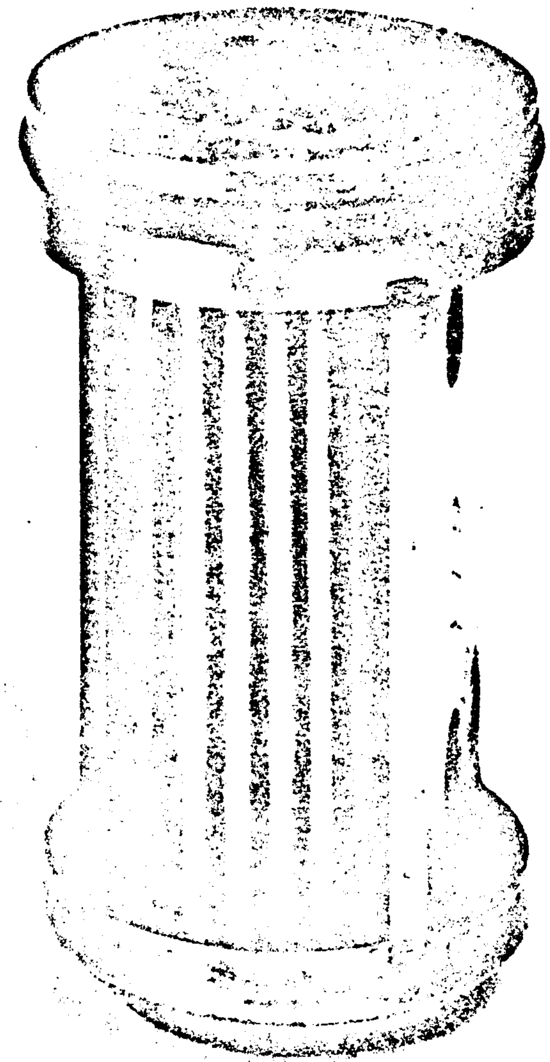

120° DOME 8 FERRULE
ONE-PIECE BURNERS
2189 HOURS OF OPERATION
150
HOLE BURNED THROUGH
1385 HOURS OF OPERATION
157

Figure 2

COOLER TUBE ASSEMBLIES
4063 HOURS OF OPERATION



116



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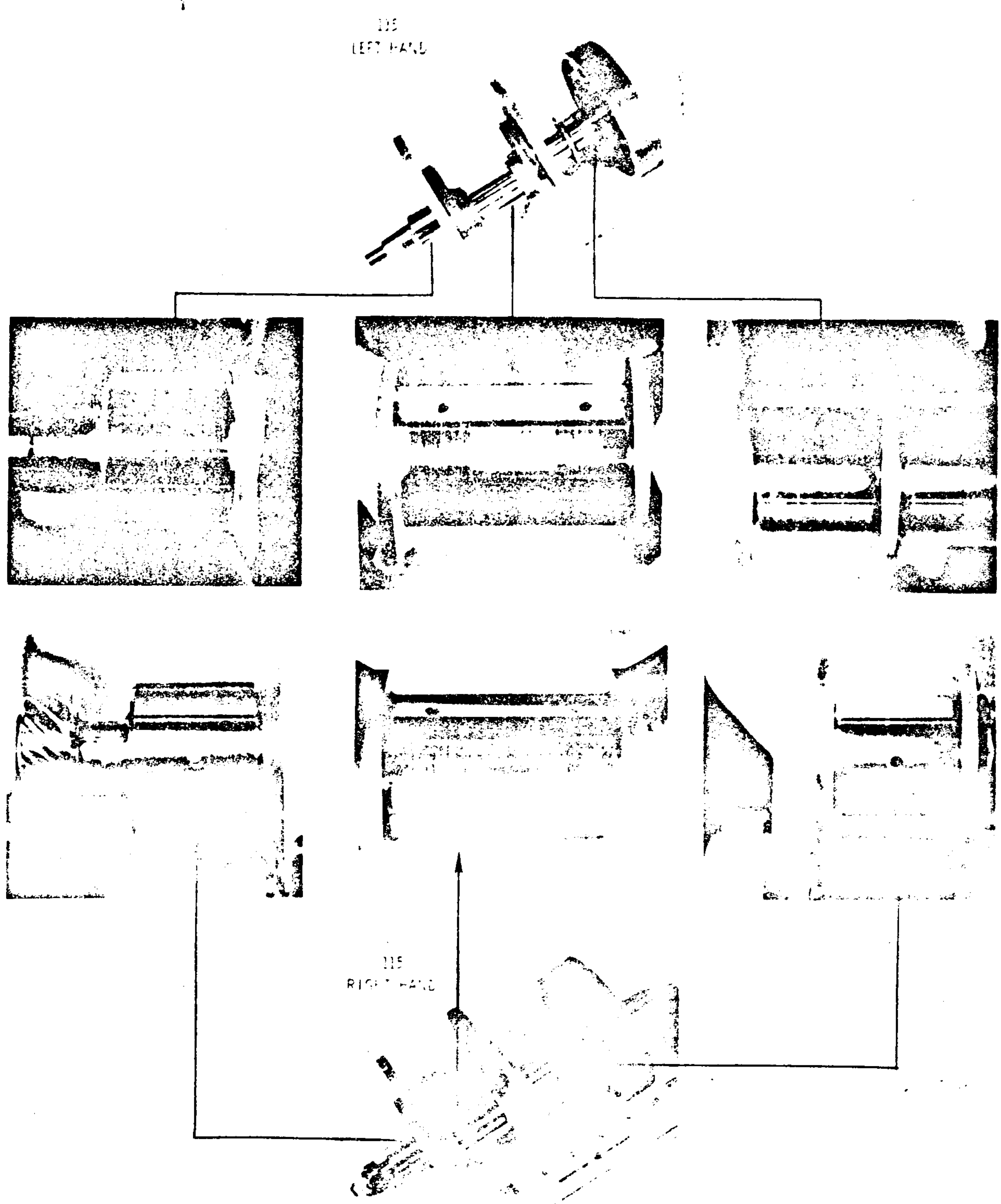



CRANKSHAFTS
4746 HOURS OF OPERATION

Figure 4

TIMING GEAR TOOTH WEAR

2475 HOURS OF OPERATION

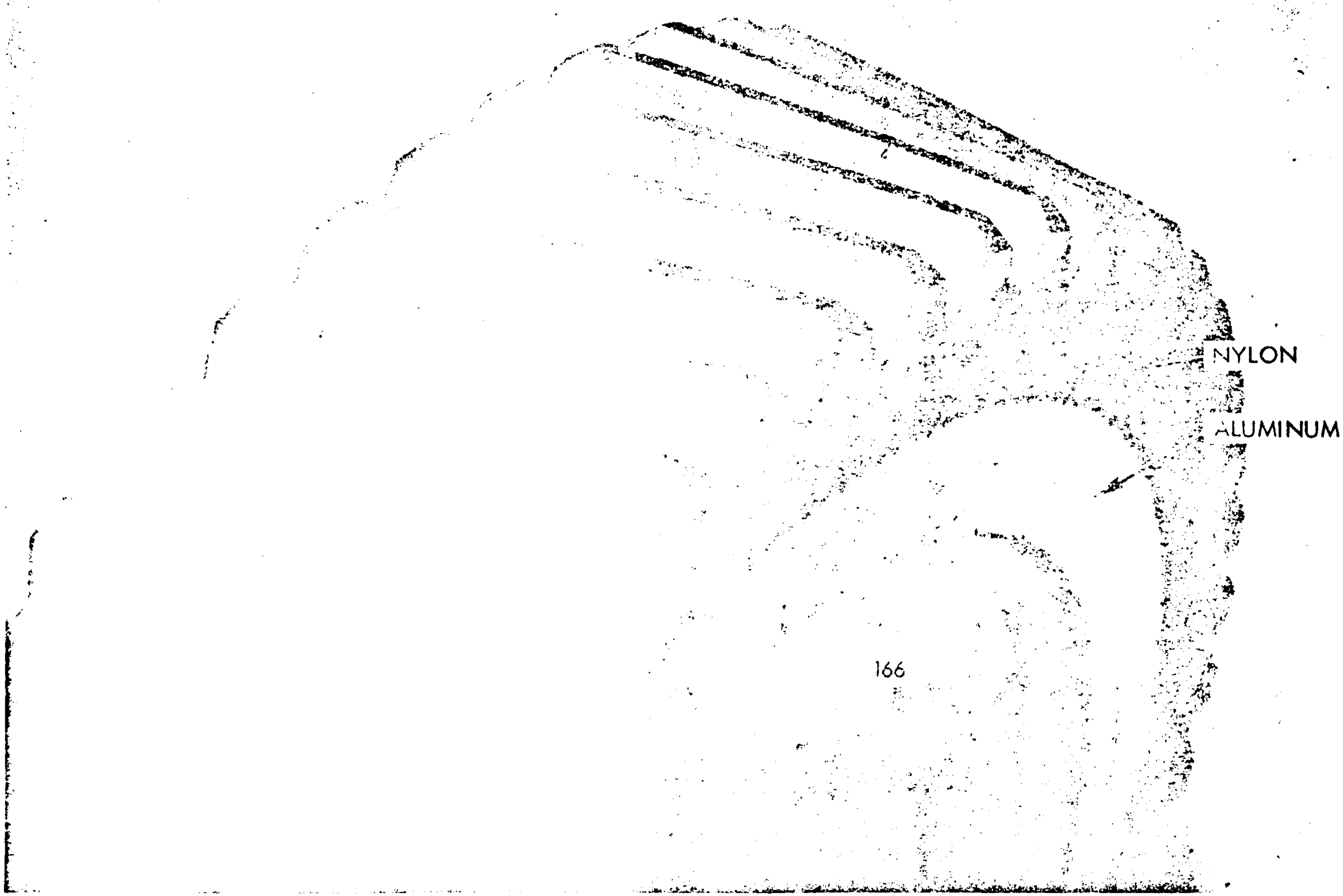
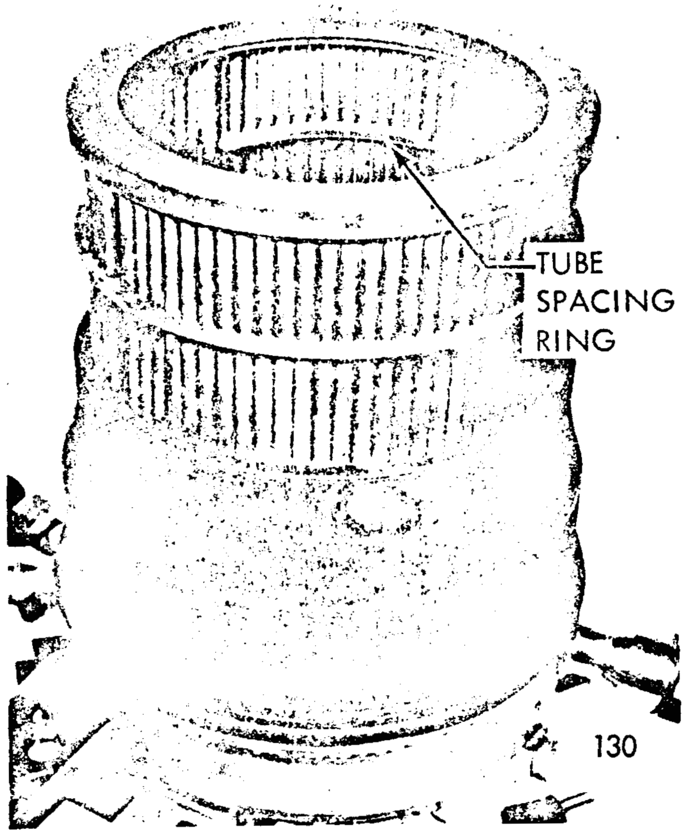



Figure 5

## HEATER AND DISPLACER CYLINDER
## 4803 HOURS OF OPERATION



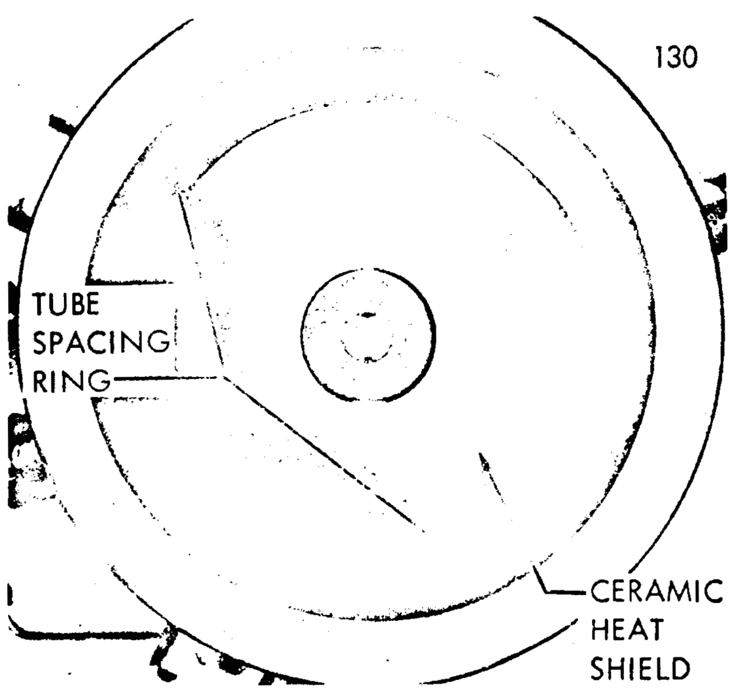
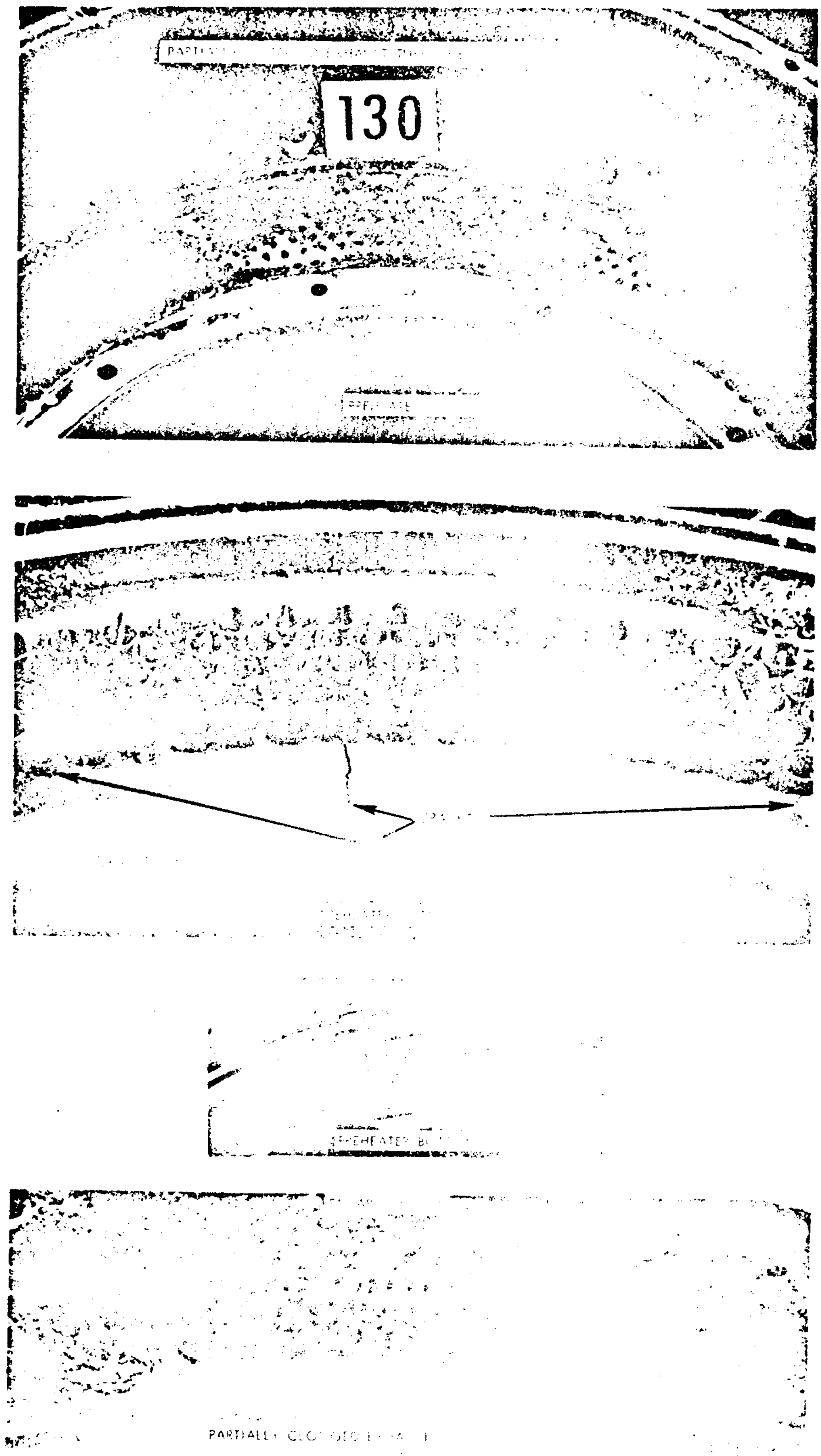



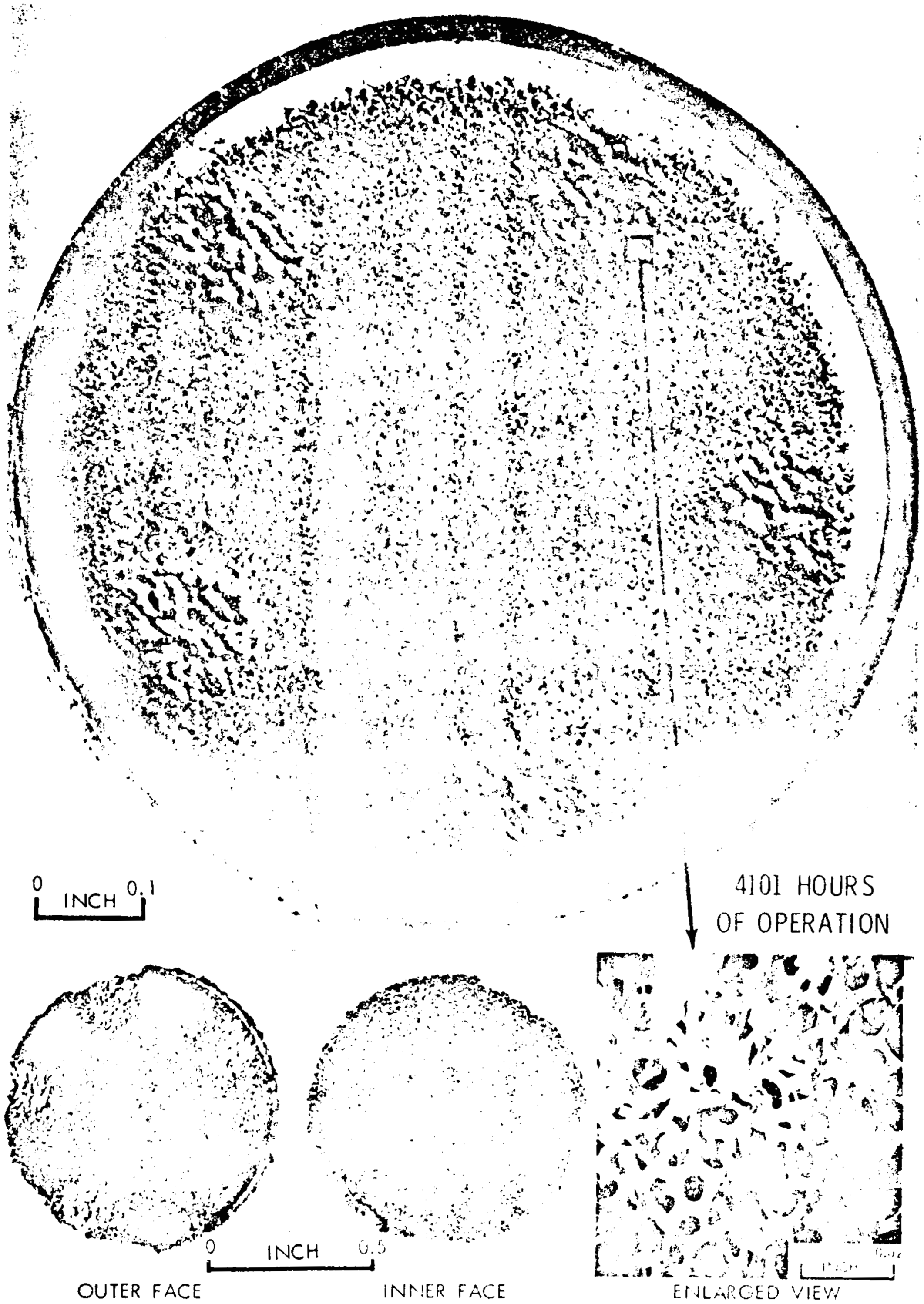

Figure 6




PREHEATER SECTIONED FOR STUDY
4410 HOURS OF OPERATION

Figure 7

REGENERATOR MATRIX 0.05-0.20 METNET
NUMBER 3
SIDE HOT TEMPERATURE END




OUTERMOST DISK VIEWS OF REGENERATOR MATERIAL

Figure 8

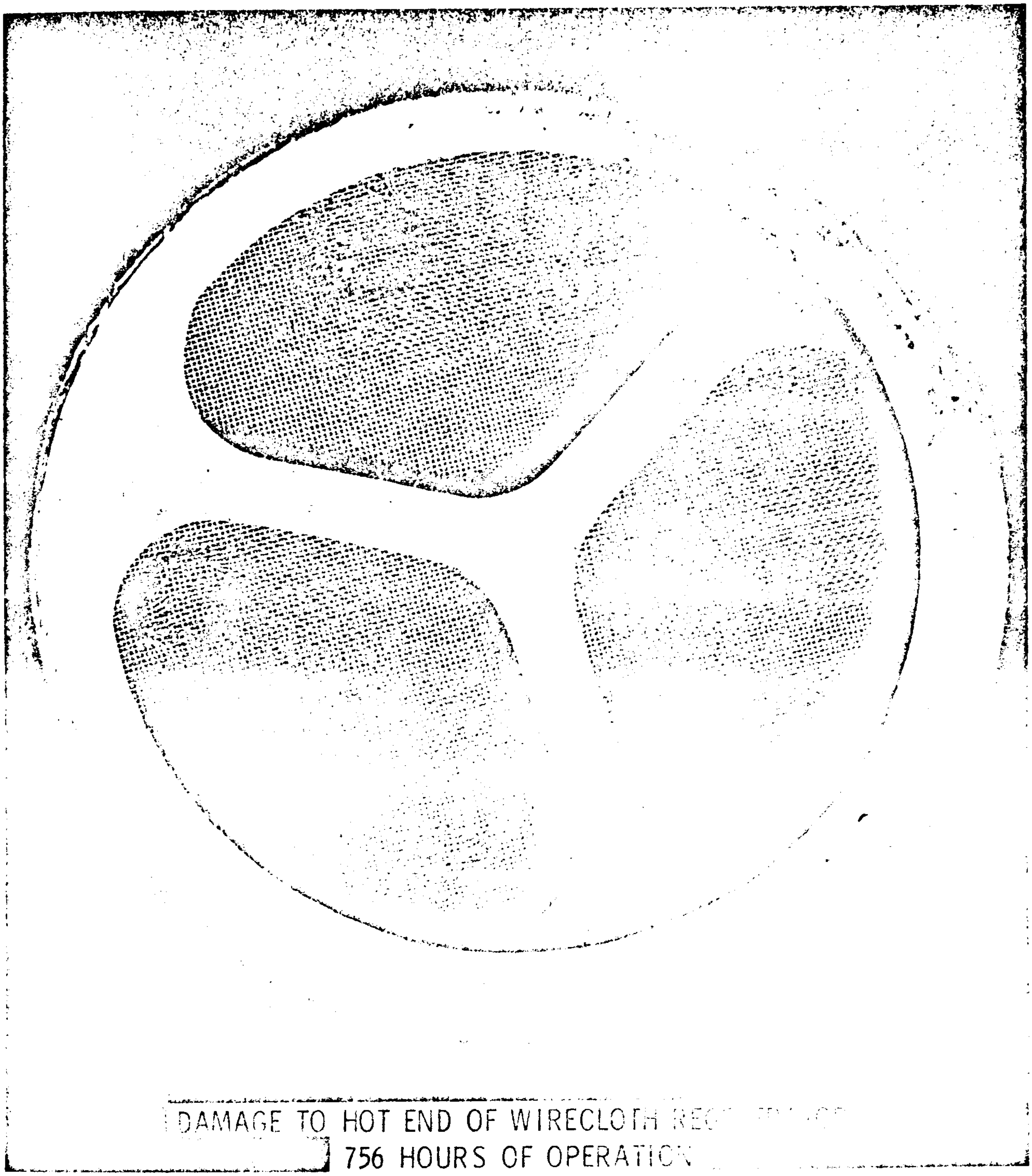
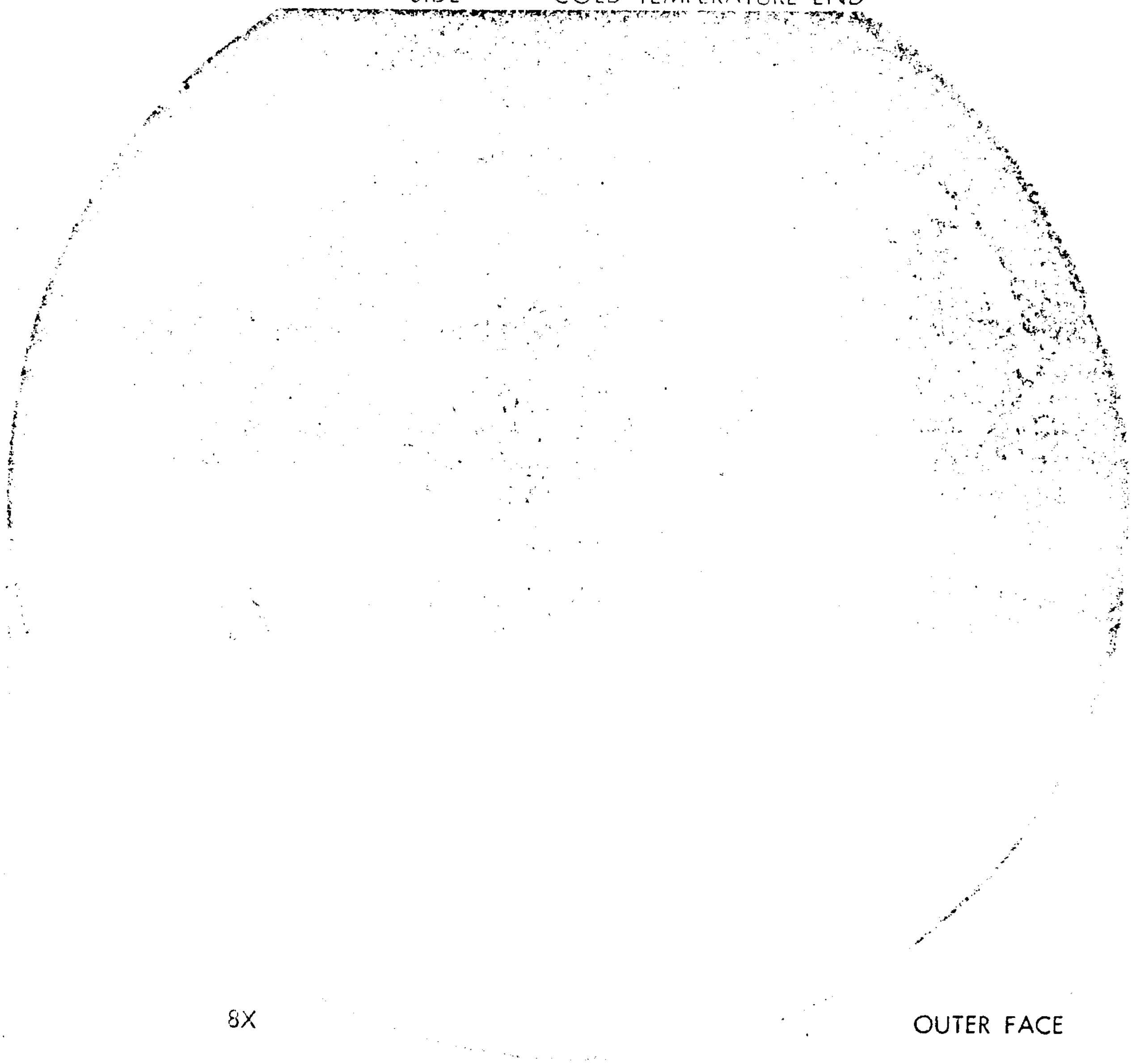



Figure 9

REGENERATOR MATRIX 0.05-0.20 METNET
NUMBER 3
SIDE COLD TEMPERATURE END



8X OUTER FACE

COLD TEMPERATURE END VIEW OF REGENERATOR
4101 HOURS OF OPERATION

Figure 10

Reproduced from best available copy.

RULON PISTON RINGS
4374 HOURS OF OPERATION

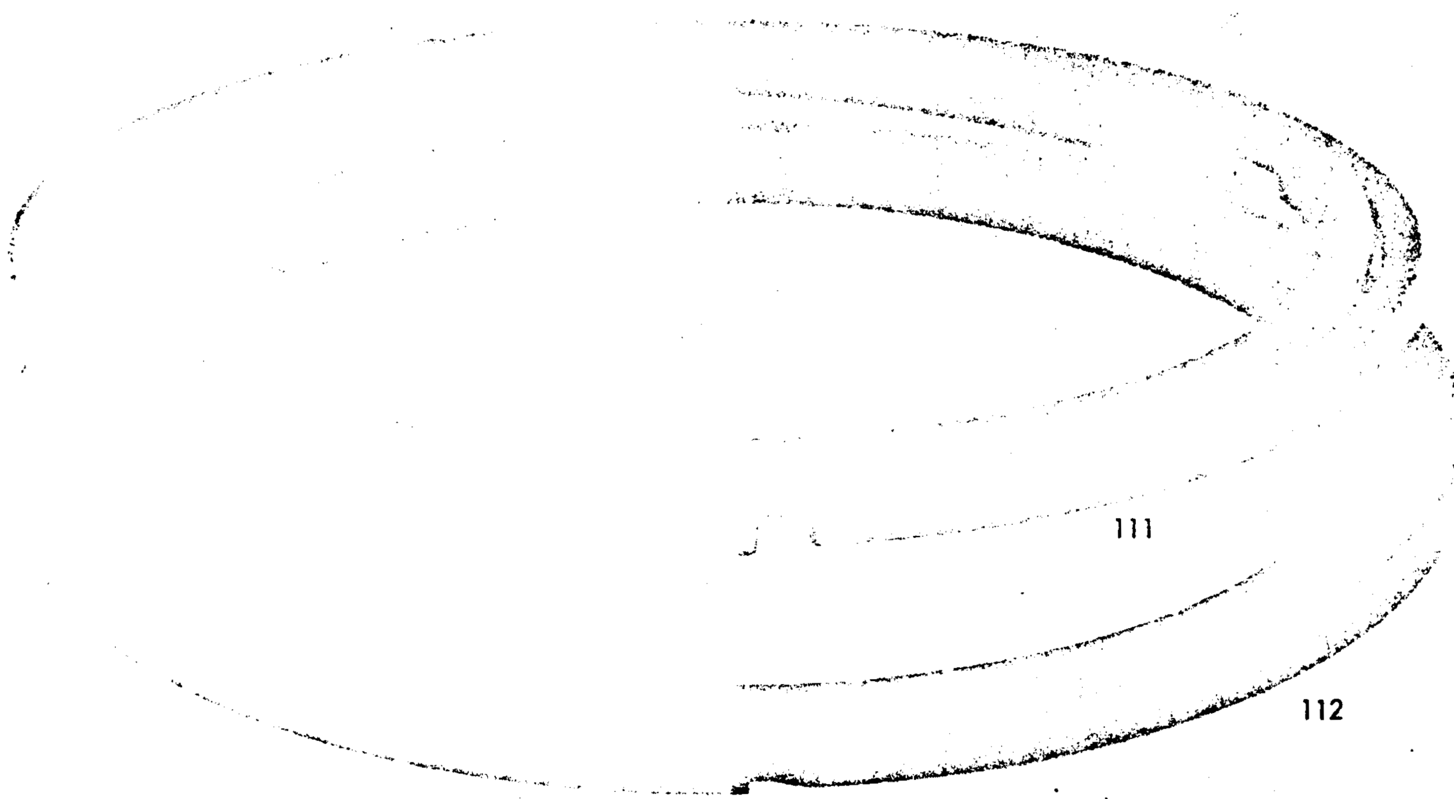



Figure 11

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## Technical Problems Remaining to be Solved for the Stirling Engine Automotive Applications and Potential Solutions to these Problems

The author has endeavored to partially answer these questions in the historical sections of the report. The author is also not fully informed at this time of the progress which has been made by Philips and its licensees in Sweden and Germany since 1970 in solving these problems.

The following appear to be the major problems for automotive applications of the Stirling engine; 1) Sealing the piston rods and loss of hydrogen; 2) Piston rings in double acting engines; 3) Heat rejection in vehicle installations; 4) Engine cost and high temperature materials.

### 1) Sealing the Piston Rods and Loss of Hydrogen.

Philips has continued to work vigorously on the rolling seal problems while United Stirling, Sweden, has concentrated on sliding seal technology. It is believed that routine operation of both types of seals in excess of 1000 hours is being realized for elastomer temperatures below 200°F. This time period is satisfactory for passenger car applications and is probably superior to the lives of other elastomer parts such as lip seals and face seals in automobiles. However, for commercial vehicles, a minimum service life of 3000 hours appears necessary. Quality control of elastomer production and molding is considered just as important as was noted in programs at General Motors in the 1960's. The author has no recent information on Philips progress on rolling seals.

The section on sliding seals discusses hydrogen leakage rates, and only briefly, hydrogen diffusion. It was concluded that a commercial engine could be built with an acceptable dynamic leakage rate; from brief discussions between the author and representatives of United Stirling, it is understood that U.S.S. also believes the present sliding seals are acceptable for commercial engines.

The question of hydrogen diffusion was never settled by either GM or Philips. It is now understood that Philips has a special ceramic coating for the interior of heater tubes which, for all practical purposes, eliminates the diffusion problem. There continues to be questions about hydrogen safety and controls. The appendix includes a letter to GMR Management which discusses the question of hydrogen safety, leakage and flammability limits. While GM had problems with hydrogen pump and control valve reliability, it is believed that Philips has solved these problems by the incorporation of the system within the buffer space.

2) Piston Rings in Double Acting Engines

The historical section on piston rings concluded that wear rates for rings in displacer engines were very satisfactory and would permit running for 10,000 hours or more. There was some indication from preliminary testing that wear rates for double acting engines might reach 3 to 4 times that for displacer engines; if this is true today it could severly limit the range of engine applications. The author has no recent information on wear rates from Philips or their licensees.

3) Heat Rejection in Vehicle Installations

Some comparisons of heat rejection to the coolant for Stirling and internal combustion engines are of interest. The author had collected these data on IC engines in 1966 when interest in Stirling vehicle propulsion was just starting.

The only measured Stirling heat rejection data available to the author are for the 10 hp model. Heat rejection to the cooler, buffer, and engine oil at full load is included. Accessory power was taken as 10% of the engine gross power. This is believed valid since the IC engine data do not include radiator fan power. Engine efficiency, and therefore heat rejection, is based on 150°F water temperature to the engine and 160°F out. Under these conditions, the 10 hp engine rejects about 5500 BTU per BHP hour.

HEAT REJECTION---INTERNAL COMBUSTION ENGINES

| Diesel Engine | Model | | BTU per BHP hour |
|---|---|---|---|
| Gardner (4 stroke) | 6 LX | (Incl. water + oil) | 1250 |
| Cummins (4 stroke) | V6-200 | " " " | 1530 |
| Gen Motors (2 stroke) | 6-71 | " " " | 2100 |
| Gen Motors (2 stroke) | 8-268A | " " " | 3000 |
| **Spark Ignition** | | | |
| Average of 10 truck engines | | (excluding oil) | 2050 |

Water temperature was not given; but it is believed heat rejection values would not vary widely for a ± 50° water temperature change.

The ratio of heat rejection for the Stirling 10 hp and the larger IC engines ranges from 4.4 (Gardner) to 1.83 for the 8-268A diesel, and 2.68 for the spark ignition truck engines.

For a modern double acting Stirling of 150 gross hp (less 10% for accessories) operating at 1500°F heater temperature and 150°F water temperature, the calculated heat rejection amounts to 4230 BTU per BHP hr -- a considerable improvement over the smaller engine. Thus the ratio of Stirling to the spark ignition truck engines now becomes 2.06.

There are several possible solutions to the radiator space problem in Stirling powered vehicles. For passenger cars, the Philips laminar-flow self-cleaning folded "V" design may be the best solution. For trucks, it may be possible to move radiators away from the conventional location up front. There is a trend for larger diesel tractor units to locate radiators on the roof or behind the cab in a special vertical cooling tunnel. If these methods are feasable, the space and area for radiators would appear to be adequate for the additional Stirling cooling requirement. If elastomer seals are still temperature limited, then a separate seal cooler system operating at a lower temperature than the engine cooler would not be difficult to incorporate. Heat rejection would be a small fraction of the main cooling requirement.

4) Stirling Cost and High Temperature Materials.

Little was done at GMR in this area other than to discuss the question in numerous meetings. Several metalurgical specialists outside the Mechanical Development Department believed that because of progress being attained in precision castings for gas turbines, the Stirling heater assembly could eventually be completely cast -- without porosity, which had been a serious limitation. Also discussed were attachment of separate heater tubes to the cylinder and regenerator cups by a stud-welding process; and fabrication by spin-friction welding of the upper part of the displacer cylinder, made of a high nickel alloy, to a lower section of the cylinder, made of plain carbon steel. Design studies were made of heater assemblies not requiring separate tubes -- the most expensive unit of the assembly.

It was also assumed that future air preheaters would be constructed of ceramic materials using the rotary regenerator principle for multi-cylinder engines, and a recuperative type heat exchanger for single cylinder engines.

It is understood that Philips had contracted with Ricardo & Co. of England to study heater costs, but the results are unknown to the author. Studies are being made and hardware is being fabricated by suppliers to both Philips and U.S.S. of ceramic heater assemblies. (Silicon carbide??) This method appears to offer the greatest possibilities for lower cost and higher temperature operation, if porosity can be eliminated.

Potential Suppliers and/or Developers for Critical Engine Components and Materials

A realistic approach to this task requires a thorough discussion with many individuals and companies who are potential suppliers and/or developers. Such a survey must also be kept up-to-date. The author has prepared a partial list from personal acquaintance and contacts with persons who are knowledgeable in these fields. He cannot attest to the accuracy of the information.

| ELASTOMER PRODUCTS | CONTACT PERSON |
|---|---|
| Chicago Rawhide Co.<br>Elgin, Ill. | Robert Brink |
| Crane Packing Co.<br>Morton Grove, Ill. | Harry Tankus |
| Dixon Corp<br>Bristol, R. I. | ---- |
| Parker Seal Co.<br>Culver City, Calif. | John Stone |
| Precision Rubber Products<br>Lebanon, Tenn. | Howard Gillette |
| Vernay Laboratories<br>Yellow Springs, Ohio | ---- |
| Victor Mfg. and Gasket Co.<br>Chicago, Ill. | Steven Lillis |

| METALS AND CERAMICS | |
|---|---|
| Advanced Materials Engineering Ltd.<br>Gateshead, England | R. B. Caws |
| Eaton Corp.<br>Eaton Research Center<br>Southfield, Mich. | Robert Richardson<br>(active in materials, fabrication, and potential manufacturing) |
| Pure Carbon Co.<br>St. Marys, Pa. | Robert Paxton |

METALS AND CERAMICS (cont.)

| | |
|---|---|
| Allegheny Ludlum Steel Corp.<br>Research Center<br>Brockenridge, Pa. | R. K. Pitler |
| Special Metals Corp.<br>(Div. of Allegheny Ludlum)<br>New Hartford, N. Y. | F. N. Darmara |
| Carpenter Technology Corp.<br>Research and Dev. Center<br>Reading, Pa. | N. J. Culp |
| Crucible Steel Co.<br>Materials Research Center<br>Pittsburgh, Pa. | E. J. Dulis |
| International Nickel Co.<br>New York, N. Y. | R. F. Decker |
| Huntington Alloy Products Div.<br>Int. Nickel Co.<br>Huntington, West Va. | J. W. Cundiff |
| Stellite Div.<br>Cabot Corp.<br>Kokomo, Ind. | S. G. Wlodek |
| Teledyne-Allvac<br>Lockhaven Park<br>Waxhaw, North Carolina | W. Wm. Dyrkacz<br>(consultant) |
| Universal Cyclops Steel Corp.<br>Research and Dev. Laboratory<br>Bridgeville, Pa. | F. M. Richmond |

TUBING SPECIALISTS

| | |
|---|---|
| Babcock and Wilcox Corp.<br>Research Center<br>Alliance, Ohio | J. P. Rowe |
| Superior Tube Co.<br>Norristown, Pa. | J. B. Giocobbe |

Huntington Alloy (above)

Allegheny Ludlum (above)

Crucible Steel Co., Trent Tube Div., Milwaukee, Wis.

Carpenter Technology, Union Tube Div., Union, N. J.

## CERAMIC SPECIALISTS

| | |
|---|---|
| Corning Glass Works<br>Corning, N. Y. | David Duke |
| Norton Corp.<br>Industrial Ceramics Div.<br>Worcester, Mass. | E. W. Hauck |
| Owens-Illinois<br>Toledo, Ohio | R. Ramelmeyer |
| Coors Porcelain Co.<br>Golden, Colo. | M. Fenerty |

## APPENDIX A

## SELECTED REFERENCES ON STIRLING ENGINES

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Mechanical Research Department
General Motors Research Laboratories


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APPENDIX B

OFFICE OF
JUL 28 1969
P. F. CHENEA
Vice President

RESEARCH LABORATORIES
GENERAL MOTORS CORPORATION

DATE July 25, 1969

SUBJECT Hydrogen in the Stirling Powered Bus

TO Dr. P. F. Chenea
Vice President
Research Laboratories

In a meeting on June 4, 1969 in Mr. Underwood's office, concerning the GMC Truck & Coach bus program, you requested that I investigate the possible restrictions on the use of hydrogen working fluid in the 150 BHP 4L23 Stirling engine for the bus installation. I have examined the possible restrictions for such a bus on public highways, tunnels, etc. In addition, I believe it is of interest to determine the quantity of hydrogen which will be contained and to compare the potential danger of hydrogen with some of the more usual flammable materials common to our homes and industries.

Our Safety Director, Jack Leggat, contacted the Michigan State Police and Detroit Police Headquarters for information on such vehicles in Detroit. The Detroit City Code, Article XVI, Section 38-16-1, which refers to "Hauling and Distribution in Bulk of Compressed, Oxidizing, Toxic, Flammable Gases or Flammable Liquids Upon the John C. Lodge Freeway," where the walls are high as well as under Cobo Hall, does not apply to our situation nor to casual hauling of a few bottles of flammable materials. For example, welding trucks for on-the-job work carrying several bottles of acetylene, oxygen, hydrogen, etc., as well as touring trailers hauling one or two bottles of LPG are not restricted in these areas, nor are such vehicles restricted in passing through the Windsor-Detroit Tunnel nor the Pennsylvania Turnpike tunnels. I contacted the Turnpike Authority directly in order to confirm that such vehicles are permitted through their tunnel system.

Also contacted was the U. S. Department of Transportation Office in Detroit who now have the authority for interstate transport in place of the former ICC jurisdiction. They recommended talking with Mr. C. A. Leuttenbacher of Chevrolet Traffic as one who is an authority on interstate shipping restrictions. He was consulted and, after hearing about our projected installation, concluded that none of the regulations of which he was aware applied. He then volunteered to disucss it with certain Legal Staff people, specifically Mr. W. D. Brusstar. The enclosed copy of the letter from Mr. Malone's office is the result of this discussion.

We have estimated the quantity of hydrogen which may be contained in the 4L23 Stirling powerplant for the bus as follows:

| | Weight | Free Volume 70°F |
|---|---|---|
| Hydrogen in engine only[1], excluding exterior plumbing | .03 lbs. | 5.77 cu. ft. |
| Plumbing, manifolds, pumps, etc. | .012 lbs. | 2.31 cu. ft. |
| Reserve | .008 lbs. | 1.54 cu. ft. |
| TOTAL | .050 lbs. | 9.62 cu. ft. |

Previous to starting up, and with only 1 atmosphere of hydrogen in the engine and plumbing system, the reservoir bottle will contain the entire .05 pounds. Our present plan calls for a storage bottle in the form of an accumulator, having an elastomer diaphragm with hydrogen on one side and an inert gas such as nitrogen or Freon 14 on the other side. The maximum storage pressure will be about 2000 psi at an operating temperature estimated to be around 200°F (warm weather and after several pumping cycles). When the bottle discharges to a minimum pressure of about 1200 psi, the diaphragm will be near its extreme position and the bottle will contain only a small amount of reserve hydrogen. The minimum storage pressure needed to fill the engine to full load working pressure is about 1150 psi. The use of an accumulator type storage reservoir reduces the quantity of hydrogen stored to about half that required if a simple non-partitioned type container were used.

Propagation of a flame in a hydrogen-air mixture depends on the percent of hydrogen and the direction of propagation[2]. For combustion vertically upward,

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1 At normal full load mean pressure of 1500 psi.

2 Bureau of Mines, Bulletin #503.

4% by volume of $H_2$ in the mixture will just propagate, but will only consume a portion of the "fuel". It requires a 6% mixture to propagate horizontally, and 9% for vertically downward. About 10% is required to guarantee all the hydrogen will be burned. Tests have shown that for 5.6% mixture, half the hydrogen was consumed for upward burning.

The volume of free hydrogen at 70°F which might escape from the powerplant to the atmosphere could range from about 7 cu. ft. if certain hardware on the engine failed to about 9.6 cu. ft. from a failure on both the engine and the storage bottle. Assuming a uniform mixture 5% hydrogen-in-air and leakage of 9.6 cu. ft., a room having a volume of air in excess of 192 cu. ft., (5' x 5' x 7-2/3'), would assure negligible flame propagation in case of ignition. If ignition occurred and half the hydrogen was consumed, which is highly unlikely, the heat release would be approximately 1300 BTU. This is equivalent to less than 1/14 lb. of gasoline or about 1/6 the contents of a "bantam" can of propane, the smallest normally stocked in hardware stores. Flame speed at this low hydrogen concentration is not known precisely, but is less than 1 ft. per sec. In contrast to hydrogen, the lower flammability limit for propane lies between 2.2 and 2.4%; for methane, it lies between 5.5 and 6%; for butane, between 1.6 and 2.2%; for gasoline, about 1.4%. The lower flammability limit of hydrocarbon gases is substantially independent of the direction of propagation, in contrast to the strong dependence for hydrogen. Therefore, at the lower limit, hydrocarbon gases are more likely to be entirely consumed if ignition takes place. It is true that the upper flammability limit is much greater for hydrogen than for the other gases (74% vs. 6 to 14%), but only in the case of non-habitable spaces would this be of serious concern.

If a leak does occur, it is important to know how quickly the hydrogen gas will disperse into the surrounding air. The diffusion coefficient for hydrogen ranges from 4 to 8 times that of most hydrocarbon gases. Numerous tests concerned with spillage of liquid hydrogen have shown that the edge of the gas cloud moves away from the source at a rate roughly between 3 and 6 ft. per sec., depending somewhat on the initial concentration. Similar results were found for closed room and open air experiments. Therefore, in a large room, an instantaneous leakage of all the hydrogen from a 150 BHP engine would be dissipated to a non-flammable mixture in a few seconds. Hydrocarbon gases disperse more slowly and, being heavier than air, tend to remain concentrated in low areas. It would appear that chances for flame propagation or explosion with hydrocarbon gases are certainly greater than for hydrogen, even if the volume of an enclosure were sufficient to insure less than minimum concentration after uniform mixing.

In summary, I believe that the potential danger of fire or explosion resulting from hydrogen leakage from the 4L23 powerplant in the bus installation is extremely small. Certainly it presents no danger from the standpoint of vehicle storage in a garage and substantially none from leakage into the passenger compartment or even into the engine compartment itself. If a person smoking a cigarette were standing very near the engine when a leak occurred, and by chance the hydrogen jet was concentrated directly in his face, then a flash of fire could likely occur. The chances of this happening seem very remote. The possibility of fire from a liquid hydrocarbon fuel source would appear to be a far greater threat.

Worth H. Percival, Head
Mechanical Development Dept.

WHP:sdw

Att.

cc: A. F. Underwood - GMR
R. F. Thomson - GMR
H. O. Flynn - GMC T&C
A. T. Baynes - DDED

# APPENDIX C

## CONTRIBUTIONS TO THE STIRLING ENGINE TECHNOLOGY 1959-1970 BY GENERAL MOTORS RESEARCH LABORATORIES AND OTHER GENERAL MOTORS' DIVISIONS (WHERE NOTED)

1. Developed smokeless, turbine-type burners and direct ignition of diesel fuel.

2. Developed complete self-contained Stirling Ground Power generator sets and introduced these to the U.S. Army Research Laboratories for their evaluation. These were the only modern low-noise engine packages to pass formally specified military performance tests.

3. Developed the first precision control constant speed governor for Stirling engines, which embodied entirely new concepts to the field of governing and which accomplished the following:

    a. Speed variation at constant load $\pm$ 1/3%.

    b. No load to full load droop 1/4%.

    c. 100% sudden load change off speed surge of 4% and recovery in 6 seconds.

4. Developed the first automatic fuel controls for starting a Stirling engine over a wide ambient temperature range without smoke emissions.

5. Developed first engine to be manually started and operated entirely from hydraulic controls.

6. Developed two new hydrogen compressor systems having zero leakage capability.

7. Developed the first mechanical temperature control of proven reliability for many thousands of hours.

8. Initiated and completed analysis and tests of low cost regenerator materials particularly the proprietary MetNet material, which have shown a reduction in cost per horsepower over 50 to 1.

9. Made detailed cost and design studies with several General Motors' Divisions of a 50 horsepower single cylinder engine, which included cost reduction techniques in cylinder construction, preheater sheet metal construction, and simplified crankcase construction.

10. Made the following contributions to the analysis of the Stirling cycle:

   a. Correction of error in cooler calculation so that thermodynamic cycle heat rejection and cooler heat transfer rate were equal.

   b. Introduced routine flow measurements of heater, cooler, regenerators, and assemblies to provide correct data for calculation of flow losses.

   c. Developed theory for heat flux distribution on the Stirling heater tubes and ran experiments to show effect of radically different heat flow pattern.

   d. Made first analytical and experimental determination of heat transfer film coefficients on the heater tubes and concluded they were 2 to 3 times greater than could be correlated from the literature.

   e. Developed original analysis of combined thermal and pressure stresses in non-uniform wall cylinders.

   f. Added and refined calculations related to drive mechanisms.

11. Developed piston sealing technology which included the following items:

   a. Initiated systematic piston seal studies to eliminate white metal clearance seals which were Philips' standard at the beginning of program.

   b. Made first screening of Teflon and other self-lubricated materials (Allison).

   c. Developed method of measuring piston leakage in operating engine.

   d. Provided experimental verification of accuracy of piston seal leakage power loss.

   e. Developed data indicating optimum combination of piston seal leakage and friction.

   f. Introduced concept of controlling relationship between working space and buffer space pressures by slots in cylinder walls.

   g. Conceived basic idea for present Rulon piston rings independently and simultaneously with Philips. The development of the rings used in all Philips and GMR engines is a closely integrated result of contributions from both sources.

12. Developed new engine drive mechanisms and explored different types of cylinder-piston arrangements, including the following:

   a. Designed, built, and tested the first modern, four-cylinder, 350 HP phase angle control engine, which could also be directly reversed (Electro-Motive Division).

b. Designed, built, and tested the first "W" configuration, double-acting engine of 140 HP (Electro-Motive Division).

c. Introduced the idea of swash plate type axial Stirling engines and provided analytical and experimental verification of low friction characteristics of properly designed swash plate mechanisms with hydrodynamic type bearings.

d. Designed, built, and tested first hermetically-sealed 10 HP engine with pressurized crankcase (Allison).

e. Designed and tested single crank displacer engine of 10 HP.

f. Designed and tested first minimum weight, 2 HP "V" type displacer engine, with separated cylinders, which could be adapted to isotope heat sources.

g. Designed and built the first 120 HP, four cylinder, in-line, double-acting Stirling engine for a proposed bus installation. The General Motors program was cancelled as testing just started.

13. Developed special fuels and heat sources as follows:

a. Designed, built, and tested the only potassium-sodium (NAK) heated engine, under a Government contract (Allison).

b. Introduced the concept of thermal energy (heat) storage with the Stirling engine (initial work on heat storage began at GM Research in the early 1950s).

i. Designed, built, and operated the first heat storage powerplant using special shaped aluminum oxide pellets provided by AC Division.

ii. Operated the first Stirling engine automobile (the Calvair) with heat supplied from an aluminum oxide heat storage container.

iii. Introduced the concept and conducted tests to establish suitability of lithium-fluoride and lithium hydroxide as energy storage materials, including a joint program with Oak Ridge National Laboratory which determined suitability of various alloys for containment of lithium fluoride.

c. Introduced the concept and performed first tests to demonstrate controlled combustion of lithium fuel with Freon oxidizers, and operated the first Stirling engine from lithium combustion heat.

d. Operated the first Stirling engine with natural gas fuel.

14. Designed, built, and operated the 10 HP engine generator sets for the first Stirling-electric hybrid passenger cars, known as the Stir-Lec I and II. Also made the first exhaust emission tests of a Stirling powered vehicle.

15. Performed first endurance tests approaching "Military Standard" severity on a Stirling generator set.

16. Made the following application and design studies:

    a. Designed complete Stirling engine submarine powerplant with LiF heat storage and submitted proposal to the U.S. Navy in 1959.

    b. Made design study of high performance swash plate drive torpedo engine of 600 HP.

    c. Made design studies of 10 and 30 HP Stirling-LiF thermal energy storage plants for small research submarines, and 1000-5000 HP plants for larger military submarines incorporating aluminum oxide heat storage.

    d. Designed rhombic engines for installation in buses.

    e. Made conceptual design of swash plate automobile engine for Oldsmobile.

    f. At the Allison Division, made conceptual design studies of a solar heater powerplant, a chemically fueled space powerplant, an isotope heated space powerplant, a rhombic drive torpedo engine, an ASW powerplant to operate from hydrogen peroxide, a residential air conditioner engine, and a back-pack powerplant to operate from indigenous fuels.

17. Published for general public, eight engineering papers on Stirling engines, including their heat sources and application studies.

18. Published internally, 330 research reports and technical memoranda at the Research Laboratories.

W. H. Percival
Oct. 6, 1970

SEARCH / GENERAL MOTORS GM RESEARCH LABORATORIES

The Stirling external-combustion engine has now shown the high efficiency, quiet operation, and high specific output long predicted for it. Its low exhaust emission characteristics are

# THE NEW STIRLING ENGINE

currently an added attraction. GM Research is building and designing a variety of experimental Stirling powerplants ranging from a few horsepower to several hundred horsepower, with an eye to commercial product applications.

More than 150 years after its invention, and 30 years since it was revitalized by modern technology, the Stirling engine has shown its inherent capabilities. GM Research engineers have now built and extensively tested various configurations of this external combustion engine and demonstrated that it has

● High efficiency and output. Efficiencies are equal to or better than those of the best internal combustion engines. It has about twice the thermodynamic efficiency of steam or vapor engines, the other external combustion engines being discussed these days. For engines of several hundred horsepower, the Stirling will be about the size and weight of a similar diesel engine; in small sizes, of tens of horsepower, it will be larger but more efficient than a gasoline engine; in special applications, it is capable of as much as 200 hp per cubic foot of engine (more than 10 times the specific power of normal mobile gas turbines or turbocharged diesels).

● Quiet operation. A three kilowatt engine-generator set built by GM Research for the Army is virtually inaudible at a distance of 100 yards.

● Clean exhaust. With heat provided by a burner using diesel fuel, the Stirling engine has carbon monoxide and hydrocarbon exhaust emissions far below the limits set by federal standards.

Fuel versatility, typical of external combustion engines, can be an added advantage. Actually, "heat source versatility" is more broadly descriptive, since the

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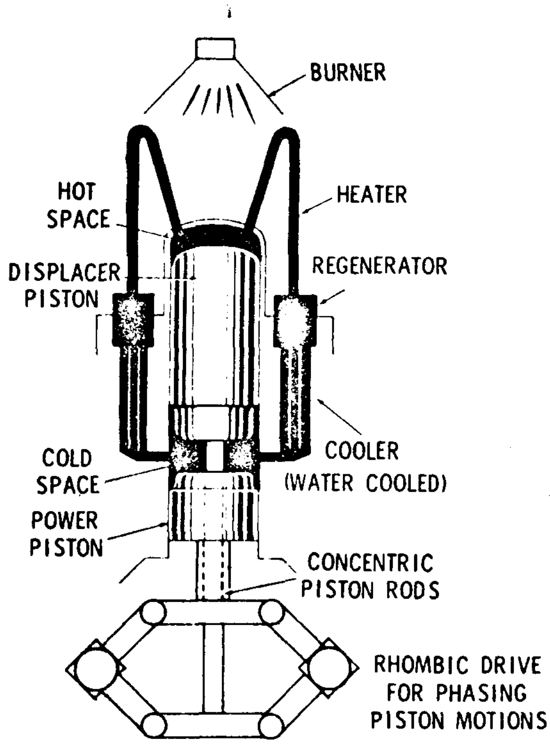



The displacer piston moves the gas back and forth. The displacer may be in the same cylinder as the power piston or in a separate cylinder (as in the VEE-1), or, in a multicylinder engine the gas motion can be handled by the power pistons in proper phase.

Stirling can use any high-temperature supply to heat the working fluid in the engine. Thus stored heat, radioisotope heat, or other nonburning sources can operate the engine with no exhaust emissions at all and keep it independent of conventional fuel supplies (for space or underwater application, for example).

## How It Began

The "old" St'rling engine was invented in 1816 and was manufactured in small quantities for many years. It served in a variety of applications requiring simple, safe power sources. But then electric motors and gasoline engines took over the market and the Stirling faded. Why? It was too inefficient and too bulky. A lack of knowledge of thermodynamics left the Stirling a victim of competition.

Then in 1937 the Philips Co. of The Netherlands (N. V. Philips' Gloeilampenfabrieken) revived the Stirling, with the help of modern technology. Over the next 20 years, Philips developed a Stirling cycle engine that could compete in efficiency and output with internal combustion engines.

In the meantime, GM Research had begun an extensive study of external combustion engines, including both vapor and gas cycle engines. After some 10 years of independent research, GMR made a formal agreement with Philips (in 1958) for an intensive cooperative research and development program on Stirling engines.

The "new" Stirling evolved from the application of modern thermodynamics and heat transfer principles and a new understanding of how the engine really works – along with improvements in materials, bearings, lubricants, and seals, to increase output, efficiency, and durability. In the past 10 years GM Research has

## The Stirling Cycle

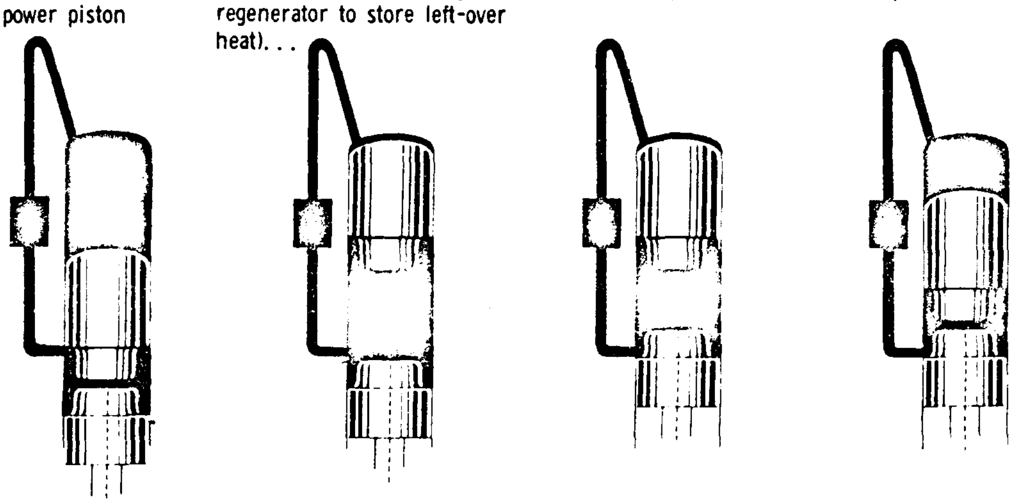
The Stirling engine uses a working fluid -- usually hydrogen or helium -- that is contained in the engine.

The compressed fluid is expanded in the hot space to drive the power piston down . . .

. . . then is moved temporarily to the cold space (through a regenerator to store left-over heat). . .

. . . is recompressed by the power piston. . .

. . . and is returned to the hot space for the next cycle.



# The Stirling-Electric Car

To demonstrate how a low-emission Stirling engine might be used in conjunction with an automotive electric drive system, a small Stirling was installed in a 1968 Opel Kadett. The vehicle uses an ac electric drive system powered by batteries, with the Stirling engine driving an alternator for battery charging. This Stirling-Electric Hybrid, Stir-Lec I, was first revealed to the public last spring and has since been demonstrated extensively.

The Stir-Lec car development features an ac electric drive system that is a further refinement of the general type used in the GM Electrovair II.

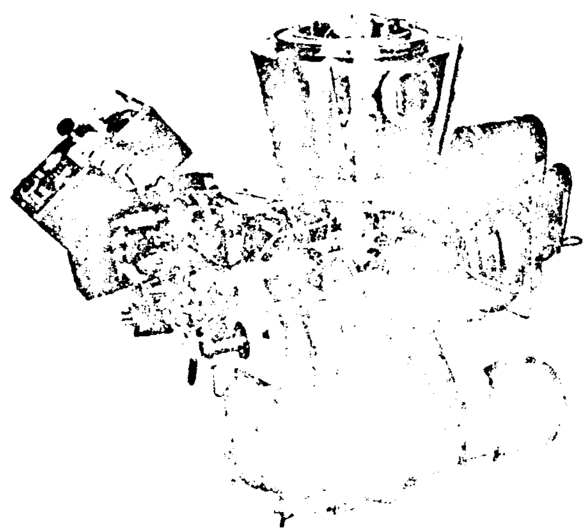
The Stirling engine for the battery-charging package is a modified GPU Stirling equipped -- for the first time -- with remote controls, so that it can be operated from the driver's compartment. Controls were converted to electrical operation and an electric starter was added.

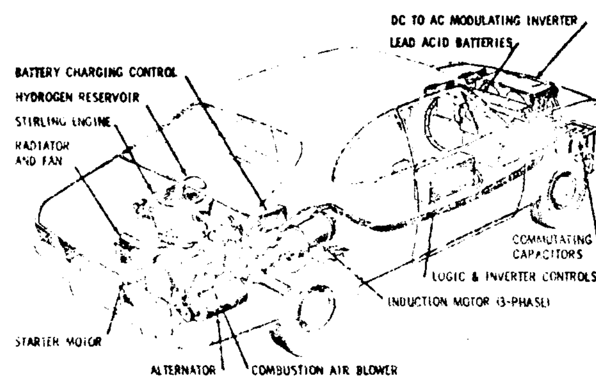
The 7 hp Stirling engine drives a three-phase alternator, which has the output rectified to charge the 14 lead-acid batteries. Battery output then goes through a modulating inverter to provide variable frequency ac to the vehicle's drive motor -- a 20 hp three-phase induction motor.

Stir-Lec I, which weighs 1100 pounds more than a standard Opel Kadett, has a top speed of about 55 mph and can accelerate from zero to 30 in 10 seconds.

For completely emission-free operation, the car can run on batteries alone (range depends on the mode of operation and, thus, the battery discharge rate). However, with the Stirling engine running, exhaust emissions are still very low. Carbon monoxide emission is typically about 0.01% (federal standards limit such emission to 1.5%); exhaust hydrocarbons typically are only 2 parts per million (federal limit is 275 ppm); emission of oxides of nitrogen is currently about half that of present gasoline engines.

This vehicle was built primarily to demonstrate that such a general system was possible with existing technology and equipment. It can serve as a test bed for further development.






developed, built, and tested efficient engines for a diversity of applications, and has brought the Stirling engine close to commercial usefulness.

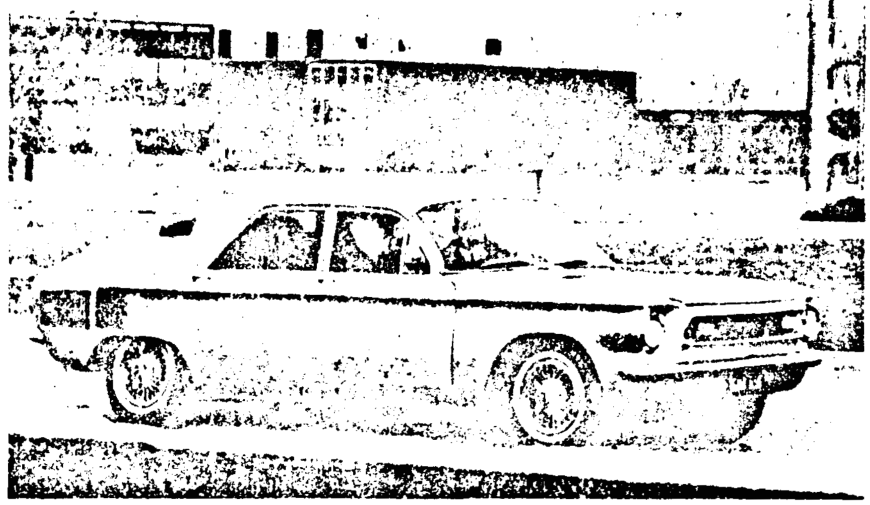
Like any heat engine, the Stirling produces power by compressing the working gas when it is cold and letting it expand -- do work -- when it is hot (see illustration). A metal mesh regenerator is the key to the modern engine's efficiency. The heat it recovers would otherwise be dumped into the cooling system and lost. Years of experience in heat transfer and regenerator design were behind this development.

## Engines Built—or Underway

GM Research has built some 25 experimental engines using the basic rhombic drive system. Most of these have been used for research; however, several have served to demonstrate the Stirling in particular applications. In 1966, the Army evaluated a three-kilowatt portable GMR Stirling engine-generator set for field use. This Ground Power Unit, GPU-3, was preceded by earlier versions in 1961 and 1963. Fuel economy, quiet operation, and durability have continued to improve with each model even though the 1966 model satisfied the requirements of the government contract.

An adaptation of the GPU-3 engine is combined with an electric drive system in an experimental hybrid car, Stir-Lec I.

This is the second Stirling engine installed in an automobile by GM Research. In 1964 GMR installed a



"CALVAIR" Car under test at the GM Technical Center in 1964.

30 hp Stirling in a Corvair, supplying the heat energy to the engine from a tank of heated alumina. Such a system is too heavy and expensive for a practical vehicle.

The principle of running a Stirling from stored heat has primarily been investigated for other applications -- current attention in this program is directed toward a propulsion system for a small submarine. In this submarine application the combination of rechargeable heat storage "batteries" plus the Stirling engine can store at least six times the useful energy of a lead-acid battery-motor system of equal weight.

GMR engineers have studied a number of possible multiple-cylinder Stirling engine configurations. All things considered -- efficiency, size and weight, cost and ease of manufacture, vibration, etc. -- a single-crank in-line seems the most promising basic configuration for a new line of compact engines in a variety of horsepowers. This type of engine can be made to deliver more pow r for its size than the rhombic engines (it has about one third as much engine volume per horsepower as the GPU-3 engine).

Although the GPU engines were designed for essentially constant-speed operation for generators, the Stirling is not limited to this type of operation. The in-line engines are being explored for a variety of traction applications and would operate over a wide speed range. Engine output is controlled by changing the amount of the working fluid in the cylinder. If the engine output is increased, by adding fluid supplied from a reservoir, more fuel is automatically supplied to keep the heater temperature constant.

## Where We Stand

At the present time the situation looks like this: The Philips Co. is working with German and Swedish manufacturers on Stirling engines for world markets. Philips also is doing military sponsored engine research. GMR is continuing basic Stirling engine and component research and is concentrating on demonstration of potential engine products. Currently there is market potential for

- Compact 4 cylinder, in-line, commercial powerplants -- for quiet boat propulsion in the 120 to 300 hp range; for low-smog intracity truck and bus propulsion in the 300 hp range; and for quiet engine-generator sets for the recreational vehicle market (boats, trailers) in the 15 to 20 hp range.

- Military specialty engines for torpedoes, submarines, and other quiet and lightweight powerplant uses.

GM engineers have now accumulated some 22,000 hours of operating time on experimental Stirling powerplants ranging from 3 hp single-cylinder engines to 400 hp four-cylinder engines. These tests have shown the efficiency, economy, and reliability of the new Stirling engine.

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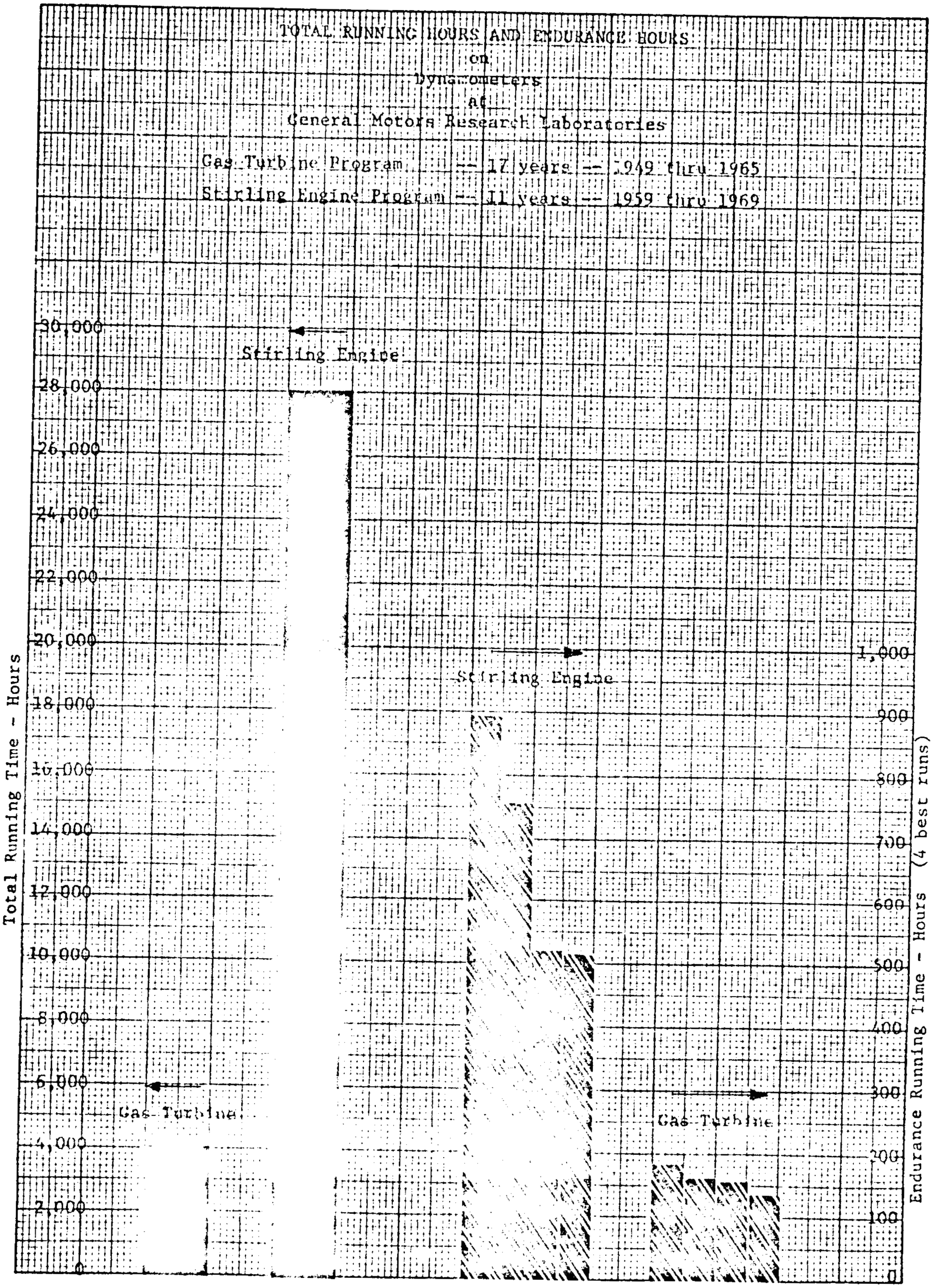
*prepared by D. W. Borland*

APPENDIX E



TOTAL RUNNING HOURS AND ENDURANCE HOURS
on
Dynamometers
at
General Motors Research Laboratories
Gas Turbine Program -- 17 years -- 1949 thru 1965
Stirling Engine Program -- 11 years -- 1959 thru 1969
Total Running Time - Hours
Endurance Running Time - Hours (4 best runs)
Stirling Engine
Stirling Engine
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