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GRAPHITE HELICAL-SCREW EXPANDERS AND COMPRESSORS

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HIGH-TEMPERATURE TESTING AND EVALUATION OF GRAPHITE HELICAL-SCREW EXPANDERS AND COMPRESSORS*

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Abstract

A graphite expander, utilizing intermeshing helical rotors and following concepts of Lysholm and the Svenska Rotor Maskiner firm, is proposed for potential use with inert gas Brayton cycles. Two demonstration assemblies were tested. Each assembly consisted of a graphite expander and an identical compressor. Each assembly was installed in an argon atmosphere furnace where the motor-driven compressor provided pressurized argon to drive the expander. The graphite-on-graphite sliding bearings used were adequate for demonstration purposes but probably not for application. Work is under way on graphite roller and gas bearings. In the first assembly the compressor failed catastrophically due to rotor dragging. This defect was corrected in the second assembly which ran for 274 hr at furnace temperatures between 2350 and 2730° F before voluntary shutdown. Expander inlet temperatures are estimated to have exceeded 2900° F. Speeds to 14,100 rpm in the compressor and efficiencies to 37% in the expander were achieved. Shaft power to the compressor exceeded 10 hp and from the expander approached 1 hp. Pressure ratios in excess of 1.5 were achieved. For space application, expander specific weights below 1 lb/electrical kW appear plausible.

Introduction

Ultra-high-temperature uncooled turbines made of graphite seem to be feasible. Adaptation of existing helical screw machine designs, to allow utilization of the outstanding high-temperature properties of graphite, appears to offer significantly greater hope for success than a similar effort using bladed turbine designs. While the efficiencies of helical screw machine designs do not match the best achievable in bladed turbines, in most situations a tradeoff is possible between turbine efficiency and turbine inlet temperature. Calculations indicate that for large space power systems an expander specific weight of 1 lb/kWe is plausible.

Description of Helical Screw Machine

The most highly developed helical screw machine is that held under patent by the Swedish firm, Svenska Rotor Maskiner, A. B. The machine^{1, 2, 3} is manufactured, principally as an industrial compressor, under license with that firm by a number of U. S. and foreign companies. Compressor efficiencies as high as 85% have been obtained.^{1, 2} Communications with the

*Work performed under the auspices of the U. S. Atomic Energy Commission.

Swedish firm reveal that expander efficiencies are practically identical to compressor efficiencies except at high pressure ratios where expanders are more efficient. There is reason to believe that high temperature graphite engines can do as well. Optimum pressure ratios for these machines appear to be in the range of 2 to 5.

Figure 1 illustrates the essential parts of the helical screw machine operating as a compressor. Shown schematically are inlet and outlet ports;

FIGURE 1. HELICAL SCREW MACHINE OPERATING AS A COMPRESSOR: (A) OUTLET PORT; (B) SIDE VIEW OF ROTORS; (C) INLET PORT

also illustrated is the manner in which internal compression (or expansion) is obtained. Most commonly these machines are run with timing gears to insure no contact between rotors. The ratio of the speed of the male rotor to that of the female rotor shown in Figure 1 is 1.5.

Of the nonmetallic refractory materials, graphite is a logical choice of materials for this

application because it has an excellent strength-to-weight ratio to very high temperatures (it has an order-of-magnitude density advantage over refractory metals). Other advantages of graphite are its exceptional thermal stress resistance, ready availability, and good machinability. Pyrolytic graphite has the disadvantage of very low strength along one crystallographic axis. This disadvantage appears to outweigh the advantage of the very high strength along the other axes. Thus, the more isotropic common grades are favored. The short-time strength of most grades of graphite increases with temperature to a maximum at about 5000° R.

Test Assembly and Furnace

In order to assess the feasibility of a graphite helical-screw expander driven by high temperature inert gas, a demonstration engine was built and operated. The advice and assistance of the Svenska Rotor Maskiner firm were secured in designing this engine. Two identical rotor-pair units were assembled into the same housing inside a furnace (2800° F capability) having an argon atmosphere. One unit was shaft-driven as a compressor. The pressurized gas so derived was used to drive an expander which, in turn, drove a generator serving as shaft load. The two units were not coupled mechanically except through the hot gas. Figure 2 gives the basic dimensions of the rotors. Figure 3 is a section through the first test assembly. The high pressure

FIGURE 2. BASIC ROTOR DIMENSIONS (INCHES)

flow passage between the compressor and expander is drawn with dashed lines on the thrust bar side of the engine (Figure 3). The low pressure flow passage on the other side forms a venturi to permit measurement of the circulating flow. This venturi was calibrated in a separate test program using a full scale model of the flow passages together with room temperature air as the metered fluid. In order to apply the calibration in cold air to hot argon, the data were re-plotted in dimensionless parameters of flow coefficient vs Reynolds number. To aid in on-the-spot evaluation of flow, a set of calibration curves were calculated and plotted to give flow rate in the test engine as a function of temperature, inlet pressure, and differential pressure. The low scatter and good repeatability of the calibration data permit a high level of confidence in the flow data generated by the high temperature helical-screw engine test. Total error is $\pm 6\%$ or less for all but the very low flow range of the data. No change in venturi dimensions was noted after nearly 400 hr of operation.

Bearings within the furnace in all cases were graphite-on-graphite. This decision was made after several high temperature bearing tests were completed at LRL. The most common

form of helical screw engine has relatively large lateral loads due to the pressure differential acting on the rotors near the high pressure port. Because conventional bearings with high efficiencies are not designed for the temperature range of which the graphite engines are capable, the need developed for a 3000° F bearing, which had suitable wear characteristics and losses low enough to give a meaningful demonstration. All bearing tests used a 1-in. diam graphite shaft riding in two 1-in. long journal bearings with a load applied through a third journal bearing 2 in. long. All tests were in an argon atmosphere. From the bearing test program it was concluded that the dry graphite-on-graphite bearing gave a usable coefficient of friction as well as simplicity for the demonstration engine.

Two units (demonstration engine) were assembled into the furnace for the high temperature tests. A brief account of the tests follows, they have been described in greater detail in a separate report.⁴

First High Temperature Assembly Test

The first test assembly did not progress very far toward interesting operating conditions, but enough useful information was obtained, both negative and positive, to warrant a brief description.

The intention in designing this assembly was to have only one bearing in the furnace for each rotor, specifically that on the short stub shaft (see Figure 3). This single bearing proved inadequate, as discussed later. Tubing for the three pressure taps was made of a platinum-rhodium alloy. Both the case and the rotor were fabricated from ATJ grade graphite,^{*} and the bushings were grade M-85 graphite.[†] The thermocouple located in the venturi block was unsheathed platinum-rhodium. The pyrometer port was a closed-end rhenium tube inserted into the high pressure gas stream and brazed to the graphite with a mixture of platinum and silicon.

While the furnace temperature was being raised to 2400° F, all three of the pressure tap tubes melted somewhat, this melting was attributed to formation of a low melting alloy of platinum and silicon from furnace insulating bricks. The melting resulted in loss of all pressure information. Loss of the thermocouple in the machine housing occurred, but was of less concern because furnace control thermocouples located near the roof of the furnace remained operative.

The compressor had been run for slightly over 25 min to speeds as high as 7000 rpm on the male rotor when internal breakage occurred. Post-run inspection revealed a number of broken corners from the blade-like portions of the female rotor, as well as a break in the shaft of that rotor. The nature of the breaks, and of marks in the compressor housing, indicated that considerable scraping of the housing by the rotor had taken place, apparently due to misalignment resulting from thermal expansion of various parts in the assembly.

^{*}Carbon Products Division, Union Carbide Corporation.

[†]Advance Carbon Products, Inc.

FIGURE 3. CROSS SECTION OF COMPRESSOR-EXPANDER ASSEMBLY

Second High Temperature Assembly Test

Modifications

Following the lessons learned from testing the first assembly, a second assembly was tested. Relative to the first assembly, changes were as follows:

1. Pressure taps to the two existing ATJ housing parts were modified to consist of graphite pipes (5/8-in. o.d. $\times$ 3/16-in. i.d.) with necessary graphite elbows.* These were fabricated from graphitite G-grade graphite.† They were attached to the graphite parts with threads and commercial cement.

2. Rotors were fabricated from H-205 graphite.‡ This grade has properties similar to the ATJ grade but is available in a somewhat larger size, which permits cutting the rotors with an orientation such that the properties in all directions normal to the axis of the rotor are the same. The expansion coefficient of this graphite is somewhat higher than that of the ATJ grade in the housing; data on both appeared to be adequate to take account of this in the designs. The shoulder on the shaft end of the rotor was increased in height by 0.002 in. for increased protection against tip dragging.

3. Bushings of M-85 graphite were fitted into the venturi block with 0.002 in. radial clearance (hot) from the shaft, as shown in Figure 4.

4. The Pt-Pt 10% Rh thermocouple in the venturi housing was encased in an alumina tube

FIGURE 4. GRAPHITE BUSHINGS INSTALLED IN VENTURI BLOCK FOR SECOND TEST ASSEMBLY

which was completely closed inside the furnace to prevent harmful interaction with silicon.

5. The 3-hp, 7500-rpm electric motor used in testing the first assembly was replaced with a hydraulic motor driving through V-belts so as to be capable of a speed of nearly 15,000 rpm on the male compressor rotor and a power of over 10 hp.

6. Alterations were made in the method of mounting the assembly to minimize uncertainties in high temperature alignment of the parts.

Average radial tip clearance in the second assembly was 0.008 to 0.010 in. at operating temperature. This clearance is rather high, both by

* One of these pipes with elbow is seen in Figure 5.

† Basic Carbon Corporation, a division of the Carborundum Company.

‡ Great Lakes Carbon Corporation.

intention and some deviation from the desired dimension in the housing assembly.

Prior to testing the second assembly, compressor rotors from the first assembly were altered by boring the rotors and pressing them onto steel shafts. In order to assess the possibility of running a graphite helical-screw engine without gears, these two rotors were run together, properly spaced, but not in a housing. They ran at a speed of 6000 rpm (male rotor) for over 100 hr. During the first 48 hr there was a weight loss of 0.90 g. During the remainder of the running time there was a very small apparent weight gain, probably due to pickup of atmospheric moisture or oil from bearings. No breakage occurred.

Test Phases

Testing of the second assembly was divided into three phases

Phase A — Furnace temperature initially at 2340° F and later raised to 2440° F (furnace shut down and engine partially disassembled and inspected after Phase A).

Phase B — Furnace temperature, 2450° F (furnace not shut down between Phases B and C).

Phase C — Furnace temperature, 2700° F (raised to 2730° F toward end of test).

Phase A Testing Furnace Temperature, 2340° F and 2440° F. After the furnace had been stabilized at temperature it was evident that torques required to turn both the compressor and the expander were higher than they should have been, apparently due to rotor expansion and some machining error. A run-in period of about 12 hr was conducted during which the expander, as well as the compressor, were motor-driven. Speeds during this run-in were gradually raised from a few hundred to 4500 rpm (compressor male rotor speed).

The remainder of the time devoted to Phase A testing was spent either running at steady conditions for long periods, or obtaining data wherein the compressor speed was held constant and the expander speed was varied. In setting conditions for the steady-state endurance running, the speed of the expander was taken to be the most important quantity, therefore, the expander was loaded very lightly. During all of Phase A, the assembly was tested with gears on both the expander and the compressor (see Figure 3)

The maximum speed achieved in the male rotor of the compressor was 11,700 rpm. The total running time on the compressor at speeds of 500 rpm or greater was slightly over 50 hr. Most of this time was accumulated at a steady compressor speed of about 9000 rpm. Total running time on the expander at substantial speeds was about 4 hr less than that on the compressor, due principally to the time at very low speeds during data taking. Shaft-to-shaft efficiencies were generally less than 10%, with individual component efficiencies in some cases exceeding 30% (efficiencies are defined in the data summary section). The highest pressure observed at the expander inlet was 7.6 psig with the expander stalled and the compressor running at 10,900 rpm. Alignment of the pyrometer well was such that sighting to the bottom of the well was partially obscured by the edge of the furnace port, making pyrometer data

of less value in Phase A than in subsequent phases where the vision blockage had been corrected.

Toward the end of Phase A testing, difficulty was encountered with torque cell bearings. The test was discontinued, and the opportunity was taken to cool the furnace, partially disassemble the engine, and inspect the parts.

There was some evidence of oxidation of graphite surfaces external to the assembly. This oxidation is readily apparent on the housing and pipe at "A" and "B" in Figure 5. Such loss of material through oxidation is also apparent on both

FIGURE 5. HOUSING BLOCK AFTER PHASE A TESTING OF SECOND TEST ASSEMBLY (OXIDATION OF GRAPHITE SURFACES AT A AND B)

the expander shafts at "A" in Figure 6. During subsequent operations increased precautions were taken to prevent air from entering the furnace

Figure 7 shows part of the second test assembly in the furnace after removal of the housing block. The polish near the tips is evidence of slight tip dragging. This end of the rotors is the high pressure end, and it is suspected that the dragging was due to hot, high pressure gas causing axial expansion preferentially near the tips. The shallow holes in the ends of the rotors in Figure 7 are for balancing.

Phase B Testing Furnace Temperature, 2450° F. Before the units were reassembled into the furnace, a few thousandths of an inch of material was removed with emery paper from regions of the parts where rubbing had occurred.

When the furnace had reached 2450° F, an attempt was made to resume operation. Repeated binding of the expander unit was encountered, apparently due to slippage in the steel-to-graphite

FIGURE 6. ROTORS AFTER PHASE A TESTING OF SECOND TEST ASSEMBLY (EXPANDER, ABOVE, COMPRESSOR, BELOW, OXIDATION OF GRAPHITE SURFACES AT A)

FIGURE 7. SECOND TEST ASSEMBLY IN FURNACE AFTER REMOVAL OF HOUSING BLOCK

shaft taper joints. Removal of the timing gears from the expander relieved this difficulty, and satisfactory operation followed for the remainder of Phase B and Phase C testing.

The thermocouple installed in the rotor housing block was no longer operative at this time, apparently having been damaged during handling of the assembly after Phase A.

During Phase B, the maximum speed of the compressor male rotor was 14,000 rpm. At this speed the centrifugal stress was about 900 psi — established in a photoelastic stress analysis of the male rotor (the most highly stressed of the two rotors).^{*} A photoelastic model was tested using the photoelastic frozen stress technique.

Total time on the compressor was 27.9 hr at speeds over 500 rpm. As in Phase A, most of this time was spent at a steady 9000 rpm. The expander operating time at all but very low speed was about 2.6 hr less than for the compressor. The highest pressure observed was 6.8 psi for a compressor speed of 10,800 rpm with the expander stalled; this reduced pressure, compared to Phase A, is probably the result of wear-enlarged leakage passages and possibly some contribution from the sandpapering away of material on rotors between Phases A and B. Efficiency data were similar to the data of Phase A.

Phase C Testing: Furnace Temperature, 2700° F and 2730° F. For the Phase C testing, the furnace temperature was brought directly up to 2700° F.

All of the Phase C testing was conducted with no gears on the expander. The total compressor running time over 500 rpm during Phase C was 124.4 hr. As before, most of this time was at a steady 9000 rpm. Gears were on the compressor the first 70.3 hr of this time; gears were not on the compressor the last 54.1 hr. Toward the end of Phase C the expander was slower for a given input pressure. This was believed due to misalignment of steel and graphite shafts arising from nonuniform creep of the bricks in the rear furnace wall. In an effort to test this hypothesis, furnace temperature was varied between 2650 and 2730° F.

The temperature decrease caused an increase in both expander and compressor torque. The temperature increase caused no observable change. The furnace was left at 2730° F for the last 19 hr of compressor operation. The maximum speed achieved at a furnace temperature of 2700° F was 14,100 rpm and at 2730° F was 12,000 rpm. Efficiencies during the early portion of Phase C were the best observed for the entire test program, with shaft-to-shaft efficiencies of over 10% and with individual component efficiencies to 37%.

Stalled expander pressures were increased in going from 2450 to 2700° F. This increase appears due in part to the fact that rotors continued to out-expand the case and in part to an increase in viscosity of the gas. Substantial wear in the compressor was evident during Phase C. Early in this phase the stalled rotor pressure was 5.59 psi at a compressor male rotor speed of 8850 rpm. Toward the end of this phase, 2.65 psi was attained for compressor male rotor speed of 9000 rpm.

The expander speed dropped during the latter part of Phase C, probably due to misalignment of the steel and graphite shafts. It seemed clear that continued operation would be possible, but that continued wear and degradation of performance would take place. Since speeds and times attained had already more than achieved goals for this test, the machine was shut down and disassembled.

^{*}This higher stress results from the fact that there are sharper reentrant corners on the male rotor to act as stress raisers and also that its speed is 1.5 times that of the female.

Post-Test Examination

Examination showed that wear had taken place in various parts of the assembly, ranging up to 0.030 in. in regions of greatest severity. The processes leading to this wear have been traced as much as is feasible, and appear to be largely involved with nonuniform creep of the rear furnace wall causing tilting of the assembly and with the difficulty of maintaining desired alignment of the assembly parts during temperature changes in the furnace. Figure 8 shows wear marks in the compressor side of the housing block, and Figure 9 shows corresponding marks on the compressor rotors.

FIGURE 8. HOUSING BLOCK SHOWING DRAG OF COMPRESSOR ROTOR TIPS ON CYLINDRICAL PORTIONS OF HOUSING BORES (PHASE C TESTING OF SECOND ASSEMBLY)

FIGURE 9. COMPRESSOR ROTORS SHOWING WEAR ON EDGES

Data Summary for Second High Temperature Assembly Test

Efficiency Definitions

Power delivered to the compressor shaft is:

$$\dot{W}_c = 2\pi \tau_c N_c, \quad (1)$$

where

τ_c is the measured compressor shaft torque
 N_c is the measured compressor male rotor speed

The same equation yields power delivered to the load when expander torque and male rotor speed are used. The overall efficiency of the machine as a fluid coupling is then

$$\eta_0 = \dot{W}_e / \dot{W}_c \quad (2)$$

where the subscript c designates compressor and the subscript e designates expander.

The separate efficiencies of the compressor and the expander are of particular interest. These relate the shaft power of each machine to the ideal power associated with isentropic compression or expansion between the high and low cycle pressures. This isentropic power can be derived directly from the perfect gas law and the isentropic expansion law $p/p_0 = (\rho/\rho_0)^\gamma$ as:

$$\dot{W}_i = p_h \frac{\dot{w}}{\rho_\ell} \left(\frac{p_\ell}{p_h} \right)^{1/\gamma} \left(\frac{\gamma}{\gamma-1} \right) \left[1 - \left(\frac{p_\ell}{p_h} \right)^{\frac{\gamma-1}{\gamma}} \right] \quad (3)$$

where

p_h = high cycle pressure, absolute units
 p_ℓ = low cycle pressure, absolute units
 ρ_ℓ = low cycle density (calculated from pressure and temperature)
 $\dot{w}$ = measured mass flow rate
 γ = ratio of specific heats for argon = 1.67

The efficiency of the compressor can be written

$$\eta_c = \dot{W}_i / \dot{W}_c, \quad (4)$$

and that of the expander

$$\eta_e = \dot{W}_e / \dot{W}_i. \quad (5)$$

By this definition, we still have the overall (fluid coupling) efficiency as

$$\eta_0 = \eta_c \cdot \eta_e = \dot{W}_e / \dot{W}_c \quad (6)$$

Volumetric efficiency of the compressor is defined as measured volume flow rate divided by theoretical flow rate. Volumetric efficiency of the expander is defined as theoretical volume flow rate divided by the actual measured flow rate. In both cases, theoretical volume flow is the swept volume displacement times the male rotor speed.

Presentation of Data

Most of the useful data from the successful tests of the second assembly were taken during operation at 2700° F furnace temperature, and only that data will be presented here. Table I lists parameters determined at various operating conditions, and the following series of figures presents relationships of special interest which have been selected from the same data.

Figure 10 shows compressor efficiency as a function of compressor male rotor speed. As reference to Table I will show, the data at each indicated value of compressor male rotor speed were taken at a variety of values of pressure ratio and gas flow rate. Nevertheless, the compressor efficiency values all fall within the rather narrow limits indicated by the height of the bars. The

TABLE I. PARTIAL DATA FROM PHASE C TESTING (2700° F FURNACE TEMPERATURE)

Data point No	Input ^a speed (rpm)	Output ^a speed (rpm)	Input power (kW)	Output power (kW)	Pressure ratio	Gas flow (lb/sec)	Compressor efficiency (%)	Expander efficiency (%)	Compressor volumetric efficiency (%)	Expander volumetric efficiency (%)
210	1500	0	0.089	0	1.014	0.0065	11.8	0	b	b
208	3000	155	.249	0.001	1.058	.0077	26.8	1.4	b	b
209	3000	0	.260	0	1.065	.0075	28.4	0	b	b
205	4500	840	.621	0.010	1.082	.0100	19.7	8.1	56.2	33.2
206	4500	600	.621	0.011	1.085	.0098	20.3	8.4	55.3	24.1
207	4500	0	.568	0	1.099	.0086	22.8	0	48.7	0
201	6000	1505	.900	0.018	1.112	.0129	24.0	8.2	54.6	46.0
202	6000	975	0.971	0.035	1.113	.0118	34.3	14.7	49.8	32.6
203	6000	610	1.042	0.032	1.140	.0115	23.3	13.4	48.7	20.9
204	6000	0	1.089	0	1.157	.0105	32.9	0	44.3	0
34	9000	4080	1.667	0.160	1.170	.0217	31.7	30.2	60.7	74.9
35	9000	3350	1.846	.178	1.184	.0210	30.1	32.1	58.7	63.5
36	9000	3050	2.024	.217	1.211	.0205	30.9	34.7	57.2	59.3
37	9000	2600	2.166	.231	1.221	.0197	29.1	36.6	55.0	52.6
38	9000	2060	2.400	.232	1.262	.0192	30.5	31.7	53.6	42.7
39	9000	1620	2.720	.230	1.289	.0187	20.0	29.2	52.3	34.4
40	9000	1100	2.826	.182	1.314	.0181	29.3	22.0	50.8	24.1
41	9000	510	3.068	0.116	1.351	.0172	28.5	13.3	48.2	13.2
42	9000	0	3.302	0	1.380	.0166	27.6	0	46.6	0
108	10500	4680	2.734	0.167	1.224	.0250	27.7	25.5	59.6	74.8
109	10500	4400	2.859	.261	1.228	.0245	27.7	32.8	58.5	71.7
110	10500	3700	3.107	.306	1.258	.0237	28.1	36.0	56.6	62.2
118	10500	3200	3.397	.303	1.272	.0220	26.2	34.1	54.6	55.8
111	10575	3100	3.505	.330	1.296	.0231	27.7	34.0	54.6	53.7
113	10500	2520	3.728	.328	1.323	.0223	27.6	31.6	53.3	45.1
112	10537	2200	3.991	.338	1.344	.0220	27.1	31.2	52.5	39.8
115	10500	1700	4.060	.282	1.357	.0211	26.6	26.0	50.5	32.1
116	10500	1030	4.474	.196	1.385	.0202	24.9	17.4	48.5	20.2
119	12150	5600	4.123	.331	1.245	.0295	24.0	20.1	60.4	76.3
120	12000	4900	3.977	.348	1.279	.0280	27.2	26.8	58.1	70.4
121	12000	4330	4.356	.359	1.300	.0270	25.8	31.9	56.1	64.3
122	12000	3720	4.924	.462	1.343	.0266	25.7	36.5	55.2	56.2
123	12150	3300	5.273	.430	1.368	.0260	25.2	32.2	53.4	50.9
124	12150	2820	5.753	.484	1.396	.0255	24.5	34.3	52.5	44.2
125	12000	2400	5.776	.440	1.422	.0247	25.2	30.2	51.5	38.9
126	12000	2170	5.776	.411	1.422	.0233	24.0	29.7	48.6	37.2
213	13500	6200	5.220	.222	1.272	.0325	22.7	18.6	59.5	77.2
214	13500	5400	5.326	.336	1.306	.0313	24.4	25.8	57.4	69.7
216	13500	4730	5.966	.392	1.276	.0295	23.1	28.5	54.4	64.5
217	13500	4100	6.392	.474	1.426	.0284	22.9	32.2	52.2	58.2
218	13500	3100	7.137	0.440	1.463	0.0276	22.7	27.1	50.9	45.0

^aAll rotation speeds refer to male rotor of unit being discussed.^bIndeterminate because of very low flow.

FIGURE 10. COMPRESSOR EFFICIENCY VS SPEED

data at 4500 rpm and below comprise relatively few points, and are considered less reliable than that at higher speeds because of difficulty in taking accurate readings at low values of pressure ratio and torque. The efficiency observed is about 25% less than that reported in private communication with Mr. H. Nilsson of the Svenska Rotor Maskiner

firm for commercial low temperature machines of under 5 hp. This efficiency is higher than might be expected in view of the crude bearings in the high temperature units and the known excessive clearances.

Figure 11 shows compressor efficiency as a function of the ratio of high to low cycle pressure. The available data for existing low temperature machines continue to show fairly constant efficiency up to pressure ratios of 3 or 4. An apparent tendency toward an earlier dropoff in the high temperature machine may be due to pressure forces causing misalignment and high mechanical losses in the crude bearing used. The low pressure ratios attained in this test were due primarily to excessive leakage past both expander and compressor through larger-than-anticipated clearances.

In Figure 12 the expander efficiency is presented as a function of expander male rotor speed for several constant compressor male rotor speeds.

FIGURE 11. COMPRESSOR EFFICIENCY VS PRESSURE RATIO

FIGURE 12. EXPANDER EFFICIENCY VS SPEED

Lines have been drawn in to aid the eye in following connected data points. Note that the expander efficiency peaks at a much lower speed than that of the compressor.

Figure 13 presents expander efficiency as a function of pressure ratio, and Figure 14 shows the relationship of pressure ratio to both expander male rotor and compressor male rotor speeds.

FIGURE 13. EXPANDER EFFICIENCY VS PRESSURE RATIO

Graphite Engine Element Development

When the high temperature engine test series had been completed, development of critical elements needed to improve efficiency, life, and simplicity of graphite expanders was undertaken. These tests show an increasing dependence on

FIGURE 14. PRESSURE RATIO DEPENDENCE ON COMPONENT SPEEDS

room temperature testing. Such demonstrations should be conservative relative to high temperature performance because graphite strength increases with temperature and because graphite-on-graphite friction decreases with temperature. In addition to the following development work, another engine is in fabrication to incorporate seal strips on the rotor tips wearing against a silicon carbide layer in the graphite case. A larger and higher pressure engine is also planned.

Roller Bearings

To date, two radial needle bearings (no cage) and one radial roller bearing with cage have been tested; it is too early to draw any firm conclusions. However, results thus far are encouraging, and indicate that bearing losses could probably be reduced by a factor of 15 with the use of graphite roller bearings (with cage) rather than sliding bearings.

The three radial bearings tested had an inner race diameter of 1-3/4 in. and 0.25-in. diam rollers with an overall length of 1 in. Both the inner and the outer races were graphite, as well as the rollers. The needle bearings had 25 rollers and the caged roller bearing had 15 rollers inserted in the graphite cage. The major difference between the first and second needle bearings tested was the location of the roller containment flange on the inner rather than the outer race.

The first needle bearing tested showed an increase in coefficient of friction with speed for all loads up to 133 lb (maximum tested). This increase was from approximately 0.01 at 1000 rpm to 0.03 at 3000 rpm. When the second bearing was first assembled and run, the coefficient of friction for speeds up to 6000 rpm and loads up to 89 lb remained approximately 0.005 to 0.006. However, when the load was increased to 111 lb, the coefficient of friction did increase with speed, and had reached 0.031 at 4000 rpm.

In all cases the increase in the coefficient of friction with speed was accompanied by an increase in thrust load, indicating skewing of the rollers.

The caged roller bearing gave a more or less constant coefficient of friction with speed. The following is the coefficient of friction at 6000 rpm for loads up to 1000 lb (maximum tested):

Load (lb)	Coefficient of friction based on inner race radius
110	0.0068
264	0.0050
420	0.0045
624	0.0057
825	0.0072
1000	0.0088

The bearing was run for 32-1/2 hr at speeds from 1000 to 10,000 rpm. Twenty-four hours of this time was at 9200 rpm with a load of 264 lb. Another 2-1/2 hr was accumulated at 6000 rpm with a load of 1000 lb. At the end of the total period the bearing was inspected. The following observations were made:

- (1) Roller diametrical wear in the range of 0.002 to 0.004 in.
- (2) Race diametrical wear less than 0.001 in.
- (3) Roller length wear: six rollers had worn between 0.010 and 0.050 in.
- (4) Cage wear only at area of contact with roller ends; wear was approximately equal to roller end wear for a given roller.

Calculated load capacities of graphite roller bearings are higher than one might anticipate because the low modulus of elasticity results in good load distribution. It seems reasonable to expect a load capacity of 25% of a similar steel bearing if the deflection does not become a limiting factor.

Gearless Rotor Wear Test

In order to obtain some estimate of the wear rate of two graphite Lysholm rotors running together without timing gears, the same pair of graphite rotors mentioned on page 4 were run together at a constant 6000 rpm and room temperature for 2003 hr. The results reported here include times given on page 4 and are all room temperature data. Only bearing friction and windage loss loads were transmitted to the female shaft for the first 1100 hr, because this should be the case for a well-designed Lysholm engine. After 1100 hr, a light electrical generator load was added in an attempt to accelerate the wear rate.

The parameter chosen to indicate wear rate was the backlash between the rotors measured in mils at the pitch circle. Backlash was measured at several angular positions, and the maximum value recorded. Initially, the value was 2 mils. As shown in Figure 15, the backlash increased rapidly up to about 300 hr after which it increased very slowly. Visual indications of wear support Figure 15 qualitatively with initial wear at highly localized spots which grew into uniform bands along each lobe by approximately 300 hr.

When the load was raised at 1100 hr, the wear rate increased initially but was decreasing when the test was terminated at 2003 hr, indicating that a new equilibrium wear surface was being formed. Over the last 500 hr an average wear rate of $5 \mu\text{-in./hr}$ was noted. This wear rate is acceptable for rotor life of thousands of hours. Rotor lives of 10^4 hr appear possible with harder graphites, better lobe profile control, and high temperatures.

Test of Timing Gears in Hot Zone

If timing gears are used, there is an obvious advantage to their being within the hot zone. To

FIGURE 15. RESULTS OF WEAR TESTING OF GEARLESS ROTOR

investigate their feasibility, a pair of gears was fabricated of graphite and assembled in a graphite case as shown in Figure 16. Power to the gears was delivered through a flexible coupling and absorbed within the case by a brake acting on the driven gear. The brake was applied through an air cylinder forcing a ram downward on the upper block of the gear case assembly.

FIGURE 16. GRAPHITE TIMING GEARS ASSEMBLED IN GRAPHITE CASE FOR HOT ZONE TEST

The gears, with 28 and 42 teeth, were of 12 diametral pitch with 1-in. face width. They were cut with a 20° stub profile (Class II precision) from EP-1924 grade graphite.* Two sets of gears were fabricated and tested.

The first set of gears was run at room temperature for approximately 100 hr of intermittent operation. During this period loads up to 25 in.-lb and speeds of 8500 rpm were attained. Polishing of the gear teeth was noted, but no measurable wear was observed.

* POCO Graphite, Inc.

After the cold run-in was completed, the assembly was placed in an argon atmosphere furnace and the temperature raised to 2750° F. The first test was abruptly terminated when the shaft coupling snapped after 10 min at 8000 rpm and no load. Upon disassembly it was found that two teeth had been broken on the large gear as shown in Figure 17. The small gear was essentially undamaged except for a few small chips knocked out of several teeth.

FIGURE 17. LARGE GEAR SHOWING BROKEN TEETH

Based on the appearance of the gears and the observed action of the assembly during the cold run phase, the following reason for failure is proposed. Due to the light load on the upper gear and the high speed at the time of failure, it is presumed that the upper gear may have lifted out of its journal and twisted or impacted sufficiently to break the teeth on the large gear. The shaft subsequently broke during the crushing of tooth material which followed. The largest pieces of material remaining are shown in Figure 17.

The second set of gears was assembled in the undamaged case, and the test was resumed. After a run-in period of 16 hr at 4000 rpm and 10-in. -lb torque, the speed was increased to 6500 rpm and held constant with 10 in. -lb transmitted for 45 hr. At this time the drive motor failed, and the gears were removed for inspection while the motor was being repaired. During this period the gears were inadvertently loaded in excess of 45-in. -lb torque (maximum scale on recorder) at 6500 rpm. A defective pressure regulator controlling the ram pressure failed. The gears survived this overload without damage.

Upon removal from the furnace the gears were inspected for wear. It was apparent that their axes were not parallel because wear was concentrated on the ends of each tooth with machining marks still visible over half the tooth width on each side. This condition resulted from excessive clearance in the bushings due to differential expansion of bushing and shaft. On inspection it was determined that each tooth had a maximum taper of 0.8 mil across its full width,

while tolerance on the tooth profile for this class of gear is 0.5 mil. An additional 23 hr at 2750° F was obtained with better load distribution and no significant increase in wear.

In conclusion, a pair of graphite gears of standard design have been successfully run at 2750° F for a total of 84 hr. The gears successfully transmitted 2-1/4 hp at 6000 rpm with an overload factor of 2. Further testing is in progress to estimate useful life of such gears.

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