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

RELEASED FOR ANNOUNCEMENT

IN NUCLEAR SCIENCE ABSTRACTS

Livermore, California

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ABSTRACT

In a simple demonstration, a graphite helical-screw compressor drove an identical expander with temperatures on the low pressure side to 2730°F. Gas temperature on the high pressure side was uncertain in detail, but the temperature in the pyrometric well was as high as 135°F above the outside case temperature. This well temperature was below the actual gas temperature, but the difference has not been assessed. Rotational speeds to over 14,000 rpm and tip speeds to 230 fps were achieved in the compressor, which ran for a total of 274 hr at speeds above 500 rpm. Shaft power to the compressor achieved values above 10 hp; shaft power from the expander exceeded 0.9 hp. Efficiency of the expander approached 40%. There were some difficulties with materials. Platinum tubing and thermocouples melted after alloying with silicon. The commercial graphite cement used behaved poorly, although the tests were not critically dependent on the joints. A review of the dependence of the strength of graphite on temperature reveals that the conditions of these tests reveal in small measure the potentialities of graphite.

INTRODUCTION

Adaptation of existing helical screw machine designs to allow utilization of the outstanding high temperature properties of nonmetallic refractory materials appeared to offer significantly greater hope for success than a similar effort utilizing 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.

Two analyses dealing with the crucial radiator weight in space power systems may be cited in support of the existence of a favorable tradeoff for such an application.

1. An unpublished Brayton cycle analysis conducted at the Lawrence Radiation Laboratory (LRL) used turbine inlet and outlet pressures of 120 and 52.5 psia, respectively. Radiator inlet and outlet temperatures were varied so as to give minimum radiator area. A turbine efficiency of 90% and a turbine inlet temperature of 2500°R yielded a radiator area 9.4 times as great as that for a turbine efficiency of 80% coupled with a turbine inlet temperature of 5000°R. The cited conditions appear plausible operating limits for a metal bladed turbine vs a graphite helical-screw expander. In order to achieve the same radiator weight as with a 90% efficient turbine at an inlet temperature of 2500°R, a turbine efficiency of 47% is required if the turbine inlet temperature is 5000°R.

2. The following comparative data for Rankine cycles have been published¹: Changing the turbine inlet temperature from 1900 to 2500°R while decreasing the turbine efficiency from 90 to 80% results in a reduction in radiator area by a factor of 2.6. Comparing these two turbine inlet temperatures reveals that the same radiator area is obtained for a turbine efficiency of 90% at 1900°R and for a turbine efficiency of 35% at 2600°R.

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. This machine^{2,3,4} 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.^{2,3} Communications with the Swedish firm reveal that expander efficiencies are practically identical to compressor efficiencies, except at high pressure ratios where expanders are more efficient. 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; also illustrated is the manner in which internal compression (or expansion) is obtained. Most commonly the 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 is 1.5.

Of the nonmetallic refractory materials, graphite is a logical choice for this application because it has an excellent strength-to-weight ratio to very high temperature (it has about an order-of-magnitude density advantage over refractory metals). Other advantages of graphite are its exceptional thermal stress resistance, ready availability, and good machinability. The disadvantage of the very low strength of pyrolytic graphite

Fig. 1. Helical screw machine operating as a compressor: (a) outlet port; (b) side view of rotors; (c) inlet port.

along one crystallographic axis appears to outweigh the advantage of the very high strength along the other axes, thus pointing to more common polycrystalline grades. 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 units were assembled in the same housing inside a furnace 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. The furnace was not hermetically sealed, so the argon atmosphere was maintained by pressurizing the furnace to 1 to 2 in. of water to insure outward leakage. Figure 2 gives basic dimensions of the rotors. Figure 3 is a cross section of the first test assembly.

A number of aspects of this demonstration were from the outset far from optimum. The furnace was an existing one originally intended for air atmosphere operation. The makeshift argon atmosphere thus provided was expected to

Fig. 2. Basic rotor dimensions (inches).

Fig. 3. Cross section of compressor-expander assembly.

contain significant amounts of oxygen and silicon. Temperatures were limited by the furnace insulation to about 2800°F (at least for extended periods). The graphite-on-graphite bearings were selected as a means to operate a demonstration engine with minimum development effort and with recognition of the fact that frictional losses in these bearings and their life would likely make them unacceptable for a final application. Also, it was known that smaller engines of this type have efficiencies lower than those of larger ones.

The high pressure flow passage between the compressor and the expander is drawn with dashed lines on the thrust bar side (Fig. 3). The low pressure flow passage on the other side forms a venturi to permit measurement of the circulating flow. Calibration of this venturi is described in Appendix A.

Bearings within the furnace in all cases were graphite-on-graphite. Tests dealing with the feasibility of such bearings are described in Appendix B. Two units were assembled into the furnace for tests. These units had the same case, with modifications, but different rotors.

FIRST HIGH TEMPERATURE ASSEMBLY TEST

The first assembly test was largely a disappointment; 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 Fig. 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 an available graphite product: ATJ grade graphite.* Also, the bushings were made from available material: 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 bonded to the graphite with a mixture of platinum and silicon.

During the period of raising the furnace to the target temperature of 2400°F, some melting occurred in all three of the pressure tap tubes. This is attributed to formation of a low melting alloy of platinum and silicon. The furnace uses silicon carbide heating elements, and the insulating bricks are known to contain silicon. In a nonoxidizing

*Carbon Products Division, Union Carbide Corporation.

†Advance Carbon Products, Inc.

environment, SiO in vapor form is frequently found (deposits from such vapor have been observed in this furnace). Such vapor transport leads to formation of the low melting alloy. The Pt-4.4% Si eutectic has a melting point of 825°C. This melting, of course, resulted in loss of all pressure information. The loss of the thermocouple in the engine housing was of less concern because furnace control thermocouples located near the roof of the furnace remained operative.

The compressor was run for slightly over 25 min to male rotor speeds as high as 7000 rpm. At the end of this time parts internal to the compressor were broken. Post-run inspection revealed that edges had broken off the blade-like threads of the female rotor as shown in Fig. 4. It was apparent that this had resulted in breakage of the

Fig. 4. Female rotor of compressor from long shaft end after post-run inspection.

graphite shaft which was integral with the rotor. Inspection revealed that the relatively sharp edges of the female rotors had been dragging on the venturi block, as indicated by the circular marks at A in Fig. 5. The direction of rotation was such that the motion of the sharp edges was analogous to a cutting action. It is reasonably clear that this scraping eventually resulted in a fractured piece which jammed, causing the indentation at C in Fig. 5 and the broken edges. There then resulted broken edges on the opposite end of the rotor, and the broken shaft. All this damage apparently occurred in less than one rotor revolution. The meshing rotor was undamaged, and the case was undamaged beyond the chip at C. A shoulder 0.002 in. high had been provided, as seen in Fig. 4, supposedly to preclude this. Contact of this shoulder with the venturi block is seen at B in Fig. 5.

As indicated earlier, the original intent had been that there be no bearing located where the shaft passed through the venturi block. A radial clearance of 0.0065 in. was provided at the point, compared to the 0.002-in. radial clearance in the bushing at the short-shaft end of the rotor. Shaft bending was greater than realized early in the design stage, and the lateral pressure load on the rotors caused bending deflection, moving the shaft to the extreme position allowed by this 0.0065-in. clearance. Tilting of the rotor thus was the principal reason for the damaging tip-dragging.

Fig. 5. Venturi block showing damage by female rotors (marks made by edges of threads of female rotor at A; contact of rotor shoulder at B; indentation from jamming of fractured piece at C).

The expander never rotated continuously, but it did make slow intermittent motion in the right direction. This action was apparently due to some combination of the following factors:

1. The O-ring drag on the output shafts was objectionable. It was noticeably high when the expander was turned by hand.
2. The graphite stock on hand, as well as available sizes from the manufacturer in that grade, had dimensions such that the rotor had two different expansion coefficients in two directions transverse to its axis. The resulting ellipticity when the rotors were heated may have caused some friction between them.
3. When the furnace was heated, the graphite assembly expanded about 0.145 in. with respect to the outside machinery, due principally to the expansion of bricks in the floor of the furnace. The scheme for shaft alignment upon heating involved prior measurement of this vertical motion and repositioning of the bearing housings without direct observation of the graphite housings or shafts. This somewhat indirect method may have resulted in misalignment, causing binding on the graphite shafts.

In addition to producing objectionable torque, the O-rings were inadequate for the speeds. Those in the compressor evidenced burning on post-run inspection.

SECOND HIGH TEMPERATURE ASSEMBLY TEST

Modifications

Applying the lessons learned from testing the first assembly and using parts existing and in process, a second assembly was tested. Relative to the first assembly, changes to and differences from the portion contained within the furnace 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 grade 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 grade of graphite

* One of these pipes with elbow is shown in Fig. 7.

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

‡ Great Lakes Carbon Corporation.

is somewhat higher than that of the ATJ grade in the housing; however, data on both grades appeared to be adequate to account for this in detail. 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 grade graphite were fitted into the venturi block with 0.002-in. radial clearance (hot) from the shaft, as shown in Fig. 6.

Fig. 6. Graphite bushings installed in venturi block for second test assembly.

4. The platinum vs platinum-10% rhodium thermocouple in the venturi housing was encased in an alumina tube which was completely closed inside the furnace to prevent harmful interaction with silicon.

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

A glove box was gasketed to the rear wall of the furnace to permit maintaining an atmosphere temporarily on the outside of the rear furnace wall, and an intermediate copper plate was installed between each needle bearing housing and

the rear wall of the furnace. During heatup, the graphite shaft ends protruded through relatively close fitting holes in the copper plate. When the furnace stabilized at temperature, and with the bearing housing removed, the copper plates were aligned to the graphite shafts visually and with the aid of dial indicators. After the copper plates had been bolted to the furnace, the bearing housings were bolted on with preinstalled dowels between housings and copper plates. Then the glove box was removed. The O-rings had been removed, and in the area of the O-ring groove the clearance between the shaft and housing was reduced to about 0.002 in. This arrangement eliminated the shaft seal torque problem but required a small continuous argon leak on each shaft. Average radial tip clearance in the second test assembly was 0.008 to 0.010 in. at operating temperature. This clearance is rather high, both by intention and some deviation from desired dimension in the housing assembly.

Prior to testing the second assembly, compressor rotors from the first assembly (including the broken female rotors) were altered. Where necessary, the integral graphite shafts were removed, and the rotors were bored and pressed onto steel shafts.

In order preliminarily 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 0.90-g weight loss. During the remainder of the running time there was a very small apparent weight gain, probably due to the pickup of atmospheric moisture or oil from bearings.

Test Phases

Testing of the second assembly was divided into three phases, and the following discussion is so divided:

Phase A - Furnace temperature initially at 2340°F and later raised to 2440°F. Furnace was shut down and engine was partially disassembled and inspected after Phase A.

Phase B - Furnace temperature, 2450°F. Furnace was 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. This condition was apparently due to the fact that the H-205 rotors, upon heating, had expanded more than the ATJ case, and the fact that some dimensional error due to cutter wear affected the clearances where the two rotors meshed. 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 period were gradually raised from a few hundred to 4500 rpm (compressor male rotor speed).

The remainder of the time (Phase A testing) was spent either running at steady-state conditions principally to accumulate running time, or obtaining data wherein the compressor speed was held constant and the expander speed was varied. In setting conditions for the steady-state endurance run, 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.

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 reasonable speeds was about 4 hr less

than that on the compressor, principally due 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 as defined in Appendix C. The highest pressure observed at the compressor discharge (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. This sighting made such pyrometer data of less value in Phase A than the data in subsequent phases where the vision blockage had been corrected.

Toward the end of Phase A testing, difficulty was encountered with torque cell bearings because the shaft attached to the torque cell caused excessive loads on this bearing. The bearing overheated, causing damage to the torque cell and to the nylon shaft coupling adjacent to this bearing. The coupling was replaced several times. The test was discontinued due to these difficulties, and the opportunity was taken to (1) cool the furnace (a 40-hr process resulting from heat capacity of the furnace bricks), (2) partially disassemble the engine, and (3) 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 Fig. 7. Some loss of material through oxidation is also apparent on both the expander shafts at A in Fig. 8. It is believed that this oxidation was associated with operation of the glove box mentioned earlier. After completion of the Phase A testing, it was recognized that the action of moving one's arm inward and outward in the gloves could pump air into the glove box and thence into the furnace cavity. During subsequent operations care was taken to move slowly and to insure a positive pressure of argon in the glove box at all times. There was no visual evidence of oxidation internal to the assembly. Apparently, potential access paths for oxygen into the interior were sufficiently torturous to provide prior reaction of the oxygen.

Figure 9 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 is the high pressure end of the rotors, and it is suspected that the dragging is due to hot, high pressure gas causing axial expansion preferentially near the tips. The shallow holes in the ends of the rotors in Fig. 9 are for balancing.

There is visual evidence of contact between rotor lobes in both Fig. 8 and Fig. 9. The polished appearance on the stub shaft and on the shaft proper near the rotor in Fig. 8 is the result of contact with the bushing and not of any deliberate finishing operation. The shaft wear given in Table B-VII of Appendix B is determined from measurements on the polished-appearing region on the shaft proper and on the shaft immediately adjacent to that region but toward the rotor.

Fig. 7. Housing block after Phase A testing of second test assembly (oxidation of graphite surfaces at A and B).

Fig. 8. Rotors after Phase A testing of second test assembly (expander, above; compressor, below; oxidation of graphite surfaces at A).

Phase B Testing: Furnace Temperature, 2450°F

Before the units were reassembled into the furnace, approximately 0.002 in. of material was removed with emery paper from the polished regions visible on the rotor tips in Fig. 9. Also, approximately 0.003 in. was removed from visible regions of contact between rotor lobes. No other changes were made to the assembly in the furnace before Phase B testing, although it was found when the furnace reached temperature that the thermocouple installed in the block was inoperative. It is suspected that this was due to an accidental blow received by the thermocouple that broke the enclosing alumina tube inside the furnace. Such a fracture could result in the low temperature melting associated with silicon.

When the furnace had reached 2450°F, an attempt was made to resume operation. It was found that the expander turned relatively freely, but (after brief operation) encountered binding, which caused the expander to stop.

Fig. 9. Second test assembly in furnace after removal of housing block.

There was a brief run-in period during which the expander was motor-driven. Following this run-in period, the same result was encountered; after a few minutes of satisfactory expander operation, binding developed which stopped the expander. During an elapsed time of about 2-1/2 days and a compressor operating time of 12.9 hr (over 500 rpm), this sequence was repeated five or six times. In the course of these trials, realignment of steel shafts to graphite shafts (requiring installation and removal of the glove box) and re-timing of the gears were accomplished.

Following this period, the gears were removed from the expander. Satisfactory operation of the expander followed for Phases B and C. The evidence makes it reasonably clear that this short cycle deterioration of running torque on the geared expander was due to slippage in one or both of the graphite-to-steel tapers. Post-run examination revealed high polish on portions of the graphite taper. Insofar as bending moments in the shaft are concerned, Fig. 3 makes it clear that there is redundancy of supports. Thus, misalignment can cause a bending moment in the tapered joint. It seems plausible that the combination of torque and cyclic bending moment can cause a very slow slippage of the taper which locks a torque into the redundant torque loop formed by two shafts, two gears, and two rotors acting as gears. Such a locked-in torque will cause lateral loads between rotors and consequent friction in bearings and between rotor lobes.

The maximum speed of the compressor male rotor was 14,000 rpm. At this speed the centrifugal stress was about 900 psi as covered in detail in Appendix D. 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 9,000 rpm. The expander operating time at all but very low speed was about 2.6 hr less than that for the compressor. The highest pressure observed was 6.8 psi for a compressor speed of 10,800 rpm with the expander stalled. The reduced pressure, compared to Phase A, is probably the result of wear affecting leakage passages and possible some contribution from the sandpapering away of material on rotors during the down time between Phases A and B. Efficiency data were similar to the data for Phase A.

Phase C Testing: Furnace Temperature, 2700°F and 2730°F

The furnace was not shut down between Phases B and C. The temperature was brought to 2700°F, the furnace was allowed to stabilize, and the steel shafts were realigned to be coaxial with the graphite shafts. The realignment was necessary because of increased thermal growth of the insulation in the floor of the furnace.

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 during 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°F 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°F to 2700°F. This increase appears due in part to the fact that the 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 a compressor male rotor speed of 9000 rpm.

The hypothesis that expander speed was dropping toward the end of Phase C due to misalignment of steel and graphite shafts appears to be supported by the following explanation.

During this period the leakage through the stalled expander was lower than it had previously been for a given pressure drop. The likely explanation for this lower leakage appears to be the fact that the graphite housing assembly tipped relative to the steel shafts so as to cause the rotors to move from their normal loaded position (as in Fig. 10) to a position such that the stub shaft moved to the other extreme within the bearing. This movement would decrease rotor tip-to-housing clearance in the critical high pressure region. Post-test examination revealed that such contact had occurred in these bearings. As indicated earlier, such misalignment would be expected to produce a uniform frictional drag resulting in the slowing down of the expander.

Fig. 10. Normal loaded position of expander rotors (bearing clearances exaggerated).

Since it became clear that continued operation seemed possible but that it would result only in increased compressor wear with further degradation of performance and since speeds and times were much better than had been anticipated, operation was shut down.

Post-Test Examination

Post-test examination revealed considerable wear in the compressor as had been anticipated from stalled expander data. The greatest wear was in the bushings supporting the stub shafts. As was expected, these bushings were badly out-of-round. Table I lists increases in the measured diameter over that measured in the inspection following

Table I. Increases in compressor bushing diameter for stub shaft bushing (maximum and minimum).

Female rotor (in.)	Male rotor (in.)
0.027	0.015
0.016	0.004

the Phase A testing. The two figures for each bushing are derived from the maximum and minimum diameters of the worn bushings.

The larger amount of wear for the female rotor bushing appears inconsistent with the fact that the male rotor turns at the higher speed. It seems reasonable that the large amount of wear in these bearings was due to misalignment

arising from creep in the rear wall of the furnace. A tipping motion of the housing assembly such as to reduce leakage through the expander, as discussed earlier, is also in the right direction to cause increased wear in these compressor bushings on the side where it was observed. The difference in wear for the two compressor rotors could be due to the inability to align the two sets of shafts identically.

Wear in the corresponding expander bushings was no more than a few ten-thousandths of an inch and was, in fact, obscured by final and pre-existing out-of-roundness, the result of the use of simple hand tools for measurement.

Post-test inspection revealed that the vertical graphite plate installed in the furnace (see Fig. 3) had tipped (due most likely to creep in the bricks of the rear furnace wall) with the top edge moving toward the rear of the furnace. The slope of this graphite plate relative to the rear wall of the furnace was 0.004. The direction of this slope is consistent with both the large wear in compressor bushings on the high pressure end and with the decreased leakage through the expander. During all running time a load of about 500 lb was maintained on the thrust bar (see Fig. 3).

The bushing in the venturi block for the female rotor of the compressor showed wear indicated by an apparent increase in diameter in one direction of 0.005 in. Corresponding wear in the bushing for the male rotor was slightly over 0.001 in. Measurements of diameter for the corresponding expander bushings indicated a decrease in their bore diameter of about 0.001 in. While error of this magnitude may be inherent in the hand tools used for measurement, it seems likely that this is real (and unexplained).

The large amount of bearing wear in the compressor caused the tips of the compressor rotors to drag on the cylindrical portions of the housing bores. This is shown in Fig. 11. Resulting wear on the rotor tips can be seen in Fig. 12.

Fig. 11. Housing block showing drag of compressor rotor tips on cylindrical portions of housing bores (Phase C testing of second assembly).

It is apparent in Fig. 12 that bearing wear was sufficient to cause contact between the shaft and the venturi housing (reduced diameter) immediately adjacent to the rotor proper. The combination of displacement of the compressor rotors due to bearing wear and of loss of tip material caused an increase in clearance on the high pressure side of the female rotor of about 0.030 in. and on the high pressure side of the male rotor of about 0.023 in. Clearances on the low pressure side were decreased essentially to

Fig. 12. Compressor rotors showing wear on edges.

zero. Clearances on the high pressure side should have a dominant effect in controlling compressor leakage, and thus the clearance changes are consistent with the observed increased leakage in the compressor.*

Shaft wear on the stub shafts was slightly less than 0.001 in. on all rotors. On the long shafts, where supported by the bushings in the venturi block, the wear was a few ten-thousandths of an inch for all rotors.

Figures 12 and 13 show that the male rotor of the compressor was in contact with the venturi block over the entire low pressure end of the rotor. Apparently, the pressure force on this rotor was greater than on the female rotor. Also, thrust loads, due to gear action in the rotors, tend to cause this contact. The pneumatic shaft thrust

*Experiments with sealing ribs and case coatings to reduce leakage flow by maintaining a minimum clearance between rotor lobes and the case are described in Appendix E.

Fig. 13. Venturi block after Phase C testing showing contact of male rotor of compressor with block.

cylinders in Fig. 3 were so arranged that equal loads went to each rotor in the compressor (or expander). In general, pressure in each of these cylinders was adjusted to maximize the speed of the unit under consideration.

The bushings in the venturi block had been cemented in place, and also the two graphite pipe elbows had been threaded and cemented in place with a commercial graphite cement. Three of the four bushings were loose upon disassembly. The fact that both of the graphite elbows were unscrewed by hand from the two housing blocks indicates the inadequacy of the cement.

There may have been additional oxidation on the exterior surfaces continuing the pattern observed in Fig. 6. In addition, a hard green coating had developed on some of the exterior surfaces near the silicon carbide heating elements. Presumably, this was a silicon carbide coating.

RESULTS

The graphite helical-screw compressor drove the graphite helical-screw expander with temperatures on the low pressure side to 2730°F. Gas temperature on the high pressure side was uncertain in detail, but temperature in the pyrometric well was as high as 135°F above the outside case temperature. This well temperature was below the actual gas temperature, but the difference has not been assessed.

Rotational speeds to above 14,000 rpm and tip speeds to 230 fps were achieved in the compressor, which ran for a total of 274 hr at speeds above 500 rpm. Shaft power to the compressor achieved values above 10 hp; shaft power from the expander exceeded 0.9 hp. Efficiency of the expander approached 40%.

Platinum tubing and thermocouples melted due to alloying with silicon. The commercial graphite cement used behaved poorly, although the tests were not critically dependent on the joints. No other difficulties with materials were encountered.

CONCLUSIONS

The essential ruggedness and durability of the compact helical screw engine have been demonstrated by the two test engines and individual component testing. The chunky rotors are surprisingly tolerant of contact with each other and with the case. All of the limitations on life and durability so far exposed by the test program are directly traceable

to the difficulty in attaining and maintaining reasonable alignment through the massive furnace wall. Improved test equipment and techniques are expected to remove these limitations.

Efficiency of the helical screw engine may be improved substantially by several methods. Development of high temperature anti-friction bearings offers an immediate improvement in efficiency of up to 50%. Closer control of case-rotor clearances, or the addition of sealing ribs, should reduce bypass leakage and significantly improve performance. The efficiency of these engines is known to increase with size, and the size of the present engine ranks it with the smallest for which data exists.

Significant increases in both efficiency and specific output may be obtained by more fully utilizing the potential of graphite as a high temperature engineering material. All limitations imposed on temperature, life, and speed during the present test program were due to the limitations of external test equipment.

The development potential of the graphite helical-screw engine is interesting for both space and ground-based power application. The ability of the engine to utilize very hot gas or vapor leads either to high thermodynamic efficiency or to low radiator mass as desired.

The engine might be used as a topping stage in a conventional Rankine or Brayton cycle, or, through compounding two or more, might constitute the entire expansion process. Use of similar engines in refrigeration systems shows a high tolerance of entrained liquid or wet vapor in the working fluid. However, it appears doubtful that graphite will be compatible with Rankine cycle fluids.

ACKNOWLEDGMENT

The contribution of R. J. Rothstein in preparing the photoelastic stress analysis of the male rotor (Appendix D) is acknowledged.

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4

Helical Screw Engine Venturi Tests

To measure gas flow in the high temperature helical-screw engine tests, a venturi flowmeter was built into the engine block. The 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. The brass model (Fig. A-1) was machined to the hot dimension of the graphite engine block flow passages. The model was calibrated utilizing dimensionless parameters to permit application of cold air data to hot argon flow.

Fig. A-1. Venturi model (low temperature).

The flow model was installed in the LRL blowdown facility downstream of the standard orifice meter section. The venturi was calibrated against the standard meters. Air flow was exhausted at atmospheric pressure as in the helical screw engine. After steady-state flow was established, readings of differential pressure, absolute inlet pressure, and inlet temperature were recorded in the standard orifice section and in the test venturi.

The absolute pressure upstream of the venturi was measured by a Heise gauge with 0.05 psi graduations. The upstream temperature was measured by an iron-constantan thermocouple and millivoltmeter. The venturi differential pressure was measured by a precision manometer with 0.002-in. graduation. Maximum flow error due to instrumentation was estimated at less than 1.7% over the entire range of calibration.

Eight flows were measured in the range from 0.015 to 0.07 lb/sec with both increasing and decreasing flow. Three standard orifices were required to cover this range of flow; therefore, where the ranges overlapped extra points were taken. The 20 data points are plotted in Fig. A-2 as venturi differential pressure vs mass flow rate. The data exhibit small scatter with a high degree of linearity. The low scatter is encouraging in view of the high turbulence that must be developed just upstream of the throat by the sharp 90-deg turn.

In order to apply the calibration in cold air to hot argon, the data were replotted in dimensionless parameters of orifice coefficient vs Reynolds number. The orifice

coefficient or pressure ratio is defined as the ratio of differential pressure to velocity head in the throat:

$$K = \frac{2 (P_e - P_t)}{\rho_t U_t^2}$$

where

P_e = venturi entrance pressure

P_t = throat static pressure

ρ_t = density of gas in throat

U_t = velocity of gas in throat.

Velocity and density in the throat are obtained from inlet conditions by assuming isentropic flow in the venturi.

The orifice coefficient, K , is correlated against the throat Reynolds number:

Fig. A-2. Venturi calibration curve for 70°F air.

$$R_e = \frac{\rho_t U_t D_t}{\mu_t}$$

where

ρ_t = is as defined above

U_t = is as defined above

D_t = throat diameter at mean furnace temperature

μ_t = absolute viscosity of gas at throat temperature.

The throat gas properties are again determined from measured pressure, temperature, and differential pressure by assuming isentropic flow.

Fig. A-3. Dimensionless venturi calibration.

The data of Fig. A-2 are replotted in Fig. A-3 using the above dimensionless parameters. The scatter is increased due to multiplication of measured quantities. The overall scatter, however, is still less than $\pm 5\%$. The curve of Fig. A-3 was obtained by a least squares fit of the data to the standard curve fitting equation:

$$\ln K = \sum_{j=0}^n a_j (\ln Re)^j \quad n = 1, 2, 3, \dots$$

This equation is merely the logical extension of the simple form (where $n = 1$):

$$\ln K = a_0 + a_1 \ln Re$$

or

$$K = e^{a_0} (Re)^{a_1},$$

which generates a straight line on log-log graph paper. The best fit of the general equation to the actual data occurred with $n = 3$. The coefficients, a_j , were computed by well known curve fitting techniques⁵ resulting in the final equation:

$$\ln K = -52.193 + 14.413 (\ln Re) - 1.3164 (\ln Re)^2 + 0.040037 (\ln Re)^3,$$

from which the curve of Fig. A-3 was generated.

This equation has been written into the data reduction code used for the engine test data, along with the definition of K and Re , given above. Gas properties for hot argon were obtained from National Bureau of Standards data.

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. Curves were plotted for temperatures from 2000 to 3000°F at 50°F intervals. On each curve, flow rate is plotted as a function of differential pressure with inlet pressure as a parameter. A typical calibration curve is shown in Fig. A-4. For actual test data points the temperature, inlet pressure, and differential pressure recorded as data are fed into the digital data reduction code, resulting in an output flow rate for the given data point.

Fig. A-4. Typical venturi calibration curve for 3000°F argon.

Errors in measured flow rates arise from several sources besides the uncertainty of calibration. Especially at low flow rates, instrument error may exceed the uncertainty in calibration. Absolute entrance pressure was measured by a water manometer with 0.1-in. graduation. Differential pressure was measured by an inclined mercury manometer with 0.01-in. graduation. Temperature was measured by a platinum vs platinum-10% rhodium thermocouple inserted into the gas stream for Phases A and B of the test. For Phase C, mean furnace temperature was used because of failure of the direct instrumentation. At some of the low flow rates in the test instrument, error may be as high as 20%. At rated flow (0.035 lb/sec) instrument error is less than 1.5%.

Change in throat diameter due to temperature variation is negligible over the range of temperatures at which the engine operated. Enlargement due to possible oxidation or erosion can be determined only by a steadily decreasing indicated flow rate or by ex-post-facto examination of the venturi. Upon examination following almost 400 hr of operation at temperature, no measurable enlargement was noted.

In conclusion, 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 over all but the very low flow range of the data. No change in venturi dimensions was noted after nearly 400 hr of operation.

APPENDIX B

High Temperature Bearings for Helical Screw Engines

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. As 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.

HIGH TEMPERATURE BEARING TESTS

Several high temperature bearing tests were completed at LRL. All tests used a 1-in. diam graphite shaft riding in two 1-in. journal bearings with a load applied through a third journal bearing 2 in. long. The test fixture is shown in Fig. B-1.

Fig. B-1. High temperature bearing test fixture.

No torque cell was available for Tests I through V, and for these tests the bearing coefficient of friction* was calculated from the electrical power minus the sum of no-load power and I^2R losses of the motor. All tests were in a positive pressure argon atmosphere.

Test I

Shaft material: ATJ graphite
Journal material: ATJ graphite
Diametrical clearance: 0.0002 in.

*The bearing coefficient of friction is defined as the torque divided by the lateral force and bearing radius (this should not be used as sliding friction data).

Table B-I.

Temp (°F)	Running time (min)	Speed (rpm)	Bearing load (lb)	Total motor power (kW)	Bearing coefficient of friction (μ)
2460-2500	13	4700-4800	50	0.61	0.147
2500-2660	17	4800-4900	100→70	1.2→1.1	0.158

Remarks: No measurable wear, although shaft and bearings were scored.

Appreciable oxidation of graphite was observed.

Total revolutions: About 6×10^5 .

Test II

Shaft material: ATJ graphite

Journal material: ATJ graphite

Diametrical clearance: 0.005 in.

Temperature: 2950°F

Table B-II.

Running time (min)	Speed (rpm)	Bearing load (lb)	Total motor power (kW)	Bearing coefficient of friction (μ)
7	2700	50	0.25	
15	2700	100	0.43→0.50	
30	6000	98	1.3 →1.2	0.14 →0.125
105	6000	100	1.2	0.128
30	6000	100	1.1	0.108
25	6000	100	1.1 →1.6	0.108→0.178

Remarks: Oxidation negligible

Alumina brick supporting bearing assembly squashed 3/4 in.

Rise in motor power during the last 25 min is thought to be due to the shaft rubbing on the furnace wall when blocks deformed 3/4 in.

Reduction in bearing losses from Test I could be due to increase in clearance.

Test III

Shaft material: ATJ graphite

Journal material: Boron nitride.

Table B-III.

Temp (°F)	Running time (min)	Speed (rpm)	Bearing load (lb)	Total motor power (kW)	Bearing coefficient of friction (μ)
2825-2870	6	3600	25	0.24	
2870	10	3400	100	1.02 → 0.83	
2800	30	6000	100	1.9 → 0.53	0.215
2800	5	6000	0	0.53	

Remarks: Near end of test no change in power when load was removed, indicating bearings were not supporting load.

Bushing wear (boron nitride): 0.170 in. in 50 min.

Test IV

Shaft material: ATJ graphite

Journal material: Advance Carbon Products M-85 graphite

Diametrical clearance: 0.0055 in.

Temperature: 2850°F

Load: 100 lb

Speed: 6000 rpm

Note: See Fig. B-2 for coefficient of friction as a function of time.

Remarks: Shaft out of round 0.001 in. after test and cold. This would approximately correspond to the differential thermal expansion of hot shaft in the two radial directions, indicating that shaft wore round when hot.

Negligible oxidation.

Some scoring on shaft and bushings.

Test V

Shaft material: ATJ graphite

Journal material: Advance Carbon Products M-85 graphite

Temperature: 2400°F

Lubricant: Molten tin injected into bearing when noted in data.

Fig. B-2. Data from Bearing Test IV.

Table B-IV.

Running time (min)	Speed (rpm)	Bearing load (lb)	Total motor power (kW)	Bearing coefficient of friction (μ)	Remarks
8	6000	52	0.815	0.141	---
4	6000	100	1.18	0.122	---
2	6000	100	1.54	0.171	Tin injected
2	6000	100	1.53	0.170	---
4	6000	100	1.43	0.154	Tin injected
2	6000	100	1.42	0.155	---
11	6000	100	1.52	0.17	Large amount of tin injected
5	6000	100	0.97	0.0915	---
6	1000	51	0.158→0.10	---	---
5	1000	51	0.081	---	Tin injected
10	6000	100	1.62	0.183	---

Remarks: More scoring than without tin.

Test VI

Shaft material: ATJ graphite

Journal material: Advance Carbon Products M-85 graphite

Torque cell used to determine coefficient of friction

Note: Test primarily to check tapered coupling

Table B-V.

Running time (min)	Temp (°F)	Speed (rpm)	Bearing load (lb)	Torque (ft-lb)	Bearing coefficient of friction (μ)
13	2350	1800	20	0.204	0.120
1	2400	1800	40	0.353	0.104
1	2400	1800	60	0.584	0.114
1	2400	1800	80	0.780	0.115
8	2450	1800	100	1.02	0.122
8	2500	1800	100	0.978	0.117
8	2200	1800	100	0.978	0.118
7	2150	600	100	0.978	0.119
8	1800	600	100	0.952	0.114
3	1775	1800	100	0.860	0.104
2	1760	3000	100	0.843	0.101

Remarks: Shaft broke when going to 6000 rpm

Test VII

Shaft material: Great Lakes (H205) coated with Microseal*

Journal material: Advance Carbon Products M-85 coated with Microseal*

Temperature: 2700°F

Torque Cell used to determine coefficient of friction

Table B-VI.

Running time (min)	Speed (rpm)	Bearing load (lb)	Torque (ft-lb)	Bearing coefficient of friction (μ)
15	6000	22	0.240	0.131
2	~ 5	50	0.364	0.088
3	1000	49	0.392	0.095
2	2000	48	0.403	0.101
4	3000	48	0.43	0.107
4	4000	48	0.403	0.101
3	5000	47.5	0.415	0.105
5	6000	100	0.858	0.103
3	5000	100	0.881	0.106
2	4000	100	0.858	0.103
15	3000	100	0.858	0.103

Remarks: No apparent advantage to Microseal for these conditions.

Shaft wear approximately 0.0002 in. on diameters, which would indicate removal of Microseal, which was about 0.0001 in. thick.

Towards end of test, squeaking was heard from bearings.

As coefficient of friction did not decrease with speed, it is assumed no lift came from the gas.

GENERAL DISCUSSION

From the above testing it was concluded that the dry graphite-on-graphite bearing gave the minimum coefficient of friction as well as simplicity for the combinations tested. It was also concluded that the Advance Carbon Products M-85 bushings gave a decrease in the coefficient of friction from the ATJ bushings. The following choice of materials was then made for the first two sets of graphite helical-screw engines:

* A commercial surface seal sometimes used to reduce dry friction.

Set 1 - Shaft: ATJ graphite (rotor material)

Bushings: Advance Carbon Products M-85

Set 2 - Shaft: Great Lakes H205 (rotor material)

Bushings: Advance Carbon Products M-85

After approximately 50 hr of operation (following Phase A testing), the second assembly was shut down due to instrumentation problems. At this time the graphite machines were disassembled and inspected for wear. Each rotor shaft was measured, both at the venturi block bearing and just off the bearing. The results are given in Table B-VII.

Table B-VII.

Rotor	Shaft diameter off bearing (in.)	Shaft diameter at bearing (in.)
Female compressor	1.3760	1.3759
Male compressor	1.3760	1.3759
Female expander	1.3759	1.3757
Male expander	1.3759	1.3759

Bearing wear following Phases B and C testing is discussed along with the description of that testing.

BEARING EFFICIENCY

In the first and second assemblies of the helical screw units so far tested, an attempt was made to reduce the thrust loads in the graphite bearings to very low levels, and to carry them, instead, in ball thrust bearings outside the furnace. The following analysis is made on the assumption that there is no loss in the graphite thrust bearings. The power for bearings carrying lateral loads in a helical screw machine is:

$$W_b = 2(2\pi \tau N^1) = 2\pi(P_2 - P_1) A \mu r N^1 \quad (\text{Eq. B-1})$$

where

r = radius of bearings

N^1 = average speed of two rotors assuming load is evenly distributed between rotors

$P_2 - P_1$ = difference between high and low machine pressures

A = effective area of differential pressure for two rotors

τ = torque

The power out of a lossless, leakless expander for isentropic expansion is

$$W_E = \frac{\gamma}{\gamma - 1} P_1 V N'' \left[\left(\frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right] \quad (\text{Eq. B-2})$$

where

γ = ratio of specific heats

V = swept volume per revolution related to low pressure side

N'' = speed of rotor on which the volume is based.

Combining Eqs. (B-1) and (B-2),

$$\frac{W_b}{W_E} = \frac{2\pi A r N' \mu \left(\frac{P_2}{P_1} - 1 \right)}{\frac{\gamma}{\gamma - 1} V N'' \left[\left(\frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right]}$$

For a given geometry of rotors, linear scaling of rotor dimensions including the bearings, a constant coefficient of friction, constant gamma, and constant pressure ratio,

$$\frac{W_b}{W_E} = \text{constant}$$

This ratio is independent of temperature, pressure, size and speed. For present helical screw engine configurations at LRL and argon gas

$$\frac{N'}{N''} = 0.833$$

$$A = 11.8 \text{ in}^2$$

$$r = 0.563 \text{ in. (average radius of both bearings)}$$

$$V = 23.5 \text{ in}^3$$

$$\frac{P_2}{P_1} = 1.25$$

and using

$$\mu = 0.1$$

then

$$\frac{W_b}{W_E} = 0.16 \text{ (isentropic expansion only).}$$

Considering real engines and adding losses for thrust bearings, this ratio would approximate 25%.

Other types of bearings being considered at this time are gas bearings, bearings with rolling rather than sliding friction, and bearings with liquid metal as a lubricant.

APPENDIX C

Data Summary for Second High Temperature Assembly Test*

Data taken during the test of the second high temperature assembly were processed by a digital code written specifically for this test program. The primary purpose of this code was to convert all measured quantities into consistent units and compute individual component efficiencies.

Power delivered to the compressor shaft is simply

$$\dot{W}_c = 2\pi \tau_c N_c$$

where

τ_c = measured compressor shaft torque

N_c = measured compressor speed[†]

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

$$\eta_o = \dot{W}_c / \dot{W}_e$$

where the subscript c designates compressor and the subscript e designates expander. The fluid coupling efficiency is usually less than 10%.

Of considerably greater interest are the efficiencies of the compressor and expander separately. To obtain these values the shaft power of each is related to the power available if the measured flow rate were expanded isentropically from the measured high pressure to the measured low pressure of the cycle. Because measurement of high cycle temperature is uncertain, the expander exhaust is used as the reference state. The isentropic power is then given by:[‡]

$$\dot{W}_i = P_h \frac{w}{\rho_L} \left(\frac{P_L}{P_h} \right)^{1/\gamma} \left(\frac{\gamma}{\gamma - 1} \right) \left[1 - \left(\frac{P_L}{P_h} \right)^{\frac{\gamma - 1}{\gamma}} \right]$$

*Only data for Phase C testing (2700°F furnace temperature) are reported here; insufficient data were taken for Phases A and B to present a complete set of efficiency curves.

[†]All rotation speeds in this appendix refer to male rotor of unit being discussed.

[‡]Derived from Ref. 6.

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)

w = measured mass flow rate

γ = ratio of specific heats for argon = 1.67.

Since an isentropic process is reversible, the same amount of power, $\dot{W}_i$, would be required to compress the flow, w , from the pressure, P_L , to the pressure, P_h . The separate efficiencies of the compressor and expander may now be written

$$\eta_c = \dot{W}_i / \dot{W}_c$$

and

$$\eta_e = \dot{W}_e / \dot{W}_i.$$

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

$$\eta_o = \eta_c \cdot \eta_e = \dot{W}_e / \dot{W}_c$$

All efficiencies referred to are as defined above.

Figure C-1 shows compressor efficiency as a function of compressor speed with expander speed as a parameter. This curve demonstrates the relative independence of compressor efficiency from expander speed. The two points at 3000 rpm are not considered significant, since at low speeds and torques large errors in reading data are frequent. The curve of Fig. C-1 was fitted by eye.

The magnitude of the peak efficiency is about 25% less than efficiencies reported in private communication with Mr. H. Nilsson of the Svenska Rotor Maskiner firm for commercial low temperature machines of under 5 hp. This agreement is remarkable in view of the crude bearings in the high temperature units and the known excessive clearances.

Almost the same shape is noted in Fig. C-2 where the same data are plotted as compressor efficiency vs the ratio of high cycle pressure to low cycle pressure. The available data for low temperature machines do not drop off at these low pressure ratios but continue reasonably flat to ratios of 3 or 4. It is suspected that the earlier dropoff in the high temperature engines is due to pressure forces causing misalignment and high mechanical losses in the crude bearings 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.

Fig. C-1. Compressor efficiency vs speed.

Fig. C-2. Compressor efficiency vs pressure ratio.

In Fig. C-3 the expander efficiency is presented as a function of expander speed for several constant compressor speeds. These correspond to the data in Figs. C-1 and C-2. Note that the expander efficiency peaks at a much lower speed than the compressor efficiency. This is a measure of the slip or leakage in the system operated as a fluid coupling. The tendency is for higher compressor speeds to push the expander efficiency up and to a higher speed. The data for the 13,500 rpm curve seem to deny this, but this particular set of data is suspected due to torque cell malfunction. A correction factor has been applied to correct for observed zero shift in the data and may have overcorrected. The raw data call for a peak efficiency of 46%. When these data are compared directly to Fig. C-1, the expander is seen to be almost twice as efficient as the compressor up to 3000 rpm.

Fig. C-3. Expander efficiency vs speed.

In Fig. C-4 the data of Fig. C-3 are plotted against pressure ratio with compressor speed as a parameter. Expander efficiencies exhibit a sharp maximum when presented in this manner. For any given supply of gas (compressor speed), there is a very definite optimum pressure ratio for the expander. This result is distinctly different from that in the case of the compressor shown in Fig. C-2, where the maximum was broad and fell off slowly. Again, this result may be due to the crude bearings used in the high temperature engine.

Fig. C-4. Expander efficiency vs pressure ratio.

It may be noted that Fig. C-4 is really only Fig. C-3 with the abscissa reversed and distorted. The cause may be seen in Fig. C-5, where the pressure ratio is given as a function of expander and compressor speeds. At a constant compressor speed the pressure ratio is almost a linear decreasing function of the expander speed. The pressure ratio attainable at any speed depends on the leakage past both the expander and the compressor.

The effect of time on the leakage through the expander may be noted in Fig. C-6, where venturi pressure differential is plotted against high cycle pressure for the case of a locked expander. For this case the venturi pressure differential is directly proportional to the expander leakage flow. The curve was fitted by eye, but it may be seen that most of the data points taken early in the run lie above the line, while later points are scattered on both sides of the line. The last data taken before shutdown covered a locked rotor data run, and the results are plotted here. The fact that almost all of these points lie below the earlier data is significant. This means that, for a given pressure drop, less flow was leaking through the expander at the end of the test than initially. This increase in flow resistance is consistent with the observed tilting of the engine housing and resulting misalignment, since the direction of tilt was such that the clearances in the high pressure end of the expander were reduced. There are several causes for the scatter of data in Fig. C-6. It has been found that the leakage flow is dependent on the angular position of the locked rotor as well as the axial position of the rotor relative to the case.

Fig. C-5. Pressure ratio dependence on component speeds.

Table C-I presents a partial listing of the data from the second high temperature assembly test. The data are arranged in order of increasing compressor speeds. 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 rotor speed. The other data columns are as defined previously or are self-evident. These data have been checked and corrected for obvious errors.

The extensive data produced during the very successful test of the second high temperature assembly are still being analyzed and sifted for significant conclusions. The brief description presented here represents what is known or suspected to date. Only data from Phase C, 2700°F furnace temperature, are reported here, as insufficient data were taken for Phases A and B to present a complete set of efficiency curves. Efficiencies are somewhat greater at the high temperature.

Fig. C-6. Leakage flow related to driving head.

Table C-I. Partial data from Phase C testing (2700°F furnace temperature).

Data point no.	Input* speed (rpm)	Output* 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	†	†
208	3000	155	.249	0.001	1.058	.0077	26.8	1.4	†	†
209	3000	0	.260	0	1.065	.0075	28.4	0	†	†
205	4500	840	.621	.010	1.082	.0100	19.7	8.1	84.3	33.2
206	4500	600	.621	.011	1.085	.0098	20.3	8.4	82.9	24.1
207	4500	0	.568	0	1.099	.0086	22.8	0	73.0	0
201	6000	1505	.900	.018	1.112	.0129	24.0	8.2	81.8	46.0
202	6000	975	.971	.035	1.113	.0118	24.3	14.7	74.7	32.6
203	6000	610	1.042	.032	1.140	.0115	23.3	13.4	73.0	20.9
204	6000	0	1.089	0	1.157	.0105	22.9	0	66.4	0
34	9000	4080	1.667	.160	1.170	.0217	31.7	30.2	91.0	74.9
35	9000	3350	1.846	.178	1.184	.0210	30.1	32.1	87.9	63.5
36	9000	3050	2.024	.217	1.211	.0205	30.9	34.7	85.8	59.3
37	9000	2600	2.166	.231	1.221	.0197	29.1	36.6	82.4	52.6
38	9000	2060	2.400	.232	1.262	.0192	30.5	31.7	80.4	42.7
39	9000	1620	2.720	.230	1.289	.0187	29.0	29.2	78.4	34.4
40	9000	1100	2.826	.182	1.314	.0181	29.3	22.0	76.1	24.1
41	9000	510	3.068	.116	1.351	.0172	28.5	13.3	72.2	13.2
42	9000	0	3.302	0	1.380	.0166	27.6	0	69.8	0
108	10500	4680	2.734	.167	1.224	.0250	27.7	25.5	89.3	74.8
109	10500	4400	2.859	.261	1.228	.0245	27.7	32.8	87.7	71.7
110	10500	3700	3.107	.306	1.258	.0237	28.1	36.0	84.9	62.2
118	10500	3200	3.397	.303	1.272	.0229	26.2	34.1	81.9	55.8
111	10575	3100	3.505	.330	1.296	.0231	27.7	34.0	81.9	53.7
113	10500	2520	3.728	.328	1.323	.0223	27.6	31.6	79.9	45.1
112	10537	2200	3.991	.338	1.344	.0220	27.1	31.2	78.7	39.8
115	10500	1700	4.060	.282	1.357	.0211	26.6	26.0	75.7	32.1
116	10500	1030	4.474	.196	1.385	.0202	24.9	17.4	72.7	20.2
119	12150	5600	4.123	.331	1.245	.0295	24.0	20.1	90.6	76.3
120	12000	4900	3.977	.348	1.279	.0280	27.2	26.8	87.1	70.4
121	12000	4330	4.356	.359	1.300	.0270	25.8	31.9	84.1	64.3
122	12000	3720	4.924	.462	1.343	.0266	25.7	36.5	82.8	56.2
123	12150	3300	5.273	.430	1.368	.0260	25.2	32.2	80.1	50.9
124	12150	2820	5.753	.484	1.396	.0255	24.5	34.3	78.7	44.2
125	12000	2400	5.776	.440	1.422	.0247	25.2	30.2	77.2	38.9
126	12000	2170	5.776	.411	1.422	.0233	24.0	29.7	72.9	37.2
213	13500	6200	5.220	.222	1.272	.0325	22.7	18.6	89.2	77.2
214	13500	5400	5.326	.336	1.306	.0313	24.4	25.8	86.1	69.7
216	13500	4730	5.966	.392	1.276	.0295	23.1	28.5	81.5	64.5
217	13500	4100	6.392	.474	1.426	.0284	22.9	32.2	78.3	58.2
218	13500	3100	7.137	0.440	1.463	0.0276	22.7	27.1	76.5	45.0

* All rotation speeds in this appendix refer to male rotor of unit being discussed.

† Indeterminate because of very low flow.

APPENDIX D

Photoelastic Stress Analysis of the Male Rotor for a Helical Screw Engine

Insofar as centrifugal stresses are concerned, it is clear that the more highly stressed of the two rotors is the male rotor. This higher stress results from the fact that there are sharper re-entrant corners on the male rotor to act as stress raisers and also that its speed is 1.5 times that of the female rotor. Neglecting differences in geometry, centrifugally induced stress is proportional to speed squared. Figure D-1 defines the model which was tested.

Fig. D-1. Photoelastic model.

The photoelastic model was machined from ERL 2774 epoxy material. The following are the pertinent physical properties of the material:

Material fringe value (f): 2.67 lb/in./fringe

Poisson's ratio (μ): 0.5

Young's modulus (E): 5000

Density (ρ): 0.039 lb/in³.

The photoelastic frozen stress technique was used in the investigation. This procedure involves bringing the stressed specimen to thermal equilibrium at 165°C and then cooling to room temperature in such manner as to avoid inducing thermal differential stresses. Upon completion of this treatment, fringes representative of the stresses within the model remain locked within the model material.

Fig. D-2. Stress freezing equipment setup.

The application of inertial forces at temperature required that the specimen be rotated at a relatively high speed within an environmental chamber. The stress freezing equipment arrangement with the model partially enclosed in an aluminum cage is shown in Fig. D-2. Both the model and the cage rotated as a unit. Figure D-3 is a close-up of the unit in an inverted position. Figure D-4 shows the photoelastic model.

Fig. D-3. Photoelastic model and aluminum cage.

Fig. D-4. Photoelastic model shown after stress freezing cycle.

The model was rotated at 2200 rpm. The driving motor (upper portion of Fig. D-2) was capable of a speed of 2600 rpm. However, the unit became dynamically unstable, at about 2350 rpm, limiting the testing to the lower figure. The photoelastic model was free to expand in the axial direction during the stress freezing operation. This arrangement satisfied the requirement that the applied axial force be zero.

The first slice taken from the model contained the central longitudinal axis of the rotor; i. e., the surface plane of the slice was parallel to the longitudinal axis. The orientation of this slice is illustrated in Fig. D-5. Fringe patterns observed in this

Fig. D-5. Plane of initial slice taken from model.

Fig. D-6. Principal stress direction in root-to-lobe fillet.

slice failed to reveal any noteworthy stress concentrations. This result was true for both normal and oblique light incidence observations. Normal incidence fringe patterns are indicative of the average stresses in the plane of the slice. Oblique incidence fringe patterns are an indication of the average stresses on planes that are inclined to the plane of the slice.

The highest stresses were found in the fillet where the lobes begin in the root area. These stresses were detected in a slice taken tangential to the surface in the same area. Tension values obtained from three positions of varying locations along the rotor were 11.6, 11.8, and 11.9 psi. The direction of this principal stress is illustrated in Fig. D-6. Tension values for the average stress at the center of the root area appears to be about 7 psi and at the center of the rotor about 6.5 psi. Stresses across a lobe surface are dramatically lower, and no actual value was determined.

Model-to-prototype conversion of the results is accomplished in the following manner. In the analysis that follows, subscripts m and p refer to model and prototype, respectively.

Force can be expressed in terms of stress over a given area:

$$F_m = \sigma_m (\text{area})_m = \sigma_m (L_m^2).$$

Also, force can be equated to the acceleration on a given volume:

$$F_m = M_m (a_m) = \rho_m L_m^3 (L_m T_m^{-2}) = \rho_m L_m^4 T_m^{-2}.$$

Equating the two foregoing quantities and realizing the requirements of structural similarity between the model and prototype give:

$$\sigma_p L_p^2 = \rho_p L_p^4 T_p^{-2}$$

and

$$\sigma_m L_m^2 = \rho_m L_m^4 T_m^{-2}.$$

The relationship between the model and prototype is determined by:

$$\sigma_p = \sigma_m \frac{\rho_p L_p^2 f_p^2}{\rho_m L_m^2 f_m^2}.$$

If model and prototype have the same dimensions, then:

$$\sigma_p = \sigma_m \frac{\rho_p f_p^2}{\rho_m f_m^2}$$

where

$$\sigma_m \approx 12 \text{ psi}$$

when

$$f_m \approx (110/3) \text{ cps}$$

and

$$\rho_m = 0.039 \text{ lb/in}^3,$$

so

$$\sigma_p = 0.229 \rho_p f_p^2$$

when

$$\rho_p \text{ is expressed in lb/in}^3,$$

and

$$f_p \text{ is expressed in cps.}$$

This solution assumes that no great difference exists between the Poisson's ratios of the model and the prototype when the model is at operating temperature.

SUPPLEMENT TO APPENDIX D

Approximate Stress Analysis for a Helical Screw Male Rotor and Estimate of Failure Probability Due to Centrifugal Stress

The following stress analysis provides a supplement to the photoelastic data to show the effect of changing certain variables and to provide additional background to use in the statistical prediction of rotor behavior in the helical turbine.

In order to analyze stresses in the male rotor (see Fig. D-1), two kinds of simplifying approximations were made with the intention of bracketing the actual case. These are shown in Fig. D-7. The model is a hollow drum with an outside diameter equal to that of the root surface in the male rotor. Although stresses are lower for a

Fig. D-7. Loading conditions for simplified stress analysis. Case A, uniform traction (tension); Case B, four concentrated load lines.

solid rotor than for a hollow one, it is common to base design on a hollow cylinder for conservatism. The boundary conditions on the model drum are Case A, a uniform traction equal to the force exerted by the four lobes, and Case B, concentrated forces replacing the lobes.

The tangential stresses for Case A are found by superposition of stresses due to rotation of a long cylinder and those due to uniform external traction on a thick-walled cylinder.

The tangential stresses for Case B are found by curved beam analysis, an approach which leads to infinite stresses at the center if the hole vanishes. It is believed that the situation at the center of a solid rotor is reasonable approximated by Case A in accordance with St. Venant's principle.

The following gives the formulas to which the models lead:

Case A: Uniform Traction

At I. D.

$$\sigma_{I_A} \simeq 0.026 \left[3 + \nu + (1 - \nu) \xi_i^2 + \frac{4 Ak}{\pi R_o^3 (1 - \xi_i^2)} \right] \rho R_o^2 f^2.$$

At O. D.

$$\sigma_{O_A} \simeq 0.026 \left[1 - \nu + (3 + \nu) \xi_i^2 + \frac{2 Ak}{\pi R_o^3} \left(\frac{1 + \xi_i^2}{1 - \xi_i^2} \right) \right] \rho R_o^2 f^2.$$

Case B: Concentrated Forces

At I. D.

$$\sigma_{I_B} \simeq 0.026 \left\{ 3 + \nu + (1 - \nu) \xi_i^2 + \frac{2 Ak}{R_o^3 (1 - \xi_i^2)} \left[1.414 (1 + 3 K_i) - 3.8 K_i \right] \right\} \rho R_o^2 f^2.$$

At O. D.

$$\sigma_{O_B} \simeq 0.026 \left\{ 1 - \nu + (3 + \nu) \xi_i^2 + \frac{2 Ak}{R_o^3 (1 - \xi_i^2)} \left[1.414 (1 - 3 K_o) + 3.8 K_o \right] \right\} \rho R_o^2 f^2.$$

K_i and K_o are plotted in Fig. D-8.

At Center of Solid Rotor (for Case A)

$$\sigma \simeq 0.013 \left[3 + \nu + \frac{4 AK}{\pi R_o^3} \right] \rho R_o^2 f^2.$$

The symbols appearing above have the following definitions and values:

A = total lobe area, in ² = 3.86	$\xi_i = R_i/R_o$
k = radius of gyration of lobes, in. 1.6	f = frequency, cps
R_i = inside radius, in.	ρ = density, lb/in ³
R_o = outside radius, in. 1.07	σ = stress, psi

Fig. D-8. Curved beam correction factors for stresses in inside and outside diameters.

Fig. D-9. Estimated stress bounds on outside diameter and inside diameter ($\nu = 0.25$).

Since the equations are all of the form $\sigma = C\rho R_o^2 f^2$, the coefficient C is plotted for each case in Fig. D-9, using the lobe geometry of the first turbine and a Poisson ratio of 0.25. The plots are appropriate for geometrically similar configurations; i.e., the size effect then goes as R_o^2 .

These plots can be used to estimate stresses in the first engine. Assuming 15,000 rpm and putting in the other appropriate values, the results are:

1. At inner diameter of small hole:
Uniform traction $\sigma_{ui} = (0.25) \times (0.065)(1.14)(250)^2 = 1160$ psi
2. At interior hole:
Concentrated force (min value of C) $\sigma_{ci} = 0.62/0.25 \sigma_{ui} = 2880$ psi
3. At center of solid rotor:
 $\sigma = 1/2 \sigma_{ui} = 580$ psi
4. At outside surface:
Uniform traction $\sigma_{uo} = 0.11/0.25 \sigma_{ui} = 510$ psi
5. At outside surface:
Concentrated force $\sigma_{co} = 0.35/0.25 \sigma_{ui} = 1620$ psi

In all but one of these cases the stresses are not very large, and, since operating speeds were slightly less than 15,000 rpm, the working stresses in the test engine were quite low.

Neither fatigue nor vibration appear to be of prime importance (at least for the test conditions).

The stresses in the male rotor were also investigated by photoelasticity. An identical model was fabricated from a suitable photoelastic plastic and the inertial stresses frozen in by rotating at 2200 rpm in a furnace. The properties given for this material (ERL-2774) are:

1. Poisson's ratio: 0.5
2. Density: 0.039 lb/in³

The stresses at the stations indicated in Fig. D-10 are:

1. At center: 6.5 psi (calculated value = 7.5 psi)
2. In fillet: 11.8 psi
3. At root surface: 7 psi (calculated value = 6.5 psi min. to 20 psi max.)

Fig. D-10. Male rotor showing points at which stress was measured photoelastically.

From the photoelastic results, a coefficient in the equation $\sigma = C \rho R_o^2 f^2$ can be calculated which can serve for the center as well as exterior surface with a 1.7 stress concentration to account for the fillet.

The stress formula from the photoelastic results, then, is:

$$\sigma = 0.12 \rho R_o^2 f^2$$

which describes, approximately, the stresses in the center and on the root surface away from the fillet; evidently, the uniform traction approximation is the preferred model. The stresses given by this formula at 15,000 rpm are:

$$\begin{aligned} \sigma &= 560 \text{ psi (center and root surface)} \\ \sigma &= 950 \text{ psi (fillet)} \end{aligned}$$

Because graphite, the material of construction, behaves as a typical brittle material (i. e., with relatively large uncertainty in strength), the estimation of design reliability must take into account the statistical variation of strength. For the purpose of these estimates, the statistical behavior of H-205 graphite was estimated from the behavior of similar grades for which sufficient data were available.^{7, 8} The available data for grade CFZ graphite (similar to H-205) were examined with the Weibull distribution. The form of the distribution used was the one initially given by Weibull as

$$S = e^{-\int_v \left(\frac{\sigma}{\sigma_o} \right)^m dv} \quad (D-1)$$

where

S = survival probability

σ = applied tensile stress distribution which is integrated over volume in which it acts

σ_0, m = constants (m is referred to as the Weibull modulus).

This form of the Weibull distribution can be modified into a more convenient expression which avoids an explicit determination of σ_0 :

$$S = e^{-\beta^m \Gamma^m} \quad (D-2)$$

where

β = strength of any given specimen divided by mean value of sample

Γ = Gamma function, $\Gamma(1 + \frac{1}{m})$.

The determination of m immediately defines the statistical strength distribution. Figure D-11 shows some typical data on CFZ graphite. Similar data were available for samples in bending as well as in tension at a number of temperatures; and although

the mean strength varied with temperature, the Weibull modulus was consistently between 14 and 20 (such a variation is not uncommon even in controlled, homogeneous samples).

Fig. D-11. CFZ graphite tension (W.G.).

On the basis of these results and results on another molded graphite, grade SN-4, a consistently conservative basis for design is given by the Weibull distribution with modulus $m = 10$. It is assumed that such a distribution (also shown in Fig. D-11) applies to the H-205 grade graphite. In order to produce absolute values of strength from the statistical form above, an estimate of the mean value is needed. The value chosen for the H-205 grade (at 1500°C) is 4800 psi with the grain, and 4500 psi against the grain in flexure.

In order to predict the behavior of the rotors, based on flexure tests, the integral appearing in Eq. (D-1) should be evaluated. As an alternative, a conservative assumption can be made of the relative size between the flexure specimens and the rotors, assuming thereby that the rotor acts like a giant flexure specimen (which, because of

the size effect, is weaker). The survival probability, S_r , of these assumed rotors is related to the survival probability of a flexure specimen, S_f , by the relative volume, V_r , as follows from the consequences of weakest link theory:

$$S_r = S_f^{V_r} = e^{-V_r \beta^m \Gamma^m} \quad (D-3)$$

The size ratio, V_r , is also an equivalent number of reference specimens. Allowing $V_r = 200$, the two male rotors have a strength equal to the strength of the weakest flexure specimen in a sample of 200 such specimens. Given this number and the modulus m , it is possible to predict the most likely value of strength of the rotor. The relation is plotted in Fig. D-12. Entering at $n = 200$ and going to the line $m = 10$, the expected rotor strength is approximately 0.6 of the strength of the reference specimens. For H-205, the reference strength (with grain) at the test conditions is 4800 psi, and, therefore, the mean strength of the rotors is 2880 psi.

Fig. D-12. Graph for estimating strength of multiple specimens or large specimens.

To estimate the probability of survival of the rotors at any applied stress, Eq. (D-4) is used:

$$S_r = e \left(\sigma / 2880 \right)^{10} \left[\Gamma(1.1) \right]^{10} = e^{-0.605} \left(\sigma / 2880 \right)^{10}, \quad (D-4)$$

or, simply, the graph for $m = 10$ in Fig. D-11.

From the results of the photoelastic study it is estimated that approximately 1000 psi is the maximum stress at 15,000 rpm. Putting this stress in Eq. (D-4) gives

$$S_r = e^{-0.605 (0.35)^{10}} \geq 0.999 \quad \text{probability of survival}$$

However, no account is taken of the effect of time, vibration, shock, or creep.

The question remains as to what level of survival probability should be used. If 95% is arbitrarily chosen, then, on this basis, stresses in excess of 2000 psi should be tolerable.

APPENDIX E

Preliminary Experiments with Sealing Ribs and Case Coatings

A possible refinement in the graphite helical-screw engine is to reduce leakage flow by maintaining a minimum clearance between rotor lobes and the case. One

approach to this end is the use of small ribs and ridges along the tips of the lobes (see Fig. E-1), although, to reduce friction loss, there should be no contact between rib and case.

Fig. E-1. Male rotor with sealing ribs along tips of lobes.

The proposition is that the interior surface of the rotor-case be coated with a hard abrasive surface, probably a metal carbide, and when interference occurs, the ribs will wear until they just clear. Some initial experiments have been conducted toward establishing the coating formulation and technique and to test the wear-in characteristics of a simulated system.

Table E-I summarizes the initial efforts to form coatings with various coating materials and techniques. The materials included silicon powder, platinum powder, titanium hydride powder, titanium, and various other metals plasma-sprayed on graphite samples.

The coated specimens were examined visually to note uncoated areas and scratched with a steel scribe to test hardness. A slightly coarse texture was desired to aid the wear-in capability and to reduce the filling or clogging tendency. Coatings which seemed promising were tested for abrasion behavior at room temperature and under simulated operational conditions.

Table E-I. Summary of coating materials and techniques.

Coating material (proportion by weight)	Vehicle (proportion by volume)	Coating cycle (all heating in graphite vacuum furnace)	Results - remarks
(1) W, Zr, Ta, Si, Ti	Plasma- sprayed	Commerical process	Extremely rough surface; some poor adhesion, but the surface preparation requirement by itself is sufficient to reject these samples.
(2) Pt-Si 1:1 weight	Acetone and resin	2800°F 30 min.	Hard, fairly smooth coat; some small uncoated spots. The acetone vehicle was not satisfactory be- cause of its volatility. Used in first room temperature wear tests.
(3) Sodium silicate	Itself	2800°F	Very little coating on surface.
(4) Silicon powder	Sodium silicate & H ₂ O (2:1 by vol)	2800°F 30 min.	Extremely hard, smooth and tenacious coating, but not uniform. Uncoated areas in several places.
(5) Pt powder	Sod. Sil. & H ₂ O (2:1 by vol)	2800°F 30 min.	Hard, shiny coat, but very non- uniform. No flow to speak of.
(6) Pt & Si	Sod. Sil. & H ₂ O (1:1)	3100°F 10 min.	Shiny & smooth, not so hard as (4): areas of nonuniform coverage.
(7) Pt & Si	Sod. Sil. & H ₂ O (3:1)	3100°F 10 min.	Uniform, smooth coat, good cov- erage, not very hard.
(8) Pt & Si	Sod. Sil. & H ₂ O (3:1)	3100°F 10 min.	Hard, <u>rough coat</u> (texture of a medium emery); looks usable for abrasion.
(9) Pt & Si	Sod. Sil. & H ₂ O (3:1)	3100°F 10 min.	Similar to (8) although some smooth spots; used in second high temper- ature test.

The room temperature tests consisted of turning a cone, on which representative ribs were machined (Fig. E-2), at fairly high speed in a conical cup with a coated surface. These tests were run in a conventional metal lathe, the male, ribbed cone being mounted on a tailstock and run in. As interference occurred, abrasion quickly relieved it. This procedure could be repeated until the whole rib was worn away. In one case where the surface was not completely coated, coating (2), residual dragging occurred at the uncoated areas.

The simulation of the high temperature wear-in process was accomplished by using a graphite shaft having ribs machined on its surface as shown in Fig. E-3. This shaft rotated within a split graphite ring, (Fig. E-4) the interior of which was covered with the candidate coating. With no load on the split ring, the condition is here defined as free-running. As the load gradually increases, this split ring deflects until interference occurs and the ribs wear against the coated inner surface of the split ring. If

wear is complete, the drag should return to this free-running value. All parts are of graphite and are placed in the cavity of a furnace capable of 2850°F.

Fig. E-2. Male cone with ribs for machining.

Fig. E-3. Graphite shaft for Rib Wear Test.

Fig. E-4. Rib wear simulation apparatus.

There have been two tests in this high temperature environment using the coating and vehicle as in (2) and (8) of Table E-I. During the heating of the furnace for Run No. 1, the argon flush was lost and not restarted until 2200°F. A great deal of oxidation occurred in the exposed graphite surfaces; however, wear data were accumulated at 2500°F. Run No. 2, with Coating (8), was at 2700°F.

The results of rib-wear for Run No. 1 are shown in Fig. E-5 in which the oxidized surfaces are evident. Several steps in the rib section are seen, corresponding to shifting of the position of the split ring (this was not intentional movement and was prevented in Run No. 2 by the stake shown in Fig. E-4). Rather complete wear occurred, and the drag behavior during the wear is shown in Fig. E-6. This figure indicates that though wear occurred, time on the order of minutes was required, and the frictional drag did not return to the critical, free-running condition.

Fig. E-5. Rib specimen after Run No. 1 (platinum-silicon coating).

The results of Run No. 2 are shown in Fig. E-7, and the torque behavior in Fig. E-8. The observed torques were quite low and similar to the observations in Run No. 1. Examination of the shaft, Fig. E-7, shows that a maximum wear of about 0.020 in. on the radius had occurred. All of the interior of the split ring retained the coating, although the portion where interference occurred became very smooth. Apparently, it was this smoothing which resulted in the parasitic drag and fairly slow wear rate.

The behavior when no coating was present is shown in Fig. E-9, when the wear was negligibly small, amounting to less than 0.001 in.

Fig. E-6. Torque and load as functions of time for High Temperature Run No. 1 (chart speed, 3-3/4 in./min, i.e., 16 sec/in.).

Fig. E-7. Rib specimen after Run No. 2 (platinum-silicon coating).

Fig. E-8. Torque and load as functions of time for High Temperature Run No. 2 (chart speed, 3-3/4 in./min).

Fig. E-9. Post-run condition of shaft with no coating.

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