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**EXPERIMENTAL PERFORMANCE CHARACTERISTICS
OF THREE IDENTICAL BRAYTON ROTATING UNITS**

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TECHNICAL PAPER proposed for presentation at Fifth Intersociety
Energy Conversion Engineering Conference sponsored by the
American Institute of Aeronautics and Astronautics
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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

EXPERIMENTAL PERFORMANCE CHARACTERISTICS OF THREE IDENTICAL BRAYTON ROTATING UNITS

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Abstract

Extensive hot performance testing has now been completed on three identical Brayton Rotating (turboalternator-compressor) Units on gas bearings. Each unit (BRU) was tested in a different type of test installation. The first in a BRU performance test rig, the second as part of a Brayton gas loop test (BRU with engine heat exchangers and ducting), and the third as part of a full Brayton engine test powered by an electric heat source.

At design conditions of 1600° F turbine inlet temperature and 80° F compressor inlet temperature, BRU output power exceeds the design objectives in all cases. Gas bearing performance in all three units was satisfactory and differences from unit to unit were minor. The thermal design was generally very good, particularly in the area of the journal bearings. However, alternator hot spots ran 15° to 20° F higher than predicted. Added cooling which would be required to run at powers higher than design (10.5 kW) is readily obtainable by liquid cooling of the alternator end bells.

Overall BRU performance and operation has been very successful.

Introduction

The NASA-Lewis Research Center is presently developing a closed loop Brayton cycle engine (Refs. 1 and 2) for space applications. This engine was designed to deliver from 2 to 10 kW of electrical power continuously over a 5-year life with shutdown and re-start capability. At the present time, ground tests of all major components, subsystems, and the complete engine have been or are being conducted.

The "heart" of this engine is the Brayton Rotating Unit (BRU). It consists of a turbine, alternator, and compressor mounted on a single shaft, supported on gas lubricated bearings. Extensive testing of the BRU has been conducted and is continuing. This test activity started with individual component tests of the turbine, compressor, and alternator utilizing individual component research packages and dynamic tests of all the bearings in a BRU simulator (Refs. 3 to 5). It has included the hot testing of three identical BRU's in three different test installations.

The first unit was tested in a test rig for evaluation of the BRU only (Ref. 6). In this test the objective was to evaluate the operation and performance of the BRU as a component. These tests were intended to exercise the unit over its complete operating range, determine any operational limitations, and evaluate the behavior of the BRU during startup and shutdown. This unit has now been installed and is being tested in the Brayton engine, replacing the third unit discussed below.

The second unit was tested as part of the Brayton engine gas loop (Ref. 7). In this installation the BRU was installed with the flight-type Brayton Heat Exchanger Unit (BHXU), and with an electrically powered heat source. The objectives of this test activity were to evaluate the overall performance of the complete Brayton gas loop and evaluate BRU and BHXU performance as a part of the loop. This test program is still underway and includes an objective of completing a 10,000-hour endurance test.

The third unit was tested as part of a complete Brayton engine, which included a flight-type electrical control system, coolant system, and a gas management system. The primary objectives of this test (Ref. 8) were to demonstrate and evaluate the performance of the complete engine over its operating range, to explore component interactions, and to investigate engine startup and shutdown techniques.

Test data from the three units now allows a preliminary evaluation of the repeatability and consistency of performance of the BRU from unit to unit as well as a comparison with the initial design ob-

jectives. It is the purpose of this paper to present such an evaluation. Table I summarizes the total operating time on these units as of May 18, 1970. However, only data through February 11 is evaluated in this paper.

BRU Design

On June 30, 1966, the NASA-Lewis Research Center contracted with AiResearch Manufacturing Company of Phoenix, Arizona for the design and manufacture of the Brayton Rotating Unit. The BRU design requirements were as follows:

Alternator:		
Power output		2.25 to 10.5 kW
Frequency		1200 cycles ±1%
Voltage		three-phase, 120/208
Working fluid		He-Xe [Mw = 83.8 (krypton)]
Shaft speed		36,000 rpm
Turbine inlet temperature		1600° F
Compressor inlet temperature		80° F
Operating life		5 years
Turbine and compressor type		radial flow
Bearing system		gas bearings
Start-stop		hydrostatic
Normal operation at rated speed		hydrodynamic

The initial design studies included basic design of the major components, overall package design, and the development in conjunction with NASA-Lewis Research Center of a set of reference design conditions compatible with the overall Brayton engine requirements.

These reference design conditions for the BRU contract are summarized briefly as follows:

Alternator output, kW	2.25	6.0	10.5
Working fluid - Helium-Xenon			
Molecular weight	83.8	83.8	83.8
Shaft speed, rpm	36,000	36,000	36,000
System flow rate, lb/sec	0.42	0.80	1.32
Compressor pressure ratio	1.88	1.9	1.9
Compressor efficiency, %	79	80	80
Turbine pressure ratio	1.73	1.75	1.75
Turbine efficiency, %	85	87	87
Turbine inlet temperature, °F	1600	1600	1600
Compressor inlet temperature, °F	80	80	80
Alternator electrical efficiency, %	85	92	92

The BRU design, generated to meet these requirements, is shown schematically in Fig. 1. Figure 2 is a photograph of the BRU. The radial inflow turbine with a 5-inch diameter wheel and the radial flow compressor with a 4½-inch diameter wheel are mounted at the opposite ends of the shaft.

The alternator, located in the center of the unit, is a four-pole modified Lundell unit. The alternator rotor is an integral part of the shaft. It is a brazed composite of 4340 magnetic steel end sections, incorporating the pole pieces, and Inconel 718 nonmagnetic separating material between the pole pieces.

The shaft is supported by pivoted three-pad gas journal bearings located just outboard of the alternator, and a gimbal-mounted double-acting step-sector gas thrust bearing at the compressor end of the shaft.

The basic structural member of the unit is the alternator housing. It incorporates liquid cooling passages to carry away waste heat. The turbine and compressor scrolls mount to the main flanges at opposite ends of the alternator housing. The journal bearings are

supported off the alternator end bells and the thrust bearing is mounted from the compressor-end flange of the alternator housing. To provide external sealing of the alternator housing a sheet metal pressure containment shell is welded around the alternator housing. Feedthroughs for power, cooling lines, bearing jacking gas, and instrumentation are provided in this shell.

In order to provide a maximum bearing ambient pressure, particularly for the low power, low pressure conditions, the bearing cavity is connected to the compressor discharge through a filtered 1/2-inch bleed line. This also assures a positive outward flow of gas past the labyrinth seals to the back faces of the compressor and turbine, preventing leakage of hot unfiltered gas into the main BRU housing.

The most critical factor in obtaining a satisfactory overall package design of the BRU was thermal management and temperature control throughout the unit. Prime requirements of this thermal design were to limit the alternator hot spot temperature, minimize temperature levels and gradients in the bearings (to minimize thermal distortions which would destroy the load carrying capacity of the bearings), and to minimize and accommodate differential thermal growth between the journal bearing shaft and shoes.

The thermal design attempted to meet these requirements through the use of heat dams, thermal shunts, thermal isolation mounts, and convection heat transfer shoes between the shaft and appropriate stationary members.

Other significant design features include the use of fully conforming pivots for the journal bearing shoes, which act as the seal for the introduction of hydrostatic jacking gas through the pivot shown in Fig. 3; and the use of a flexure mounted gimbal to support the thrust bearing and accommodate misalignments and runouts between the thrust bearing rotor and stator. Startup and shutdown of the unit are accomplished while the bearings are hydrostatically supported by jacking gas externally supplied to the bearings at 100 to 150 psia. For normal design speed operation, the jacking gas is turned off and the bearings, both thrust and journals, operate completely hydrodynamically.

Description of Test Installations

Hot tests of the complete BRU have been conducted in three different installations. These installations are shown schematically in Figs. 4 to 6. Brayton flight-type components are shown in the figure as those connected by double-lined ducting, whereas component simulators are shown to be connected by double-lined ducting, whereas component simulators are shown to be connected by single lines.

In all installations the BRU was operated in the vertical position with the turbine end up.

BRU Test System

BRU tests were performed in an atmosphere environment in the test installation shown in a simplified schematic diagram in Fig. 4. In this installation, the BRU was the only flight-type engine hardware under test. Reference 9 describes the facility components which controlled the inlet conditions to the BRU. This system included a single controlled jacking gas supply line for both journal bearings and one for both sides of the thrust bearing. Provision was included for pressurization of the bearing cavity from the compressor discharge scroll through a compressor bleed valve. A gas injection valve was provided upstream of the heater for injection startup and a vent valve was included to control system inventory.

Brayton Gas Loop Test System

Brayton gas loop tests are being performed in an atmospheric environment in the test installation shown in the simplified schematic in Fig. 5. In this installation the BRU is installed with the flight-type Brayton Heat Exchanger Unit (BHXU). The BHXU consists of a recuperator, waste heat exchanger, and connecting ducting to the BRU and is constructed as a single unit. The BRU and BHXU, in conjunction with an electric (nonflight-type) heat source, forms the complete Brayton Gas Loop. Reference 7 describes the remaining

features of the facility. In this system, jacking gas is supplied to the two journal bearings from one common supply line and individual supply lines are provided for each side of the thrust bearing. Bearing cavity pressurization is supplied by a bleed line directly from the compressor discharge scroll. A gas injection line and a vent line are provided in the compressor discharge lines; the former is downstream of the primary flow valve, the latter upstream. Injection starts are made open loop with the primary flow valve closed and the vent valve open.

Brayton Engine Test System

Brayton engine tests are being performed in a vacuum environment in the test installation shown in the simplified schematic in Fig. 6. In this installation (Ref. 8) the BRU is installed as a part of a complete flight-type Brayton engine. The gas loop is the same as in the Brayton Gas Loop Tests except that the electric heat source is a different design. The gas supply system including jacking gas supply valves, injection valve and system vent and bleed valves are included as part of the flight-type Gas Management System. An Engine Control System is included which monitors and controls engine operation, including startup and shutdown functions, inventory control, and overspeed protection.

Thermal Insulation

A major difference in the above installations is that the Brayton Engine Test is being run in vacuum conditions while the BRU Test and Brayton Gas Loop Tests are conducted in normal atmosphere environment. Thermal insulation on the exterior surface of the BRU turbine reflects the different environmental conditions of the installations. The turbine insulation system of the Brayton engine consists of two Inconel shells, gold-plated for low emissivity, enclosing two layers of extra low conductivity fibrous insulation. The layers are separated by a stainless steel foil spacer. The shells are embossed for stiffness. In addition, the inner shell is ridged to minimize thermal contact between the scroll and the insulation. The insulation of the Gas Loop BRU turbine consisted of a fibrous blanket, somewhat higher in conductivity than in the engine installation, but 6 inches in thickness. The BRU Test turbine insulation was similar to that on the Gas Loop turbine.

Instrumentation

Internal instrumentation of the BRU was identical in each of the three test installations. Measurements of the shaft and bearing motions were made by use of capacitance-type proximity probes. Twenty-two probes were installed throughout each BRU. Internal instrumentation also included thermocouples throughout each unit. Their purpose was to measure the hot spot temperatures and to ascertain temperature distribution. In addition to internal instrumentation, there were measurements external to the BRU of gas temperatures and pressures in the turbine and compressor inlet and discharge lines. The locations and types of these temperature probes and pressure taps varied with the installation due to the different nature and test objectives of each installation.

Results and Discussion

At the time of this writing, testing of one unit in the BRU Test System was completed, while Brayton Gas Loop Tests and Brayton Engine Tests are still underway. Data reduced and evaluated for this paper covers test activities in each installation through February 11. A summary of these tests including number of operating cycles, is shown in Table II. As can be seen, data were obtained in the BRU Test at the design turbine inlet of 1600° F while operating on the helium-xenon working fluid. Brayton Gas Loop tests have been conducted over a fairly complete range of temperatures and pressures, including the 1600° F turbine inlet condition. However, this testing to date has been limited to the use of krypton gas as the working fluid. Brayton engine testing has been conducted on both krypton and helium-xenon and up to 1600° F turbine inlet temperature. As mentioned previously, however, the data available for this paper does not include the 1600° F condition, though the lower turbine temperatures are included.

In all three test installations initial checkout operations have been conducted on krypton due to the high cost of the helium-xenon

mixture. Since the molecular weight and critical thermodynamic properties of krypton and the helium-xenon mixture are the same, no effect should be seen on turbine and compressor performance. However, the mixture should perform better in the complete engine, due to the effect of its better heat transfer coefficient on heat exchanger performance.

Output Power and Aerodynamic Performance

In comparing and evaluating the performance of the BRU relative to the design objectives it is necessary to look primarily at the BRU Tests and the Brayton Gas Loop Tests since the design turbine inlet temperature data (1600° F) are not included from the Brayton engine tests. Figure 7 shows plots of gross alternator power versus compressor discharge pressure at the design condition of 1600° F turbine inlet and 80° F compressor inlet temperature.

In the BRU Tests both the helium-xenon and krypton points form essentially a single plot. This would be expected since the aerodynamic turbine and compressor performance should be the same with either working fluid. The data from the Brayton Gas Loop Test falls about 700 watts below or $2\frac{1}{2}$ psi to the right of the BRU Test data within the range of values covered. This is due, at least in part, to the higher pressure drops in the Brayton Gas Loop which results in a lower turbine inlet pressure and relatively lower turbine pressure ratio (see Fig. 8). In any case, the BRU performance as indicated by the gross alternator power obtained in the BRU and Brayton Gas Loop Tests substantially exceeds the reference design conditions established early in the program. The $10\frac{1}{2}$ kW power level, predicted by extrapolation of the Gas Loop Tests, would be obtained at a compressor discharge pressure of only $37\frac{1}{2}$ psia instead of 45 psia. Or one might expect to achieve 12 to 13 kW at the 45 psia condition.

Since pressure ratios were found, as predicted, to be essentially constant over the range of compressor discharge pressures tested, they have been tabulated rather than plotted in Fig. 8. The figure compares turbine and compressor performance in the design range. As indicated, the difference in turbine and compressor pressure ratios in the BRU Tests are appreciably smaller than in the Gas Loop Tests; and the results from the Gas Loop Tests are quite close to the reference conditions. In comparing efficiencies, it is found that both turbine and compressor efficiencies for the helium-xenon and krypton plot on common curves from the BRU Test data again indicating no aerodynamic effects from the two different working fluids. Compressor efficiencies shown in Fig. 8 were low in the BRU Tests compared to the reference design while the Brayton Gas Loop data was found to correspond very well with the reference design. These results correspond well with unpublished test data from the Compressor Research Package. It indicates that the compressor is operating at high equivalent mass flows in the BRU Tests moving the operating point to a lower efficiency condition. This indicates that the compressor is operating nearer to open throttle conditions than design. This is caused by the lower pressure loss between the turbine and compressor. The turbine efficiency data shows higher than the reference design performance in the BRU Tests and lower than reference design performance in the Gas Loop Tests. Since the efficiency data in Fig. 8 assumes an adiabatic expansion, heat losses from the turbine would result in indicating higher than actual efficiency, thus the plot of data from the BRU Test would seem reasonable especially since constant heat losses would tend to indicate higher efficiencies at the lower power levels. However, the turbine efficiency data plotted from Brayton Gas Loop Tests is unexplainably low. So far, attempts to reconcile the turbine efficiency data from these tests with cold gas turbine test data from the Turbine Research Package has been unsuccessful. The high operating temperature of the turbine is probably a significant contributing factor in this difficulty. Both temperature measurements and the control of heat losses is very difficult and substantial differences may exist from one installation to the next.

A comparison of the performance of the BRU's in the three different installations is shown in Figs. 9 and 10 for 1400° F turbine inlet and 80° F compressor inlet. Figure 9 shows Gross Alternator Power as a function of Compressor Discharge Pressure. In this case the Brayton engine tests were performed on both working fluids (Kr and He-Xe). In the Brayton engine tests the helium-xenon performance falls inexplicably below that of the krypton, while in all BRU testing no significant difference has been detected. At the same time it can be seen that Brayton Gas Loop Tests and Brayton Engine Tests

using krypton yield almost the same power outputs although again somewhat below the BRU Tests as would be expected with the difference in turbine and compressor pressure ratios as indicated in Fig. 10. Again in Fig. 10, compressor efficiencies are closely grouped for the Gas Loop Tests with krypton, and Engine tests with krypton and helium-xenon. The BRU Test results (1400° F data is limited) again show lower compressor performance as expected. Turbine efficiencies again show a wider spread than do the compressor efficiencies. However, the differences seem more reasonable at 1400° F turbine inlet than they did at 1600° F. The turbine efficiencies for the Gas Loop and Engine tests are fairly close, though this might be primarily coincidental. It is of interest to note that the highest indicated turbine efficiency is obtained in the Engine test on helium-xenon which also shows the lowest power output (Fig. 9). This might be explained, at least partially, by thermal losses between turbine inlet and turbine exit thermocouple locations (again remembering that these indicated efficiencies assume no thermal losses).

Bearing Performance

In evaluating BRU performance it is essential to review the performance of the Gas Bearings. In all three tests, indicated shaft orbits and pad motions were small and not considered to be a problem. Perhaps more important, though test time (Table I) is still much less than the 5-year life objective, no apparent changes have been observed in shaft orbits or pad motions as a function of time or number of start-stop cycles. Changes in orbits might indicate a shift of rotor balance, while changing pad motions could reflect a journal pivot wear problem.

Thrust Bearing. Critical to performance of the thrust bearings is the question of applied loads. The thrust bearing loads come from two sources when the unit is operated with the shaft vertical. The first, of course, is the weight of the shaft of 21.8 pounds. The second source is aerodynamic thrust loads from the turbine and compressor wheels. These were predicted during the design phase to be from 9 to 31 pounds net toward the turbine pad over the $2\frac{1}{2}$ to $10\frac{1}{2}$ kW power range.

The resultant predicted force on the thrust bearing is shown in Fig. 11 for comparison with actual loads. As can be seen the indicated actual loads from tests of all three units fall in the range of 12 to 21 pounds toward the compressor. These actual loads were obtained using a calibration curve for the thrust bearing obtained in testing the bearing simulator which gives load as a function of hydrodynamic pressure minus cavity pressure. It is obvious from Fig. 11 that the predicted aerodynamic thrust loads were not as large as anticipated. In fact, the aerodynamic loads appear to be in the range of zero to 10 pounds over the entire range of compressor discharge pressures. Also, within the accuracy of the data it would appear that all three units are quite similar and that the effects of different turbine inlet temperatures and use of either krypton or helium-xenon is negligible. Another method of evaluating the thrust loads yields values about 5 pounds larger toward the compressor, but the trend of the data (slightly decreasing loads with increasing compressor discharge pressures) remains the same. In either case, the aerodynamic thrust load and its change with compressor discharge pressure levels is much less than predicted.

This variation from the predicted values is not surprising, since the prediction involves the differences in calculated forces on each side of the turbine and compressor wheel. These individual forces are quite large. Thus the predicted loads are the result of small differences in large numbers.

Thrust bearing film thicknesses on the loaded (compressor) side were on the order of 0.0004 to 0.0005 inch for two of the units over the range of conditions tested. Reliable film thickness data was not obtained in the Brayton Engine Test. Film thickness of this mode is appreciably less than originally predicted value of 0.0006 inch minimum. While this has not affected operation of the BRU, it would indicate lower than predicted maximum load capability for the thrust bearings.

Journal Bearings. Journal bearing pad loads from tests of the three units are listed in Table III. These loads appear to be essentially constant with variations of compressor discharge pressure, tur-

turbine inlet temperature and use krypton or helium-xenon. Thus, only the range of pad loads for each unit are shown in Table III. These loads are obtained by taking 75 percent of the difference between the hydrodynamic bearing pressure and the cavity pressure. This computation method (Ref. 9) is based on experimental data using argon and the BRU bearing configuration tested in an earlier radial flow turbocompressor which was the forerunner of the BRU.

Local measurements made using proximity probes to measure flexure mount deflections and thus pad loads, indicated somewhat higher values. These values are thought to be less reliable, however, due to potential thermal distortion errors.

Whether the indicated differences are real or merely reflect potential instrumentation inaccuracies is difficult to say. However, based on jacking gas pressures required to lift the bearings prior to startup one would have anticipated higher pad loads in the BRU installed in the Brayton engine than in the other two units. Again, the variations are not significant and the results indicate that the journal bearing system design, including selection of the flexure mount spring rate, and the overall thermal design of the bearing mounts was very successful.

The thermal design of the bearing mounts was critical to the success of the BRU. Failure to adequately accommodate thermal growth of the shaft could have overloaded the bearings and caused failure. On the other hand, overcompensation could have unloaded the bearing pads and resulted in bearing instabilities and failure.

Thermal Performance

As stated earlier, the most critical element in a successful design of the BRU was good thermal management. As far as the bearings are concerned this objective has apparently been met as indicated in the above bearing discussion. Another critical area of temperature is the alternator stator hot spot temperature. The initial design objective was to limit the hot spot temperature to 180° C (356° F). While initial thermal analyses indicated this would be met, it was found upon rechecking the thermal design, after completion of detailed drawings, that the hot spot was predicted to be about 400° F at 10.5 kW and 0.85 power factor. This was considered adequate for the required 5-year life. The BRU's were fabricated without further modification.

Table IV presents a comparison of key package temperatures obtained in BRU Tests and are typical of the other two units (thermocouple numbers are included for reference to other BRU reports). The conditions for which measured temperatures are presented are for 10.5 and 11.7 kW, both at 0.95 power factor. At 10.5 kW, 0.95 power factor, the three alternator hot spot temperatures are all about 400° F. Decreasing the power factor from 0.95 to the design value at 0.85, all other factors remaining constant, raises the alternator hot spot temperatures about 15° F (according to results from the BRU Tests and substantiated by the Brayton Gas Loop Tests). Thus, at 10 kW (gross power) and 0.85 power factor, the alternator hot spot is indicated to be 415° F. Unpublished statistical data for alternators with similar insulation and potting compounds show that a 5-year life may be expected for this temperature.

The turbine end labyrinth seal is indicating several hundred degrees above predicted values. No apparent adverse effects on BRU performance can be found and it is not believed to be a problem. It may result from lower than predicted labyrinth seal leakage flow which would reduce the cooling effects on the seal.

Overspeed Capability

No specific tests were planned to determine the overspeed capability of the BRU. However, two units were inadvertently run to speeds in excess of 50,000 rpm (design value is 36,000 rpm). Both instances resulted from sudden removal of the alternator load external to the unit. In both cases, the bearings contacted at speeds above 50,000 rpm and stopped rotation within a few seconds. Damage to the units was limited to scored bearings and labyrinth seal rubs. The units involved were Unit 3 in the Engine tests (after 663 hr) and a fourth unit which was being checked out in the Gas Loop Test System. This fourth unit was replaced as a result of the incident before any significant testing was performed.

These incidents, while not planned, show substantial overspeed margin above the 120 percent design goal and also demonstrate that a runaway condition (loss of load) does not result in catastrophic damage to the unit.

Problem Areas and Potential Improvements

Testing in the three installations revealed several areas that indicated a need for correction. These problems, however, do not appear to require any major design change. In addition, several minor modifications are apparent which, if put into effect, can improve performance.

Alternator Hot Spot. A recent study effort performed by AiResearch under the BRU contract indicates that redesign of the stator to provide additional cooling and add copper to reduce losses could result in a BRU capable of about 25 kW gross alternator output at the desired 400° F hot spot temperature. This modification would require a completely new stator design with a larger housing diameter. However, it is estimated that the addition of liquid-cooled end bells and other minor modifications should be sufficient to bring the present alternator stator hot spot temperature to about 400° F (5-yr life) for gross power outputs on the order of 14 to 15 kW.

Pneumatic Hammer. In the BRU Tests, the journal bearings developed a "pneumatic hammer" type of instability when the pressure ratio - prior to startup - between the bearing jacking gas supply and the cavity exceeded 10:1 to 15:1 (Ref. 6). This pressure ratio existed when the compressor bleed valve (Fig. 4) was in the open position, exhausting the cavity to the evacuated lines in the system. With the compressor bleed valve closed, however, and the hydrostatic flow from the bearings vented only through the labyrinth seals, the cavity pressure quickly built up to about 17 psia thus holding the pressure ratio to less than 10:1. While the other two units have not exhibited this phenomenon, future units will incorporate a check valve in the compressor bleed line to the cavity. During startup the check valve will then serve the same function as the compressor bleed valve did in the BRU Tests.

Thrust Bearing Oscillations. During BRU testing at reduced turbine inlet temperature it was observed that as the pressure in the loop was increased the film thickness of the loaded side of the thrust bearing (compressor) increased slightly. At a compressor outlet pressure of 30 psia a subsynchronous motion with a frequency of about 1/6 rotative speed could be seen superimposed on the relative motion between the thrust runner and stator. It could also be seen in the relative motion between the thrust stator and the frame of the BRU. The subsynchronous motion increased as the compressor outlet pressure was increased to 45 psia. At this pressure and a turbine inlet temperature of 1200° F the peak-to-peak subsynchronous motion was about 0.00025 inch. With this oscillation, the minimum film thickness was of the order of 0.00027 inch. Brayton Gas Loop testing showed similar results at low turbine inlet temperatures and high pressures. These oscillations were evidently not a problem for these conditions. In addition, they occurred in a region well outside the normal operating range. It might be assumed that this oscillation was connected with the reduced thrust loads and increased film thicknesses encountered under these conditions. However, these changes were not large and bearing tests conducted by the contractor in the BRU simulator, where load conditions were investigated from a full 30-pound load through the neutral load condition, showed no such oscillations.

Another possible explanation is that the oscillations may be introduced by pressure pulsations in the compressor or turbine, or a pressure cross-linking between compressor and turbine. However, no evidence to support such an explanation has been noted to date.

Turbine and Compressor Performance. Minor modifications in the turbine and compressor diffusers should result in one to two point improvement in the efficiencies of these components. Research package testing has shown that the compressor performance can be improved by an adjustment of 3° in diffuser vane angle setting. While this has been known for some time it was decided not to incorporate the change into the BRU until complete engine tests verified the match point of the turbine and compressor. This has not been done and it is planned to incorporate this change in future units.

Turbine component testing (Ref. 3) indicated room for improvement in design of the turbine exhaust diffuser. Preliminary testing of the Turbine Research Package with a modified diffuser design, based on a linear variation in static pressure, instead of a linear variation in area, indicates improvements in overall turbine efficiency of about one point. If these results are verified this change would also be incorporated into future units.

Conclusions

Three Brayton Rotating Units have now been operated in three different test installations. While the differences in the test installations makes accurate comparisons difficult, it is believed that, in general, the performance evaluations presented in this paper are valid. The following conclusions have been drawn from this evaluation:

1. The overall BRU performance objectives have been exceeded based on measurements of gross alternator output power.
2. Compressor performance objectives have been met.
3. A good direct evaluation of turbine performance is not possible based on the data so far available. However, it certainly follows from conclusion 1 above that turbine performance as a part of the overall BRU has at least been adequate.
4. Bearing performance has been satisfactory. Aerodynamic thrust loads are lower than predicted. Journal pad loads are close to predicted values indicating excellent thermal growth compensation.
5. Alternator hot spot temperatures are higher than predicted and some design adjustment (such as liquid-cooled end bells) is required for power conditions exceeding $10\frac{1}{2}$ kW gross.
6. Except for the alternator, no thermal design problems are apparent.

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TABLE I

Unit number	Test installation	Operating, time, hr	Maximum inlet temperature, °F	Gas
1	BRU test Engine test ^a	1012 <u>1412</u> 2424	1600 1600	Kr, He-Xe He-Xe
2	Gas loop test	236	1600	Kr
3	Engine test	668	1450	Kr, He-Xe

^aEvaluation of data from this test was not completed for this paper.

TABLE II. - RANGE OF BRU OPERATING CONDITIONS

	BRU 1		BRU 2		BRU 3	
	BRU Tests		Brayton Gas Loop		Brayton Engine	
	Kr	He-Xe	Kr	He-Xe	Kr	He-Xe
Compressor discharge pressure, psia	^a 25-29	17-45	15-56	-----	25-44	25-45
Compressor inlet temperature, °F	^a 70-80	60-120	55-100	-----	42-95	70-90
Turbine inlet temperature, °F	^a 1200-1600	1200-1600	1140-1600	-----	1200-1450	1200-1500
Number of operational cycles	^b 8	^c 15	3	-----	5	5

^aPreliminary testing conducted with larger than design turbine and compressor clearances is not included.

^bIncludes preliminary testing with argon.

^cIncludes eight motor-start hot tests.

TABLE III. - JOURNAL BEARING PAD LOADS

Reference design, lb	15
BRU 1 (BRU Test), lb	13.8 - 15
BRU 2 (BRU Gas Loop), lb	13.8 - 14.2
BRU 3 (Brayton Engine), lb	12 - 13

TABLE IV. - KEY BRU TEMPERATURES, °F

Thermo-couple number	Location	Predicted	Measured	
		10.5 kW, 0.85 PF	10.5 kW, 0.95 PF	11.7 kW, 0.95 PF
1	Alternator end turn I. D. (turbine end)	375	402	435
2	Alternator end turn I. D. (compressor end)	375	392	429
5	Alternator slot at I. D.	400	398	436
3	Alternator end turn O. D. (turbine end)	300	263	283
10	Alternator field winding	160	144	150
31, 32	Thrust bearing gimbal	420	395	402
34	Labryinth seal (turbine end)	550	901	882
11	Main alternator housing (turbine end)	155	149	153
12	Main alternator housing (compressor end)	155	127	132
23	Journal bearing pad (turbine end)	385	387	397
25, 26	Journal bearing carrier (turbine end)	---	315	325
27	Journal bearing pad (compressor end)	355	339	349
29, 30	Journal bearing carrier (compressor end)	---	282	288

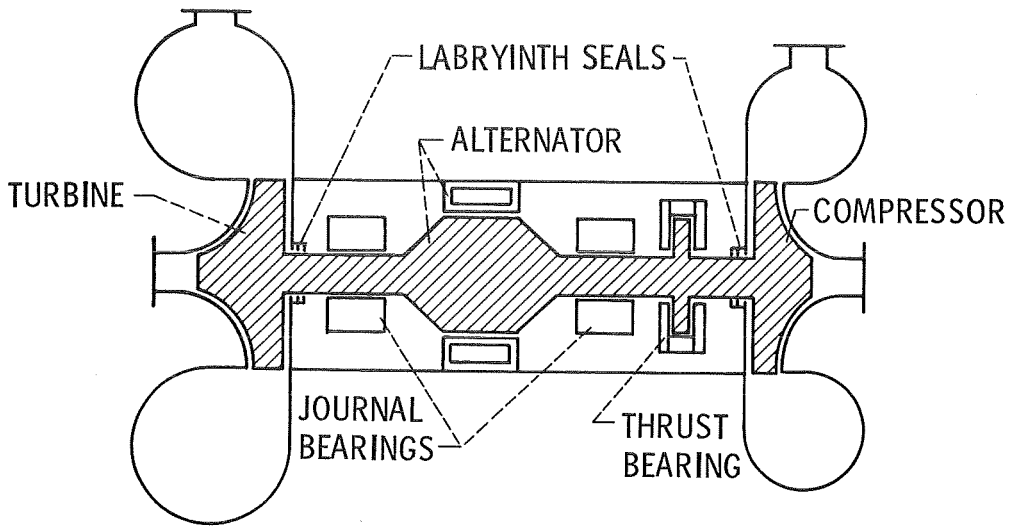
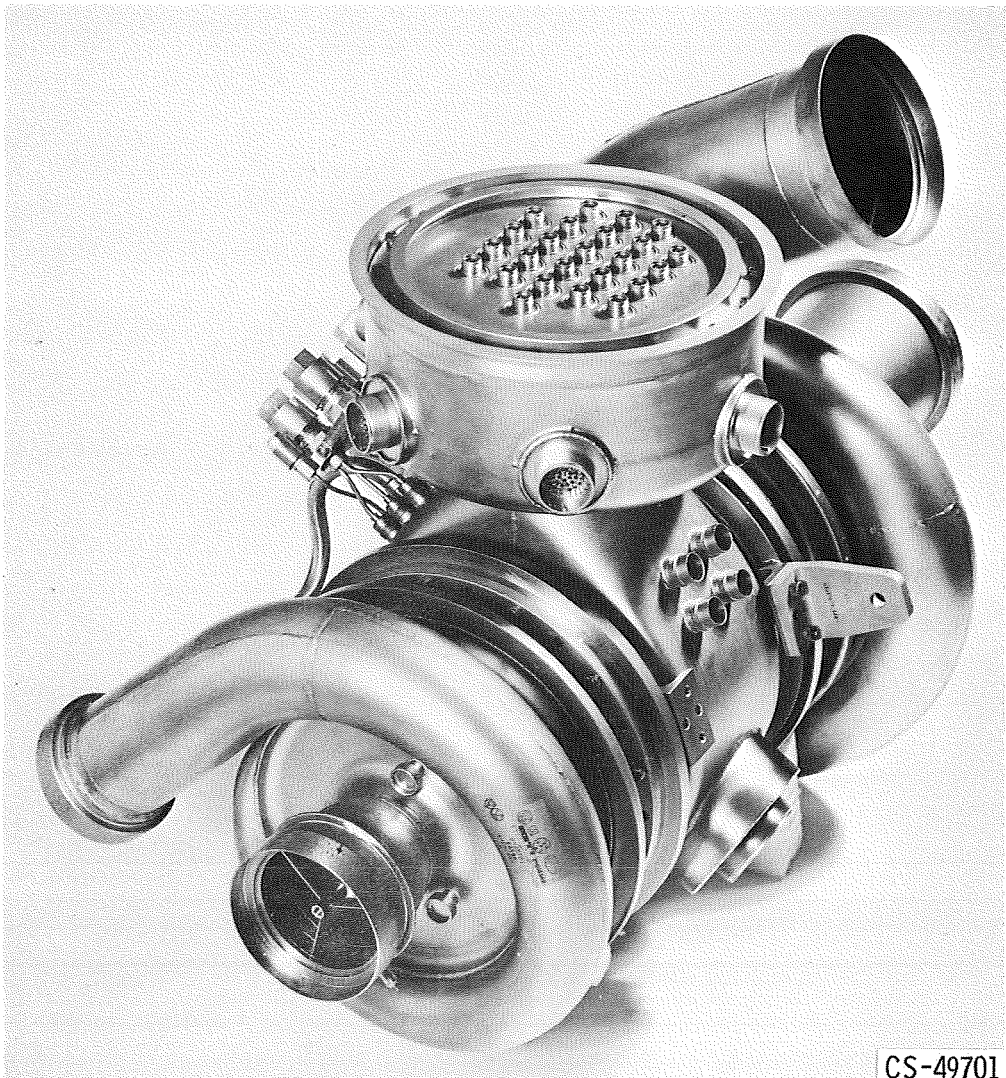
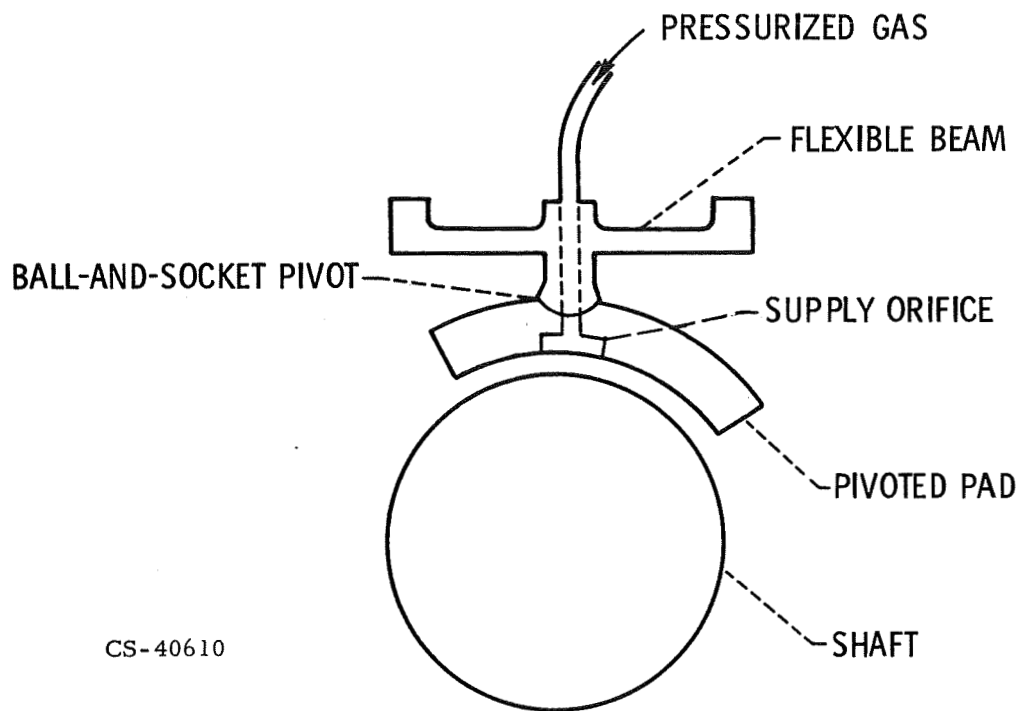


Figure 1. - BRU schematic.



CS-49701

Figure 2. - Brayton rotating unit.



CS-40610

Figure 3. - Pivoted-pad gas bearing.

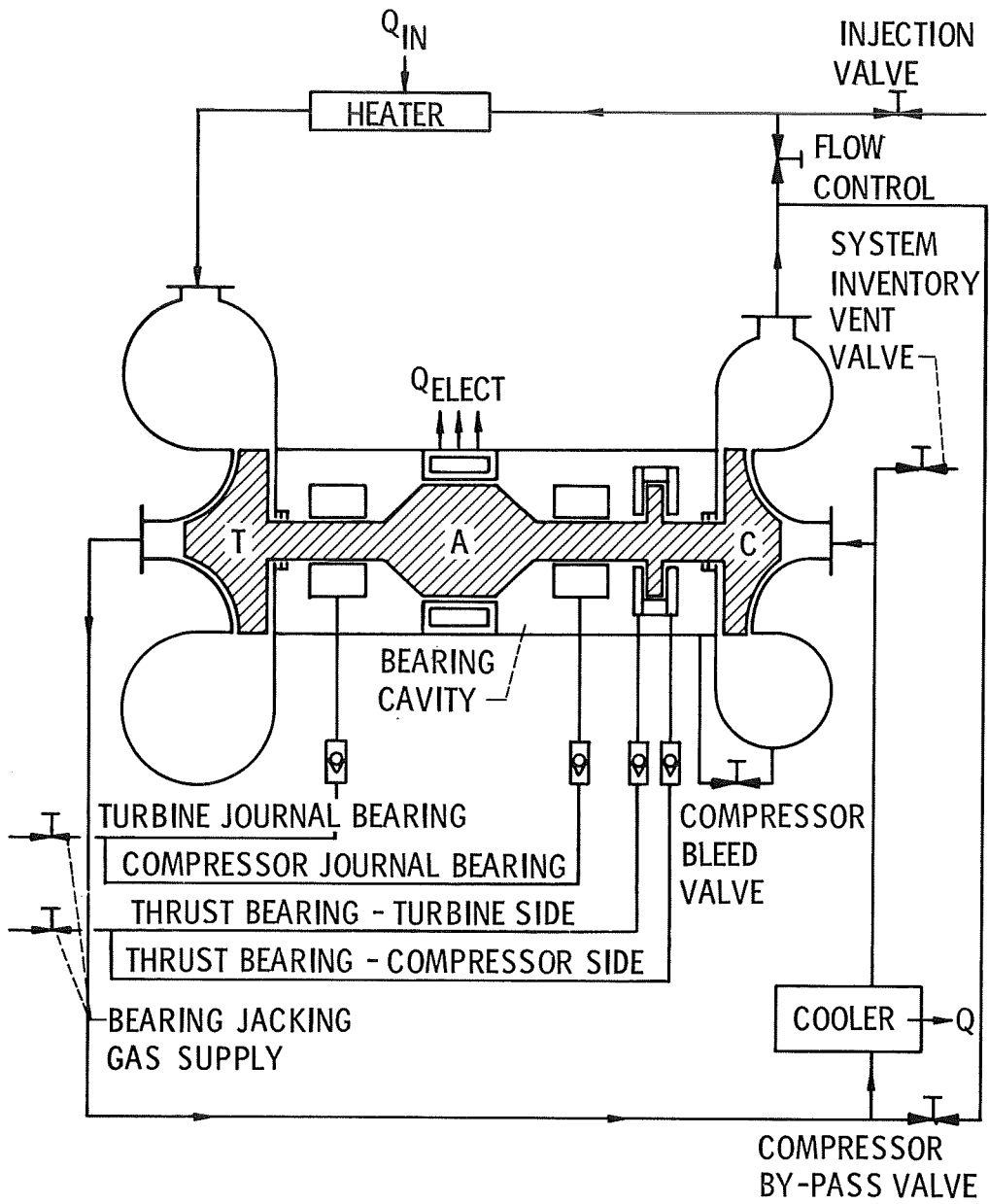


Figure 4. - BRU test system.

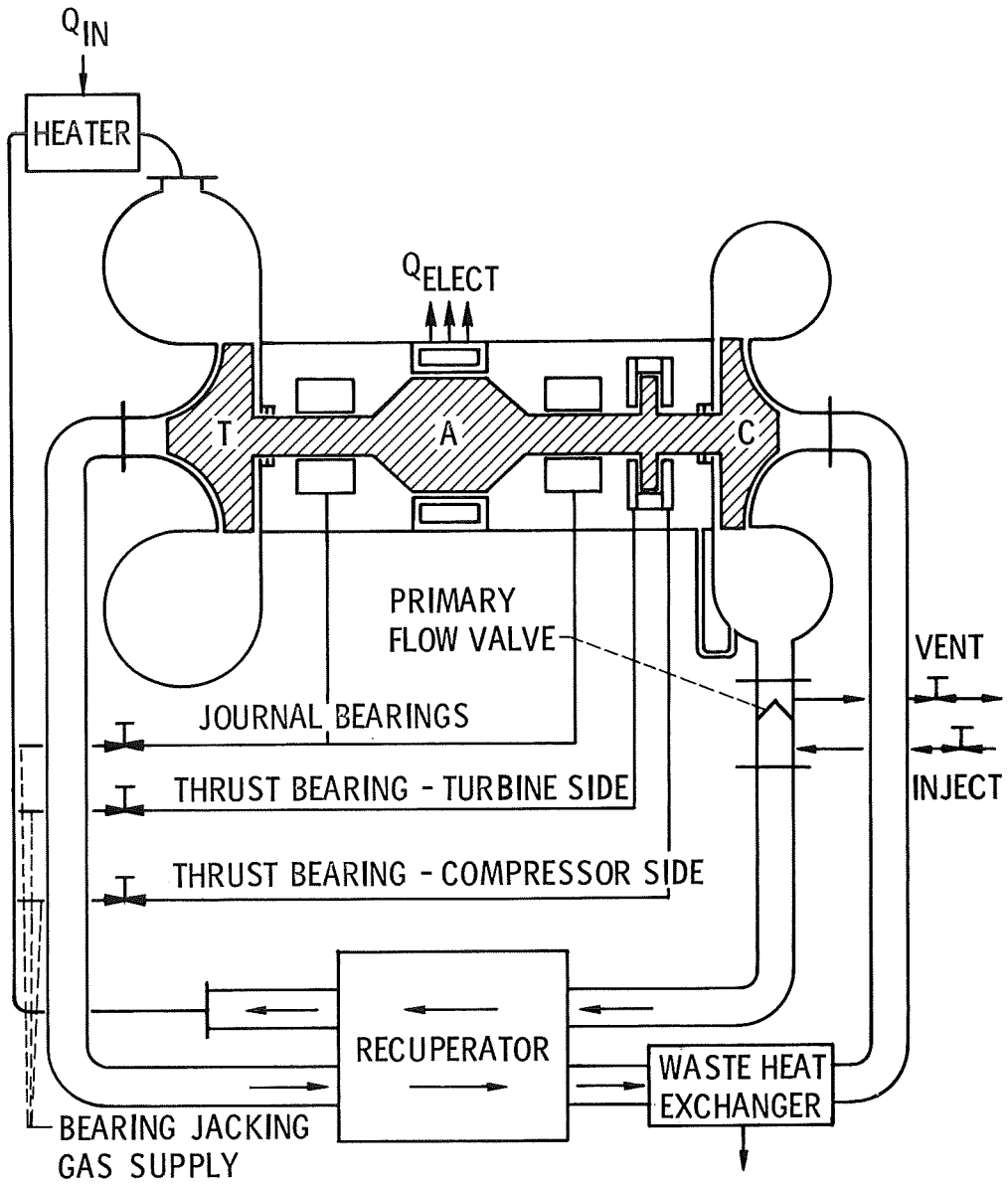


Figure 5. - BRU in Brayton gas loop test system.

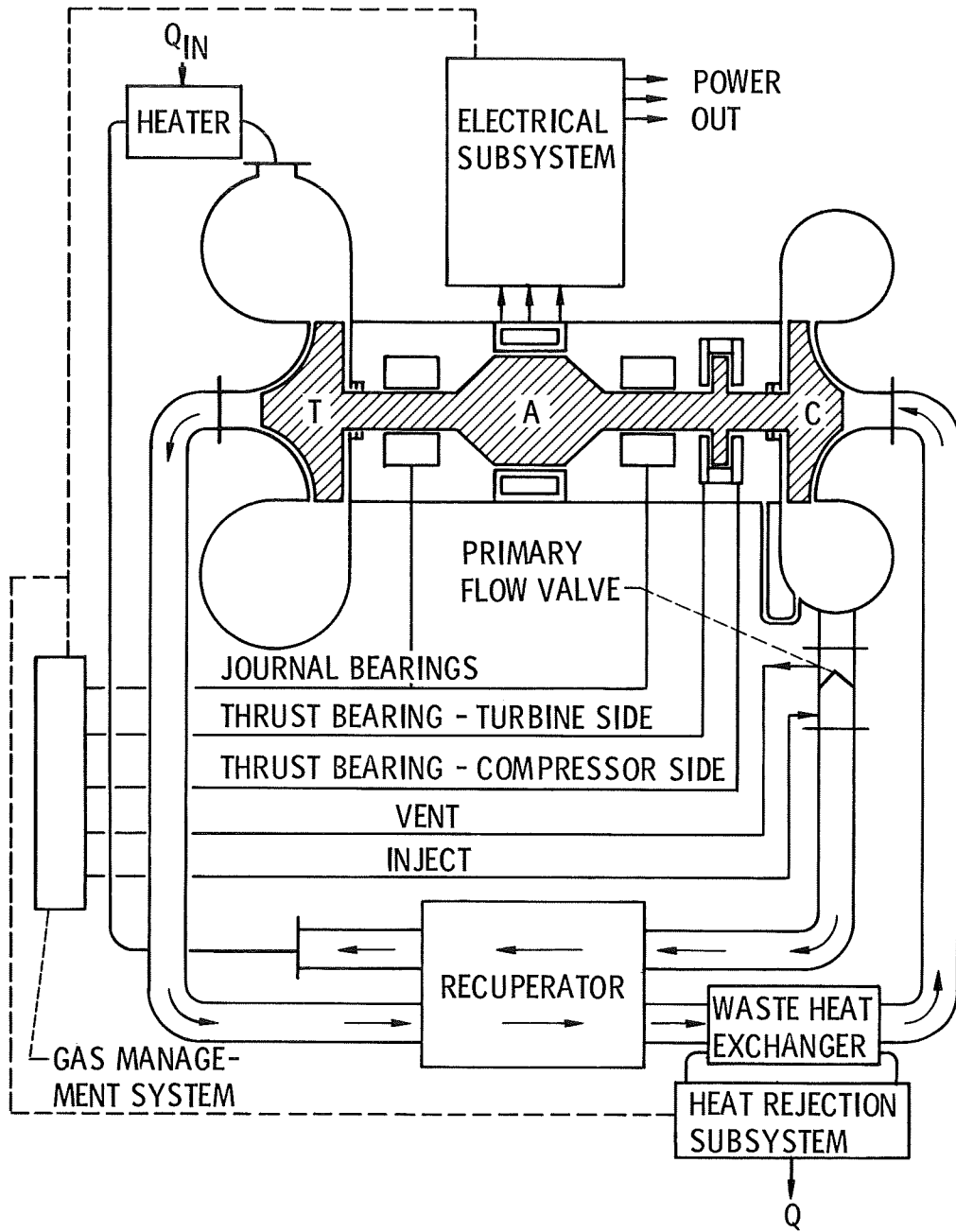


Figure 6. - BRU in Brayton engine test system.

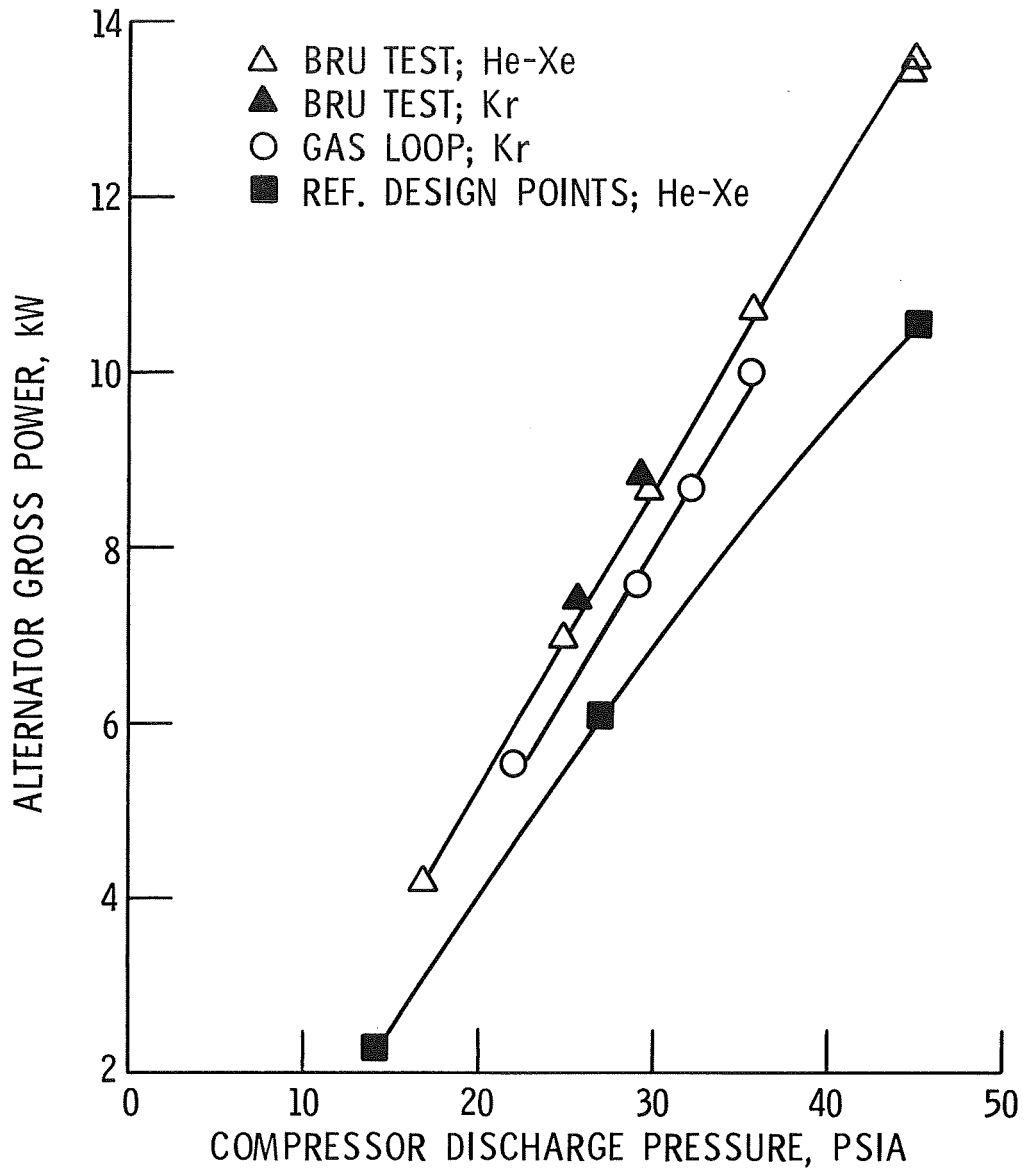


Figure 7. - BRU gross power output. Turbine inlet temperature, 1600° F (1577° to 1609° F); compressor inlet temperature, 80° F (74° to 83° F).

	PRESSURE RATIO		
	REF. DESIGN	BRU TEST	BRAYTON GAS LOOP
COMPRESSOR	1.9	1.9	1.94
TURBINE	1.75	1.85	1.78
DIFFERENCE	0.15	0.05	0.16

- △ BRU TEST; He-Xe
- ◇ BRU TEST; Kr
- GAS LOOP; Kr
- REF. DESIGN POINTS

SOLID SYMBOLS DENOTE COMPRESSOR
OPEN SYMBOLS DENOTE TURBINE

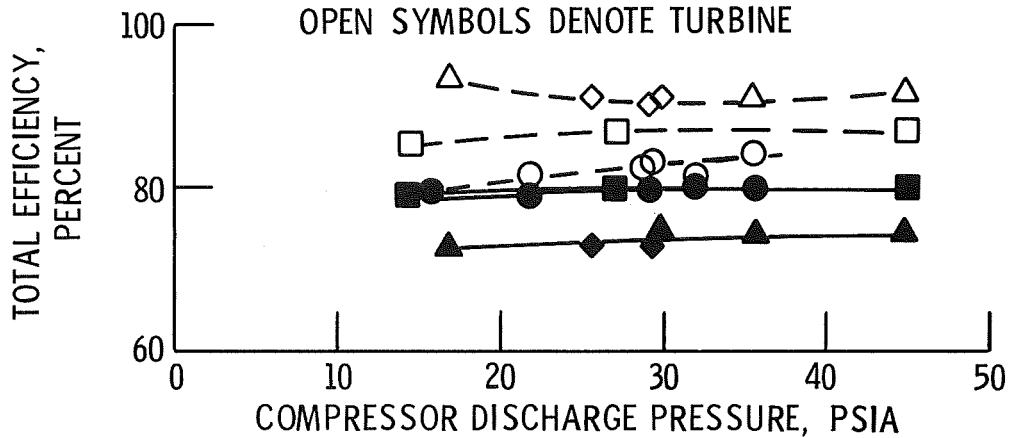


Figure 8. - Turbine and compressor performance comparison. Turbine inlet temperature, 1600° F (1577° to 1609° F); compressor inlet temperature, 80° F (74° to 83° F).

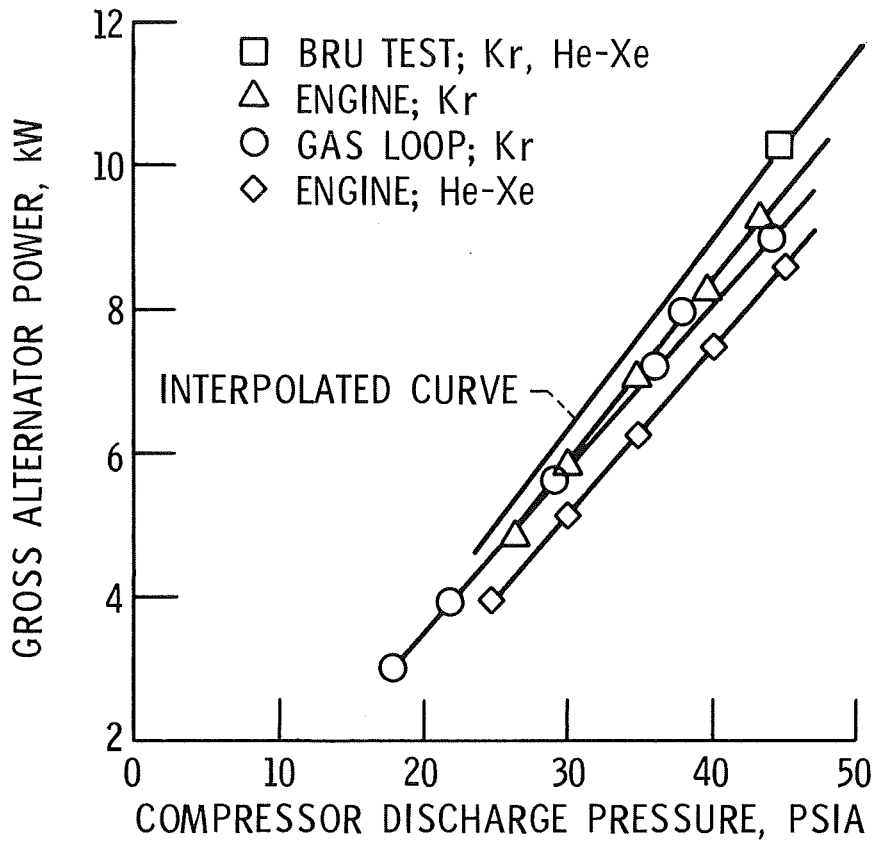


Figure 9. - BRU gross output power comparison. Turbine inlet temperature, 1400°F (1393° to 1406°F); compressor inlet temperature, 80°F (74° to 85°F).

	PRESSURE RATIO		
	BRU TEST	BRAYTON GAS LOOP	BRAYTON ENGINE
COMPRESSOR	1.87	1.91	1.89
TURBINE	1.82	1.77	1.72
DIFFERENCE	0.05	0.14	0.17

TURBINE COMPRESSOR

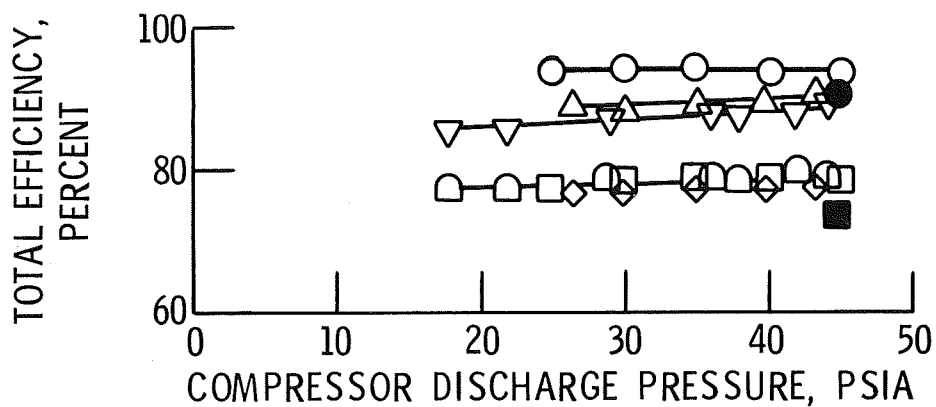
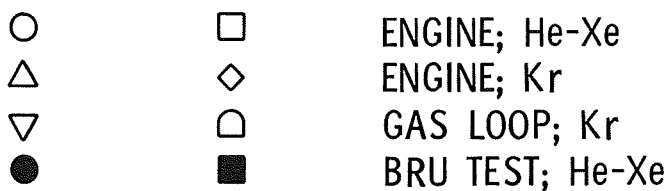


Figure 10. - Turbine and compressor efficiency comparison.

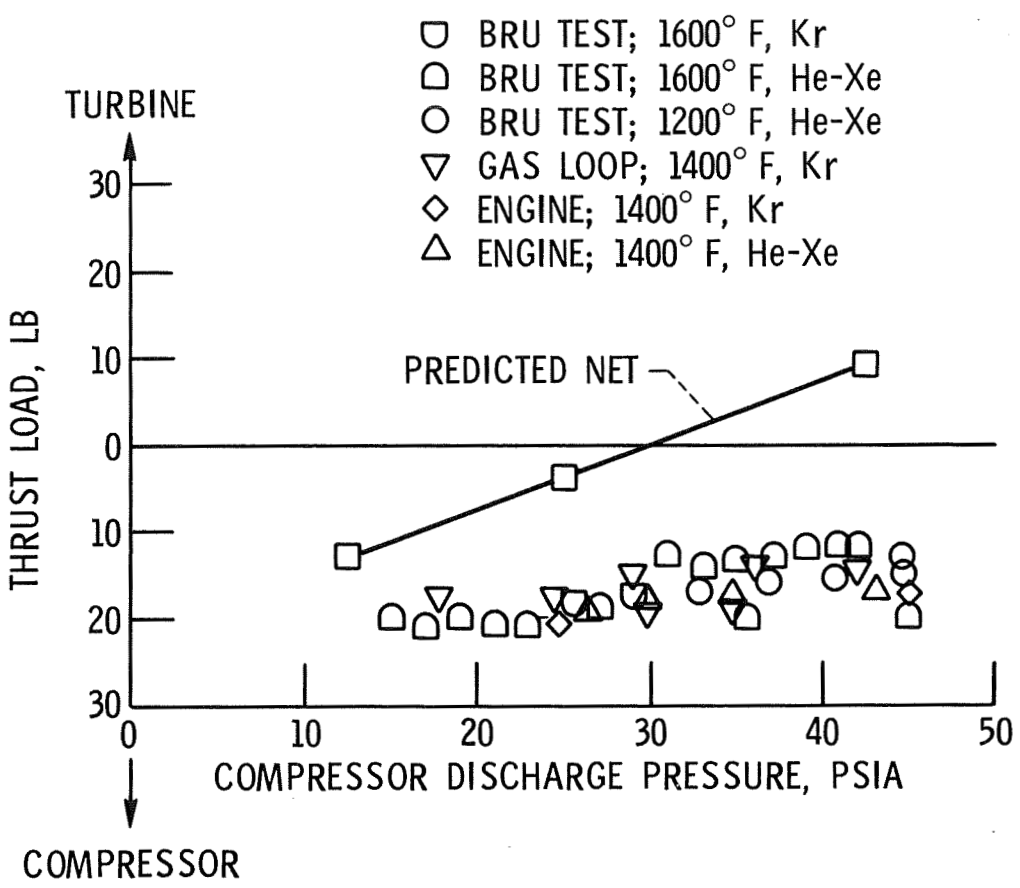


Figure 11. - Thrust bearing loads.