

FINAL TEST REPORT

APS-5270-R

AIRESEARCH BRAYTON CYCLE DEMONSTRATOR

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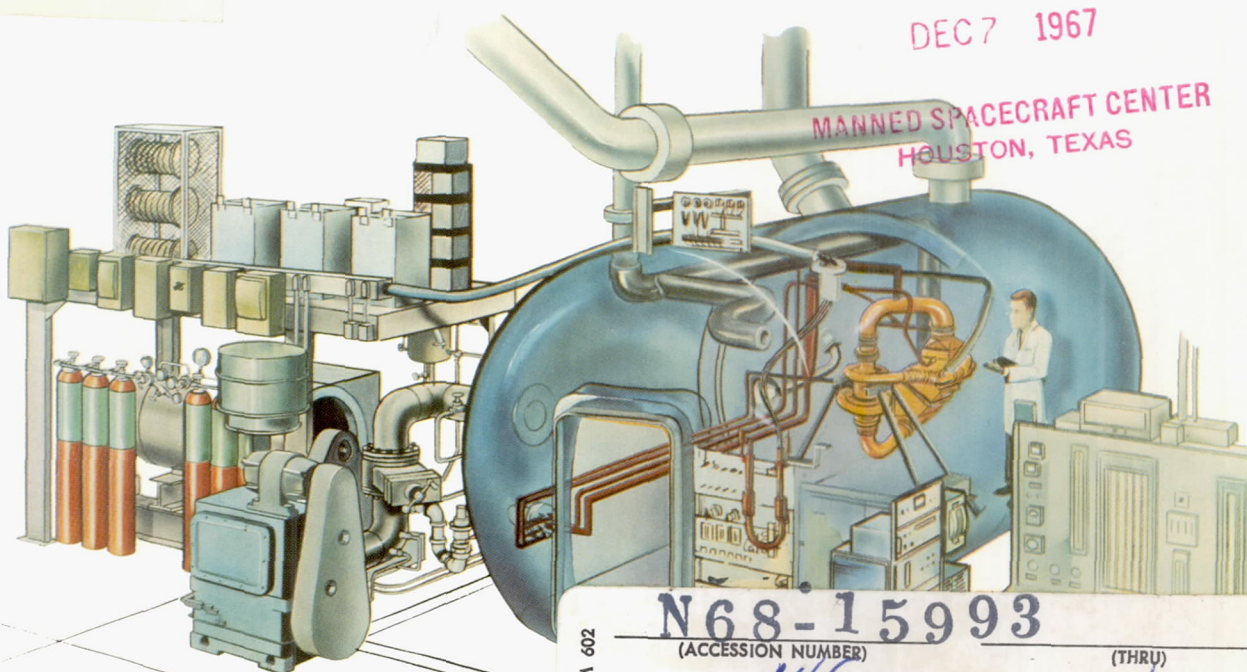
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MANNED SPACECRAFT CENTER
HOUSTON, TEXAS

AIRESEARCH BRAYTON CYCLE DEMONSTRATOR

Electrical Power Output 3KW
Turbine Inlet Temperature 1540° F
R. P. M. 64,000
Cycle Fluid Argon

FACILITY FORM 602

N68-15993

(ACCESSION NUMBER)

(THRU)

(PAGES)

(CODE)

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AIRESEARCH MANUFACTURING DIVISION OF ARIZONA

Phoenix, Arizona

FINAL
TEST REPORT

AIRESEARCH BRAYTON CYCLE DEMONSTRATOR
TEST PROGRAM
CONTRACT NAS9-6116

APS-5270-R

November 15, 1967

Prepared by Engineering Staff

Initial Issue

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AIRESEARCH MANUFACTURING COMPANY

A DIVISION OF THE GARRETT CORPORATION

PHOENIX, ARIZONA

REPORT NO. APS-5270-R

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ATTACHMENTS: Appendix I - Data Reduction Program (8 pages)
 Appendix II - Computer Program 532 (Compatible with
 CDC 3400 Computer - Fortran IV) (10 pages)
 Appendix III - Summary Test Log BCD Test Program (7 pages)

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A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

CONTRACTOR REPORT

ASSEMBLY AND CHECKOUT TESTING OF A 3-KILOWATT BRAYTON CYCLE DEMONSTRATOR

The work reported herein was accomplished under Contract NAS9-6116 by the AiResearch Manufacturing Company of Arizona, Phoenix, Arizona.

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

Manned Space Center

Houston, Texas

Program Monitor
A.E. Redding

APS-5270-R

November 15, 1967



LIST OF SYMBOLS

g_o, g	= conversion factor = 32.2 ft lb _f per lb _m sec. ²
$\dot{m}_o$	= molal gas flow rate, lb-mols per sec.
r_c	= compressor pressure ratio
r_t	= turbine pressure ratio
C_p	= specific heat with constant pressure
C_v	= specific heat with constant volume
D_c	= compressor-wheel diameter, inches
D_t	= turbine-wheel diameter, inches
E_R	= recuperator effectiveness
M	= molecular weight, lbs per lb-mole
N	= shaft speed, rpm
N_{S_c}	= compressor specific speed
N_{S_t}	= turbine specific speed
P_1	= compressor inlet pressure, lbs per sq in.
P_2	= compressor outlet pressure, lbs per sq in.
P_5	= recuperator outlet pressure, lbs per sq in. (high-pressure side)
P_6	= turbine inlet pressure, lbs per sq in.
P_9	= turbine outlet pressure, lbs per sq in.
P_{10}	= recuperator outlet pressure, lbs per sq in. (low-pressure side)
Q	= Heat flow or flux, BTU per sec.
R	= Universal gas constant = $\frac{1,545}{M}$ ft-lbs per lb-mole °R
S	= Slip factor
T_1	= compressor inlet temperature, °R



LIST OF SYMBOLS (Contd.)

T_5	= recuperator outlet temperature, $^{\circ}\text{R}$ (high pressure side)
T_6	= turbine inlet temperature, $^{\circ}\text{R}$
T_9	= turbine outlet temperature, $^{\circ}\text{R}$
T_{10}	= recuperator outlet temperature, $^{\circ}\text{R}$ (low pressure side)
T_r	= temperature ratio
U	= tip speed, ft per sec.
θ	= $\frac{\text{turbine pressure ratio}}{\text{compressor pressure ratio}} = \frac{r_t}{r_c}$
γ	= ratio of gas specific heats = 1.667 for monatomic gases
θ	= $(\gamma - 1)/\gamma = 0.4$ for monatomic gases
η_{cy}	= cycle efficiency
η_c	= compressor adiabatic efficiency, total-to-static
η_t	= turbine adiabatic efficiency, total-to-static
η_{co}	= cooler efficiency
η_h	= heater efficiency
Δp	= pressure difference (psi)



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PHOENIX, ARIZONA

FINAL TEST REPORT
AIRESEARCH BRAYTON CYCLE DEMONSTRATOR
TEST PROGRAM
CONTRACT NAS9-6116

1.0 INTRODUCTION AND SUMMARY

1.1 Introduction

Substantial quantities of electrical power will be required for long periods of time for space applications. Presently, rather modest amounts of power are required for communications, life support, navigation, etc., on a short-term basis. The future long-term needs, however, include power for auxiliary systems to support larger manned crews and electrical propulsion. To achieve these goals, mission planners are considering the use of dynamic power systems that convert nuclear or solar energy to electrical power and can operate without interruption for a period of years.

The closed Brayton-cycle power-conversion system is considered to be an attractive method for supplying electrical power from a few kilowatts to several hundred kilowatts for space-vehicle use. This report describes the testing that has been completed on this type of power conversion system.

The 3-kilowatt closed-Brayton-cycle unit that was tested in this program was originally built in 1963 to demonstrate system feasibility. The primary motivation at that time was to design, fabricate, and operate a complete system with the use of currently available hardware, and thereby demonstrate the adequacy of contemporary technology to produce such a system. Minimum analytical effort was expended at that time and the budget permitted only minimal development effort.



The BCD system, in the 1963 configuration, was designed, fabricated, and tested in less than one year. The system at that time demonstrated an overall efficiency of 18 percent, based on total electrical power output from the alternator and total electrical power supplied the heater elements. The cycle temperature ratio ($T_1/T_8 = 0.25$) was not optimized for any particular space application but is consistent with isotope applications.

The performance level achieved during the 1963 tests was impressive because the aerodynamic components were never matched during component testing. Component matching is normally done in turbine engine development when high performance is the objective.

It must also be pointed out that the compressor and turbine still have not been matched, since performance level was not a primary objective of this program and funding was limited.

The original combined rotating unit (CRU) which was tested in 1963, had conventional oil-lubricated rolling element bearings. In the early part of 1966 the CRU was modified by AiResearch to include gas-lubricated bearings, which use argon working fluid as the lubricant, and an improved brushless, solid-rotor alternator. This work was underway under company sponsorship when the NASA became interested in the effort.

As a part of the total effort at the NASA Manned Space Center (MSC) to keep abreast of space electrical power systems technology, plans were formulated in early 1966 to acquire a "breadboard" closed-Brayton-cycle demonstration system for in-house testing and evaluation. The AiResearch Brayton-cycle demonstrator satisfied



the requirements for this system; therefore, a program was initiated with AiResearch to modify, check out, and consign the demonstrator to MSC for evaluation testing. This program was conducted under Contract NAS9-6116.

1.2 Summary

During the test program conducted at the Contractor's facility several important accomplishments were made. These include:

- (1) Application of magnetic shock loads and maximum continuous magnetic loads to the CRU gas bearings, thereby proving the feasibility of gas bearings for this type of application.
- (2) Development of successful gas injection start-up procedures for the closed cycle gas turbine engine supported on gas bearings. Over 40 gas injection starts have been made. Start-up times of less than 30 seconds to operating speed have been demonstrated.
- (3) Many hours of continuous BCD operation was successfully completed, including one test lasting for 75 hours. This test conclusively demonstrated the insensitivity of the system to thermal transients and to load changes.
- (4) During the BCD check-out phase of this program, over 100 hours of closed loop testing were accumulated.
- (5) Control systems which provide for safe operation during start-up and steady-state operation were developed and demonstrated.



- (6) A wide range of operating parameters was investigated during the tests, demonstrating the high degree of flexibility existing with the Brayton cycle system.
- (7) Twenty percent cycle efficiency was achieved with the CRU rotating at speeds from 50,000 rpm to 60,000 rpm.

The program at AiResearch lasted 14 months and was completed within the original contract funding (\$75,000). The BCD is ready for shipment to the NASA test facility in Houston, Texas.

During the test program conducted at AiResearch, considerable system experience was gained. Test procedures, system control methods, and data acquisition approaches used during this test program are thoroughly described in a document which has been prepared for transmittal to the NASA, MSC. This document is entitled "BCD Operating Manual and Test Instructions," Report Number SY-6016-R.



2.0 DESCRIPTION OF THE 3-KW BRAYTON-CYCLE SYSTEM

2.1 Thermodynamic Considerations

The theoretical closed gas cycle can be represented on T-S and P-V diagrams, as shown in Figures 1 and 2.

Process 1-2 Reversible, adiabatic compression

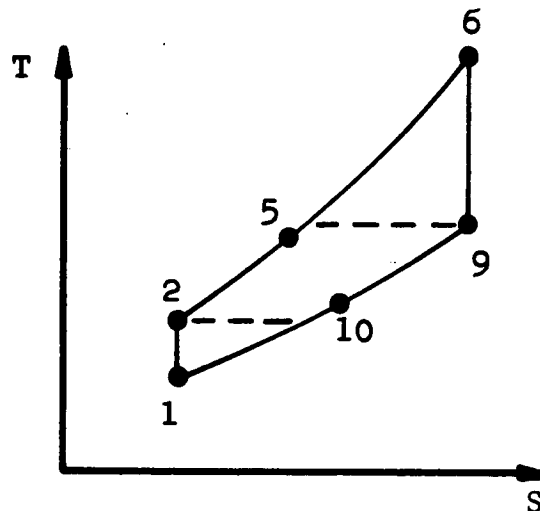
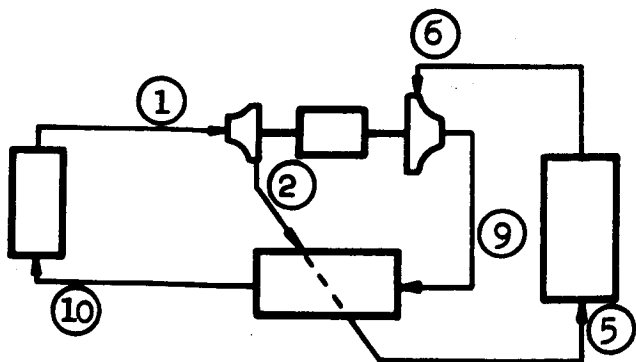
2-6 Reversible, constant-pressure heat addition

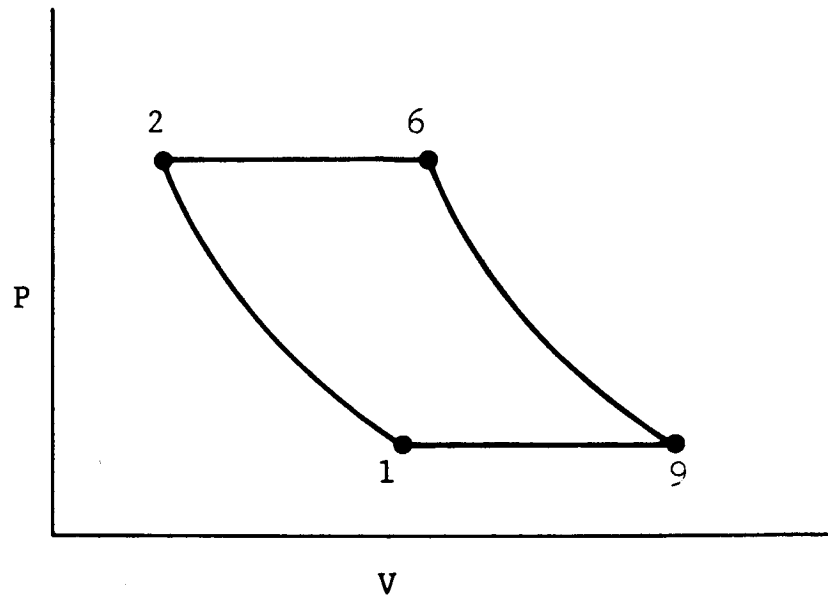
6-9 Reversible, adiabatic expansion

9-1 Reversible, constant-pressure heat rejection

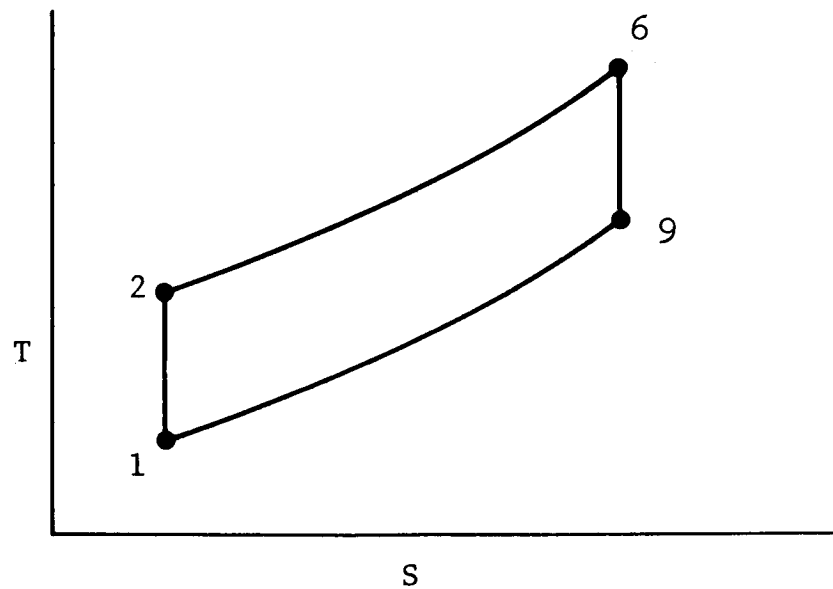
The recuperated cycle has an additional heat exchanger, which transfers heat that would otherwise be rejected from the cycle, from the low-pressure side of the system to the high-pressure side of the system. The T-S diagram is therefore modified as shown in Figure 3.

A schematic diagram of the loop is shown with its relationship to the T-S diagram in the sketch below.

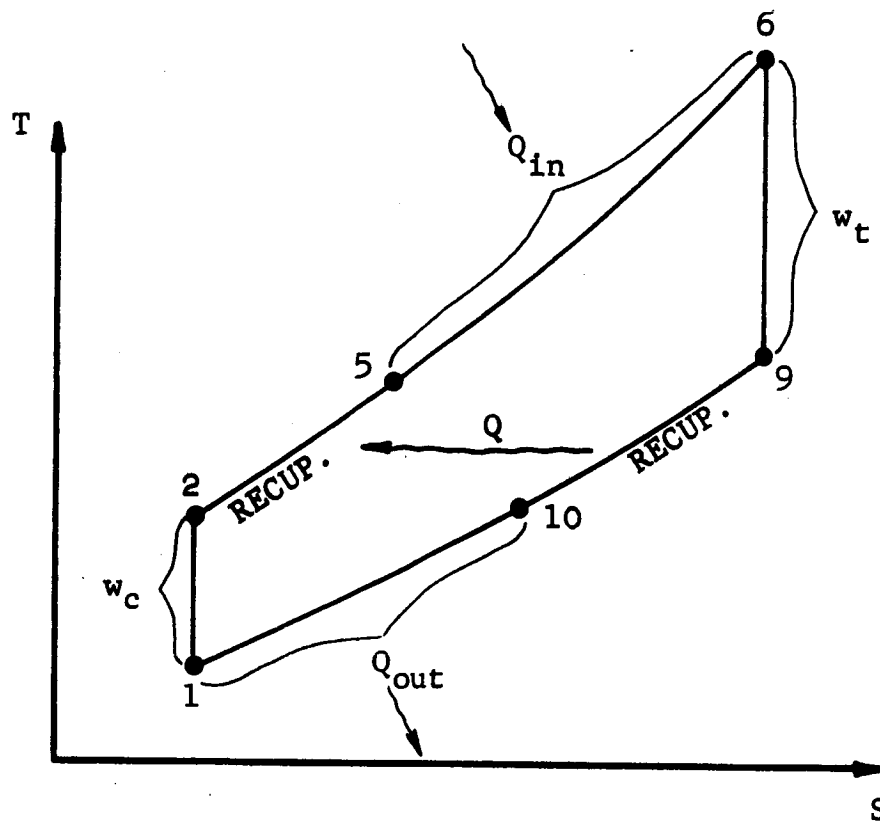




P-V DIAGRAM, IDEAL BRAYTON CYCLE
FIGURE 1



T-S DIAGRAM, IDEAL BRAYTON CYCLE
FIGURE 2



$$Q_{\text{recup}} = \dot{m} C_p (T_5 - T_2) = \dot{m} C_p (T_9 - T_{10})$$

$$\eta_{\text{cy}} = \frac{(Q_{5,6} - Q_{1,10})}{Q_{5,6}} = \frac{Q_{\text{in}} - Q_{\text{out}}}{Q_{\text{in}}}$$

T-S DIAGRAM, IDEAL RECUPERATED BRAYTON CYCLE

FIGURE 3



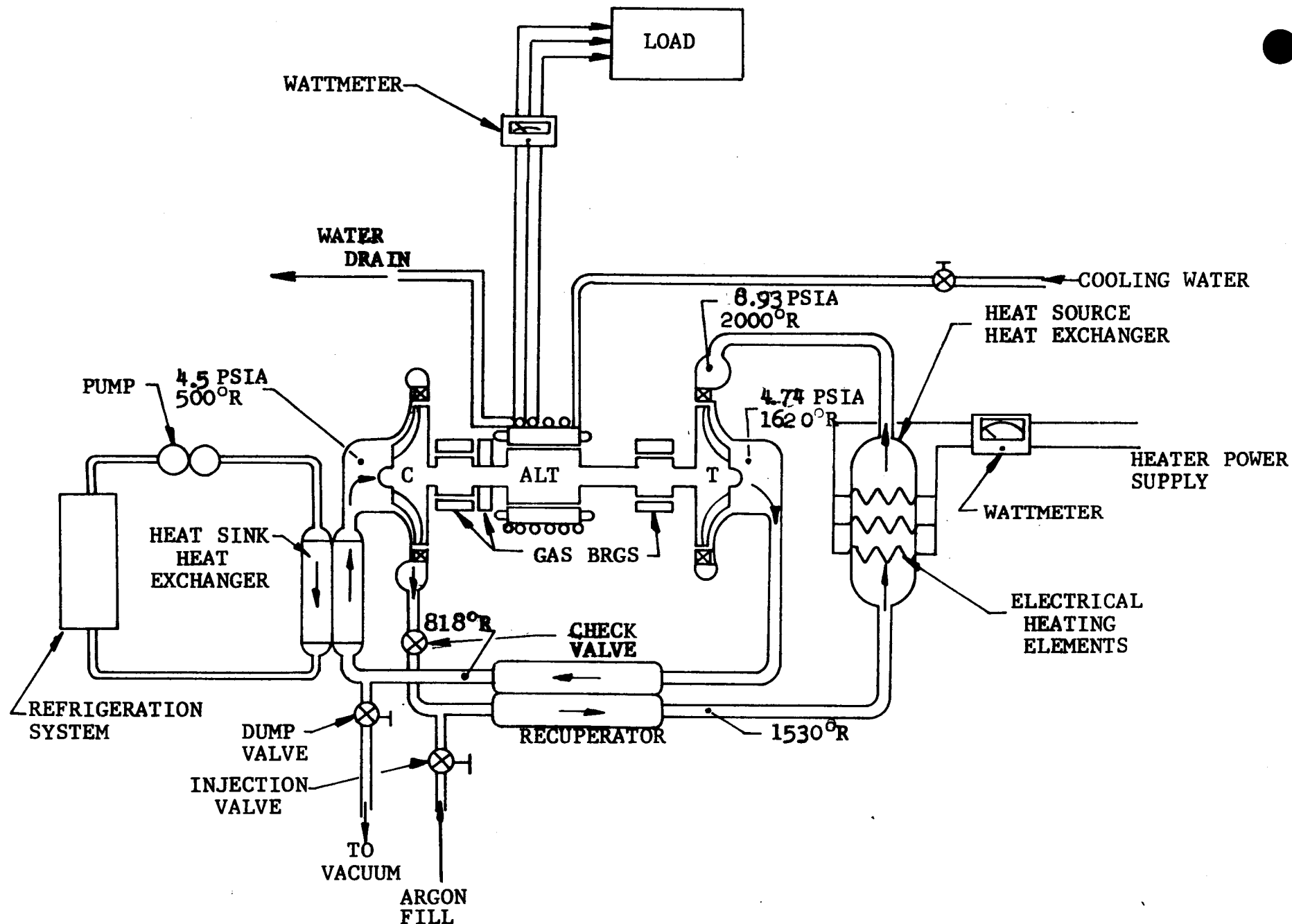
The major components of the thermodynamic loop proper are:

- 1-2 A radial-outflow compressor
- 2-5 A high-effectiveness argon-to-argon recuperator
- 5-6 An electrical heater
- 6-9 A radial-inflow turbine
- 9-10 The low-pressure side of the recuperator (References
 2 through 5)
- 10-1 A gas-to-liquid cooler

The general operation of the loop requires a supply of energy to the cycle through the electrical heater, expansion of heated, pressurized gas through the turbine, which drives the compressor and alternator on a common shaft; recuperation of heat energy from the turbine discharge flow to the compressor discharge flow, rejection of waste heat from the cycle through the cooler, and conversion of mechanical energy to electrical energy by the turbine-driven alternator. A resistance load bank was utilized to absorb the alternator output.

2.2 Reference Cycle

The reference cycle is shown schematically in Figure 4. The original design speed of the BCD was 64,000 rpm. During tests at this speed, the effects of unbalance were unacceptable due to excessive motions of the existing rotating group. Satisfactory operation was achieved at speeds up to 60,000 rpm. Since



SCHEMATIC DIAGRAM, 3 KW CLOSED-
BRAYTON-CYCLE SYSTEM

FIGURE 4



funding was not available to redesign and reconstruct a new rotating group it was decided to investigate the performance loss incurred by operating at a lower maximum speed. An analysis of the system was made with the objective to reoptimize the cycle at a lower operating speed. The results of this analysis are shown in Table 1. Very little performance penalty is incurred with this change. It should be reiterated that the primary objective of the program was to acquire closed loop Brayton-cycle operating experience.

Other pertinent data for the system follows. The total weight of argon in the loop at operating conditions is:

$$M_T = 0.071 \text{ lb of argon with a flow rate of } 0.133 \text{ lb per sec.}$$

The time required to make a completed circuit of the loop is:

$$T = 0.535 \text{ seconds per complete circuit, or}$$
$$f = 1.87 \text{ circuits per second}$$

2.2.1 Loop-Component Physical Description

2.2.1.1 Combined Rotating Unit (CRU)

The CRU, shown in cross section in Figure 5, includes the dynamic components of the power-conversion system. This system consists of a compressor, turbine, and alternator, on a common shaft supported by gas-lubricated bearings. The CRU operates at a speed of 60,000 rpm with a design turbine inlet temperature of 1540°F.

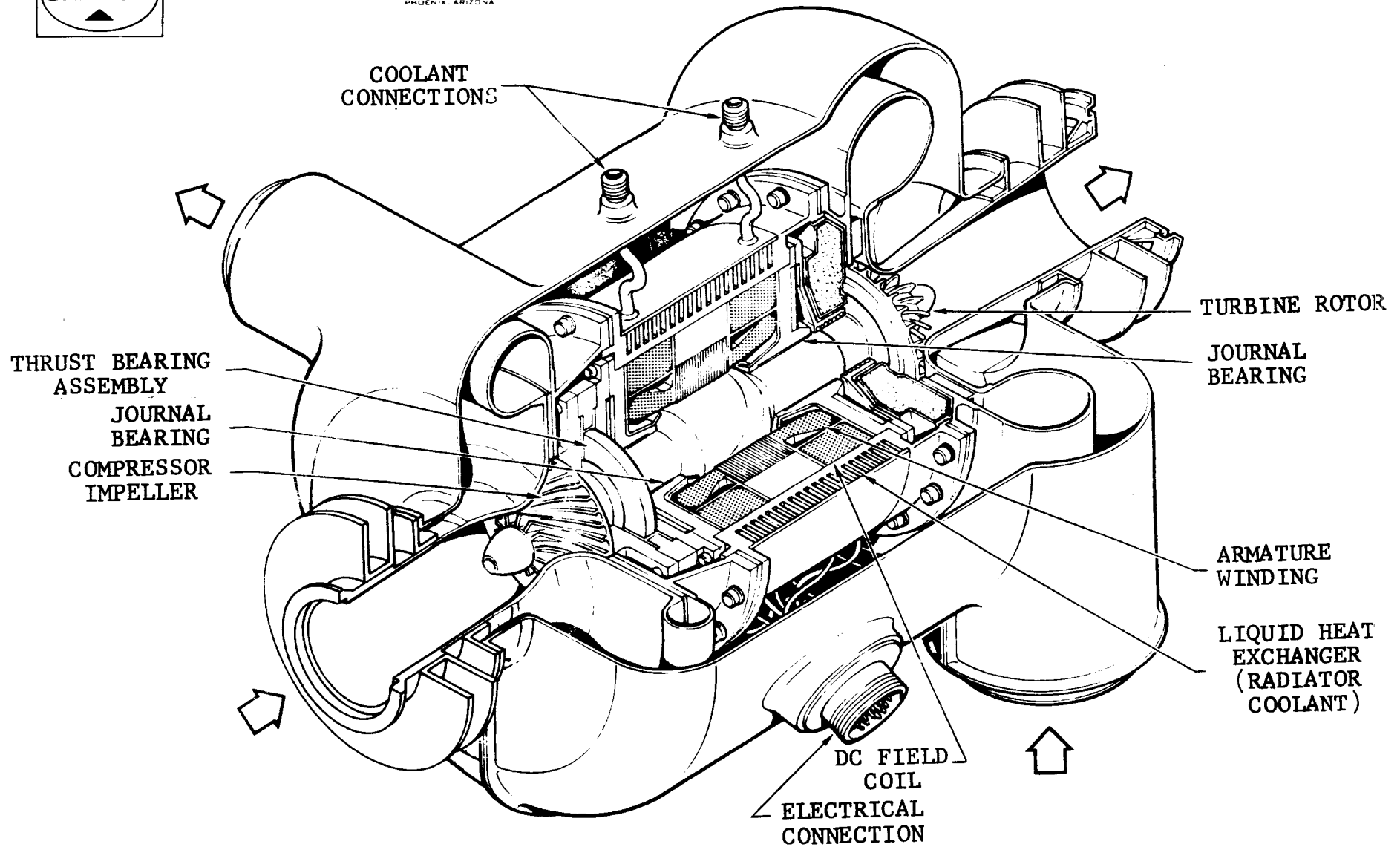


TABLE I
CYCLE REOPTIMIZATION ANALYSIS

	<u>Original Cycle</u>	<u>Increased P₁</u>	<u>Decreased KW_e</u>
Electrical power output (kw _e)	3.0	3.0	1.88
Rotating speed (rpm)	64,000	60,000	60,000
Compressor inlet temperature (°R)	500	500	500
Compressor outlet temperature (°R)	760	729	729
Heater inlet temperature (°R)	1,486	1,530	1,530
Turbine inlet temperature (°R)	2,000	2,000	2,000
Turbine discharge temperature (°R)	1,567	1,620	1,620
Cooler inlet temperature (°R)	841	818	818
Compressor inlet pressure (psia)	4.5	6.7	4.5
Compressor outlet pressure (psia)	10.4	14.0	9.4
Turbine inlet pressure (psia)	10.0	13.3	8.93
Turbine discharge pressure (psia)	4.9	7.05	4.74
Working fluid	Argon	Argon	Argon
Compressor mass flow rate (lbs per sec.)	0.170	0.197	0.133
Compressor specific speed	0.108	0.10	0.10
Compressor pressure ratio	2.30	2.09	2.09
$\eta_{cy} = \frac{\text{shaft output}}{Q_{in}}$	0.336	0.322	0.322
$\eta_{cy_{net}} = \frac{\text{electrical output}}{Q_{in}}$	0.257	0.246	0.229



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COMBINED ROTATING UNIT

A33100

FIGURE 5



The critical speeds for the CRU are as follows:

First - 6,000 rpm

Second - 10,000 rpm

Third - 110,000 rpm

It is clear that the design speed of the CRU (60,000 rpm) is safely between the second and third critical speeds.

In the early stages of the program difficulty was encountered in achieving proper alignment of the CRU bearing bores during successive rebuilds. Analysis determined, therefore, that an alternator stator housing modification was required.

The housing was redesigned to provide a rigid structural member upon which both bearing carriers could be piloted and attached. This modification resulted in an improved method for establishing proper alignment of the CRU and for maintaining dimensional stability of the bearing bores during operation in the closed loop.

Figure 6 shows an exploded view of the CRU, including aerodynamic scrolls.

Figure 7 shows the CRU with the original stator housing.

Figure 8 depicts the modified stator housing.

Figure 9 shows the CRU with the heavier stator housing.

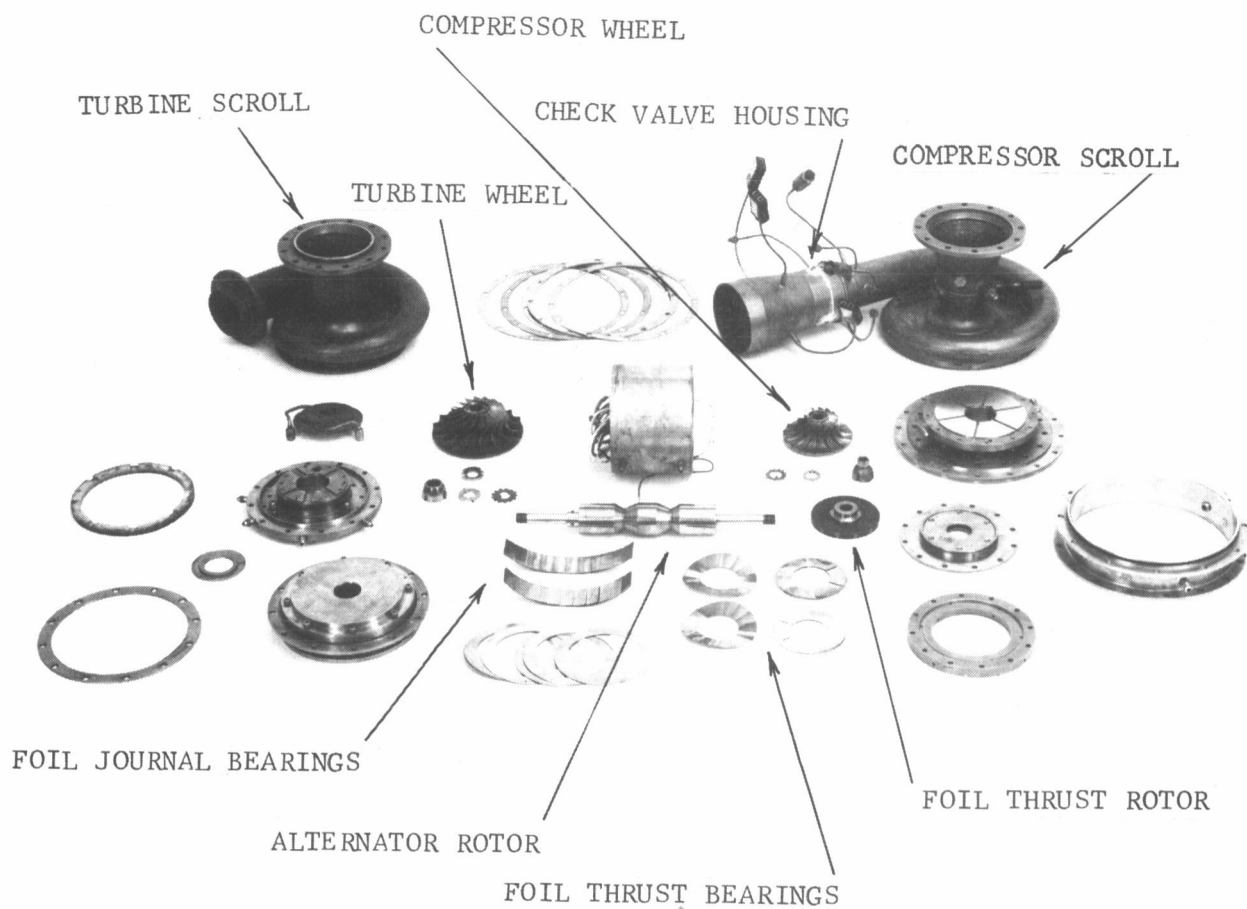


FIGURE 6

BCD TEST, PHOENIX
EXPLODED VIEW,
FOIL BEARING CRU

PHOTO NO. P-30433-12

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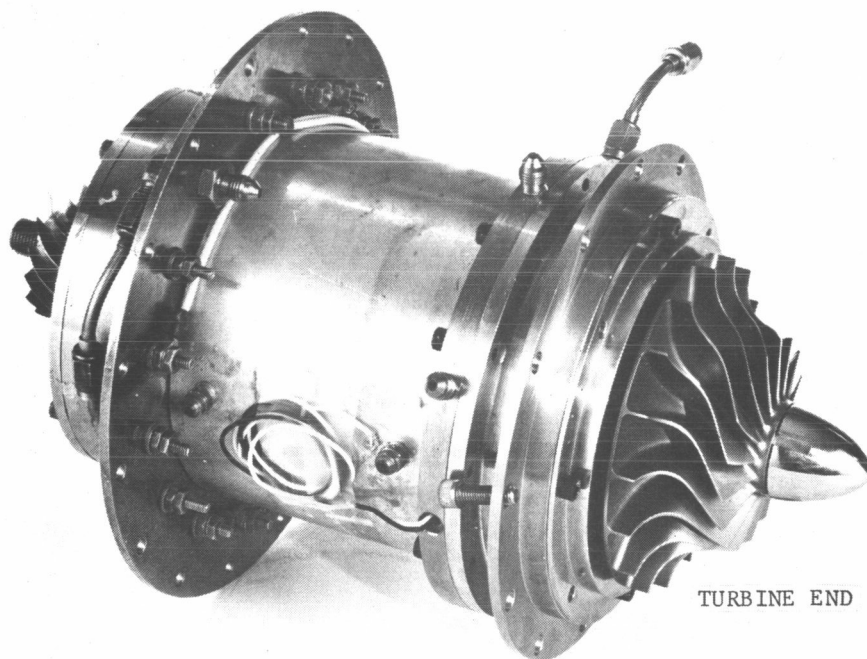
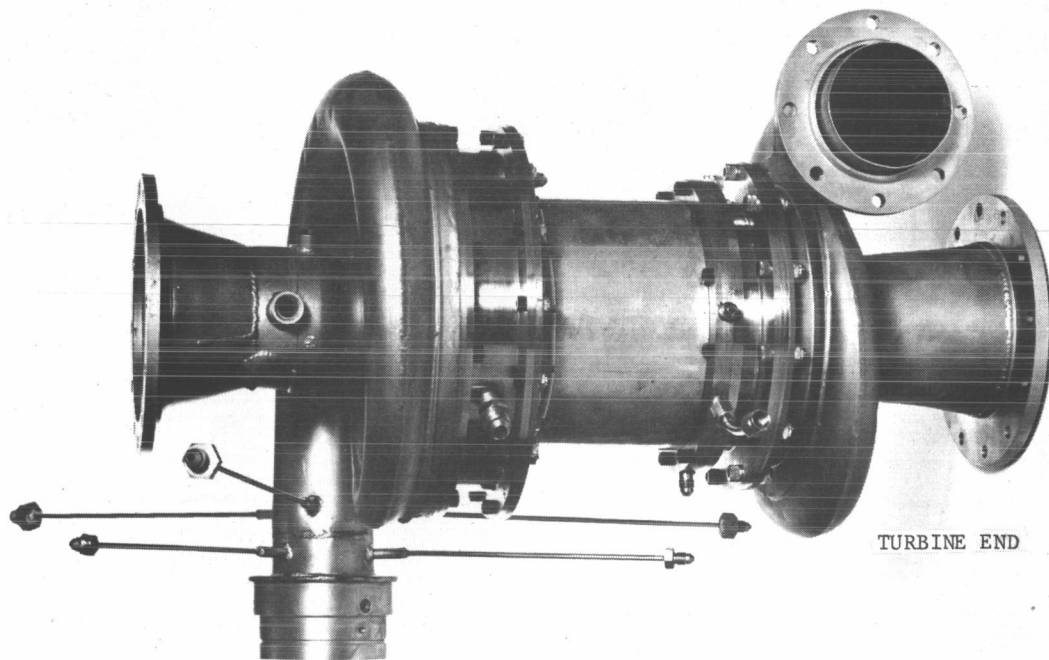


FIGURE 7
CRU WITH ORIGINAL STATOR HOUSING

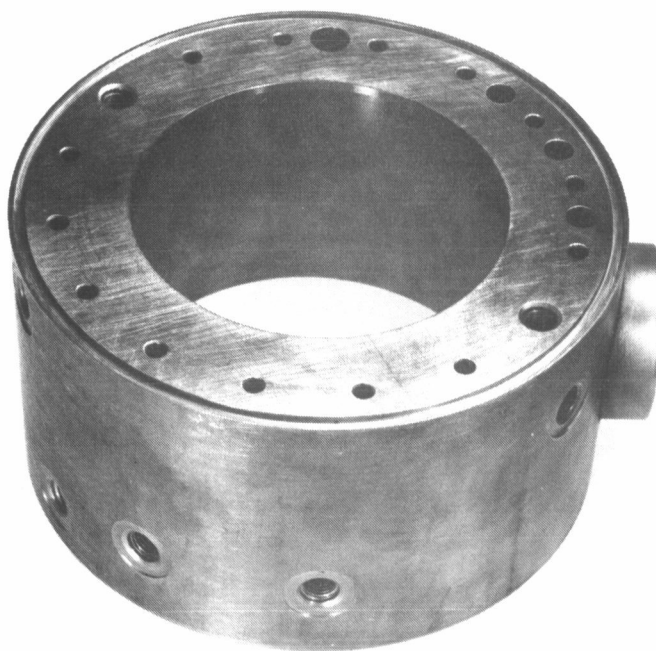


FIGURE 8



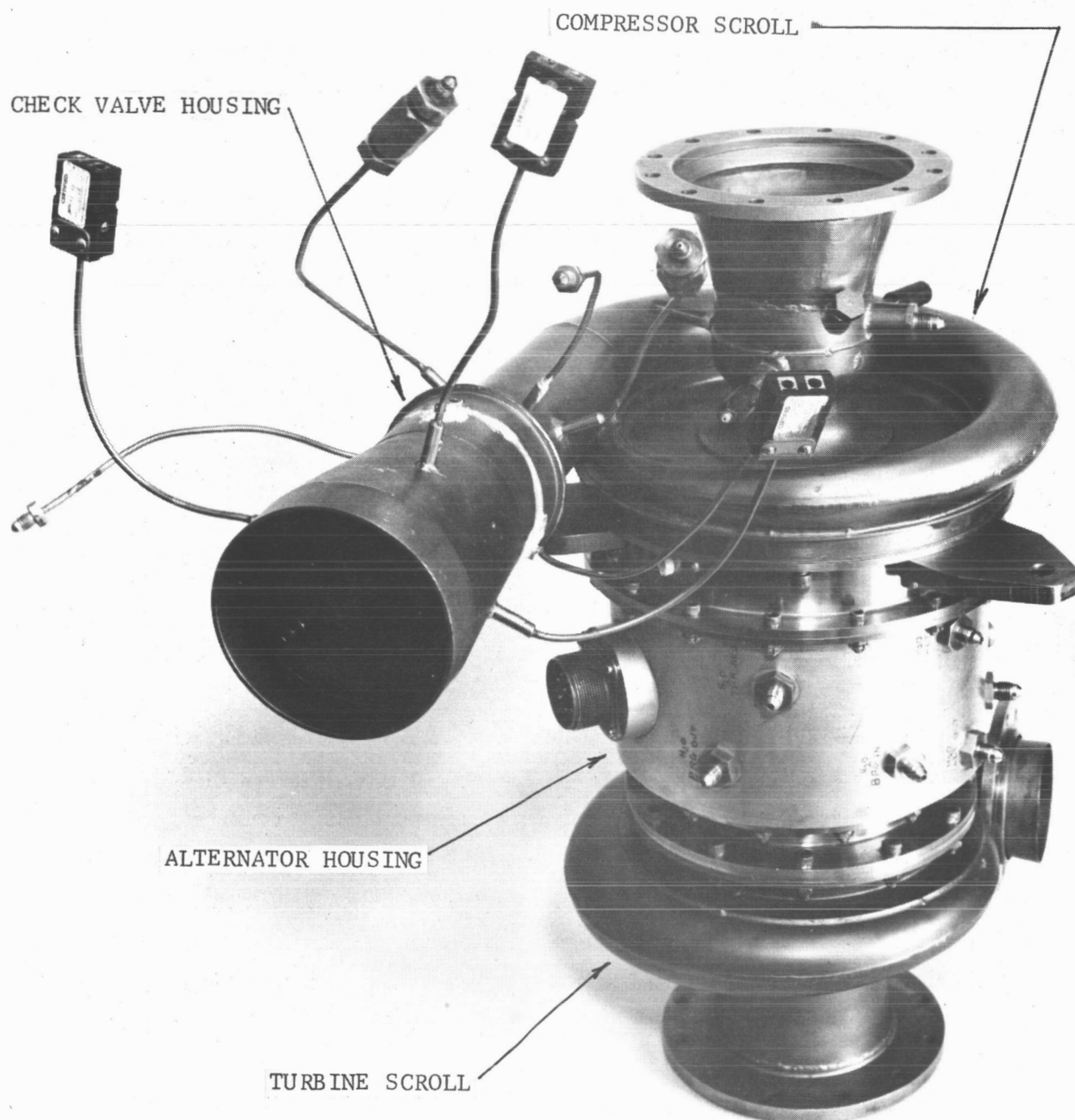


FIGURE 9

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COMBINED ROTATING UNIT
BCD P/N 699461

PHOTO NO. P-30865-1
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2.2.1.2 Compressor

The compressor, shown in Figure 10, is a single-stage, radial-outflow impeller with an outside diameter of 3.5 inches. The material is 17-4PH.

2.2.1.3 Turbine

The turbine, also shown in Figure 10, is a single-stage, radial-inflow wheel with an outside diameter of 4.6 inches. The turbine material is Waspaloy.

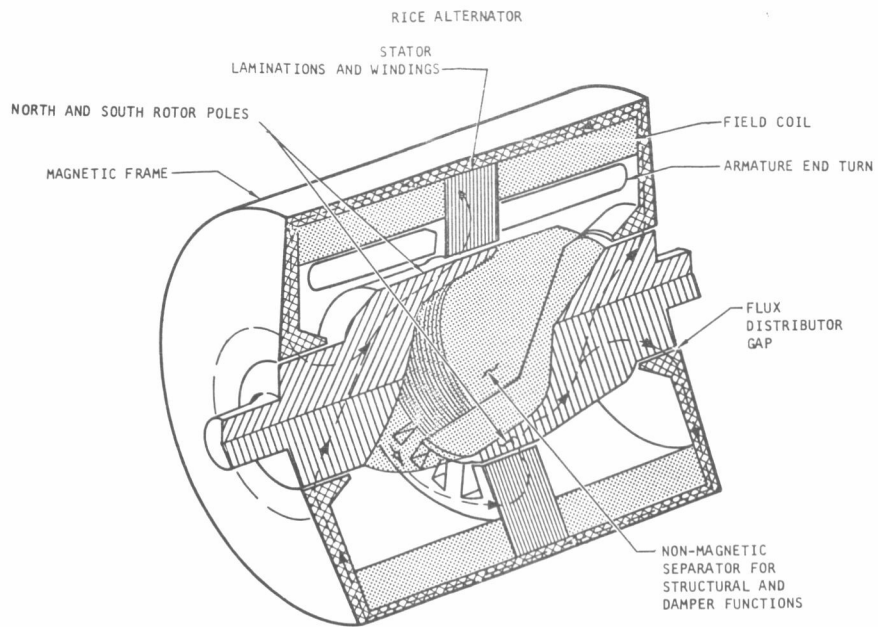
2.2.1.4 Alternator

The alternator in this machine is a two-pole Rice alternator, the cross section of which is shown in Figure 11. The Rice machine is a special type of synchronous alternator. The stator has a conventional three-phase a-c winding. The rotor is solid, and the two stationary field coils are located outside of the stator. The magnetic flux generated by the field coils passes into the rotor across a secondary air gap at the end of the rotor, through the north pole of the machine, around the magnetic frame, and re-enters the rotor at the south pole. Then it passes again across a secondary air gap at the end of the rotor to complete the flux circuit.

The construction of the rotor is shown. The two magnetic pole pieces are machined half pieces of the shaft. They are positioned with respect to each other by the use of special tooling while the nonmagnetic center separator is brazed into place. The individual pieces are subsequently fused into an



FIGURE 10
BCD COMPRESSOR AND TURBINE WHEELS



RICE TYPE ALTERNATOR

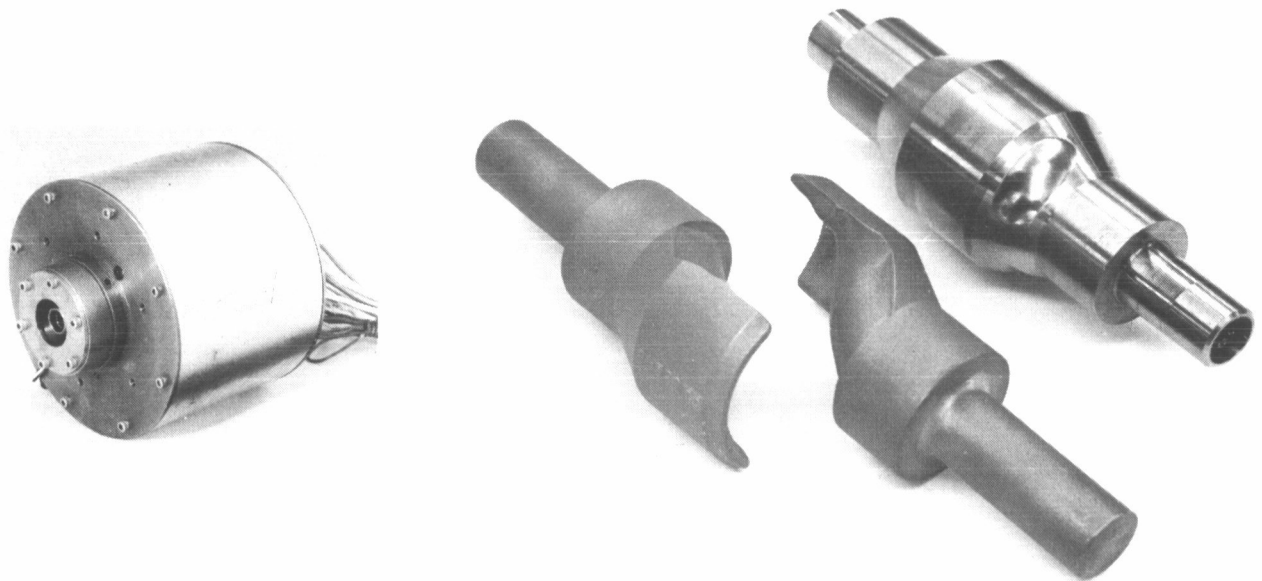


FIGURE 11

CROSS-SECTION OF BCD RICE ALTERNATOR
SHOWING ROTOR CONSTRUCTION



integral rotor member. Following this process, hydrostatic gas manifolds and shaft stub ends for the turbine and compressor are welded on and the assembly is then final-machined. The final rotor assembly is shown in Figure 12. This rotor is very rugged and has less windage loss than a salient pole rotor.

2.2.1.5 Gas Bearings

The foil gas bearings that support the CRU rotating group resulted from development work that has been done and is being continued by the AiResearch Manufacturing Company of Arizona. Both thrust and journal bearings are shown in Figure 6.

2.2.1.6 Heat Exchangers

The heat exchangers used in the BCD tests include a heater, cooler, and recuperator. These heat exchangers were either copies of production units or were fabricated from existing core designs.

For the BCD, the three heat exchangers are all mounted in close proximity on a subassembly. The heat exchangers are shown disassembled in Figure 13. The assembled subassembly with ducting is shown in Figure 14. The complete BCD, including heat exchangers, is shown in Figure 15. The BCD installed in the test cell ready for test is shown in Figure 16.

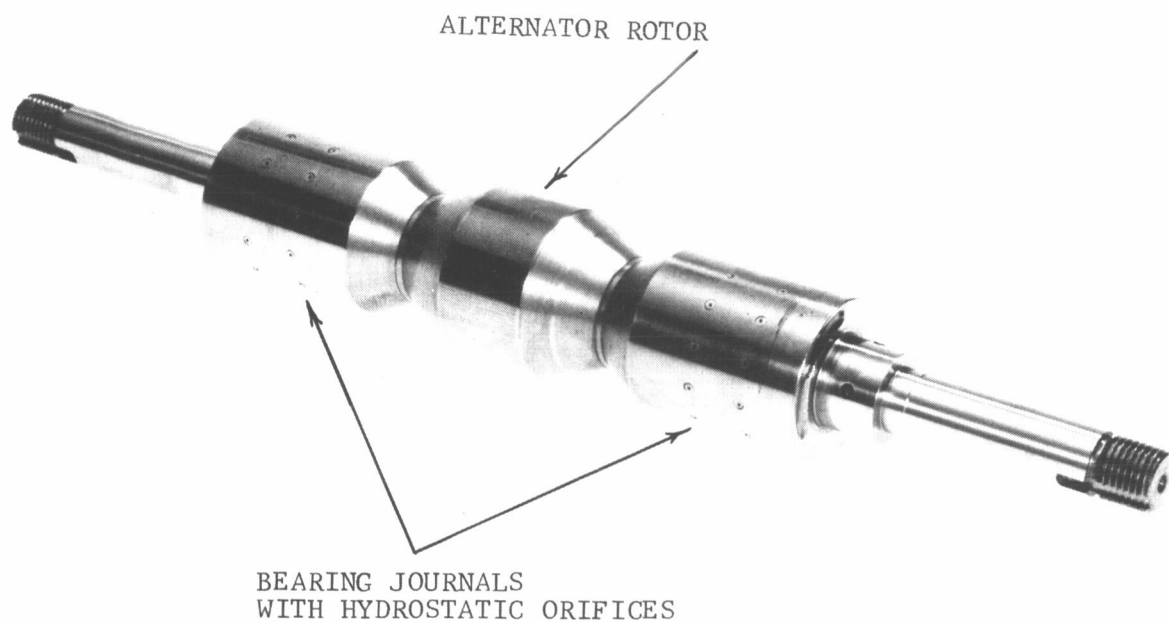


FIGURE 12

BCD TEST, PHOENIX
RICE ALTERNATOR SHAFT

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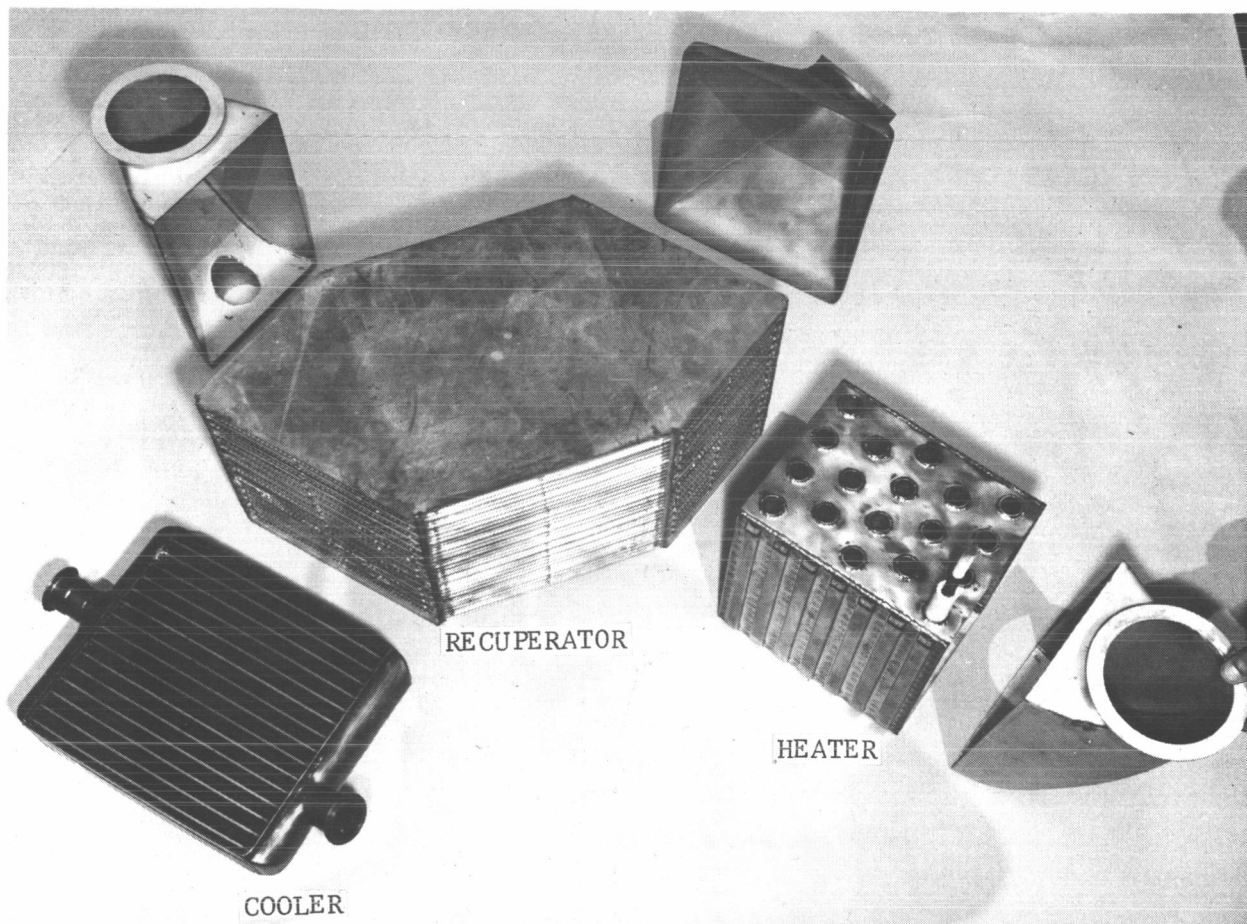


FIGURE 13

BCD HEAT EXCHANGERS (DISASSEMBLED)

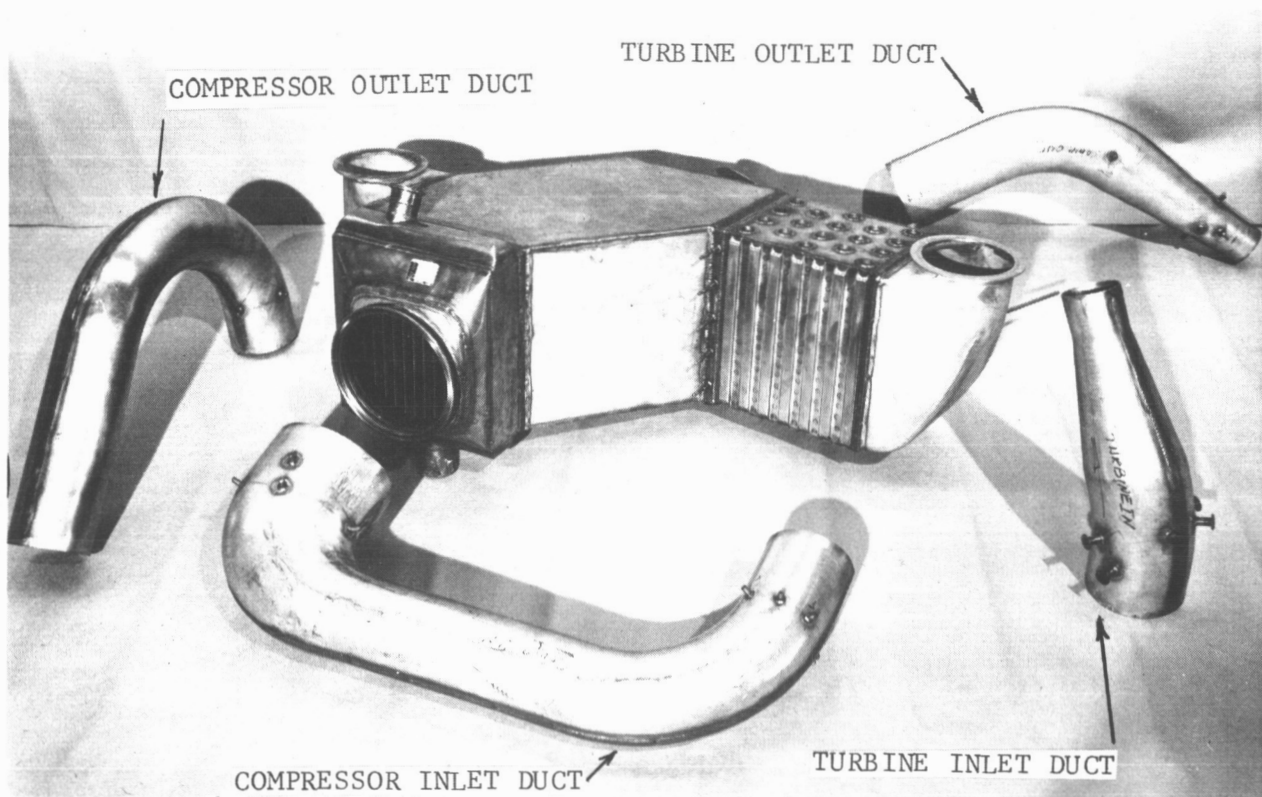
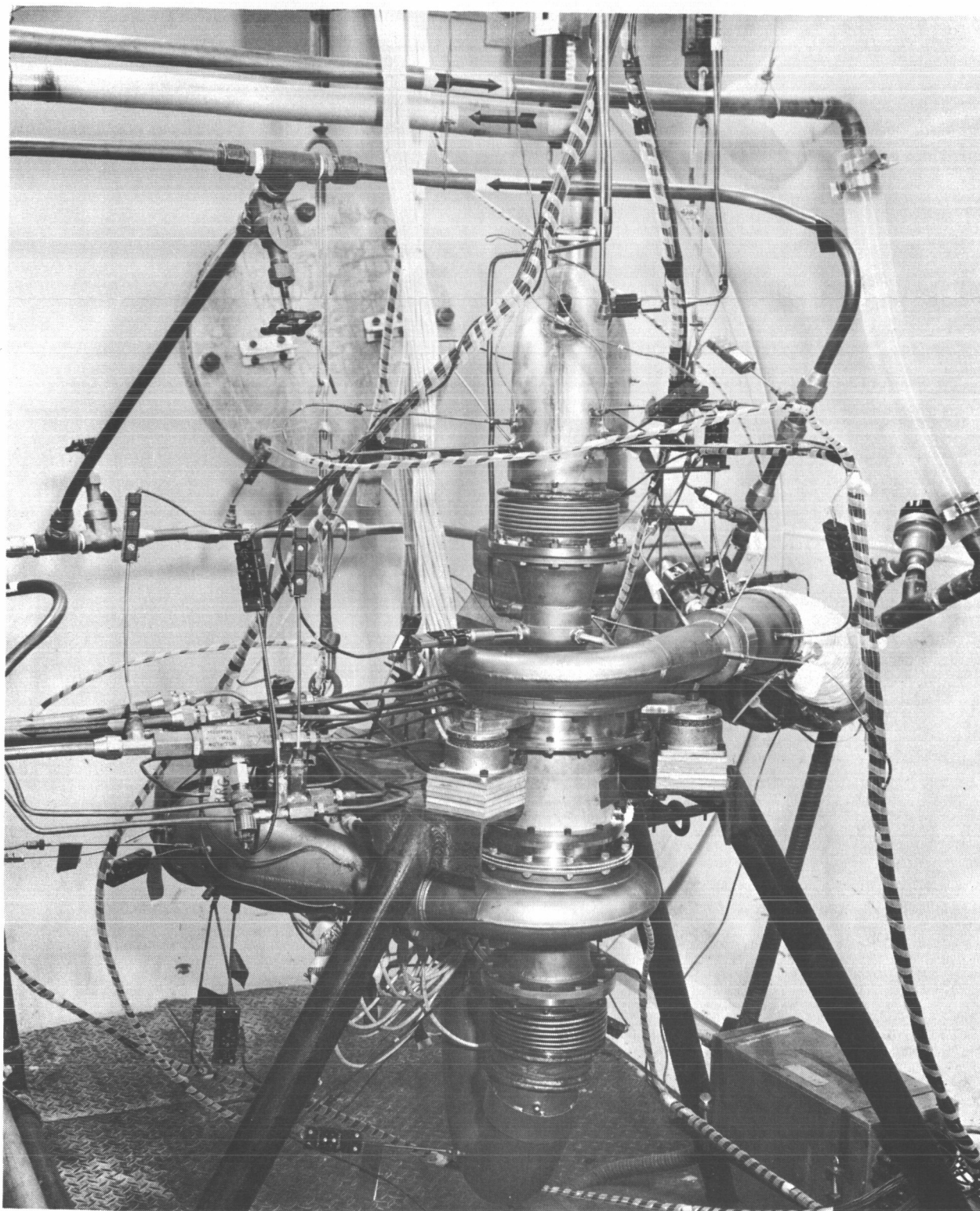


FIGURE 14

BCD HEAT EXCHANGER SUBASSEMBLY



BCD TEST, PHOENIX
FOIL BEARING CRU INSTALLED
IN CLOSED LOOP

PHOTO NO. P-29093-2

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FIGURE 15



COMPLETE BCD WITH HEAT EXCHANGERS

MP-15542

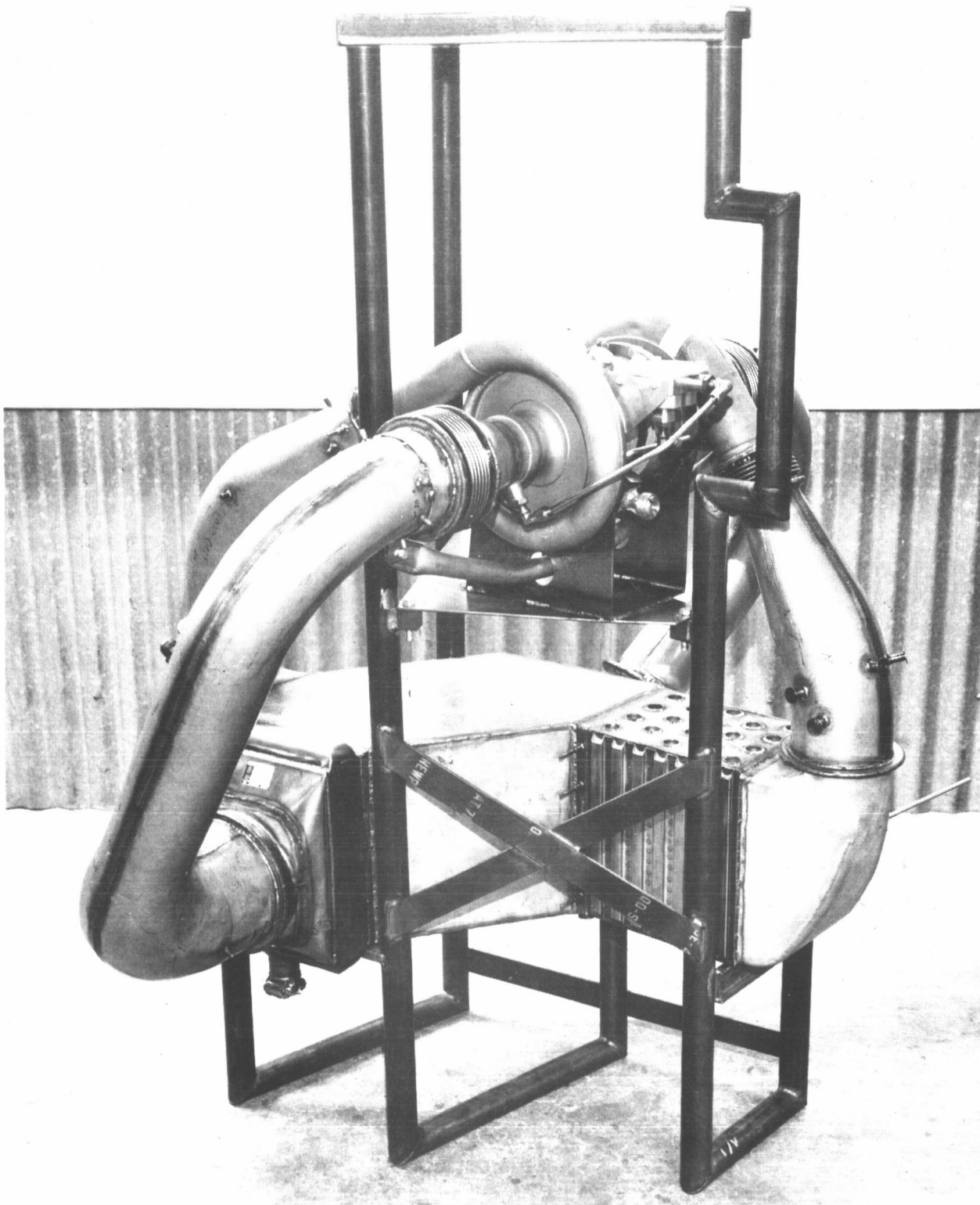


FIGURE 16



2.2.1.6.1 Heat-Source Heat Exchanger

The heat-source heat exchanger is of the single-pass, straight-through, finned-tube configuration. It has been designed to transfer the radiant heat from a series of electric heating elements to the argon working fluid of the Brayton-cycle system. The electric heating elements are installed inside metal finned tubes. As the argon flows across the metal tubes, heat is transferred from the tubes to the argon fluid by forced convection. Heat from the heating elements is transferred to the finned tubes by radiation.

Eighteen elements are installed in the heat exchanger, one in each of 18 finned tubes. Fins are brazed on the outside of the Hastelloy X tubes to increase the heat-transfer area exposed to the argon stream by a factor of 3 -- that is, the wetted area for the argon is 3 times greater than that of a plain tube of the same diameter. The 18 tubes are installed in a rectangular duct and are staggered, with triangular pitch.

The detailed design geometry of the heat exchanger is shown below:

<u>Core Dimensions</u>	<u>Inches</u>
Overall length	9.7
Overall width	7.7
Overall height	10.3
Tube length	10.0
Tube inside diameter	0.875
Fin outside diameter	1.38
Fin thickness	0.082
Fin stack	15 per inch
Transverse spacing	1.75
Longitudinal spacing	1.75



2.2.1.6.2 Heat-Sink Heat Exchanger

The heat-sink heat exchanger is of the single-pass, cross-flow, plate-and-fin configuration. The core of this heat exchanger is fabricated from stainless steel and nickel. Only the argon-side fins are nickel, the remainder of the core being steel. High-conductivity nickel is not required for fins used on the liquid side because the heat transfer coefficient is large.

The dimensions and core geometry of this unit are tabulated below.

<u>Core Dimensions</u>	<u>Inches</u>
Argon flow length	1.81
Coolant pass	9.57
<u>Liquid Side</u>	
Fin thickness	0.0078
Fin height	0.101
Fin stack	26
<u>Argon Side</u>	
Fin thickness	0.0078
Fin height	0.251

2.2.1.6.3 Recuperator

The recuperator is mounted between the compressor discharge and the turbine inlet. This unit (Figure 17) is a simple, single-pass, counterflow plate-and-fin exchanger fabricated by utilizing an existing fin design. The argon-to-argon recuperator is designed for an effectiveness of 0.90.



The core assembly is constructed of stainless steel and nickel. Of the three basic components of the core, the plates and strips are steel and the fins are nickel. Nickel is used for the fins because of its higher thermal conductivity, which increases the fin effectiveness of the core and, thus, reduces the core size and weight. The fins used on both sides of the core are 0.313 inch high and spaced at 10 fins per inch.

A nickel-based alloy is used for brazing cores of this type. It provides oxidation resistance and strength equal to or better than the parent metal. The core geometry is as follows:

<u>Core Dimensions</u>	<u>Inches</u>
Fin thickness	0.016
Fin height	0.414
Fin stack	10
Core width	10.4
Core length	10.1
Core height	11.3

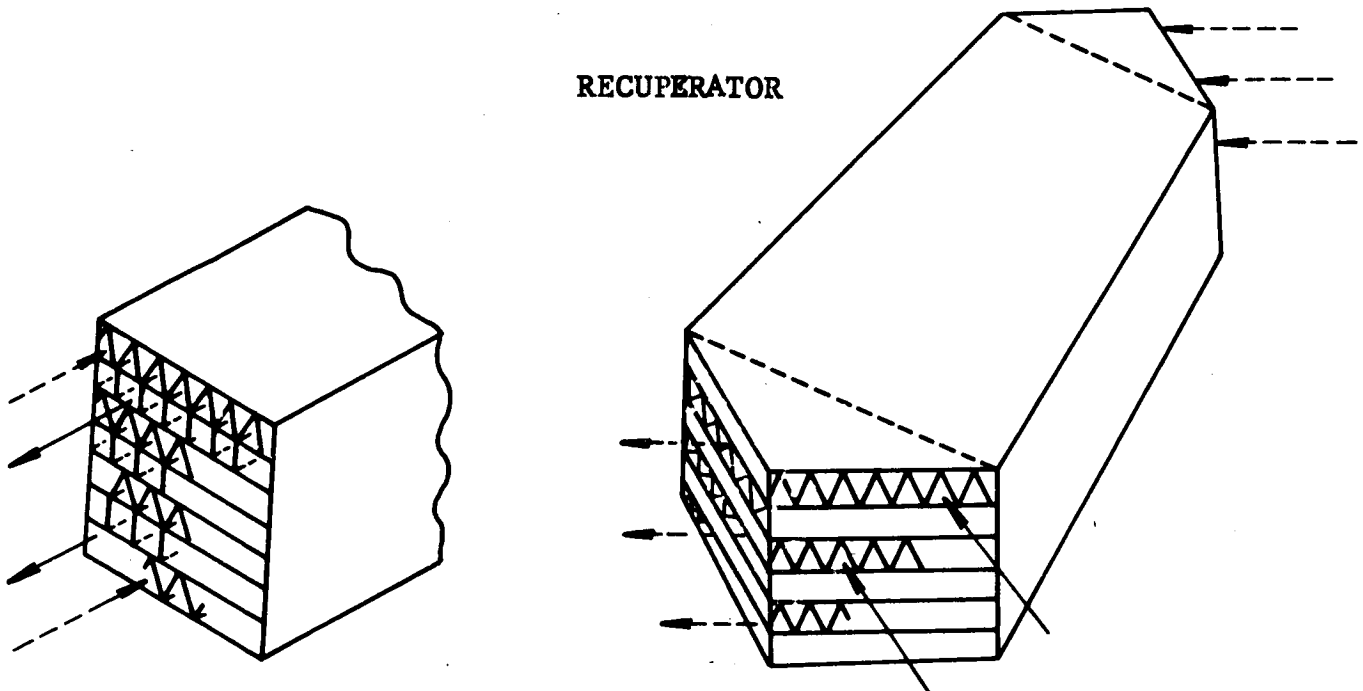


FIGURE 17



2.2.2 Loop Physical Data

The weight and volume breakdown for the BCD is as follows:

	<u>Weight, lbs</u>	<u>Volume, cu ft</u>
Recuperator	95	1.3
Heater	38	0.280
Cooler	13	0.060
CRU	46	0.040
Ducts	<u>28</u>	<u>0.928</u>
Total	220	2.608

The weights were determined by actually weighing the individual components. The volumes were determined by calculation.



3.0 TEST FACILITY AND SETUP

3.1 Phoenix Test Facility

The BCD was tested at the Contractor's facility in Test Cell CA-2, a small altitude chamber with pressure-control capabilities.

This altitude chamber was chosen for several reasons. Adequate electrical power is available for the system heaters. The refrigeration supply required for the heat-sink heat exchanger is connected to this chamber. In addition, the test cell can be evacuated, which results in reducing heat losses from the closed loop and minimizing working fluid contamination from air-inflow leakage. Figure 18 shows the BCD and test-cell layout.

3.2 Instrumentation and Recording

In order to facilitate accurate and rapid data acquisition, all temperatures and pressures were connected to a digital recorder shown in Figure 19. In addition, a Sanborn recorder was provided for use during start-up and shut-down, to monitor thermodynamic system parameters and rotating speed during transient conditions.

3.2.1 Instrumentation

The temperatures and pressures may all be read out manually or automatically. All temperatures are tied into a push-button selector panel and read out on a single meter. All pressures are similarly fed into a single calibrated transducer through a rotary selector switch. This permits "push-button" selection of any specific pressure readout at the control panel by use of a digital voltmeter which monitors the pressure transducer signal.

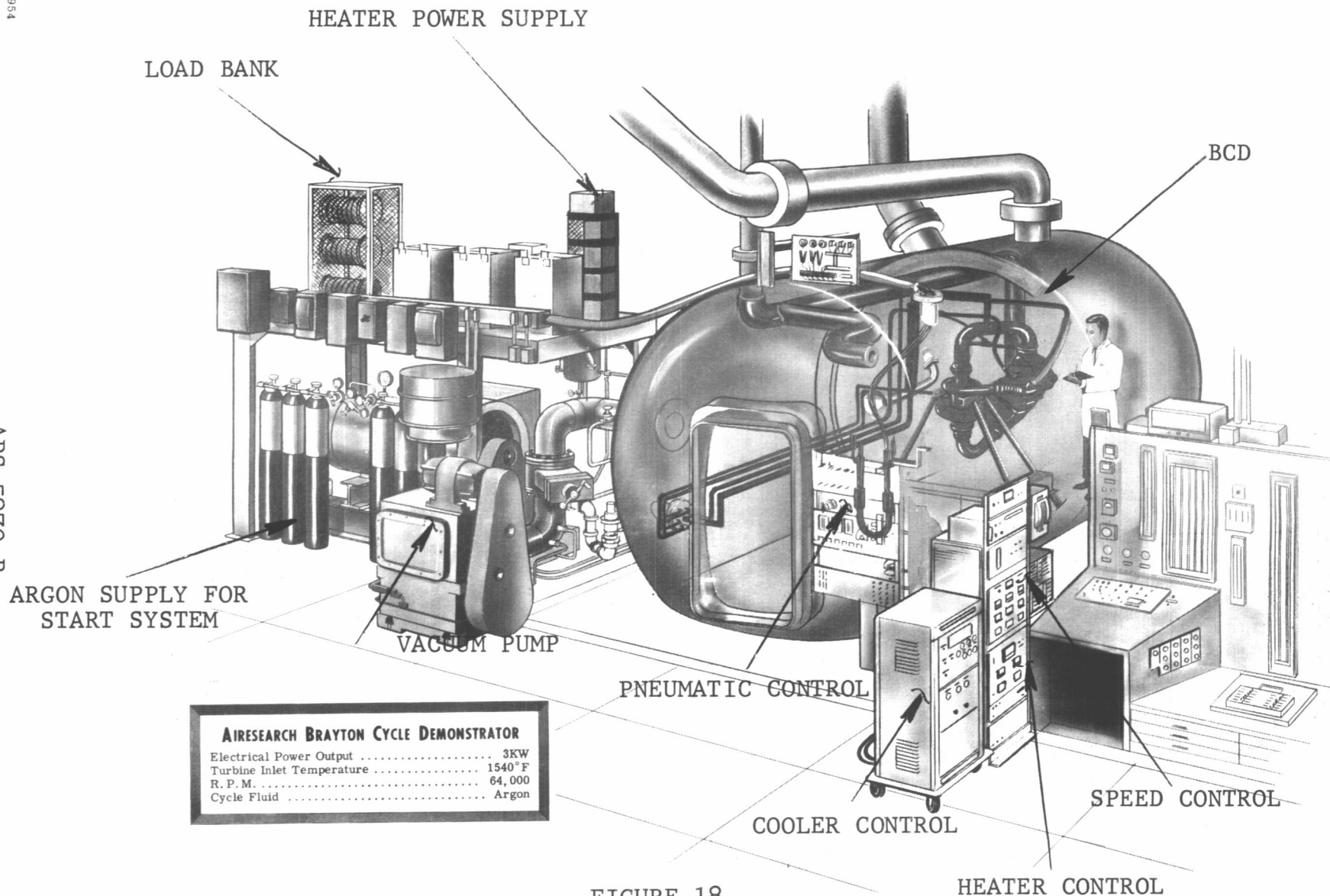


FIGURE 18
BRAYTON-CYCLE DEMONSTRATOR TEST SETUP



FIGURE 19

BCD TEST, PHOENIX
DIGITAL DATA ACQUISITION SYSTEM

APS-5270-R
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PHOTO NO. P-29637-1
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Continuous recording of selected temperatures, pressures, speed, etc., is accomplished with use of a Sanborn recorder. Figure 20 shows the unit in the cell, completely instrumented, with typical instrumentation labeled.

Figure 21 shows the general locations for the loop instrumentation. On the pages following is a list of the instrumentation used.

INSTRUMENTATION REQUIREMENTS

<u>CRU REQUIREMENTS</u>	<u>QUANTITY</u>	<u>NOTES</u>
<u>Compressor:</u>		
Inlet total temperature	3	Compressor inlet conditions Station (1)
Inlet total pressure	3	
Inlet static pressure	3	
Inlet converging section for flow measurement, static pressure taps	2	Argon flow measurement
Outlet total temperature	3	Compressor outlet conditions Station (2)
Outlet total pressure	3	
Outlet static pressure	3	
<u>Turbine:</u>		
Inlet total temperature	3	Turbine inlet conditions Station (6)
Inlet total pressure	3	
Inlet static pressure	3	
Outlet total temperature	3	Turbine outlet conditions Station (9)
Outlet total pressure	3	
Outlet static pressure	3	

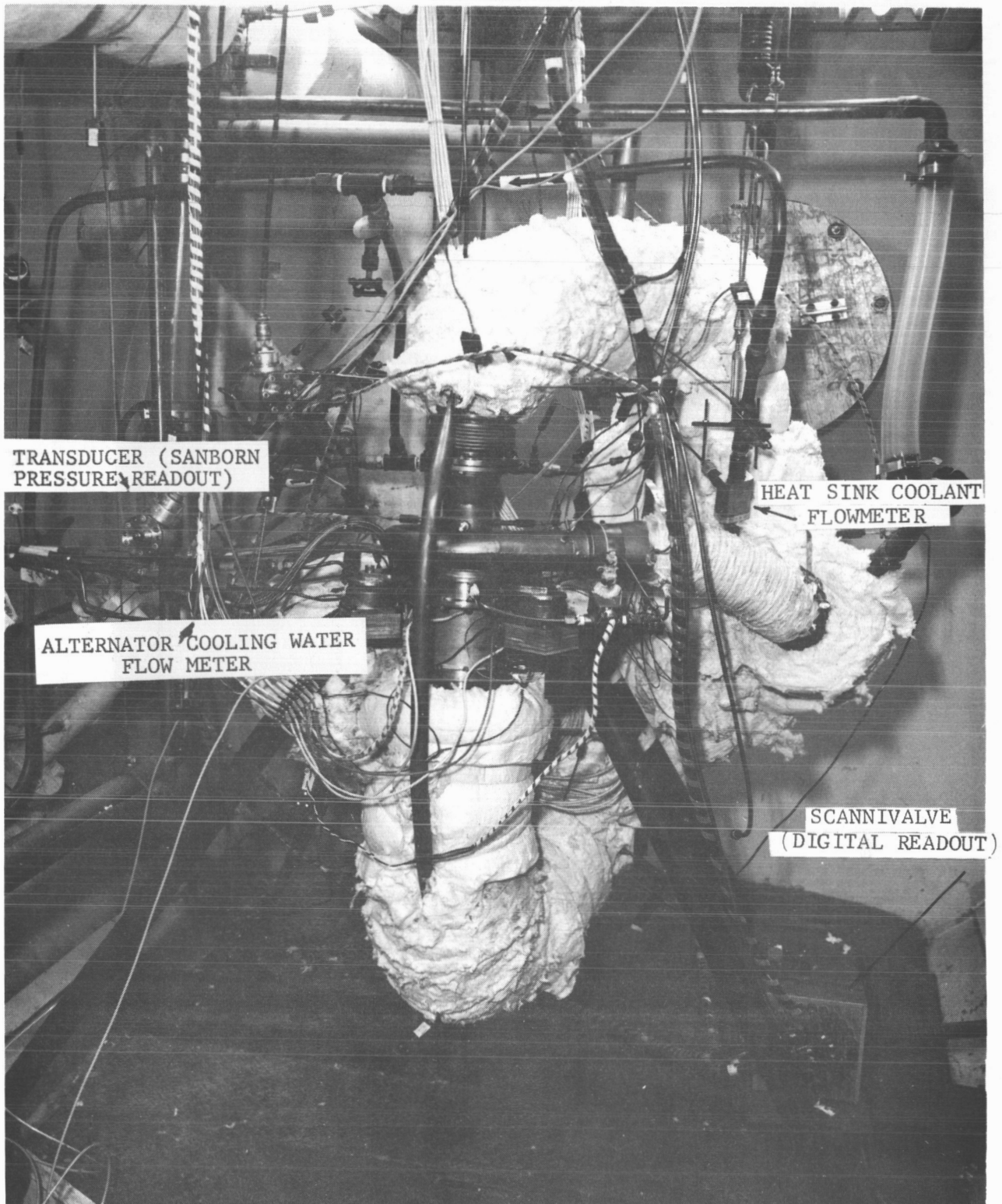


FIGURE 20

INSULATED CLOSED LOOP
BRAYTON CYCLE DEMONSTRATOR

PHOTO NO. P-29975

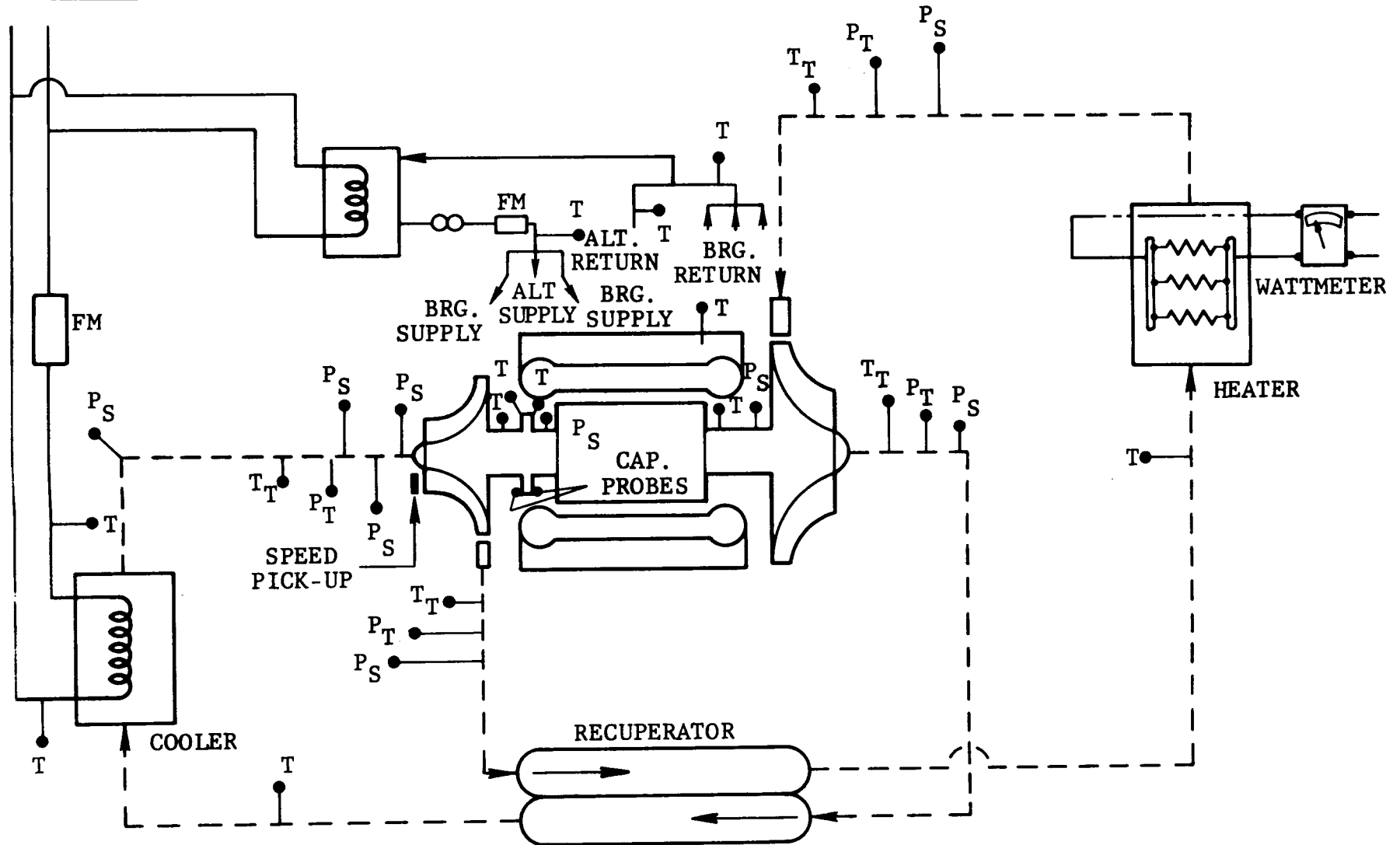
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LOCATION OF BCD LOOP INSTRUMENTATION

FIGURE 21



INSTRUMENTATION REQUIREMENTS (Contd.)

	<u>QUANTITY</u>	<u>NOTES</u>
<u>Alternator:</u>		
(Refer to loop requirements)		
(Speed-control design will include frequency signal for overspeed switch trip)		
<u>Shaft:</u>		
Speed pickup	1	Primary signal for speed control
<u>Bearings:</u>		
Journal bearing temperature	2	Bearing carrier temperature
Journal bearing static pressures	2	Bearing cavity pressure
Thrust bearing temperatures	3	Thrust plate temperature
Thrust bearing static pressure	1	Cavity pressure
Thrust bearing capacitance probe	1	Axial movement measurement
Shaft position capacitance probes	2	Shaft orbit
<u>LOOP REQUIREMENTS</u>		
<u>Electrical Output:</u>		
(Alternator out)		
A-C ammeter	1	Phase current measurement
Line-to-neutral voltmeter	1	Phase voltage measurement
Line-to-line voltmeter	1	
Wattmeter	1	Total alternator output power



INSTRUMENTATION REQUIREMENTS (Contd.)

	<u>QUANTITY</u>	<u>NOTES</u>
<u>Heat Source:</u>		
<u>Heater power input:</u>		
A-C ammeter	1	
A-C voltmeter	1	
Heater inlet gas temperature	3	
Heater inlet gas static pressure	1	
Heater skin temperature	2	
<u>Heat Sink:</u>		
Cooler inlet gas temperature	3	
Liquid coolant inlet temperature	2	
Liquid coolant inlet pressure	1	
Liquid coolant outlet temperature	2	
Liquid coolant outlet pressure	1	
Liquid coolant flow rate	1	
<u>Bearing and Alternator Liquid Coolant Flows and Temperatures</u>		
Inlet flow temperature	2	
Inlet flow pressure	1	
Alternator outlet flow temperature	2	
Bearing outlet flow temperature	2	
Total pump flow rate	1	
Alternator outlet flow rate	1	
Coolant pump discharge pressure	1	
Coolant reservoir temperature	1	



INSTRUMENTATION REQUIREMENTS (Contd.)

	<u>QUANTITY</u>	<u>NOTES</u>
<u>Other Loop Skin Temperatures</u>		
Recuperator skin temperature	9	Measure temperature gradients in recuperator
Turbine inlet duct upstream	2	Heat flow in duct
Turbine inlet duct downstream	2	
Compressor inlet duct	2	For determination of heat leak from compressor
<u>Pneumatic Control System</u>		
U-tube reference system pressure	1	
<u>Vacuum System</u>		
Vacuum-pump header pressure	1	
Vacuum tank pressure	1	Loop external ambient pressure
Bearing scavenging line pressure	2	Needed only during start-up and shut-down
<u>Argon Supply System</u>		
Regulated loop supply pressure	1	
Regulated bearing supply pressure	2	



3.2.2 Instrumentation Accuracies

The data presented below indicates the estimated error possible with the instrumentation used in this test.

A. Temperatures

<u>Station</u>	<u>Temp. Level</u>	<u>Accuracy</u>	
		(a)	(b)
(1) Compressor inlet	500°R	-1°F	±2°F
(5) Recuperator inlet (high pressure side)	800°R	-4°F	±4°F
(6) Recuperator outlet (high pressure side)	1500°R	-7°F	±4°F
(9) Recuperator inlet (low pressure side)	1600°R	-7°F	±4°F
(10) Recuperator outlet (low pressure side)	800°R	-4°F	±4°F

(a) Denotes reduced reading due to heat loss from the probe by radiation and conduction.

(b) Denotes accuracy range due to nonuniformity in the wire. (This is guaranteed value, which has been found to be quite conservative.)

At several cycle-state-point locations, severe temperature stratification exists, which makes determination of the mean bulk temperature difficult.

B. Pressures

Accuracy

Static pressure taps	±1/2 of 1 percent
Total pressure probes	±5° misorientation--
(1/8-inch impact tubes)	1 percent of the velocity head



C. Readout Equipment

The readout equipment inaccuracy is in addition to the probe inaccuracies stated above. This factor depends upon the readout equipment used. For the tests reported herein, Brown temperature recorders, manometers, Bourdon-type pressure gauges, a Sanborn recorder and a digital voltmeter recorder were used. The data reduction program employed for computing system efficiency used data acquired on the digital system. The following digital system readout accuracy is as follows:

- (a) Pressure readout accuracy, including the scanning valve is ± 0.21 psia.
- (b) Temperature readout accuracy, using the digital voltmeter is $\pm 1^{\circ}\text{F}$. For high temperature there is also an inaccuracy of $\pm 3/4$ of one percent of the lead wire temperature drop.

3.3 Controls

3.3.1 General

The philosophy upon which the control design was based involves assignment of all variations from a desired steady-state condition into one of two categories.



(a) First Category - Variations in loop conditions that remain within or are controlled within established, tolerable limits. These are either small, rapid changes which do not significantly upset the system; or slowly occurring, potentially large changes, which can be detected and easily checked at their outset. The change will be sensed and the proper control will be actuated to effect a correction (the control systems can be either automatically or manually operated).

(b) Second Category - Large, uncontrolled perturbations caused by gross environmental changes or equipment failure. A condition of this type will trigger an alarm system. If the system is being attended, actuation of an alarm requires the attending technician to follow the established emergency-shutdown procedure. When the loop is under automatic control, the alarm signal simultaneously actuates an emergency-shutdown control system.

A controlled shutdown includes turning off the heaters, evacuating the closed loop (thereby eliminating further power production by the turbine) and supplying gas to the bearings from an auxiliary system during turbine coast-down.

The total control system is comprised of five basic control functions (refer to Figures 22 and 23). Figure 24 shows the control system assembled on the Contractor's control panel.

- (1) Heater control
- (2) Cooler control
- (3) CRU speed control
- (4) Pneumatic control (Figure 25)
- (5) Overtemperature control

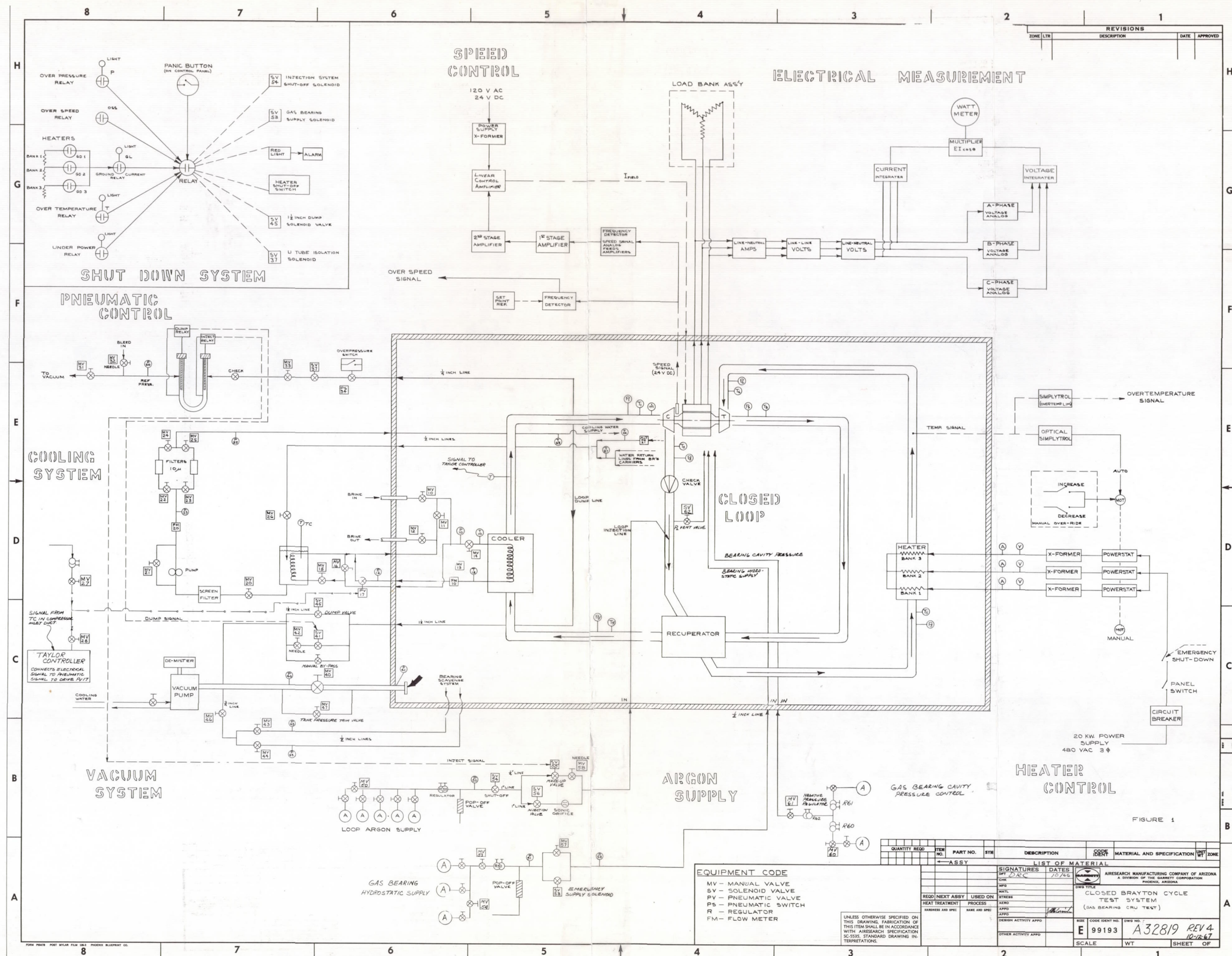
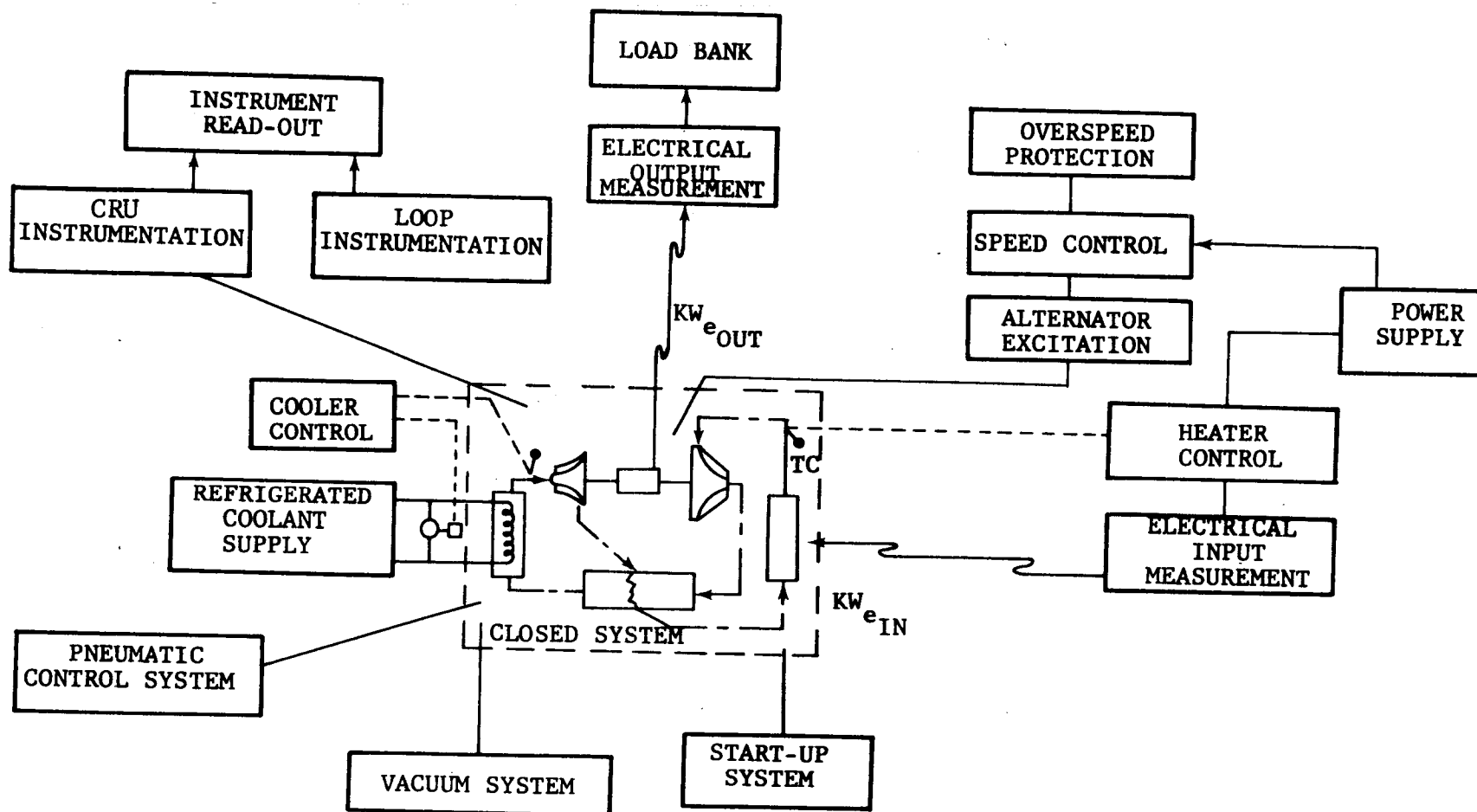


FIGURE 22
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CONTROL AND SUPPORT SYSTEMS FOR CLOSED LOOP GAS TURBINE POWER PLANT

FIGURE 23

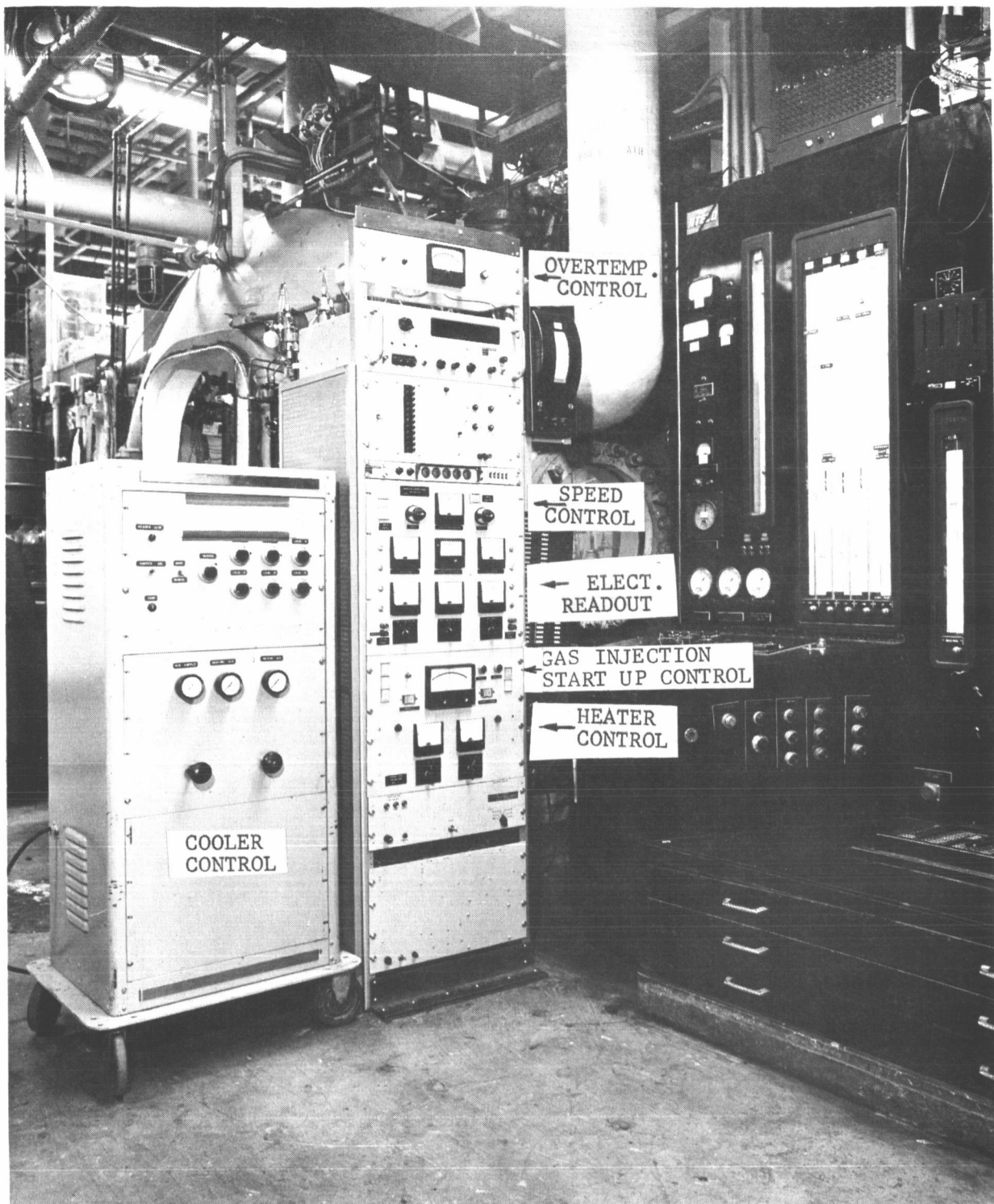


FIGURE 24

BCD TEST, PHOENIX
CONTROL SET UP

PHOTO NO. P-28277-4

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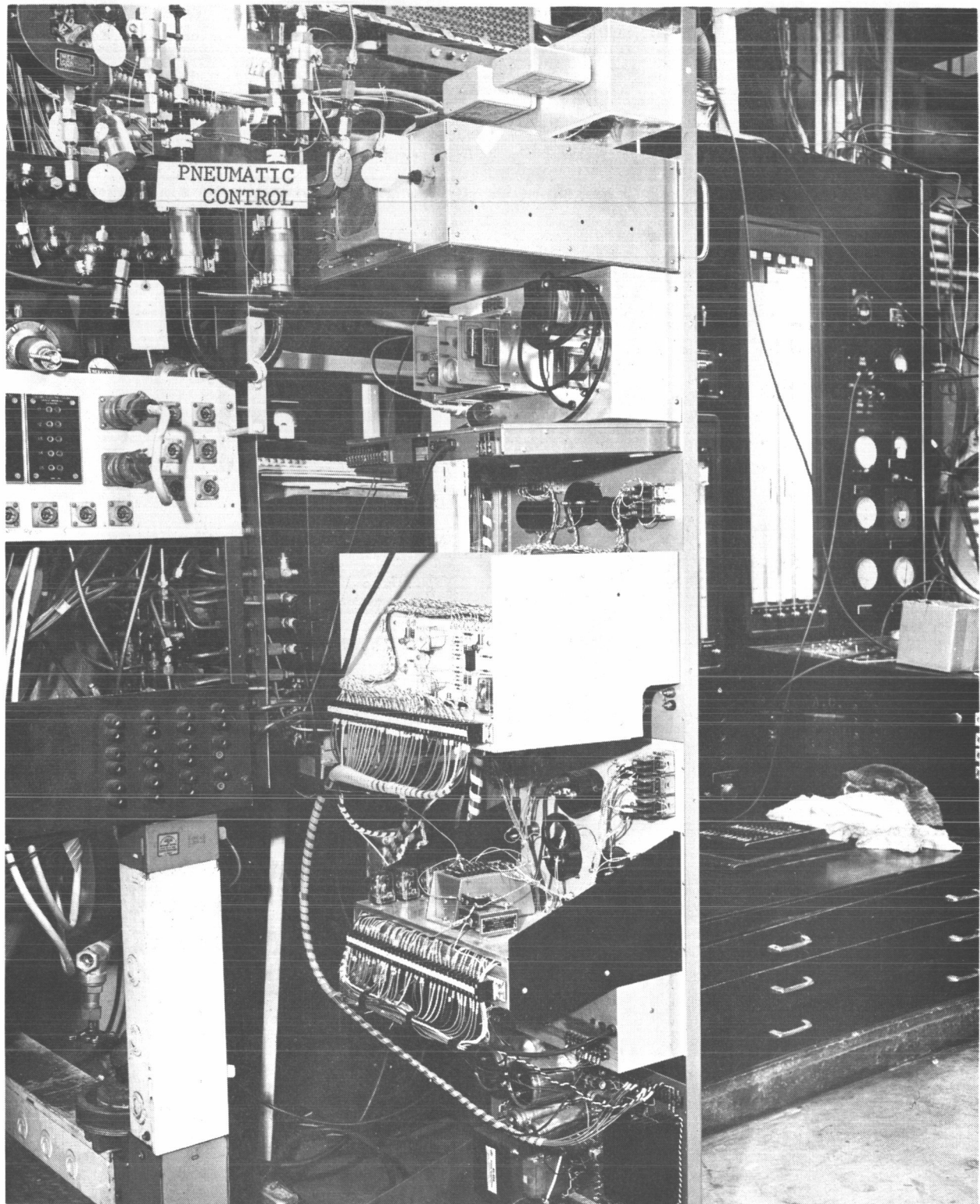


FIGURE 25

APS-5270-R
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BCD TEST, PHOENIX
REAR VIEW OF RELAY RACK
MOUNTED CONTROLS

PHOTO NO. P-28277-7

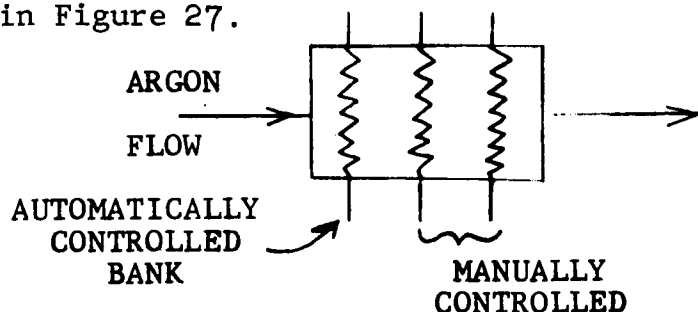
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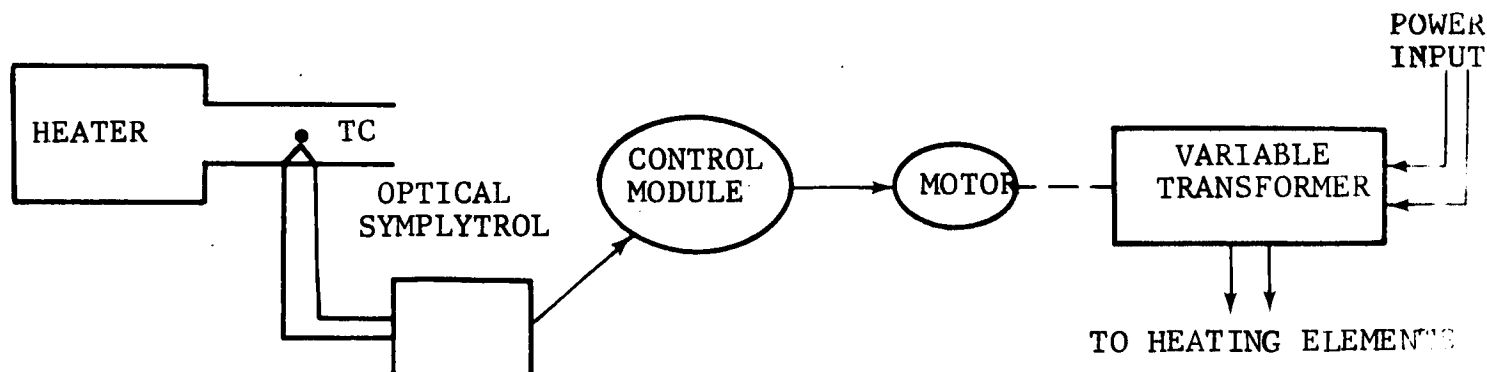


3.3.2 Heater Control

The primary function of the heater control is to maintain a constant turbine-inlet temperature. Control will be exercised over a fraction of the total heater input and is, therefore a temperature "trim control." The sketch below shows the arrangement of the three heater banks with respect to the heater. Figure 26 shows the heater transformers. The heater terminals are shown in Figure 27.



An error signal is generated by a thermocouple placed in the heater discharge flow. The signal is fed to an OPTICAL SYMPLYTROL, on which are established the high and low operating limits. A temperature condition outside of the established limits will result in actuation of a motor-driven variable transformer, which will increase or decrease the power input to the controlled heater bank. The control module, shown in the following sketch, incorporates a rate-of-change control timer and a fail-safe feature that causes heater shutdown if the control malfunctions or if the thermocouple burns out.



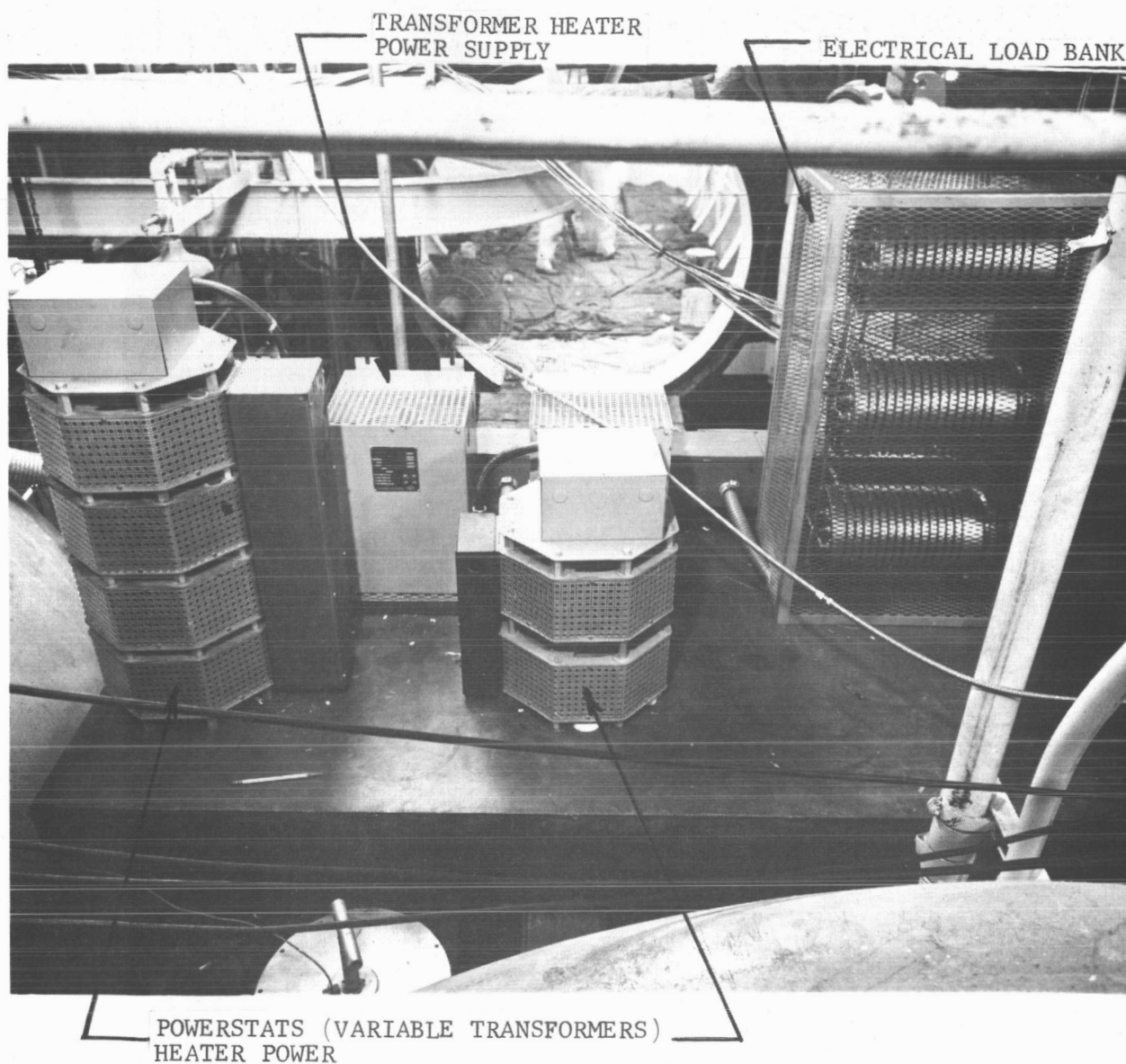


FIGURE 26

APS-5270-R
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BCD TEST, PHOENIX
HEATER POWER SUPPLY TRANSFORMERS
& POWER STATS, AND RESISTANCE LOAD BANKS

PHOTO NO. P-28277-12



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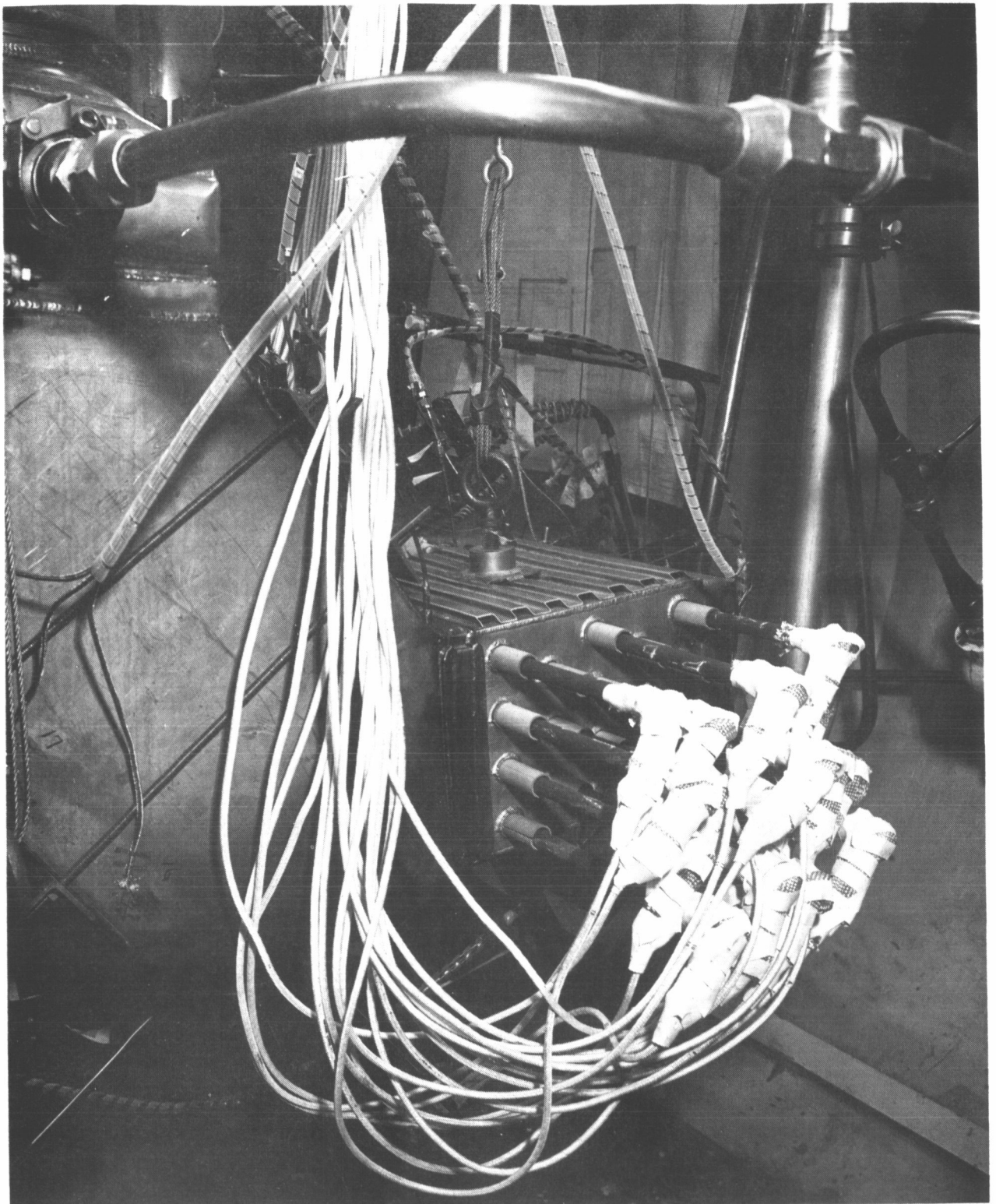


FIGURE 27

BCD TEST, PHOENIX
REAR VIEW SHOWING HEATER
GLOBAL CONNECTIONS

PHOTO NO. P-28277-3

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3.3.3 Cooler Control

The primary function of the cooler control is to maintain a constant compressor inlet temperature. Compressor inlet flow temperature is monitored and an error signal will result in activation of the cooler bypass valve.

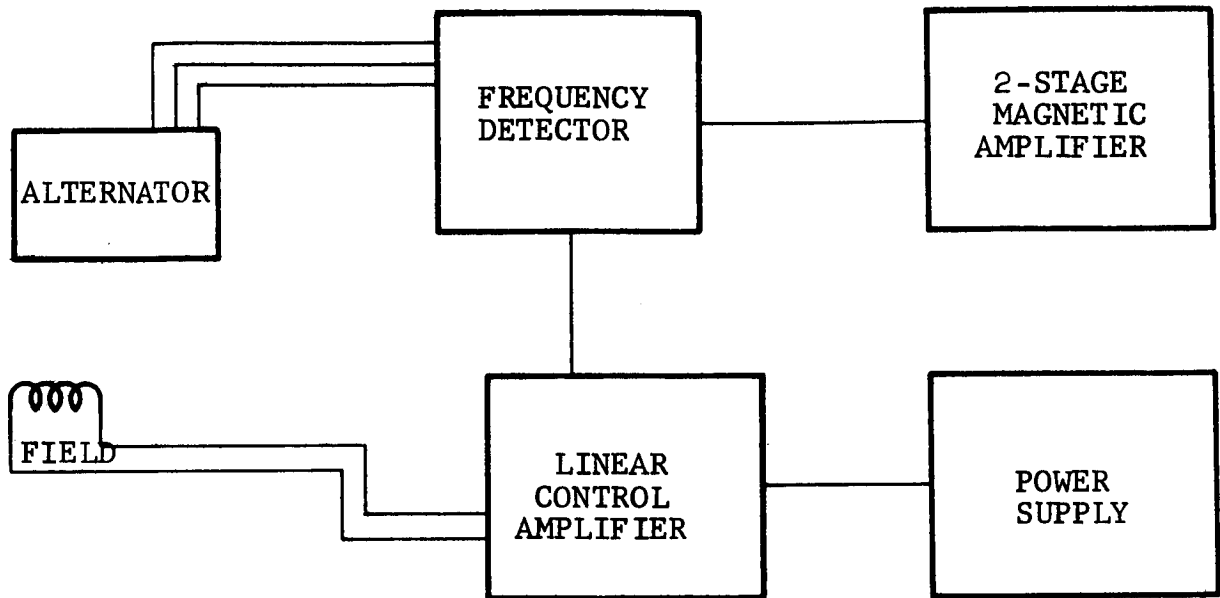
The electrical-pneumatic controller and a pneumatically actuated valve used for modulating the coolant liquid flow are the primary elements of the cooler control systems. Both the heater and the cooler are manually controlled during start-up and during establishment of design steady-state conditions. Hence, during operating periods when large perturbations are likely to occur in the thermodynamic system, the heat sink and heat source are under complete manual control. Once steady-state conditions have been established, automatic controls can be employed to handle small fluctuations in loop conditions.

3.3.4 Speed Control

The two methods of high-speed turbomachinery speed control that can be employed in this application are: (1) parasitic load control and (2) control of the generator output by variation of the generator field excitation. The latter is used in the BCD. Both methods cause a change in generator load to maintain a zero accelerating torque on the rotating group. Rotating speed is monitored using Bentley pickups at the compressor end of the shaft.



The speed-control module is shown in Figures 28 and 29. Speed control can be effected by either manual or automatic modes. The operation is shown schematically below.



As speed changes, the frequency of the alternator output changes. This output frequency is compared with a set reference frequency in the detector circuit. The error signal is amplified and fed to the linear control amplifier that controls the alternator field excitation level. The details of operation of this system are included in the Test Instructions and Operator's Manual that has been prepared by the Contractor.

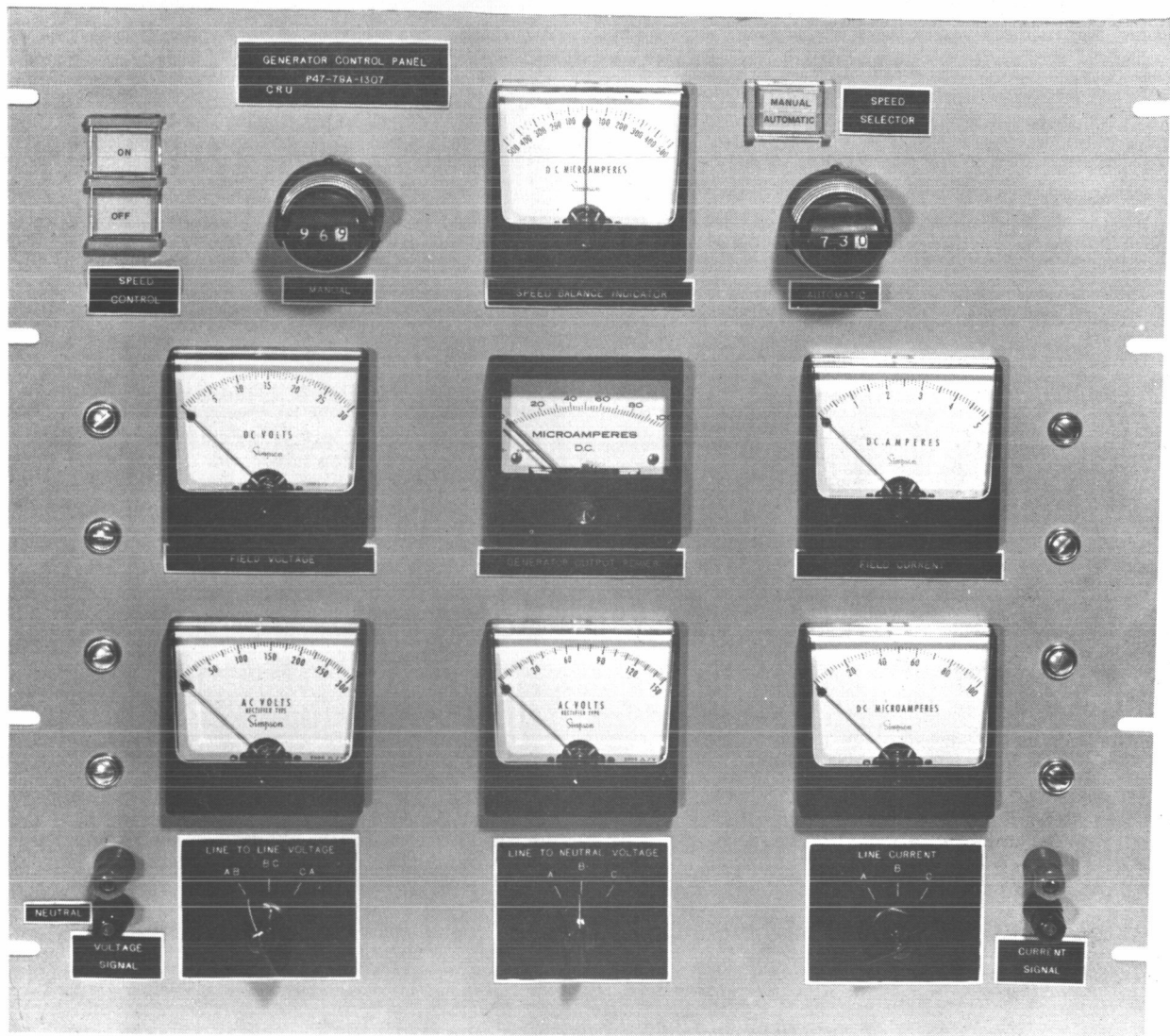


FIGURE 28



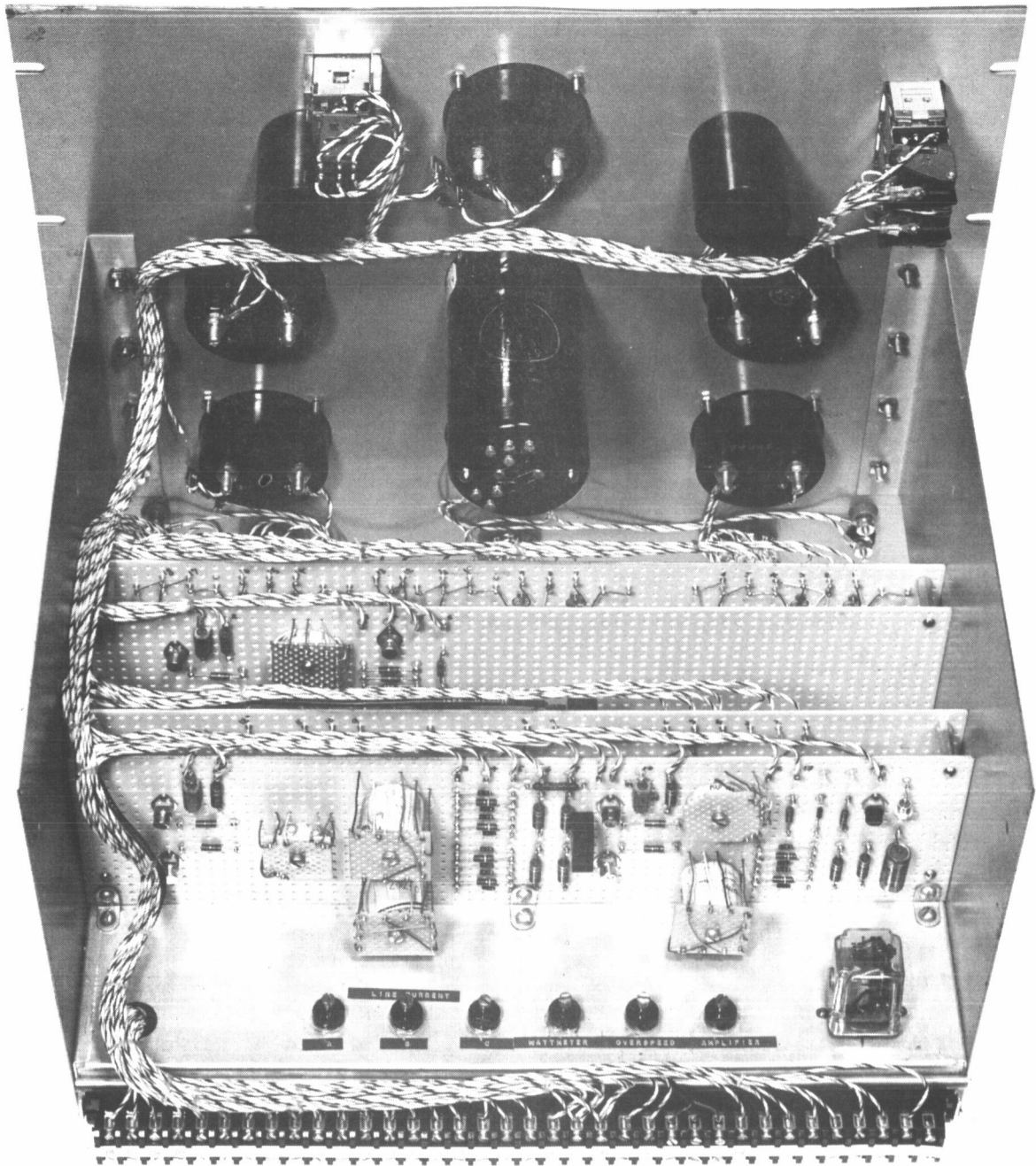


FIGURE 29

GENERATOR READOUT AND SPEED
CONTROL SYSTEM
BRAYTON CYCLE DEMONSTRATOR UNIT

PHOTO NO. P-27939-2

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3.3.5 Emergency Shutdown System

The design philosophy requires that any uncontrollable condition change in the loop must ultimately result in initiation of a safely controlled shutdown of the CRU. Hence, failures of such items as supporting equipment, loop components, or even facility power must cause actuation of the emergency shutdown system.

Many of the possible failures will result in an uncontrolled speed increase. Examples are: loss of alternator load, speed-control failure, heater overtemperature condition, loop overpressure, and the like. If events such as these occur, a speed signal from any one of several parallel speed sensors will, at a preset speed, trip an overspeed switch, which initiates the simultaneous occurrence of the following events:

- (a) Triggering of an external alarm.
- (b) Opening of the dump valve, to evacuate the closed system.
- (c) Closing of the shutoff solenoid in the pressurization system.
- (d) Turning off of the heater power.
- (e) Providing a supply of gas to the bearings.

The emergency shutdown logic can be seen in the upper left corner of Figure 22.



3.4 Start-up

Start-up is accomplished by first applying hydrostatic gas to the bearings and then spinning the rotating group by injecting pressurized gas into the turbine. A check valve located at the compressor discharge prohibits reverse flow; hence, the gas entering the loop at a point between the compressor and the recuperator must pass through the recuperator and the heater before it enters the turbine. The heated, pressurized gas expanding through the turbine brings the rotating group to self-sustaining speed very quickly. During initial spin-up to self-sustaining speed there is no load on the alternator and very little compressor work, since the compressor cavity is at a low pressure. While gas injection is in process, the dump valve is held open manually. When self-sustaining speed has been reached, the injection system and the dump system are simultaneously closed. At this time the pneumatic control takes over, and compressor inlet pressure is regulated to approximately 4.5 psia.

During the initial spin up it was stated that the bearings are operating hydrostatically (externally pressurized). During acceleration of the BCD, a changeover must be made from hydrostatic to hydrodynamic operation. This procedure is explained in detail in the "Test Instruction," Report SY-6016-R, that has been prepared by the Contractor for the NASA.

The remainder of the start-up sequence involves increasing turbine inlet temperature to achieve design speed and then bringing the CRU up to full electrical power output. The cooler, heater, and alternator output is controlled manually until design steady-state conditions have been reached. At this point total control can be accomplished automatically.



3.5 Support Systems

The support system for the BCD test setup was designed with the objective of providing an adequate means of controlling the environment of the BCD during test. The elements of this system include the argon supply system, vacuum system, pneumatic control system, and alternator coolant system. Gas, trichloroethylene coolant, and water lines enter and exit from the cell shown in Figure 30, by hermetic feed-through in the tank wall.

3.5.1 Argon Supply System

The argon supply system, shown in Figure 31, consists of several argon bottles, manifolded together, used for argon-gas-injection starts. Bottles for supply of the gas bearings and the control hardware are shown in Figures 31 and 32.

The gas-injection supply consists of two parallel systems--one line for gas-injection starting and one line for pneumatic control. The gas-injection start system consists of a sonic orifice, a solenoid valve, and a pressure regulator. The pneumatic control system consists of an adjustable needle valve, and a solenoid valve actuated by the pneumatic control system to increase the loop pressure level when this level falls below the operating set point.



FIGURE 30

BCD TEST, PHOENIX
VACUUM TANK BULKHEAD
SUPPORT LOOP EXITS

PHOTO NO. P-28277-2





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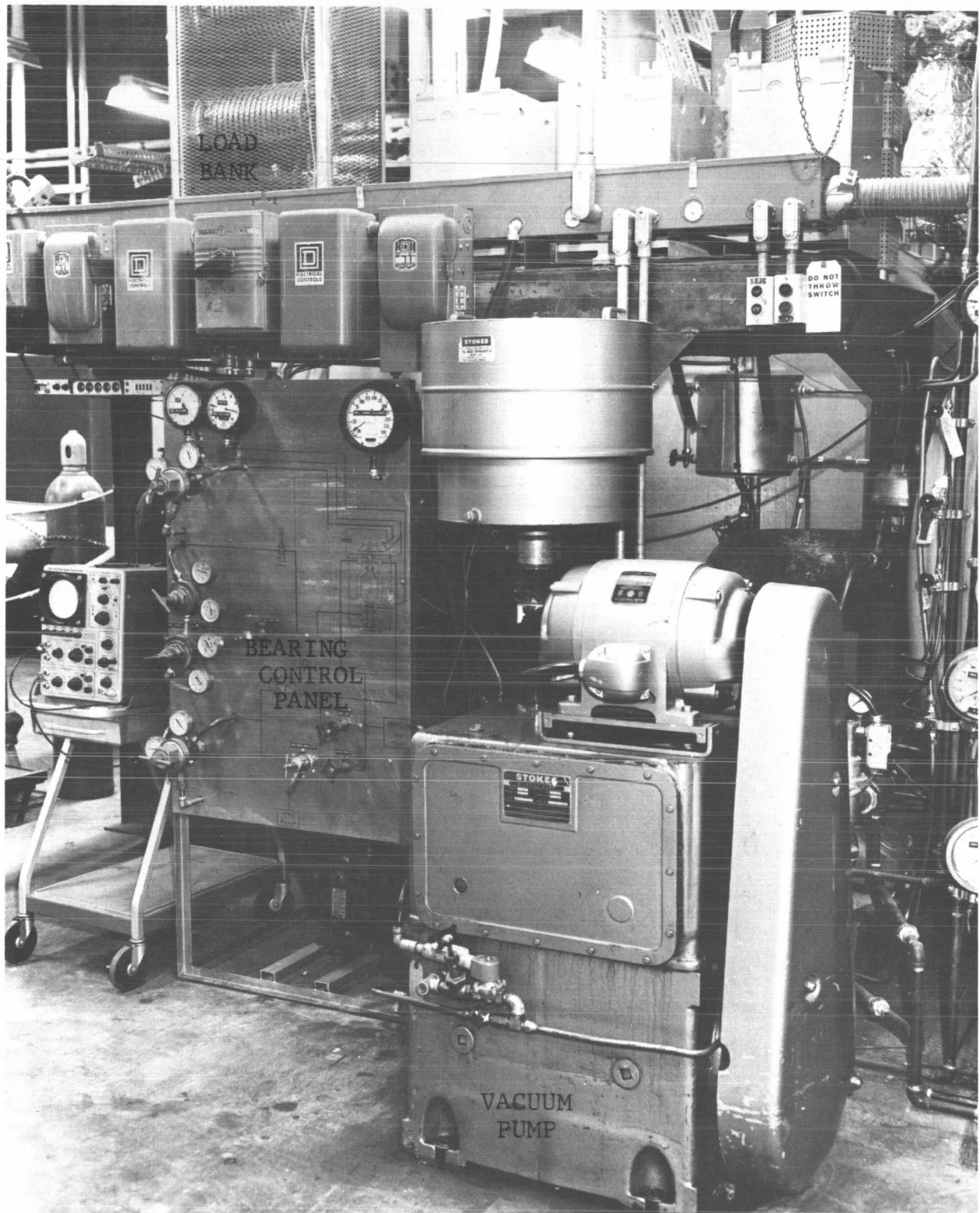


FIGURE 31

A VIEW OF THE ARGON SUPPLY SYSTEM
AND OTHER BCD SUPPORT SYSTEMS
ADJACENT TO THE TEST CELL

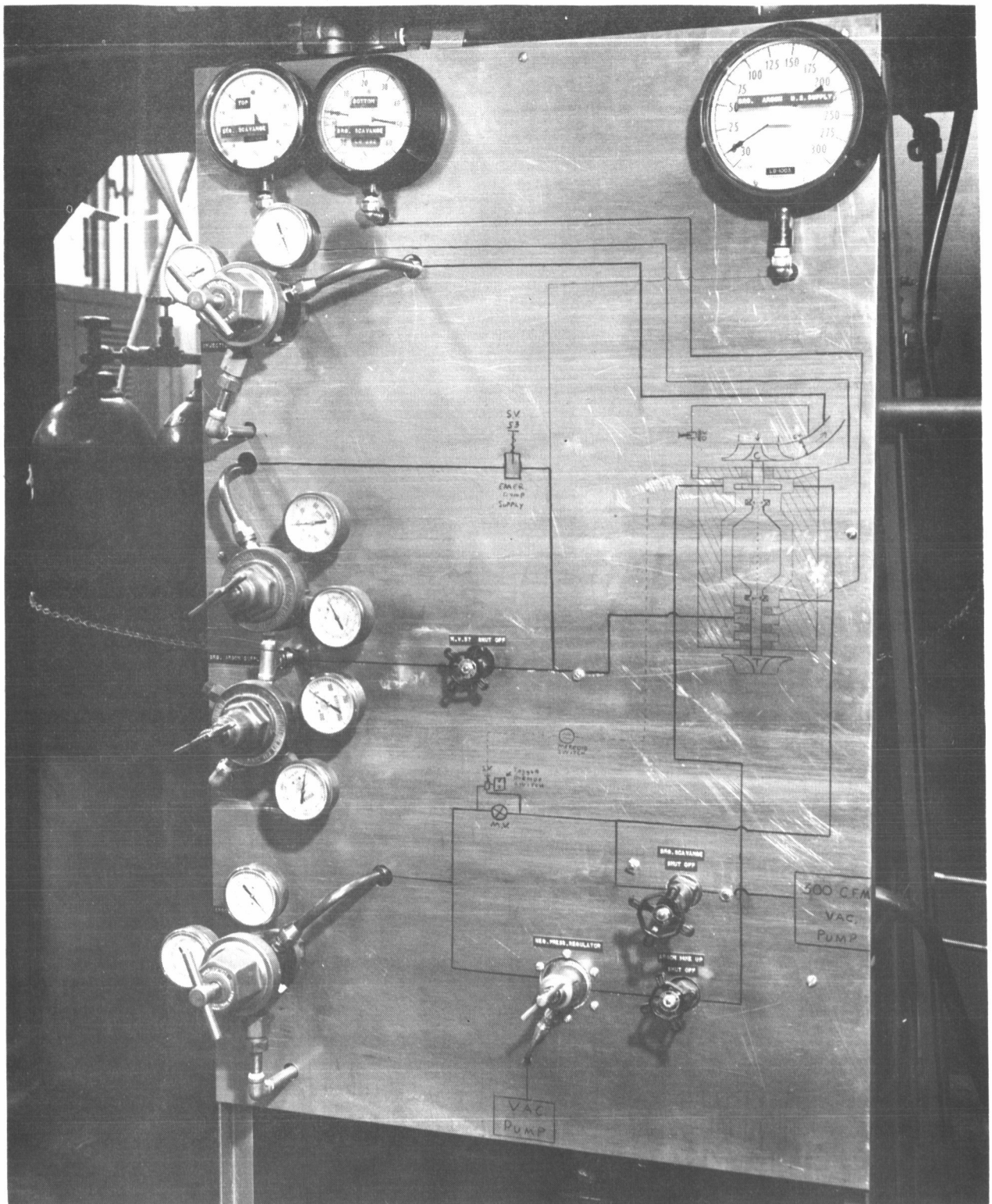


FIGURE 32

BCD ENVIRONMENTAL CONTROL SYSTEM
OPERATING PANEL



3.5.2 Bearing Supply System

The bearing supply system consists of a means to (a) provide hydrostatic gas to the bearings during start-up and shutdown and (b) ensure that the ambient pressure in the bearing cavities is maintained above a safe minimum level (approximately 9 - 10 psia). The bearings were designed for an ambient operating pressure of 10 psia.

In the hydrostatic supply system, argon is supplied at a regulated pressure through a manual control valve. In the event of an emergency, hydrostatic gas can be supplied instantly by opening the solenoid valve.

The bearing cavity pressure-control system provides gas through an absolute pressure regulator at any time the pressure level in the bearing cavities falls below the regulator set point. This system is especially useful when experiments are being performed at off-design conditions.

3.5.3 Vacuum System

In parallel with the primary control valve (MV40) is a smaller bypass valve, which is used for trim control of the tank pressure level. Also, the 300-cfm roughing pump provides a sink for bearing scavenging and pressure relief of the closed loop.



3.5.4 Water Cooling System

The alternator is cooled by water that has been precooled by circulation of a refrigerated liquid (trichlorethylene) through a stainless-steel reservoir. The reservoir is shown in Figure 33. The water pump runs continuously and the flow rate is regulated by manual valves. The water is circulated through the alternator housing and returned to the reservoir.

3.5.5 Pneumatic-Control System

The pneumatic-control system, shown in Figure 34, is designed to maintain a constant loop pressure level which, if CRU speed remains constant, will result in constant argon mass-flow rate.

In the automatic control system, pressure is controlled by a pressure switch. This switch, installed in a U-tube, is actuated by a 0.1 in. Hg (0.5 psia) variation in pressure.

This switch actuates either the pressure bleed solenoid or the make-up solenoid, depending on whether the upper or lower pressure limit has been reached.

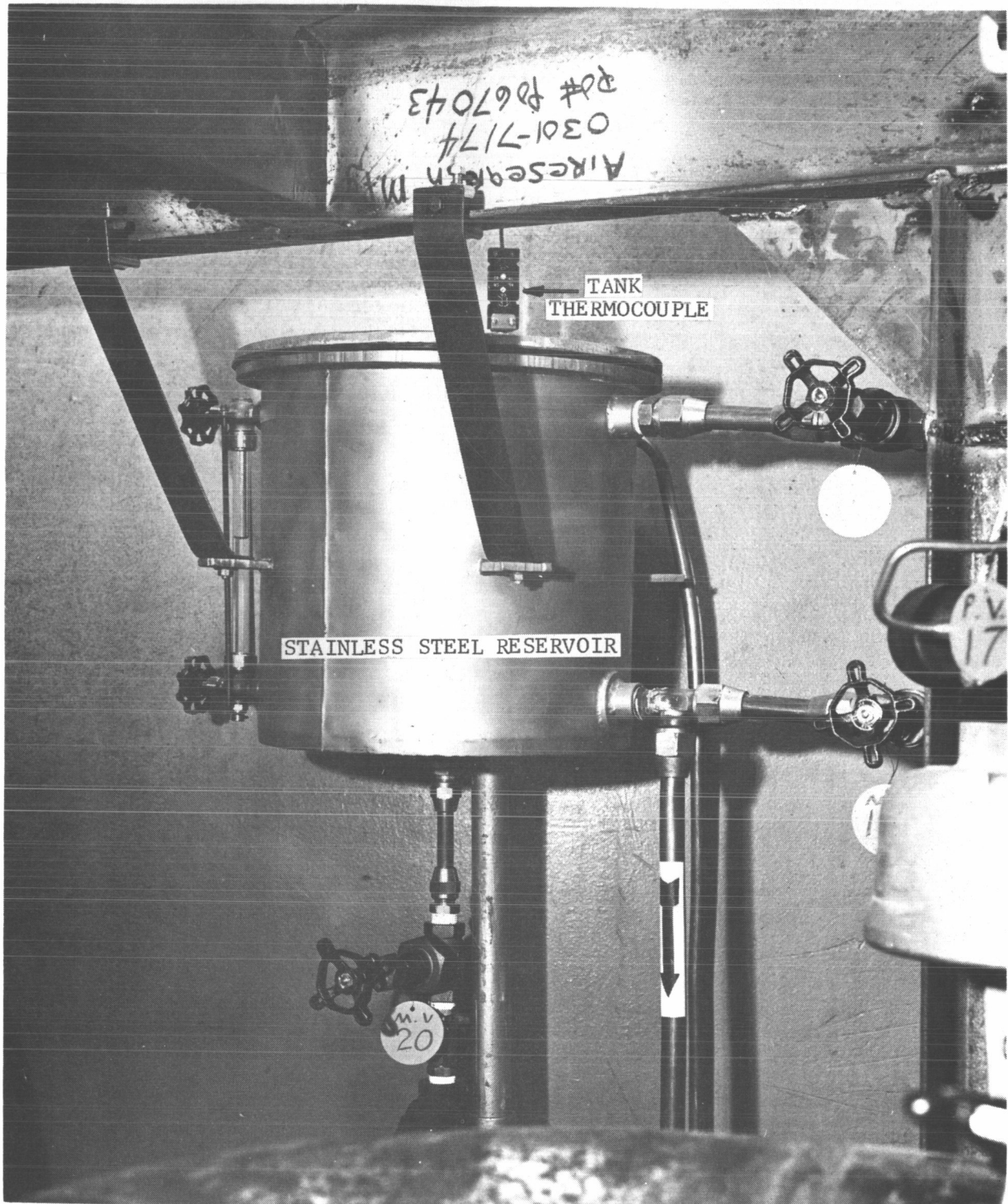


FIGURE 33

BCD TEST, PHOENIX
COOLING WATER LOOP RESERVOIR

PHOTO NO. P-28277-10

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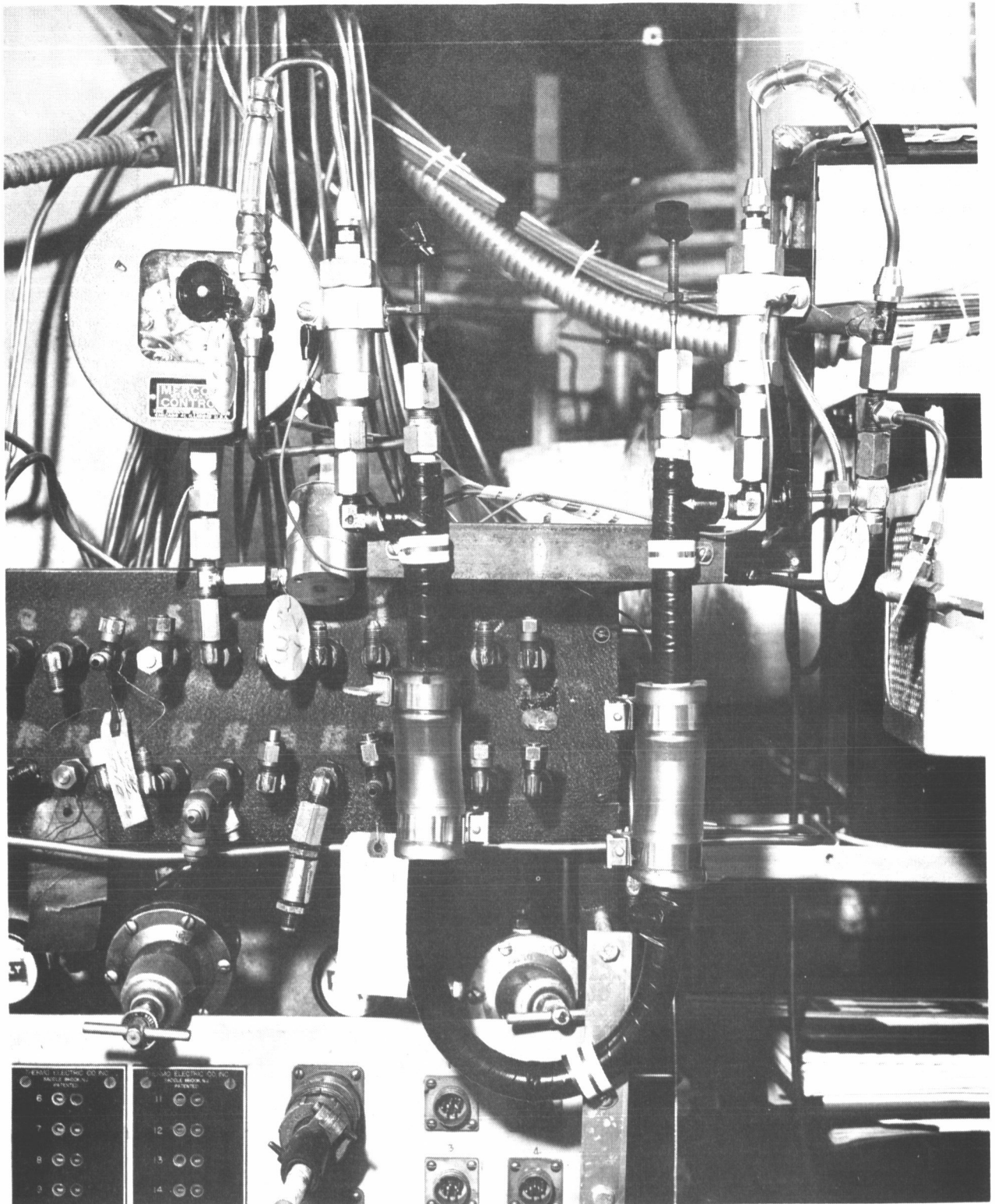


FIGURE 34

BCD TEST, PHOENIX
U TUBE - PNEUMATIC LOOP
CONTROL DEVICE

PHOTO NO. P-28277-8

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4.0 TEST RESULTS

4.1 General

When the BCD program was initiated under company-sponsorship, three objectives were set forth. These are reiterated below to place the test results in the proper perspective.

- (a) Modify the CRU to include a brushless alternator and change from oil-lubricated bearings to gas-lubricated bearings.
- (b) Integrate the modified CRU configuration with the closed loop, and develop a technique for making gas-injection starts. Demonstrate controlled closed loop operation of the BCD.
- (c) Checkout of the BCD system to establish operating limits.

As was stated in the introduction of this report, the contracting agency was desirous of having a dynamic power-conversion system, typical of hardware designed for space application, with which operating experience could be obtained, and for which performance evaluation could be accomplished. Since the components had never been matched and were "off the shelf," the ultimate



performance level itself was not an objective. As can be seen in the following test results, performance levels were high, even though virtually no system development work was done.

4.2 Summary of Tests

Over 50 successful gas-injection starts were made, and over 100 hours of closed-loop operation were accumulated during the course of BCD checkout testing. The tests are summarized in the "Summary Test Log," Appendix III. They are listed in chronological order with comments describing the salient results of each test. A comparison of BCD performance during tests performed in 1963 at AiResearch, Los Angeles and in 1967 at AiResearch, Phoenix, is presented in Table 2.

Over the period that BCD checkout testing was performed, a great deal of valuable operational experience was obtained. For the first time an opportunity existed to gain experience regarding the control systems associated with the operation of a complete gas turbine engine, supported on gas lubricated bearings, in a closed-cycle power conversion system.

Observation and evaluation of the operational characteristics of the CRU in both open loop and closed loop environments has substantially increased the technology base in gas bearing-supported turbomachinery. A new gas bearing (foil bearing) technology has been furthered as a result of the tests completed to date, and the application of foil bearings to gas turbomachinery demonstrated.



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION
PHOENIX, ARIZONA

TABLE 2
PERFORMANCE COMPARISON

Parameter	TEST POINT NO. Data Taken 1963, Los Angeles			TEST POINT NO. Data Taken 1967, Phoenix			
	4	5	6	8-7-67-2	9-29-67-2	9-29-67-3	9-29-67-4
Time, hr:min	16:00	16:40	17:35	--	--	--	--
Compressor inlet temperature, °R	498	500	501	468.6	457	456	457
Compressor inlet pressure, psia	4.6	6.6	6.7	4.8	4.95	4.95	4.94
Compressor outlet temperature, °R	682	695	688	528.7	660	640	641
Compressor outlet pressure, psia	8.9	12.7	12.8	8.43	9.61	9.08	9.11
Heater inlet temperature, °R	1,593	1,577	1,484	1,436	1,307	1,304	1,464
Heater tube temperature, °R	2,190	1,974	1,865	--	--	--	--
Heater power, watts	11,400	16,000	14,750	9,497	13,020	12,804	12,853
Turbine inlet temperature, °R	1,958	1,974	1,855	1,738	1,755	1,849	1,909
Turbine inlet pressure, psia	8.7	12.5	12.7	8.21	9.36	8.85	8.83
Turbine outlet temperature, °R	1,626	1,628	1,545	1,433	1,400	1,494	1,547
Turbine outlet pressure, psia	4.7	6.6	6.8	4.85	5.15	5.16	5.13
Recuperator outlet temperature, °R	753	781	768	698	729	715	660
Turbine rotational speed, rpm	57,700	58,300	58,000	50,030	55,000	52,000	52,160
Alternator frequency, cps	1,911	1,919	1,913	833	915	867	871
Alternator output power, Phase A, watts	520	960	820	365	436	456	504
Alternator output power, Phase B, watts	520	960	820	353	436	456	504
Alternator output power, Phase C, watts	528	980	840	365	436	456	504
Alternator rotor case pressure, in. Hg abs	10.96	15.36	15.66	--	--	--	--
Bearing slinger outlet pressure, in. Hg abs	8.76	12.96	13.36	--	--	--	--
Turbine end bearing temperature, °F	109	121	119	115	104	99	110
Ambient pressure, in. Hg abs	29.96	29.96	29.96	28.77	28.72	28.72	28.72
Turbine temperature drop, °R ΔT_{5-6}	332	346	310	305	356	355	362
Turbine temperature drop - compressor temperature rise, °R	148	151	123	145	153	171	180
Alternator output power, watts	1,568	2,900	2,480	1,085	1,310	1,368	1,512
Alternator efficiency,	0.821	0.884	0.875	0.912	0.917	0.920	0.919
Mass flow, $W = \frac{\text{Turbine output}}{1055 C_p \Delta T_{\text{turb}} - \Delta T_{\text{comp}}}, \frac{\text{lb}}{\text{sec.}}$	0.216	0.1884	0.203				
Measured mass flow, $\dot{m}, \frac{\text{lb}}{\text{sec.}}$	--	--	--	0.1230	0.1449	0.1359	0.1314
Recuperator temperature drop, °R	873	847	777	807	664	767	815
Heater temperature rise, °R	403	432	390	403	448	445	445
Heat input = $1055 W C_p \Delta T_{\text{heat}}$, watts	6,430	10,680	10,390	4,860	8,500	7,910	7,630
Heat source heat loss, watts	4,970	5,320	4,360	--	--	--	--
Heat sink temperature drop, °R	255	281	267	230	272	259	262
$\eta_{cy} \text{ overall } \frac{kw_e \text{ in.}}{kw_{\text{heater}}}$	0.1375	0.1812	0.1682	0.114	0.100	0.106	0.117
$\eta_{cy} = \frac{\text{Turbine out}}{\text{Gas Heat in}}$	0.207	0.233	0.222	0.222	0.154	0.172	0.197



During the checkout testing of the BCD, new information and operational experience were accumulated in several areas of technology. These include:

- (a) Gas injection start-up techniques for a gas turbine engine supported on gas bearings in a closed loop
- (b) Gas bearing environmental control system design
- (c) Closed cycle, thermodynamic system control procedures and equipment
- (d) Automatic control and protection, including automatic safe shutdown for gas bearing supported turbomachinery

During the initial stages of testing, a great deal of difficulty was encountered in achieving successful start-up of the BCD. Experience had to be acquired regarding the thermodynamic system response to the controls, and the manner in which the gas bearing environment and operation interfaced with the thermodynamic system, particularly during the critical start-up transient period.

These problems were resolved as the test program progressed and the gas injection start-up procedure was refined. It is now a relatively simple matter to complete a start-up within 30 seconds. (The start-up is considered to be complete when gas injection argon flow has ceased and the CRU has accelerated under its own power in the closed loop to the speed set point and is generating electrical power.) A start-up includes making a transition from hydrostatic bearing operation to hydrodynamic bearing operation. This bearing



mode changeover procedure caused considerable difficulty early in the test program. The problem was satisfactorily resolved by making the appropriate modifications to the gas bearing environmental control system.

The gas bearing control system, in its present configuration, provides for safe bearing operation at BCD design conditions, at off-design conditions, during start-up transient conditions, and during emergency shutdown conditions. The control system has been satisfactorily demonstrated in all of these situations.

The gas bearings have operated satisfactorily under various conditions of magnetic loading due to the magnetic flux field emanating from the alternator field coils in the CRU. At the outset of this program much pessimism existed throughout the aerospace industry regarding the feasibility of operating a high-speed electrical alternator on gas bearings. The concern in this regard has been substantially reduced due to the results of the testing reported herein. The CRU, on its gas bearings, has been able to withstand the most severe magnetic loading, including testing during which maximum magnetic loading was applied and then removed instantaneously many times. Shock loading and load step changes were a common occurrence in the course of closed loop start-up and operation, but did not adversely affect CRU operation.

Experience has also been obtained regarding another area of particular interest to the test engineer and system controls designer. This is the area of gas turbine engine protection in the event a system malfunction occurs. The objective is to provide safety controls which will, in the event of a malfunction, provide the gas bearings with a safe operating environment and permit a safe shutdown of the engine.



The emergency shutdown system, in a somewhat rudimentary form, exists for the BCD and has been successfully implemented on several occasions. During the course of testing, several support system or control malfunctions occurred, resulting in initiation of an emergency engine shutdown. (This procedure is described in Section 3.3.1.) Specifically, the malfunctions listed below occurred and after each the BCD was successfully shut down with no gas bearing damage sustained:

- (a) Engine overspeed (due to speed control system malfunction)
- (b) Loop overpressure
- (c) Heater ground current detection (heating element shorted to heater core)
- (d) Heater power loss (burned out fuses)

In addition, the BCD was shut down on three occasions with the bearings operating hydrodynamically until the rotating group had come to a complete stop. In each instance this type of shutdown was required because of a vacuum pump malfunction (either an overload or a mechanical failure). After each of these shutdowns the test facility equipment was repaired without removing the CRU from the closed loop, and a start-up was successfully accomplished and the operating point re-established. No apparent change in the CRU performance or gas bearing performance was observed after the "hydrodynamic shutdowns" had occurred.

The BCD has been tested over a relatively wide range of operating conditions. The ability of the gas turbine engine to operate on gas bearings over a wide range of system conditions adds considerably to evidence supporting the flexibility potential of the closed Brayton cycle power conversion system.



4.3 Method of Taking Data

Several methods of taking data were employed. These include the following:

- (a) Sanborn oscillograph recorder
- (b) Digital console with paper-tape print-out
- (c) Visual, including manometers, Brown temperature meters, oscilloscopes, electrical meters, and the like

Most of the measurements were duplicated on at least two readout devices so that a means of cross-checking the data existed.

Data logging was accomplished by the following means: (The figure numbers in parentheses refer to examples of some of the data records).

- (a) Test-cell log book (record kept by technician)
- (b) Engineers' log book
- (c) Primary visual data sheet (Figure 35)
- (d) Supplementary data sheets (Figure 36 through 38)

(These data sheets are included in this report to be used as an aid in setting up data sheets to be used at NASA-MSC, Houston.

- (e) Oscillograph recording (Figure 39).

TEST IDENTIFICATION												TEST TIME									
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20		

[illegible]

GAS FLOW ΔP, IN. H ₂ O								SHAFT SPEED, RPM								SCANNER VALVE REF, IN. Hg G								BAROMETRIC PRESSURE, IN. Hg ABS								TANK PRESSURE, PSIA							
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40

WATER COOLING LOOP AND TURBINE INLET CONDITIONS																																																									
ALT. OUT TEMP, °F								TH BRG OUT TEMP, °F								J BRG OUT TEMP, °F								TURB. INLET TEMP, °F								SUMP TEMP, °F								ALT FLOW, CPS								PUMP FLOW, CPS									
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56		

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71-2

[illegible]

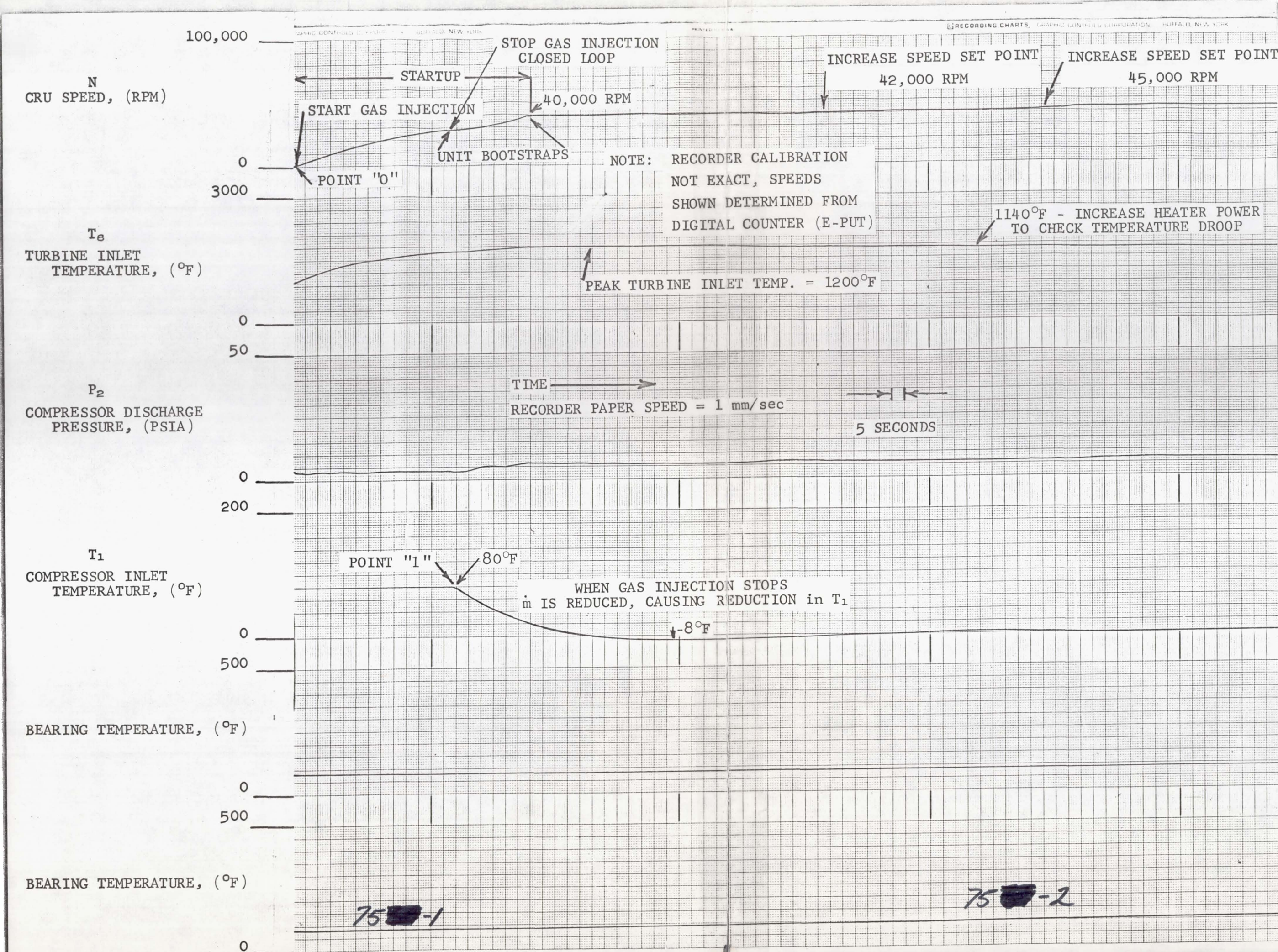
72-1

72-2

[illegible]

74-1

74-2



APPROXIMATELY 15 MINUTES OF STEADY-STATE
OPERATION HAVE BEEN REMOVED FROM
THIS TRACE TO FACILITATE PUBLICATION

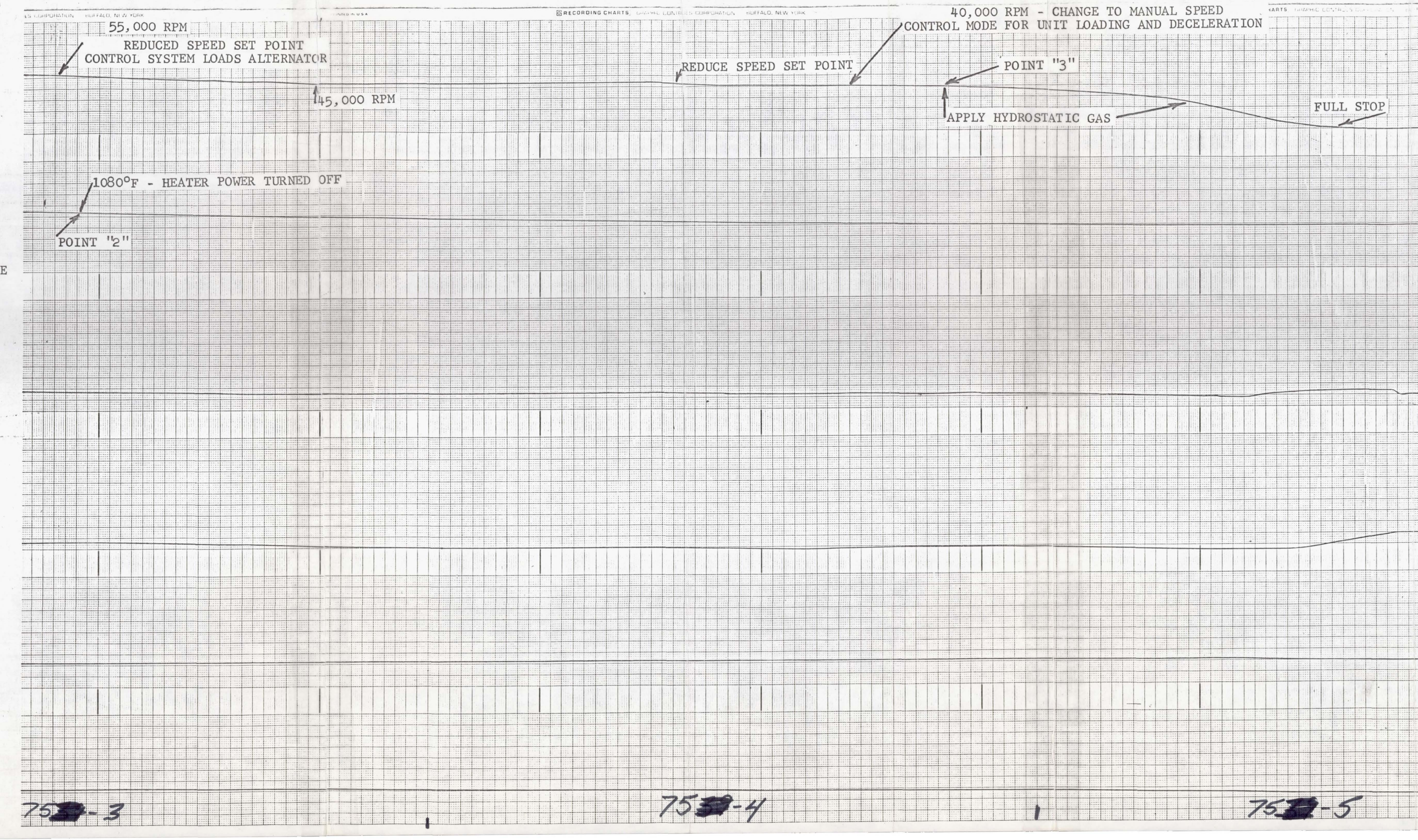


FIGURE 39
OSCILLOGRAPH RECORDING OF BCD OPERATION

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The Sanborn recorder is especially useful in monitoring transient conditions during gas-injection start-up. Shown in Figure 39 is a typical start-up and shutdown transient. Gas injection is on between points "0" and "1" and the bearings are operating hydrostatically. At Point "1" the loop is closed by simultaneously closing the injection valve and the dump valve.

Two characteristic transients can be seen which are always initiated at the moment that gas injection is stopped. If at the time the loop is closed, the CRU is above the self-sustaining speed condition, the CRU will rapidly "bootstrap" itself to operating speed, at which time the alternator is loaded by the speed-control system. Also, since the argon flowrate through the the cooler diminishes, the compressor inlet temperature (T_1) drops. This reduction of T_1 is a primary reason for the rapid acceleration of the machine. At 40,000 rpm, the hydrostatic bearing supply is removed, and the gas bearings operate hydro-dynamically.

In the example shown, T_1 is not steady. The control system for compressor inlet temperature control was modified subsequent to this run, and constant T_1 was thereafter obtained.

The shutdown transient shown is typical when the heater power is turned off (point "2").

At point "3" the alternator load has almost completely dropped off and a slight acceleration may result from application of hydrostatic gas to the bearings. Electrical load is then applied manually to bring the CRU to a full stop.



4.4 Data Reduction

A high-speed digital computer (IBM 1130) was used for data reduction. The procedure used for data recording was described in Section 4.3. Using this format, the test data was logged in a convenient manner so that computer input cards could be prepared directly from the data logs. The data was then reduced with use of the computer program, described in Appendix I.

The description, location, and accuracy of the instrumentation was discussed in Section 3.0. Additional comment is in order, however, regarding the inherent problems existing in making thermodynamic measurements in a compact closed-loop system. Determination of the mean bulk temperature at any given location in the closed loop is particularly difficult for the following reasons:

- (a) Severe temperature stratification exists at the outlet of the recuperator and cooler due to large temperature gradients which exist in the metal cores.
- (b) The limited number of probes, (three maximum), used at any one location does not permit thorough definition of the temperature profile.
- (c) Sharp turning of the gas flow in the interconnecting ducting causes a large degree of flow stratification.

Pressure measurements, especially of static pressures, are easier to make accurately than temperature measurements. Electrical measurements are also considered to be quite accurate and meter calibration was checked periodically against standards during testing.



4.5 Discussion of the Test Results

In summary, the BCD checkout test at Phoenix is significant in that it shows good correlation between theoretically predicted performance and actual performance. The measurements show cycle efficiencies up to 20 percent, in the range of 50 to 85 percent design power. These results are discussed in more detail in the following paragraphs.

One of the objectives of this program was to define the range of operation for several of the BCD parameters. The following range of operation has been demonstrated.

Turbine inlet temperature (T_6)	1660 - 2000°R
Compressor inlet temperature (T_1)	440 - 500°R
Compressor inlet pressure (P_1)	4.0 - 7.0 psia
Operating speed (N)	40,000 - 60,000 rpm
Argon flow rate ($\dot{m}$)	0.10 - 0.20 lb/sec
Power output range (kw_e)	0 - 2.0 kw

Additional funding was not available to permit further extension of these ranges at this time. However, 80 percent of objectives were met during the limited checkout testing that has been accomplished. Results are encouraging and it is felt that further extension of the operating ranges will be accomplished.

Over 50 data points were taken, where a complete data scan of the system was made using the digital data-acquisition system. All of the data was processed through the computer data-reduction program (refer to Appendix I). The results were then screened to eliminate obviously "bad" data. For instance, in some cases a visual data point reading had been erroneously made or recorded, or a punch card error made, or the like, which invalidated the performance calculation.



Figure 40, showing thermodynamic cycle efficiency versus power level, does not include all of the data taken which was considered to be reliable. There are about 20 additional data points falling at about $n_{cy} \approx 19$ percent and $kw \approx 1.1$ kw which were not plotted for the sake of preserving clarity of the chart (since they fell on or close to points already shown on the curve). Likewise, on Figures 41 and 42, the number of points plotted is a good representation of the results but does not include all the test data. There is a difference of two efficiency points in the adjusted efficiency, shown in Figure 42, from the two previous curves. For Curves 40 and 41, the efficiency was calculated with use of an argon flow rate determined from a calibrated flow section ΔP measurement. For Curve 42, the thermal input to the system was calculated by using an argon flow rate determined from an energy balance performance on the heat-sink heat exchanger.

On the liquid side of the heat-sink heat exchanger, the temperatures and liquid flow-rate measurements can be made quite accurately. The heat exchanger is, of course, at a low average temperature and well insulated, so that external heat leakage is negligible. It was found during calibration of this heat-exchanger, prior to total system operation, that nearly perfect energy balances [i.e., $(\dot{m} C_p \Delta T)_{gas} = (\dot{m} C_p \Delta T)_{liquid}$] could be made.

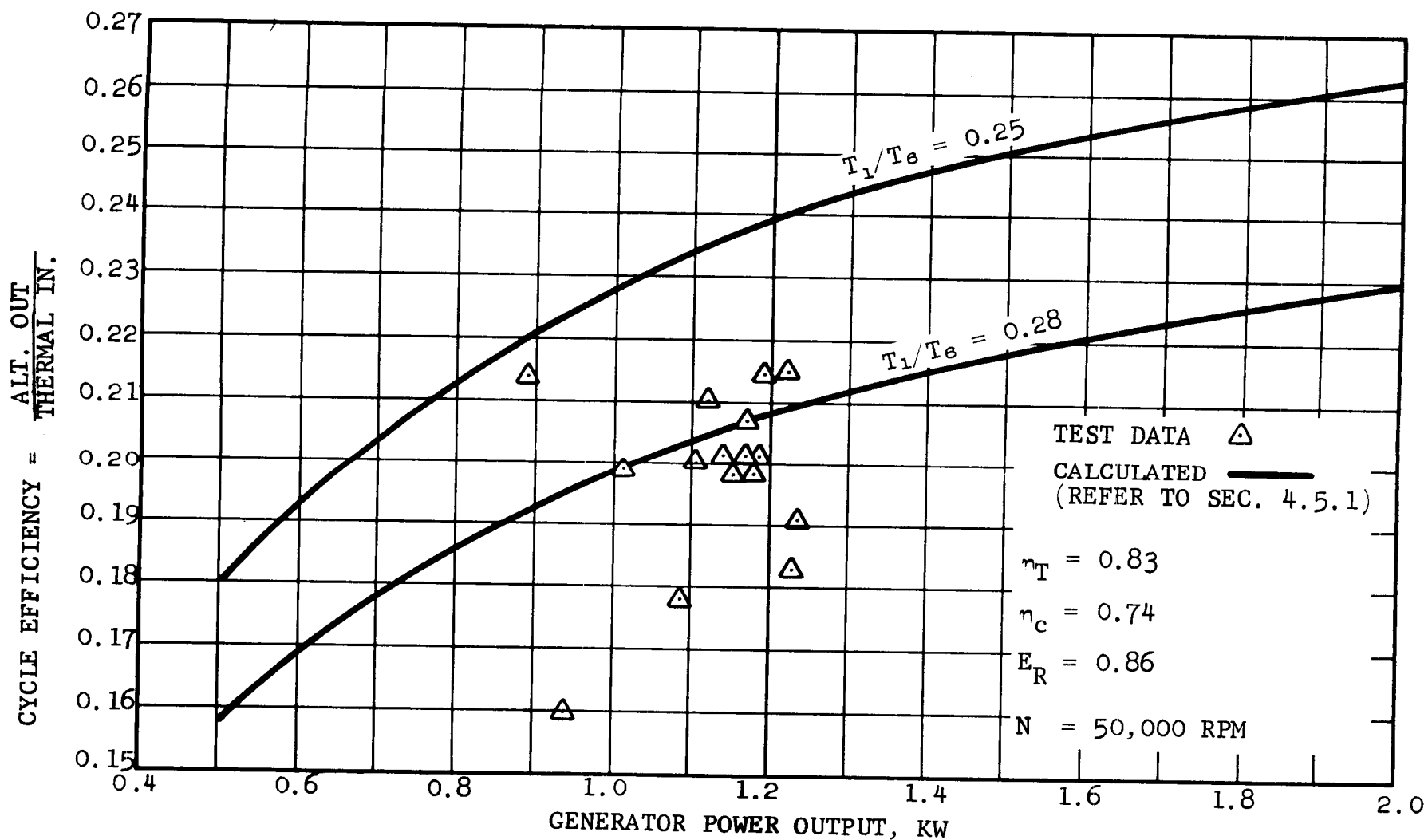


FIGURE 40

RELATION OF POWER LEVEL TO CYCLE EFFICIENCY
BCD TESTS, AUGUST 1967

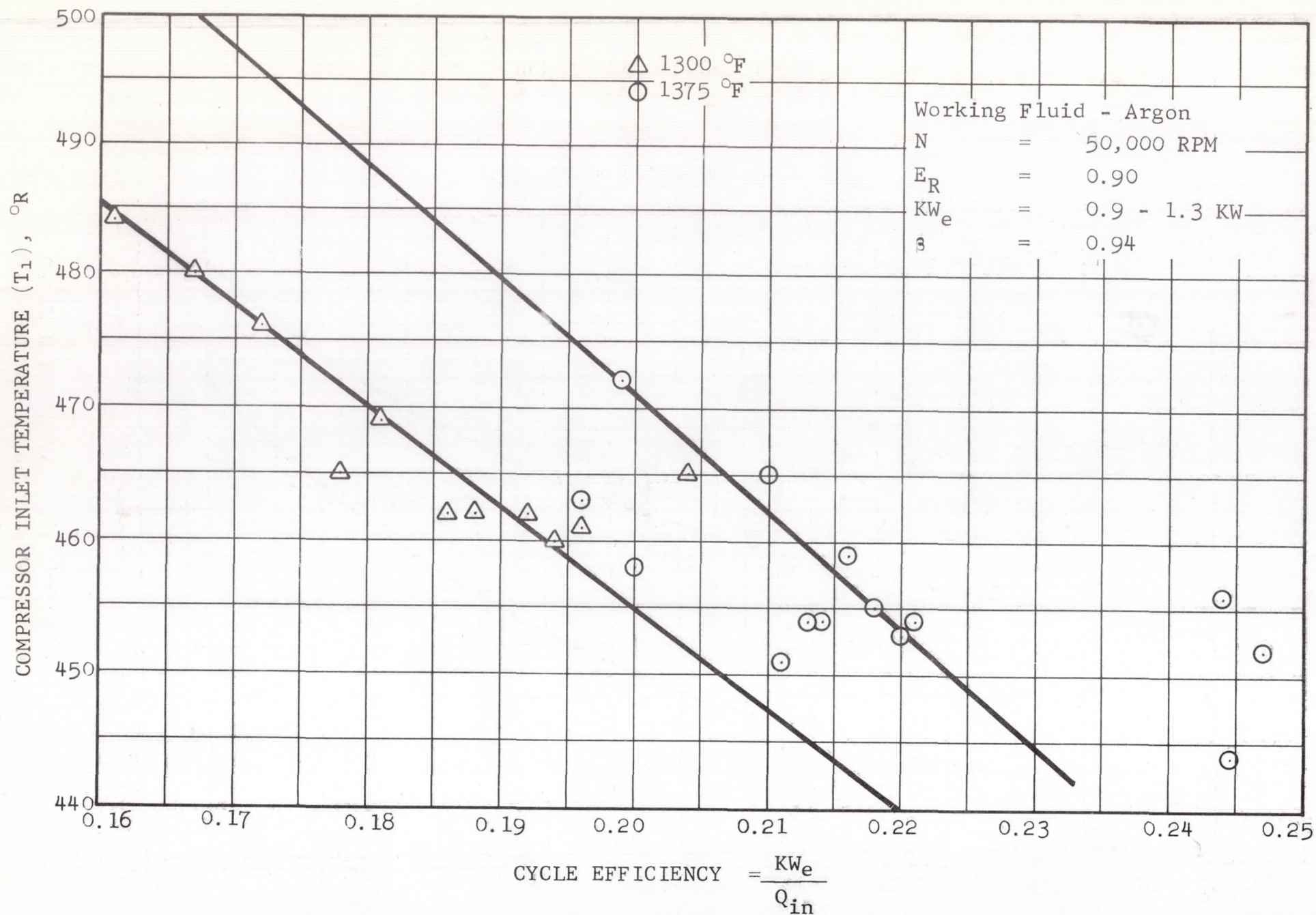


FIGURE 41
EFFECT OF COMPRESSOR INLET TEMPERATURE ON CYCLE EFFICIENCY

BCD TESTS, AUGUST 1967

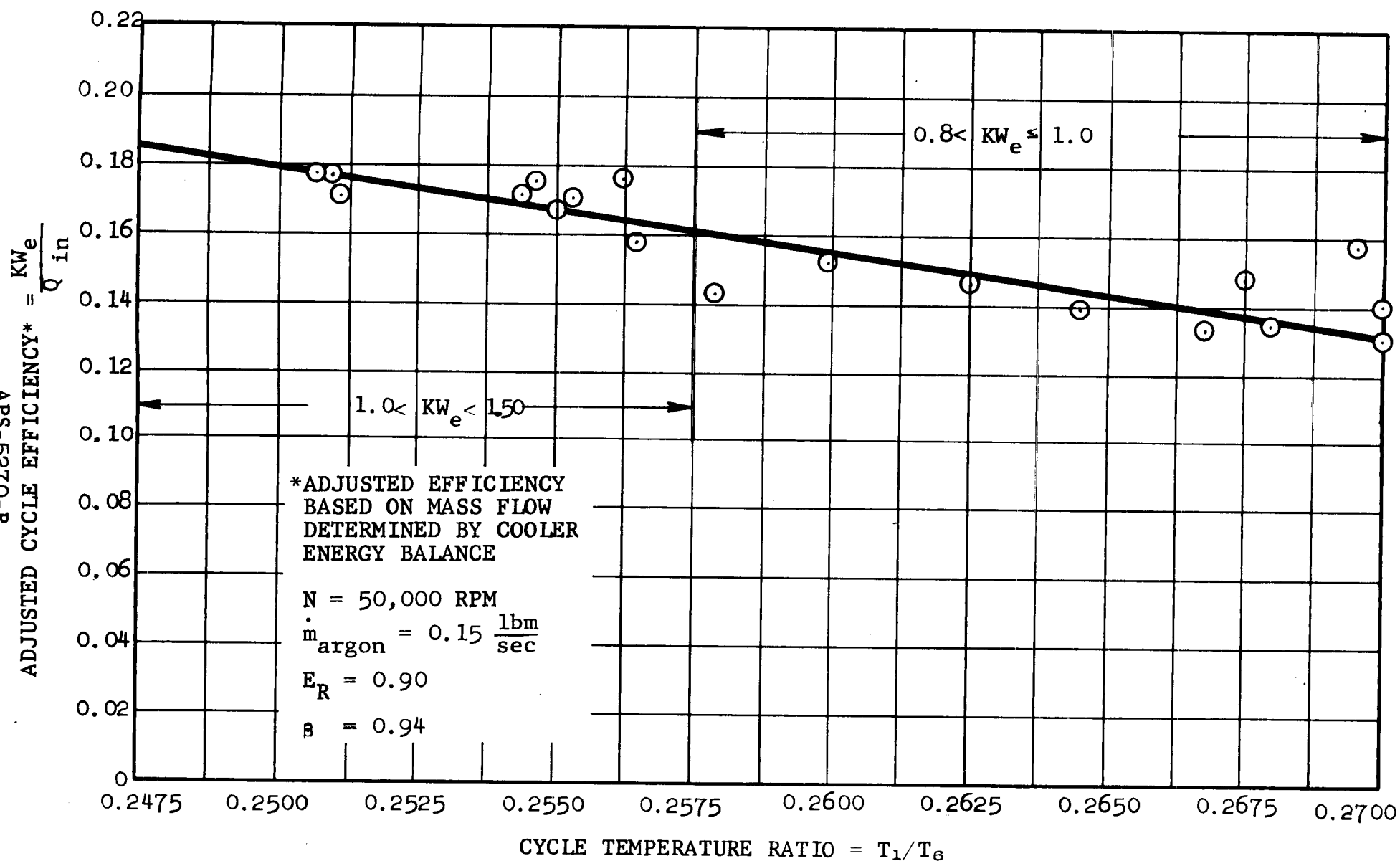


FIGURE 42

EFFECT OF CYCLE TEMPERATURE
RATIO ON CYCLE EFFICIENCY
ECD TESTS, AUGUST 1967



The cycle efficiency determined from test data reduction differs from the calculated cycle efficiency in the following manner. The calculated cycle efficiency is obtained from the following equations which do not require measured argon flow rate:

$$(a) \quad \eta_{cy}' = \frac{C_{p_{ta}} (T_6 - T_9) - C_{p_{ca}} (T_2 - T_1)}{C_{p_{ha}} (T_6 - T_5)}$$

where η_{cy}' = thermodynamic cycle efficiency

$C_{p_{ta}}$ = average specific heat of gas passing through the turbine

$C_{p_{ca}}$ = average specific heat of gas passing through the compressor

$C_{p_{ha}}$ = average specific heat of gas passing through the heater

$$(b) \quad \eta_{cy} = \eta_{cy}' \left(\frac{kw_e}{kw_t} \right)$$

where kw_e = alternator electrical output

kw_t = net turbine shaft power

$$(c) \quad kw_t = \frac{kw_e}{\eta_g} + \text{bearing losses} + \text{windage losses}$$

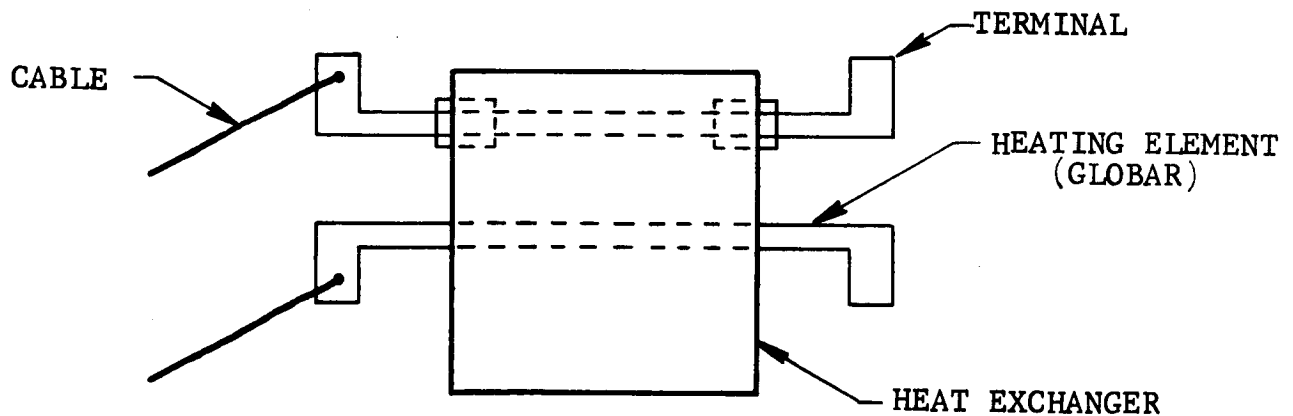
Generator efficiency, bearing losses, and windage losses are determined as described in Section 4.5.1.



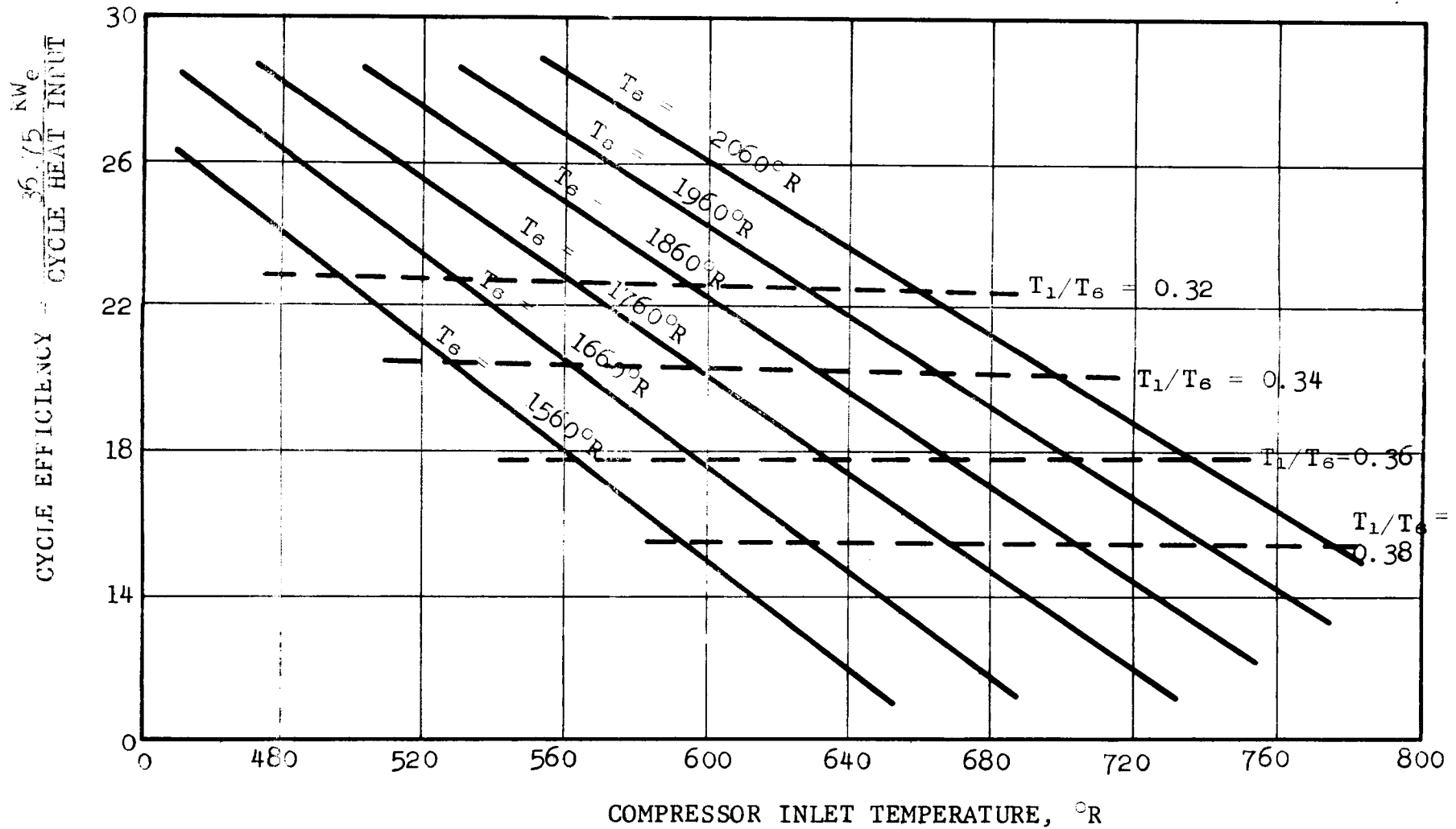
The efficiency depends largely upon the argon flow rate determination. The argon flow rate determined from a cooler energy balance is 10 percent higher than that determined by the flow-section delta P measurement. The higher flow rate results in lowering the cycle efficiency about two efficiency points. It is noteworthy that even the conservative approach to cycle efficiency determination results in impressive performance.

Figure 42 is a generalization of the data presented in Figure 41, with adjustments made in the efficiency level. It can be shown through parametric analysis that cycle efficiency is strongly dependent upon cycle temperature ratio (T_1/T_8) and only slightly affected by T_8 itself over the range of $T_8 = 1200^\circ\text{F}$ to 1500°F . This effect can be seen in Figure 43, which shows the results of a parametric study performed for a 35-kw system.

The overall cycle efficiency, defined as the ratio of electrical power output from the alternator to the electrical power supplied to the heater, was in the range of from 10 to 13 percent. A large heat loss occurs at the heat source heat exchanger. A sketch of the physical arrangement of the heating elements and the heat exchanger is shown below.



USEFUL ELECT.
 OUTPUT POWER = 36.75 KW
 $E_R = 0.90$
 $\beta = 0.88$
 $N_{sc} = .09$
 WORKING FLUID - ARGON



RELATIONSHIP OF BRAYTON-CYCLE EFFICIENCY AND CYCLE TEMPERATURES

FIGURE 43



There are 18 resistance elements, with two terminals per element. There are also 36 element ends protruding beyond the insulation. A large amount of heat is generated in and lost from the element ends and the terminals. This loss would, of course, not exist in an ultimate space application, since the entire heat source would be insulated. The end of the heating elements and the terminal connections are not completely covered by the insulation. The reason for this is that a certain amount of cooling by convection and radiation is required. Without cooling, the terminal ends overheat and fail. Unfortunately this cooling results in a substantial heat loss. In fact, the heater efficiency, with proper cooling to protect the terminals, is reduced to only 50 or 60 percent.

4.5.1 CRU Component Performance Analysis

From the quantity of test data taken during the BCD check-out program, some conclusions can be drawn regarding the individual component performances. Below is a summary of component performances during the 75-hour test, followed by a discussion of how these values were determined.

BCD Component Performance

Turbine efficiency	83 percent
Turbine speed	50,000 rpm
Turbine velocity factor	0.972
Turbine pressure ratio	1.67
Compressor efficiency	74 percent
Compressor slip factor	0.85
Compressor speed	50,000 rpm
Compressor pressure ratio	1.78
Alternator efficiency	92 percent
β (pressure loss factor)	0.940
E_R (recuperator effectiveness)	0.86



The BCD cycle efficiency was computed using the component performance values shown above as input to the AiResearch Digital Computer Design Program No. 532 (discussed in Appendix II). Good correlation for cycle mass flowrate and cycle efficiency resulted. The component efficiencies were assumed to be constant over the system power range, from 0.5 to 2.0 kw_e. The results are shown on the solid lines of Figure 40.

4.5.2 Aerodynamic Component Performance

The compressor and turbine efficiencies cannot be accurately determined from an actual temperature-difference measurement, due to heat losses from the ductwork and scrolls and internal heat leaks from the scrolls to the alternator cooling jacket. Other means are available, however, to determine these efficiencies.

4.5.2.1 Compressor Performance

The compressor performance is a function of corrected speed ($N/\sqrt{T_1}$) and corrected flow ($\frac{\dot{m}}{P_1} \sqrt{\frac{T_1}{P_1}}$). The mass flowrate can be determined rather accurately by performing an energy balance analysis on the cooler. Speed, inlet temperature, and inlet pressures are measured directly.

Another approach that results in good agreement with the above approach is:

- (a) Determine the compressor slip factor for the compressor. (Slip factor for a straight radial outflow compressor is essentially a function of the number of compressor blades.)



- (b) ΔT is related to slip factor, diameter, and speed in the following manner:

$$S \propto \frac{\Delta T}{D_c^2 N^2}$$

- (c) This ΔT and pressure ratio is then used to compute compressor efficiency.

4.5.2.2 Turbine Performance

Radial turbine performance can be accurately estimated from the turbine velocity factor.

$$\text{Velocity factor } F_v = \frac{U_{\text{tip}}}{(gH)_{\text{isen}}} = \frac{ND_t N}{g_o C_p T_s} \left[1 - \left(\frac{1}{r_t} \right)^{0.4} \right]$$

4.5.2.3 Alternator Off-Design Performance

The alternator electrical efficiency is estimated by using the following equation:

$$N_g = \frac{\text{electrical output}}{\text{output} + \text{losses}} = \frac{KW_e}{KW_e + 0.179 \left(\frac{N}{N_D} \right)^2 + 0.239 \left(\frac{I}{I_D} \right)^2 + KW_{\text{Field}}}$$

where N_D and I_D are design-point speed and output current, respectively.



4.5.2.4 Relationship Between N_{sc} and $\dot{m}$

The compressor specific speed and volumetric flowrate are related in the following way:

$$N_{sc} = \frac{NQ^{1/2}}{(gH)^{3/4}_{isen}}$$

$$Q = \frac{\dot{m}}{\rho} \left(\frac{ft^3}{sec} \right)$$

$$H_{isen} = \text{isentropic head (ft)}$$

$$\dot{m} = \text{mass flowrate (lbs/sec)}$$

$$\rho = \text{density (lbs/ft}^3\text{)}$$

Hence:

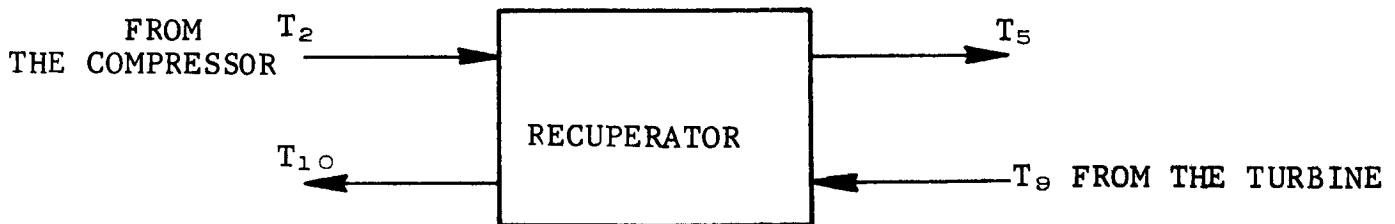
$$N_{sc} = \left(\frac{N}{60} \right) \frac{Q^{1/2}}{(778 g_c \Delta H)^{3/4}_{isen}} = \frac{NQ^{1/2}}{119,800 (0.124T_1[r_c^{0.4}-1])^{3/4}}$$

4.5.2.5 Recuperator Effectiveness

Recuperator effectiveness is defined as:

$$E_R = \frac{T_5 - T_2}{T_9 - T_2}$$

where the temperatures are defined as shown in the sketch below.





4.5.2.6 Pressure Loss Factor

The pressure loss factor (β) is defined as $\beta = r_t/r_c$. The pressure loss factor can be shown to be approximately equal to one minus the total system fractional pressure drop, i.e.:

$$[\beta \approx 1 - (\frac{\Delta p}{P})_{\text{total}}].$$



5.0 CONCLUSIONS AND RECOMMENDATIONS

In this report the equipment, procedures and methods used in the BCD check-out tests performed by the Contractor have been described. The data acquisition and reduction methods have been reviewed. Also a complete presentation of the test results has been included. The following conclusions are drawn from the results of the test program completed by the Contractor to date.

- (a) The feasibility of utilization of a high-speed combined rotating unit, supported on gas bearings, has been proven by actual demonstration testing.
- (b) Magnetic loads imposed on the CRU gas bearing system are not a limiting factor operationally.
- (c) Gas injection start-up technique, and control systems operation for the starting of gas bearing supported turbomachinery in closed loop applications have been demonstrated.
- (d) The BCD test results confirm the high performance potential of the Brayton cycle system. This was done without extensive development and at low cost.
- (e) The range of testing demonstrated the high degree of system flexibility, both in power level, and operating conditions available with the Brayton cycle system.



The following recommendations are made:

- (a) BCD testing should be continued at the NASA, Manned Spacecraft Center test facilities. The BCD represents an excellent tool for obtaining operating experience with Brayton Cycle Systems. Also, further performance evaluation is needed.
- (b) Development work should be initiated to upgrade the BCD. This would include the following:
 - (1) Aerodynamic matching of the compressor and the turbine.
 - (2) Further development of the foil gas bearing concept.
 - (3) Analysis and design studies resulting in an upgraded heat source for the system.
 - (4) CRU design analysis to improve the dynamic and thermal characteristics of the machine.
- (c) System studies should be performed to define programs for coupling the BCD with advanced control system and power conditioning equipment for evaluation testing.
- (d) Endurance testing should be initiated to establish reliability data.

APPENDIX I

DATA REDUCTION PROGRAM

```

*LIST ALL
*EXTENDED PRECISION
*ONE WORD INTEGERS
*IOCS(CARD,TYPEWRITER,KEYBOARD,1132 PRINTER,PAPER TAPE,DISK)
  DIMENSION DATE(5),ID(8),DATA(8),P(50),T(50),T1(3),T2(3),T5(3),
  DT6(3),T9(3),T9P(2),T10(3),PS1(3),PT1(3),PS2(3),PT2(3),PS6(3),PT6(3
  1),PS9(3),PT9(3)
  EQUIVALENCE (T(1),T1(1)),(T(4),T2(1)),(T(7),T5(1)),(T(10),T6(1)),
  E(T(13),T9(1)),(T(16),T9P(1)),(T(18),T10(1)),(P(1),PS1(1)),
  Q(P(4),PT1(1)),(P(7),PS9(3)),(P(10),PT9(1)),(P(13),PS6(1)),
  U(P(16),PT6(1)),(P(19),PS5), (P(20),PS10), (P(21),PS2(1)),
  I(P(24),PT2(1)),(P(27),PSCAN)
  IN=2
  IO=3
  1 DO 5 I=1,50
    T(I)=0.0
    P(I)=0.0
  5 CONTINUE
  READ(IN,10) DATE
  10 FORMAT(5A4)
  READ(IN,20) GP1V,GP1A,GP2V,GP2A,GP3V,GP3A,GFV,GFA
  20 FORMAT(10F8.3)
  READ(IN,20) HB1V,HB1A,HB2V,HB2A,HB3V,HB3A,TBIN,TBOT,BFLW
  READ(IN,20) GFDP,SPEED,SCANR,PAMB,PTANK
  READ(IN,20) TALOT,TTBOT,TJB1,TUIN,TSMP,AWFLW,WPFLW
  30 READ(IN,40) ( ID(J),DATA(J) ,J=1,8)
  40 FORMAT(8(I3,F7.2))
  DO 50 J=1,8
    IF(ID(J)=45) 42,100,42
  42 IF(ID(J)=100) 44,46,46
  44 K=ID(J)-45
    P(K)=DATA(J)
    GO TO 50
  46 K=ID(J)-100
    T(K)=DATA(J)
  50 CONTINUE
  GO TO 30
  100 DO 150 K=1,50
    IF(T(K)=2.31) 110,110,105
  105 IF(T(K)=7.91) 120,120,106
  106 IF(T(K)=19.6) 130,130,140
  110 T(K)=149.84124+T(K)*(43.375764-T(K)*(0.18753457-T(K)*(0.06719977+
    T(K)*(0.0017053186+T(K)*(0.0015862062+T(K)*0.00005555254))))))
    GO TO 150
  120 T(K)=149.87084+T(K)*(42.095214+T(K)*(0.65716552-T(K)*0.04826927))
    GO TO 150
  130 T(K)=141.54296+T(K)*(46.920898-T(K)*(0.23217773-T(K)*0.003713608))
    GO TO 150
  140 T(K)=171.00000+T(K)*(43.492187-T(K)*(0.10913085-T(K)*0.002472877))
  150 CONTINUE
  IF(P(1)=0.0) 152,152,154
  152 P(1)=P(3)
  154 IF(P(2)=0.0) 156,156,158
  156 P(2)=P(3)
  158 T6(3)=T6(1)
    T5(1)=T5(3)
    T5(2)=T5(3)

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160 T1A=(T1(1)+T1(2)+T1(3))/3.0
PT1A=(PT1(1)+PT1(2)+PT1(3))/3.0
PS1A=(PS1(1)+PS1(2)+PS1(3))/3.0
T2A=(T2(1)+T2(2)+T2(3))/3.0
PT2A=(PT2(1)+PT2(2)+PT2(3))/3.0
PS2A=(PS2(1)+PS2(2)+PS2(3))/3.0
T5A=(T5(1)+T5(2)+T5(3))/3.0
T6A=(T6(1)+T6(2)+T6(3))/3.0
PT6A=(PT6(1)+PT6(2)+PT6(3))/3.0
PS6A=(PS6(1)+PS6(2)+PS6(3))/3.0
T9A=(T9(1)+T9(2)+T9(3))/3.0
PT9A=(PT9(1)+PT9(2)+PT9(3))/3.0
PS9A=(PS9(1)+PS9(2)+PS9(3))/3.0
T9PA=(T9P(1)+T9P(2))/2.0
T10A=(T10(1)+T10(2)+T10(3))/3.0
GAFLW=10.0**((0.50232*ALOG(518.7*PT1A/(14.694*(T1A+459.7))*GFDP)/
G2.302585-0.62289)
GFPWR=GFV*GFA/1000.0
GP1PR=GP1V*GP1A/1000.0
GP2PR=GP2V*GP2A/1000.0
GP3PR=GP3V*GP3A/1000.0
GENPR=GP1PR+GP2PR+GP3PR
HB1PR=HB1V*HB1A/1000.0
HB2PR=HB2V*HB2A/1000.0
HB3PR=HB3V*HB3A/1000.0
HTRPR=HB1PR+HB2PR+HB3PR
BFLW=(0.008200*BFLW+0.04009)*0.1337*(95.0-0.0546*TBOT)/60.0
CPB=0.2203+0.0000667*(TBIN+TBOT)/2.0
ARFLW=BFLW*CPB*(TBOT-TBIN)/(0.1244*(T10A-T1A))
ETAHT=(GAFLW*0.124*(T6A-T5A)*1.055)/HTRPR
ETACR=BFLW*CPB*(TBOT-TBIN)/(GAFLW*0.1244*(T10A-T1A))
PRC=PT2A/PT1A
PRT=PT6A/PT9A
BETA=PRT/PRC
ETATU=(T6A-T9A)/((T6A+459.7)*(1.0-(1.0/PRT)**0.4))
ETACO=(T1A+459.7)*(PRC**0.4-1.0)/(T2A-T1A)
ETAGE=GENPR/(GENPR+0.179*(SPEED/85000.0)**2+0.239*((GP1A+GP2A+GP3A
1)/45.0)**2+GFPWR)
ETAGS=GENPR/(GAFLW*0.124*(T6A-T5A)*1.055)
ETAGY=ETAGS*ETAHT
ERMOT=(T9PA-T10A)/(T9PA-T2A)
ERCLD=(T5A-T2A)/(T9PA-T2A)
SCANR=(SCANR+PAMB)*0.4912
COVFL=ARFLW*39.94*(T1A+459.7)/(PT1A*144.0)
DHIS=0.1244*(T1A+459.7)*(PRC**0.4-1.0)
CHNSC=SPEED*COVFL**0.5/(119800.0*DHIS**0.75)
CSLIP=13.36*(T2A-T1A)/(SPEED/1000.0)**2
CYTR=(T1A+459.7)/(T6A+459.7)
T2C=T1A+(T1A+459.7)*((PRC**0.4-1.0)/0.72)
T9C=T6A-0.83*(T6A+459.7)*(1.0-(1.0/PRT)**0.4)
T5C=0.905*(T9C-T2C)+T2C
ETAG1=GENPR/(ARFLW*0.124*(T6A-T5A)*1.055)
ETAG2=GENPR/(ARFLW*0.124*(T6A-T5C)*1.055)
WRITE(IO,300)
300 FORMAT(32H1BRAYTON CYCLE DEMONSTRATOR TEST/1X,31(1H*))

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WRITE(10,310) DATE,PAMB,PTANK,SCANR,PSCAN,CYTR
310 FORMAT(/10H TEST NO, 3A4,6HTIME, 2A4,4X,24H AMBIENT PRESS, IN HGA =
F 6.2,5X,24HTANK PRESSURE, IN HGA = F 6.2/35H SCANNER VALVE REF PRE
OSS, IN HGA = F 6.2,13H VISUAL = F 6.2,8H DIGITAL,8X,19HCYCLE TEMP
R RATIO = F 6.4/)
WRITE(10,320)
320 FORMAT(25H THERMODYNAMIC CONDITIONS/1X,24(1H-)/36X,5H INLET,38X,9HD
FISCHARGE/20X,2(7HTEMP, F,6X,10HPTOT, PSIA,5X,11HPSTAT, PSIA,6X)/)
WRITE(10,330) ( T1(I),PT1(I),PS1(I),T2(I),PT2(I),PS2(I) ,I=1,3)
330 FORMAT(20X,F7.2,9X,F5.2,10X,F5.2,9X,F7.2,9X,F5.2,10X,F5.2)
WRITE(10,340) T1A,PT1A,PS1A,T2A,PT2A,PS2A
340 FORMAT(11H COMPRESSOR,9X,7(1H-),9X,5(1H-),10X,5(1H-),9X,7(1H-),9X,
F5(1H-),10X,5(1H-)/16X,2(4X,F7.2,9X,F5.2,10X,F5.2,5X)/)
WRITE(10,350) ( T2(I),PT2(I),PS2(I),T5(I) ,I=1,3)
350 FORMAT(16X,4X,F7.2,9X,F5.2,10X,F5.2,9X,F7.2)
WRITE(10,360) T2A,PT2A,PS2A,T5A,PS5
360 FORMAT(12H RECUPERATOR,8X,7(1H-),9X,5(1H-),10X,5(1H-),9X,7(1H-)/
F12H (COLD SIDE),8X,F7.2,9X,F5.2,10X,F5.2,9X,F7.2,24X,F5.2/)
WRITE(10,330) ( T6(I),PT6(I),PS6(I),T9(I),PT9(I),PS9(I) ,I=1,3)
WRITE(10,370) T6A,PT6A,PS6A,T9A,PT9A,PS9A
370 FORMAT(8H TURBINE,12X,7(1H-),9X,5(1H-),10X,5(1H-),9X,7(1H-),9X,
F5(1H-),10X,5(1H-)/16X,2(4X,F7.2,9X,F5.2,10X,F5.2,5X)/)
WRITE(10,375) T9P(1),T10(1),T9P(2),T10(2),T10(3),T9PA,T10A,PS10
375 FORMAT(20X,F7.2,38X,F7.2/20X,F7.2,38X,F7.2/65X,F7.2/12H RECUPERATO
FR,8X,7(1H-),38X,7(1H-)/12H (HOT SIDE) ,8X,F7.2,38X,F7.2,24X,F5.2)
WRITE(10,380) GFDP,GAFLW,SPEED,TUIN
380 FORMAT(/28H GAS FLOW DELTA P, IN H2O = F 6.3,5X,29HGAS FLOW RATE, L
FBS PER SEC = F 7.4,5X,19HSHAFT SPEED, RPM = F 6.0/25H TURBINE INLET
OTEMP, F = F 7.2,7H VISUAL/)
WRITE(10,390) GP1V,GP1A,GP1PR,HB1V,HB1A,HB1PR,GP2V,GP2A,GP2PR,
WHB2V,HB2A,HB2PR
390 FORMAT(22H ELECTRICAL CONDITIONS/1X,21(1H-)//23X,7HVOLTAGE,5X,7HAM
PPERES,5X,9HPOWER, KW,27X,7HVOLTAGE,5X,7HAMPERES,5X,9HPOWER, KW/
011H ALTERNATOR,50X,6HHEATER/4X,13H PHASE 1 (L-N),6X,F7.3,5X,F7.3,6X
R,F7.3,9X,6HBANK 1,13X,F7.3,5X,F7.3,6X,F7.3/4X,13H PHASE 2 (L-N),6X,
MF7.3,5X,F7.3,6X,F7.3,9X,6HBANK 2,13X,F7.3,5X,F7.3,6X,F7.3)
WRITE(10,400) GP3V,GP3A,GP3PR,HB3V,HB3A,HB3PR,GENPR,HTRPR,GFV,GFA,
WGFPWR
400 FORMAT(4X,13H PHASE 3 (L-N),6X,F7.3,5X,F7.3,6X,F7.3,9X,6HBANK 3,13X
F,F7.3,5X,F7.3,6X,F7.3/48X,7(1H-),53X,7(1H-)/48X,F7.3,53X,F7.3//4X,
05H FIELD,14X,F7.3,5X,F7.3,6X,F7.3/)
WRITE(10,410) TALOT,TTBOT,TJB1,TSMP,AWFLW,WPFLW
410 FORMAT(30H WATER COOLING LOOP CONDITIONS/1X,29(1H-)//14X,7HALT OUT
F,7X,8HTBRG OUT,6X,10HJBRG 1 OUT,5X,10HJBRG 2 OUT,8X,4HSUMP,9X,
09HALT FLOW,5X,10HPUMP FLOW,10X,5(4X,7HTEMP, F,4X),2(6X,3HCPS,6X)/
R10H VISUAL ,3(4X,F7.2,4X),15X,3(4X,F7.2,4X))
WRITE(10,420) (T(I),I=21,25)
420 FORMAT(10H DIGITAL ,5(4X,F7.2,4X)/)
WRITE(10,430) ETATU,ETACO,ETAGE,ETAHT,ETACR
430 FORMAT(21H COMPONENT PARAMETERS/1X,20(1H-)//6X,25HTURBINE EFFICIEN
FCY = F 5.3,5X,25HCOMPRESSOR EFFICIENCY = F 5.3,5X,25HALTERNATOR
O EFFICIENCY = F 5.3/6X,25HHEATER EFFICIENCY = F 5.3,5X,25HCOOL
RER EFFICIENCY = F 5.3,5X,18HRECUPERATOR EFFECT)
WRITE(10,440) ETAGS,ETACY,ERHOT,ERCLD
440 FORMAT(6X,25HGAS SYSTEM EFFICIENCY = F 5.3,5X,25H OVERALL SYSTEM EF

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FF      = F5.3,5X,25H      (HOT SIDE)      = F5.3/6X,25H (ALT OUT/G
OAS HEAT IN)  ,10X,25H (ALT OUT/HEATER IN)  ,10X,25H      (COLD
RSIDE)      = F5.3)
WRITE(IO,450) PRT,PRC,BETA,TBIN,TBOT,BFLW
450 FORMAT(/6X,25HTURBINE PRESSURE RATIO = F5.3,5X,25HCOMP PRESSURE RA
FTIO      = F5.3,5X,25HSYSTEM BETA (PRT/PRC) = F5.3//6X,25HBRINE INL
3ET TEMP, F      = F6.2,4X,25HBRINE OUTLET TEMP, F      = F6.2,4X,25HBRI
RNE FLOW RATE, LB/SEC = F6.3//)
WRITE(IO,460)
460 FORMAT(21H SKIN TEMPERATURES, F/1X,20(1H-)//8X,15HCOMP INLET DUCT,
F6X,14HHEATER SURFACE,4X,18HHEATER OUTLET DUCT,2X,18HTURBINE INLET
3DUCT/)
WRITE(IO,470) ( T(I),T(I+5),T(I+7),T(I+9) ,I=27,28)
470 FORMAT(12X,F7.2,13X,F7.2,13X,F7.2,13X,F7.2)
WRITE(IO,480) (T(I),I=38,43),(T(I),I=29,31)
480 FORMAT(/17X,19HRECUPERATOR SURFACE/6X,41(1H-)/6X,F7.2,10X,F7.2,
F10X,F7.2/21X,10HFRONT SIDE/6X,F7.2,10X,F7.2,10X,F7.2/6X,41(1H-)/
021X,11HBOTTOM SIDE/6X,F7.2,10X,F7.2,10X,F7.2/6X,41(1H-)//)
WRITE(IO,490)
490 FORMAT(23H CYCLE CHECK PARAMETERS/1X,22(1H-)//5X,8HGAS FLOW,2X,2(
F4X,10HCOMPRESSOR),6X,10HGAS SYSTEM,12X,14HASSUMED VALUES,13X,10HGA
OS SYSTEM/3X,12HRATE, LB/SEC,4X,10HSPEC SPEED,4X,11HSLIP FACTOR,4X,
R12HEFFICIENCY-1,4X,7HETACOMP,4X,7HETATURB,4X,7HRECPEFF,4X,12HEFFIC
MIENCY-2//)
WRITE(IO,500) ARFLW,CNSC,CSLIP,ETAG1,ETAG2
500 FORMAT(5X,F7.4,9X,F6.4,8X,F6.4,10X,F6.4,9X,4H0.72,7X,4H0.83,6X,
F5H0.905,8X,F6.4)
GO TO 1
1000 CALL EXIT
END

```

APPENDIX I (Contd.)

BRAYTON CYCLE DEMONSTRATOR COMPUTER CALCULATION FORMULAS

1. $T1(AVE) = \frac{T1_1 + T1_2 + T1_3}{3}$
2. $PT1(AVE) = \frac{PT1_1 + PT1_2 + PT1_3}{3}$
3. $PS1(AVE) = \frac{PS1_1 + PS1_2 + PS1_3}{3}$
4. $T2(AVE) = \frac{T2_1 + T2_2 + T2_3}{3}$
5. $PT2(AVE) = \frac{PT2_1 + PT2_2 + PT2_3}{3}$
6. $PS2(AVE) = \frac{PS2_1 + PS2_2 + PS2_3}{3}$
7. $T5(AVE) = \frac{T5_1 + T5_2 + T5_3}{3}$
8. $T6(AVE) = \frac{T6_1 + T6_2 + T6_3}{3}$
9. $PT6(AVE) = \frac{PT6_1 + PT6_2 + PT6_3}{3}$
10. $PS6(AVE) = \frac{PS6_1 + PS6_2 + PS6_3}{3}$
11. $T9(AVE) = \frac{T9_1 + T9_2 + T9_3}{3}$
12. $T9(AVE) = \frac{PT9_1 + PT9_2 + PT9_3}{3}$
13. $PS9(AVE) = \frac{PS9_1 + PS9_2 + PS9_3}{3}$
14. $T9P(AVE) = \frac{T9P_1 + T9P_2}{2}$
15. $T10(AVE) = \frac{T10_1 + T10_2 + T10_3}{3}$

APPENDIX I (Contd.)

$$16. \text{ ARGON FLOW RATE (GAFLW)} = \frac{10 \text{ EXP} \left[\frac{0.50232 \times \text{LOG}^{-1} \left[\frac{518.7 \times \text{P1A}}{14.694 \times (\text{TIA} + 459.7) \times \text{GFDP}} \right]}{2.302585 - 0.62289} \right]}{\text{(Based on duct calibrated } \Delta P)}$$

Taken from σ_p (vs) flow
log - log plot

$$17. \text{ FIELD POWER (KW)} = \frac{V_F I_F}{1000}$$

$$\text{GENERATOR FIELD POWER} = \text{GENERATOR FIELD VOLTAGE} \times \frac{\text{GENERATOR FIELD AMPERAGE}}{1000}$$

$$18. \text{ GENERATOR PHASE 1 POWER (KW)} = \text{GENERATOR PHASE I VOLTAGE} \times \frac{\text{GENERATOR PHASE I AMPERAGE}}{1000}$$

Similar calculation for Phases (2) and (3)

$$19. \text{ GENERATOR POWER (KW)} = \text{GENERATOR PHASE 1 POWER} + \text{GP2PR} + \text{GP3PR}$$

$$20. \text{ HEATER BANK 1 POWER (KW)} = \frac{\text{HEATER BANK 1 VOLTAGE} \times \text{HEATER BANK 1 AMPERAGE}}{1000}$$

SIMILAR CALCULATION FOR HEATER BANK (2) and (3)

$$21. \text{ HEATER TOTAL POWER (KW)} = \text{HEATER BANK 1 POWER} + \text{HB2PR} + \text{HB3PR}$$

$$22. \text{ BRINE FLOW RATE (LB/SEC)} = (0.017142) \times (\text{BRINE FLOW RATE}) + 0.07429 \times \frac{(0.1337) \times (95.0 - (0.0546) \times (\text{BRINE OUTLET TEM} - 60.0))}{60.0}$$

APPENDIX I (Contd.)

$$23. \text{ BRINE SPECIFIC HEAT} = 0.2203 + 0.0000667 \times (\text{BTU/LB } ^\circ\text{F})$$

$$\frac{\text{BRINE IN TEMP.} + \text{BRINE OUT TEMP.}}{2}$$

(READ PROPERTIES MAP)

$$24. \text{ ARFLOW} = \text{ARGON FLOW RATE BASED ON COOLER HEAT BALANCE}$$

$$25. \text{ ARFLOW} = \frac{\text{BRINE FLOW RATE} \times \text{BRINE } C_p \times (\text{TEMP. IN-TEMP. OUT}) \text{ BRINE}}{0.1244 \times (T_{10A} - T_{1A})}$$

$$\dot{W}_A = \frac{\dot{W}_B C_p \Delta T_B}{(C_p \Delta T)_{\text{Cooler Gas Side}}}$$

$$26. \text{ HEATER EFFICIENCY} = \frac{\text{GAFLW} \times 0.124 - (T_{6A} - T_{5A}) - 1.055}{\text{HEATER POWER}}$$

$$\eta_w = \frac{C_p \Delta T_{6-5}}{\text{KW}_e \text{ Heater}}$$

$$27. \text{ COOLER EFFICIENCY} =$$

$$\frac{\text{BRINE FLOW RATE} \times \text{BRINE } C_p \times (\text{TEMP. IN-TEMP. OUT}) \text{ BRINE}}{\text{GAFLW} \times 0.1244 \times (T_{10A} - T_{1A})}$$

$$28. \text{ COMPRESSOR PRESSURE RATIO (PRC)} = \frac{PT_{2A}}{PT_{1A}}$$

$$29. \text{ TURBINE PRESSURE RATIO (PRT)} = \frac{PT_{6A}}{PT_{9A}}$$

$$30. \text{ BETA} = \frac{\text{TURBINE PRESSURE RATIO}}{\text{COMPRESSOR PRESSURE RATIO}}$$

PRESSURE LOSS PARAMETER

$$31. \text{ TURBINE EFFICIENCY} = \frac{T_{6A} - T_{9A}}{(T_{6A} + 459.7) \times 1 - \left(\frac{1}{\text{PRT}}\right)^{0.4}}$$

$$32. \text{ COMPRESSOR EFFICIENCY} = \frac{(T_{1A} + 459.7) \times (\text{PRC}^{0.4} - 1)}{T_{2A} - T_{1A}}$$

APPENDIX I (Contd.)

$$33. \text{ ALTERNATOR EFFICIENCY} = \frac{\text{GENERATOR POWER}}{\text{GENPR} + 0.179 \times \left(\frac{\text{SPEED}}{85000}\right)^2 + 0.239 \times \left(\frac{\text{GEN PHASE I AMPS}}{15}\right)^2 + \text{GEN FIELD POWER}}$$

$$\eta_g = \frac{KW_e}{KW_e + \frac{N}{N}^2 \text{ Field Loss} + \left(\frac{I}{I_D}\right)^2 I^2 \text{ Phase} + KW_f}$$

$$34. \text{ GAS SYSTEM EFFICIENCY} = \frac{\text{GENERATOR POWER}}{\text{GAFLW} \times 0.124 \times (T6A - T5A) \times 1.055}$$

$$\eta_{cy_g} = \frac{KW_e}{T_{5-6}}$$

$$35. \text{ SYSTEM OVERALL EFFICIENCY} = \text{GAS SYSTEM EFFICIENCY} \times \text{HEATER EFFICIENCY}$$

$$\eta_{cy} = \eta_{cy_g} \times \frac{KW_T}{KW_{e_H}} = \frac{KW_e}{KW_{e_H}}$$

$$36. \text{ RECUPERATOR EFFECTIVENESS} = \frac{(T9PA - T10A)}{(T9PA - T2A)} \quad (\text{Based on hot side})$$

$$37. \text{ RECUPERATOR EFFECTIVENESS} = \frac{(T5A - T2A)}{(T9PA - T2A)} \quad (\text{cold side})$$

$$38. \text{ SCANNER VALVE REFERENCE (SCANR)} = (\text{VISUAL SCANR} + \text{AMBIENT PRESS.}) \times 0.4$$

$$39. \text{ VOLUMETRIC GAS FLOW RATE} = \frac{\text{ARFLW} \times 39.94 \times (TIA + 459.7)}{PTIA \times 144}$$

$$40. \text{ ISENTROPIC } \Delta H \text{ ACROSS COMPRESSOR (DHIS)} = 0.1244 \times (TIA + 459.7) \times$$

$$\left[(\text{PRC})^{0.4} - 1.0 \right]$$

$$41. \text{ COMPRESSOR SPEC SPEED} = \frac{\text{SPEED} \times (\text{VOL. GAS FLOW})^{0.5}}{119800 \times (\text{DHIS})^{0.75}}$$

$$42. \text{ COMPRESSOR SLIP FACTOR} = \frac{13.36 \times (T2A - T1A)}{\left(\frac{\text{SPEED}}{1000}\right)^2}$$

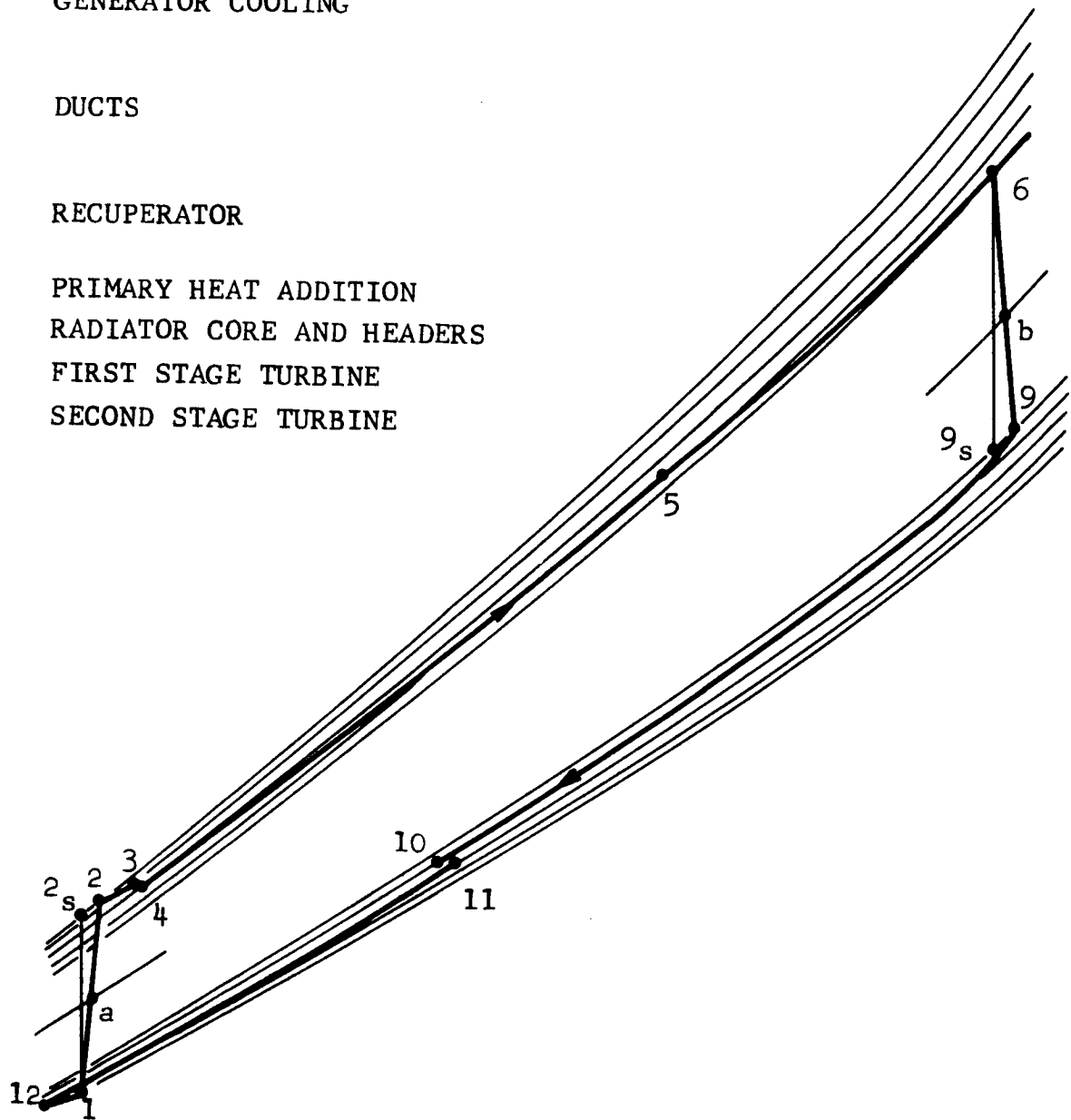
APPENDIX II
COMPUTER PROGRAM 532
(COMPATIBLE WITH CDC 3400 COMPUTER - FORTRAN IV)

Included in this appendix is the computer program flow diagram showing the calculational sequence and a brief description of the individual calculations.

This design program consists of over 120 design equations which relate the over 150 different cycle parameters. The complexity of the interrelationships between parameters in the closed recuperated gas cycle makes comprehensive cycle investigation and design optimization an impossibility without the computer. The basic cycle design program matches empirical turbine and compressor performance to the thermodynamics of the Brayton cycle to produce maximum cycle efficiency for each set of input design conditions. Pressure drops throughout the cycle are considered in the analysis, as are bearing and generator windage losses and the variations in working-fluid transport properties with temperature.

PROCESSCOMPONENT

1-a	FIRST STAGE COMPRESSOR
a-2	SECOND STAGE COMPRESSOR
2-3 or 12-1	GENERATOR COOLING
3-4 and 10-11	DUCTS
4-5 and 9-10	RECUPERATOR
5-6	PRIMARY HEAT ADDITION
11-12	RADIATOR CORE AND HEADERS
6-b	FIRST STAGE TURBINE
b-9	SECOND STAGE TURBINE



ENTROPY

CLOSED-RECUPERATED BRAYTON
CYCLE WITH COMPONENT PRESSURE DROPS

FIGURE 1

TABLE 1
INPUT QUANTITIES

ϕ	Generator type factor
C_{10}	Percent compressor bleed for cooling
E	Recuperator effectiveness
M	Gas molecular weight
T_1	Compressor inlet temperature
γ	Ratio of specific heats
N	Rotating speed
N_{s_c}	Compressor specific speed
X_1	Compressor first-stage work fraction
X_2	Turbine first-stage work fraction
T_{r_c}	Compressor total temperature ratio
T_e	Turbine inlet temperature
Z_g	Number of generator stages
$\frac{\Delta P}{P}$	Component fractional pressure drop
η_g	Generator efficiency
KW_1	Generator electrical output
KW_2	Bearing loss
KW_3	Electrical component cooling load
S_c	Compressor slip factor
S_{r_c}	Compressor polytropic work factor

TABLE 2
PRIMARY CALCULATED QUANTITIES

<u>Symbol</u>		<u>Diagram No.</u>
T_2	Compressor exit temperature	1
T_c	Average compressor temperature	2
C_{P_c}	Average compressor specific heat	3
θ_c	Average compressor $\theta = \frac{\gamma-1}{\gamma}$	4
Pr_c	Compressor total pressure ratio	5
P_2	Compressor discharge pressure	6
P_0	Turbine inlet pressure	7
P_3	Turbine exhaust pressure	8
μ_g	Generator gas viscosity	9
D_g	Generator diameter	10
T_{c1}	Average first-stage compressor temperature	11
$C_{P_{c1}}$	Average first-stage compressor specific heat	12
θ_{c1}	Average first-stage compressor θ	13
D_{c1}	First-stage compressor diameter	14
T_{c2}	Average second-stage compressor temperature	15
$C_{P_{c2}}$	Average second-stage compressor specific heat	16
θ_{c2}	Average second-stage compressor θ	17
D_{c2}	Second-stage compressor diameter	18
Re	Reynolds number	
KW_4	Generator rotor peripheral windage loss	20, 21
KW_5	Generator rotor radial windage loss	23, -26
KW_t	Total turbine output power	27
$P_{t_{t1}}$	Turbine total pressure ratio	28
T_9	Turbine exhaust temperature	29
s	rt/rc	30
T_{t1}	Average turbine temperature	31
$C_{P_{t1}}$	Average turbine specific heat	32
θ_{t1}	Average turbine θ	33
$\dot{m}_t$	Turbine mass flow rate	34
$\dot{m}_c$	Compressor mass flow rate	35
P_1	Compressor inlet pressure	36
Pr_{c1}	Compressor first-stage pressure ratio	37
Pr'_{c1}	Reference compressor first-stage pressure ratio	38

TABLE 3
CALCULATED QUANTITIES

<u>Item</u>	<u>Item</u>	<u>Diagram No.</u>
η'_{c1}	Reference compressor first-stage efficiency	39
η'_{cr1}	Reference compressor first-stage rotor efficiency	40
η'_{pc1}	Reference compressor first-stage polytropic efficiency	41
η_{pc1}	Compressor first-stage polytrop effieicncy	45
η_{cr1}	Compressor first-stage rotor efficiency	46
η'_{cd1}	Reference diffuser efficiency	47
η_{cd1}	Design diffuser efficiency	48
η_{c1}	Compressor first-stage efficiency	49
	(subscripts c_2 for second stage)	50 - 64
η_c	Overall compressor adiabatic efficiency	65
	(subscripts T11 for first-stage turbine)	63 - 84
	(subscripts T12 for second-stage turbine)	85 - 102
η_{t1}	Overall turbine adiabatic efficiency	103
T_4	Recuperator inlet temperature	106
T_{12}	Radiator exist temperature	108
T_5	Recuperator exit temperature	109
T_{10}	Radiator inlet temperature	110
η'_{cy}	Gas cycle thermal efficiency	112
η_{cy}	Net cycle electric efficiency	113

- (1) Calculate compressor discharge temperature, using a given compressor temperature ratio.
- (2)(3)(4) Calculate average compressor temperature, specific heat, and isentropic exponent.
- (5)(6) Calculate compressor pressure ratio and discharge pressure from assumed temperature ratio and compressor efficiency.
- (7) Starting with P_2 , calculate turbine inlet pressure P_8 , using the fractional pressure drops in the high-pressure-side components and ducts.
- (8) Starting with P_1 , calculate turbine discharge pressure P_9 , using the fractional pressure drops in the low-pressure-side components and ducts.
- (9) Calculate average generator viscosity as a function of temperature.
- (10) Using electrical output requirement and rotating speed, size the generator.
- (11)(12)(13) These parameters will differ from (2), (3), and (4) only if a two-stage compressor is desired.
- (14) Size the first-stage compressor rotor.
- (15)(16)(17) Same as (11), (12), and (13) for the second stage of the compressor.

- (18) Size the second-stage compressor rotor,

Between (18) and (19), a few checks are made as to the type of generator and relative rotor diameters of the compressor and generator. If the generator rotor is smaller than the compressor rotor, and if an axial-gap inductor generator has been selected to be integrated with the compressor, the generator windage loss calculations are by-passed. If not:

- (19) through (26) Calculate the peripheral and side windage losses chargeable to the generator.

- (27) Calculate the total turbine shaft power output.

$$KW_t = \frac{KW_1}{\eta_g} + \text{windage} + \text{bearing} + \text{parasitic losses}$$

- (28) through (33) Calculate turbine temperatures, pressures, and gas properties.

- (34) From the turbine shaft power requirement, compute the turbine mass flow requirement.

- (35) Compressor mass flow rate same as (34) unless bleed gas is required for generator or bearing cooling.

- (36) From $\dot{m}_c$, calculate the compressor inlet pressure

$$P_1 \propto \left(\frac{N}{N_{s_c}} \right)^2 \dot{m}_c T_1 (M)^{1/2} \left[\frac{1}{r_c (T_2 - T_1)} \right]^{3/2}$$

This equation shows the effect of compressor specific speed on the system pressure level.

- (37) through (65) These calculations establish the pressure ratio for the reference air compressor corresponding to the pressure ratio of the system being designed for use of monatomic gas. The reference air compressor efficiency η'_{c1} is determined from empirical data. Corrections are then made for Reynolds number and change in specific heat ratio of the working fluid. Two stages can be designed in this manner. The diffuser efficiencies are included and an overall compressor adiabatic efficiency is determined, Equations (65), η_c .
- (66 through (103) These calculations are a design routine for one or two turbine stages using empirical test data for a reference air turbine. The approach is similar to the compressor design approach discussed above.
- (14) At this point the turbine matches the compressor. A check is made to see if the turbomachinery matches the thermodynamic cycle. If not, an iteration is made until the turbomachinery designs fit the thermodynamic cycle.
- (105) through (111) The remaining cycle state point temperatures are calculated.

(112)(113) The cycle efficiency is calculated. The net cycle efficiencies (η_{cy}) are plotted, a curve fit made, and the maximum efficiency determined for a particular overall cycle temperature ratio.

(114) through (123) The remaining state point pressures are calculated. Four final checks are made: turbine tip speed, compressor inlet Mach number, generator power factor, and compressor diffuser surface-finish requirement.

This completes the calculation for one case. The calculation is repeated if additional input variable values exist.



APPENDIX III

SUMMARY TEST LOG

BCD TEST PROGRAM

Initial BCD checkout tests were performed between July 1966 and January 1967. The tests during this period are summarized as follows:

- July 1966 - Loop fabrication was completed. The CRU was tested in the open loop test cell. Full field excitation was applied at a rotating speed of 42,000 rpm. A 900 watt load was applied.
- August 1966 - Completed closed loop support system checkout. Installed CRU in closed loop and achieved first gas injection spin up of the CRU to a rotating speed of 11,000 rpm. A start-up was made and the first self-sustained operation of the CRU in the closed loop was achieved. CRU removed from closed loop for further open loop testing.
- October 1966 - The CRU was operated at 64,000 rpm in the open loop. Hydrodynamic CRU starts were attempted but were not successful.
- November 1966 - Applied magnetic shock loads to CRU bearing system operating in the open loop test facility, at rotating speed of 62,000 rpm. Evaluated cycle performance at 60,000 rpm and, because of the results, changed the design point to 60,000 rpm.



January -

Worked on optimization of gas-injection starts in the closed loop. Reduced start-up time from 15 minutes to less than 1 minute. Achieved 1 hour continuous run in closed loop, with Hydrodynamic bearings. Pulled power in closed loop--800 watts at 55,000 rpm.

The following tests are numbered to show the series of tests performed before and after a major modification was made to the CRU. Tests 1 through 11 includes the tests performed before the modification to the alternator housing (for details refer to Section 2.2.1.1). The "B" series, beginning with test 12B, was performed after the modification was made.

Test No. 1, 1-5-67
Duration -- 5 min

The BCD was started and achieved a speed of 40,000 rpm. The automatic speed control system did not function, the test was aborted, and a shutdown was made.

Test No. 2, 1-5-67
Duration -- 20 min

The BCD was started and again the speed count was erratic. Hydrostatic gas was removed prematurely due to the erroneous speed count. At low speed the compressor pressure ratio and hence the bearing cavity pressure was low and a bearing failure resulted.

In all of the preceeding tests the thrust bearing configuration was single-sided. That is, the thrust bearing would carry load only in the direction of the turbine. Open loop testing of the CRU, in parallel to the closed loop tests, resulted in development of a satisfactory double-sided thrust bearing configuration. This thrust bearing was employed in all of the tests thereafter.



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Test No. 3, 1-17-67
Duration -- 5 min

Erratic speed count after the start-up caused the overspeed switch to trip, resulting in an emergency shutdown. The automatic controls performed as planned, hydrostatic gas was supplied to the bearings, the heaters shut off, and the gas dumped from the loop. A safe shutdown resulted.

Test No. 4, 1-17-67
Duration -- 5 min

For this test manual speed control was attempted. CRU acceleration was very rapid and 64,000 rpm was reached before enough load could be applied to check acceleration. The test was aborted.

Test No. 5, 1-18-67
Duration -- 30 min

Good start-up, using manual speed control. Loaded generator to about 700 watts. Improper limit settings on heater control blew heater fuses resulting in a controlled shutdown. BCD operated at 52,000 rpm.

Test No. 6, 1-18-67
Duration -- 57 min

This test resulted in the first sustained closed loop operation. The rotating speed was 47,000 rpm. An excessive temperature droop occurred during start up. Attempts to accelerate were unsuccessful. Off-design-point inefficiency and limited heater capacity prevented CRU acceleration.

Test No. 7, 1-19-67
Duration -- 5 min

Increased the starting temperature and initial power settings to effect a rapid start up. Start-up in less than 60 sec. and changed over to hydrodynamic bearing operation. When the loop was closed, very rapid acceleration occurred. Excessive alternator load was applied manually, and speed fell below safe hydrodynamic operating range. Failed thrust bearings.

Test No. 8, 1-26-67
Duration -- 10 min

Again, the automatic speed control system was employed. It performed satisfactorily but excessive T_b droop occurred and speed could not be maintained. A controlled shutdown was made.



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Test No. 9, 1-26-67
Duration -- 7 min

A good start-up was made. At 55,000 rpm and $T_8 = 1180^{\circ}\text{F}$, a turbine rub occurred causing journal and thrust bearing failures.

During February of 1967 two-closed-loop tests were conducted. In both, problems developed in the automatic load control system. Added negative regulator for bearing cavity pressure control

Test No. 10, 2-2-67
Duration -- 5 min

Excellent spin up performed in less than one minute using the gas injection system. A rotating speed of 60,000 rpm with a $T_8 = 1300^{\circ}\text{F}$ was realized. When hydrostatic gas was cut off the bearing cavity pressure dropped below safe limits. Reason unknown. Sustained journal bearing failure.

Test No. 11, 2-8-67
Duration -- Unknown

No speed count was received during attempt to start-up. Disassembly of unit revealed that the CRU had been rotating at very high speeds (estimated at over 70,000 rpm). The unit had been rotating when the bearing gas supply was shut off and the loop evacuated. Both the journal and the thrust bearings failed at very high speed. Rather extensive CRU damage was sustained, including a twisted and bent shaft.

March and April, 1967

During this period the necessary CRU overhaul was accomplished. A major design improvement was made, in the form of a redesign of the alternator stator housing. This is discussed in more detail in Section 2.2.1.1.



In addition to the CRU modification, a change was made to the speed measurement equipment. The "monopole" pickups were replaced with a "Bentley" probe system. The Bentley system involves a very high frequency (several kilocycles) carrier signal. This signal is less effected by the low frequency interference which emanates from the heater power supply and the CRU alternator. It can be seen from reviewing the tests described below that this modification did not entirely provide satisfactory speed control. The speed control system was subsequently overhauled between tests 16B and 17B. It performed very well thereafter.

Test No. 12B, 5-21-67
Duration -- 10 min

To verify CRU rpm, two additional pieces of instrumentation were added. The alternator voltage was displayed on an oscilloscope. A special Bently probe test device was built to check out the speed count system prior to start-up. A start-up of the BCD in the closed loop was made but the speed count was still erratic. A controlled shutdown was made.

Test No. 13B, 5-21-67
Duration -- 10 min

Worked on speed control system. Made a start, however automatic loading did not occur as desired. The CRU reached a rotating speed of 67,000 rpm with a $T_e = 1300^{\circ}\text{F}$ before acceleration was stopped. The machine was stopped, using manual alternator loading. Operated for a few minutes using manual loading mode before shutting down the system.

Test No. 14B, 6-5-67
Duration -- 30 min

Very good test initially, with fully automatic control at 50,000 rpm and $T_e = 1300^{\circ}\text{F}$. Began to experiment with bearing cavity control--experienced pressure cycling and consequently failed thrust bearing.



Test No. 15B, 6-29-67
Duration -- 120 min

Experienced problems with automatic speed control. Changed to manual control mode and ran for 2 hours at a speed of 47,000 rpm and $T_6 = 1350^{\circ}\text{F}$.

Test No. 16B, 6-30-67
Duration -- 30 min

Test was performed to troubleshoot the speed control system. Ran through three gas-injection starts using automatic speed control system. Encountered control system problem and sustained three loop dumps because the overspeed switch was tripped. No bearing damage resulted, however, since emergency system functioned as planned.

Following test 16B, a decision was made to extensively redesign the speed control. The control gain was consequently reduced to eliminate the instability that had been manifest in previous tests.

Test No. 17B, 7-8-67
Duration -- 4 hrs.

Good start up to a speed of 50,000 rpm and $T_6 = 1400^{\circ}\text{F}$. Excellent test, automatic speed control functioned properly. Test terminated due to burned-out heater terminals.

Test No. 18B, 7-12-67
Duration -- 2 hrs.

Very successful test. Operating temperature was 1350°F and operating speed was 50,000 rpm. Uneventful shutdown.

Test No. 19B, 7-13-67
Duration -- 3 hrs

After 3 hours of operation, attempt to increase P_1 resulted in thrust bearing failure due to overload.

Test No. 20B, 7-19-67
Duration -- 70 min.

A modification was made to the thrust bearing configuration to prohibit rotation between the thrust foils and the supporting spring. After a successful start the BCD was operated for over 1 hour at 50,000 rpm and a $T_6 = 1325^{\circ}\text{F}$. Then a thrust bearing failure occurred.



During the month of August 1967 an endurance test was accomplished. The original objective was 100 hours of continuous operation. During the course of the test problems developed with the laboratory support equipment, causing interruption of the test. These problems did not adversely effect the BCD and the CRU was not removed from the closed loop during the entire test. The test was terminated at the end of 75 hours on Saturday, August 12, 1967. The results of the test were presented to the 2nd Annual Intersociety Energy Conversion Engineering Conference on August 13th, 1967, in Miami, Florida.

Test No. 21B, 8-7-67
Duration -- 75 hrs.

Made start-up and established operating conditions at a speed of 50,000 rpm, and $T_s = 1350^{\circ}\text{F}$. At the end of 3 hours a vacuum pump failure occurred. A controlled shutdown was made. Repairs were made and the test resumed. At the end of 32 hours a heat sink coolant leak was detected and several heater elements had burned out. The test was again temporarily halted. The leak was repaired and the heater elements replaced. The test was again resumed and continued into the 75th hour on 8-12-67.

During the 75-hour test, support-system failures caused shutdown of the BCD. Three times the CRU was brought to a stop hydrodynamically (no external bearing pressurization). At the conclusion of the 75-hour test, these same bearings were still operating smoothly, and the CRU was still operable.

A considerable amount of test data was taken during the 75 hour test. The results are presented in Section 4.0.