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6 KW SYSTEM EVALUATION AND ENDURANCE

A.E.C. ORGANIC RANKINE CYCLE
TECHNOLOGY PROGRAM
TOPICAL REPORT

SEPTEMBER 8, 1972

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PREPARED BY

R. E. NIGGEMANN
GROUP ENGINEER

AVIATION DIVISION
SUNDSTRAND CORPORATION

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APPROVED:

S. S. B. 9-13-72
S. S. Baits
Chief Development Engineer

J. Johnson
Contract Coordinator

MASTER

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FOREWORD

This report was prepared by Sundstrand Aviation, a Division of Sundstrand Corporation, in accordance with the requirements of Contract AT (04-3)-651. Mr. Ron Anderson, Division of Space Nuclear Systems, USAEC, Washington, D.C., administers the program, while Mr. R. E. Niggemann serves as Project Engineer at Sundstrand. Mr. J. Petersen is the Contract Program Manager for Sundstrand. Messrs. T. J. Bland, and G. W. Hill are the significant contributors to this report.

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1.0 INTRODUCTION AND SUMMARY

The Sundstrand 6 KW organic Rankine cycle power conversion system is, in part, an outgrowth of the work performed by Sundstrand for the Air Force under Contract No. 33(615)-3664. The objective of that effort was to develop a turboalternator pump mounted on a single shaft which, through minor changes, would produce 2 to 10 KW of useful power. However, the Air Force cancelled this program in the fall of 1967, as the result of a decision to restrict its activity in the space power field.

Sundstrand proceeded independently to develop the turboalternator pump and incorporate it into a complete 6 KW fixed power output system for performance and endurance testing. Also, a newly developed jet condenser was configured into the system for evaluation.

In April, 1969, after initial testing had begun, the Atomic Energy Commission contracted Sundstrand for the remainder of the testing program and extended endurance testing.

This report describes the results of the testing program performed as an extension to Contract AT(04-3)-651 Organic Rankine Cycle Technology Program. The purpose of the contract extension was to perform the following:

- Record performance data for a 6 KWe organic Rankine cycle power conversion system utilizing both a surface condenser and a jet condenser using Dowtherm A working fluid at 650°F turbine inlet temperature and .5 psia turbine back pressure.
- Record performance data for the 6 KWe organic Rankine cycle power conversion system using a jet condenser and Dowtherm A working fluid at 700°F turbine inlet and .5 psia turbine back pressure.
- After a performance and shakedown run of approximately 500 hours the 6 KWe power conversion system was to be endurance tested for 2500 hours using the jet condenser configuration with Dowtherm A working fluid at 700°F turbine inlet condition. A pre-test and post-test documentation of the condition of the hardware was to be performed.

The 6 KWe organic Rankine cycle system was operated with both a surface condenser and a jet condenser at 650°F turbine inlet temperature. Performance measurements indicate that with reworked regenerators, and in the case of the jet condenser configuration, with a pump designed for the specific operating conditions, both systems would produce a system efficiency of over 17%.

The 700°F turbine inlet-jet condenser version was operated for over 3000 hours. All rotating equipment was in excellent condition upon completion of the test.

2.0 POWER CONVERSION SYSTEM DESCRIPTION

The Surface Condenser configured 6 KWe system shown in Figure 1 consists of the energy conversion machinery used to convert heat energy from vaporized Dowtherm A to electrical energy and the heat transfer and pumping components needed to add and remove heat from the system.

This section describes the essential components of the power conversion system configured with a surface condenser for performance evaluation at 650°F.

2.1 ENERGY CONVERSION MACHINERY

The energy conversion machinery comprises a turbine, electrical alternator and vaporizer feed pump on an integral shaft. The shaft is enclosed within the alternator stator, bearing, pump and turbine housings. The housings are sealed to preclude the inleakage of air. An exploded view of the 6 KWe power conversion machinery is shown in Figure 2.

2.1.1 Turbine and Nozzle Assembly

The turbine is a single stage, supersonic, axial flow impulse, full admission design. The nozzles are a conical, convergent-divergent design. Figures 3 and 4 show the nozzle ring and turbine wheel. Table 1 presents a summary of the turbine and nozzle system parameters.

The wheel was designed to drive a generator to produce 6 KWe of electrical power at 24,000 rpm with an inlet vapor temperature of 650°F and a back pressure of 0.5 psia.

Turbine buckets were electrical chemical machined from a MS1407 steel wheel forging. The nozzle ring is machined from AISI-303 steel.

2.1.2 System Generator and Eddy Current Brake

The electrical generators for the 6 KW system consists of two homopolar alternators on an integral shaft contained in the same enclosure. These generators were originally designed for another application and were adapted for use in the 6 KW system. One alternator provides useful electrical output and the other is an eddy current brake for speed control.

The system generator is a double stator homopolar inductor alternator with six laminated poles of 3.7 inches diameter on the rotor.

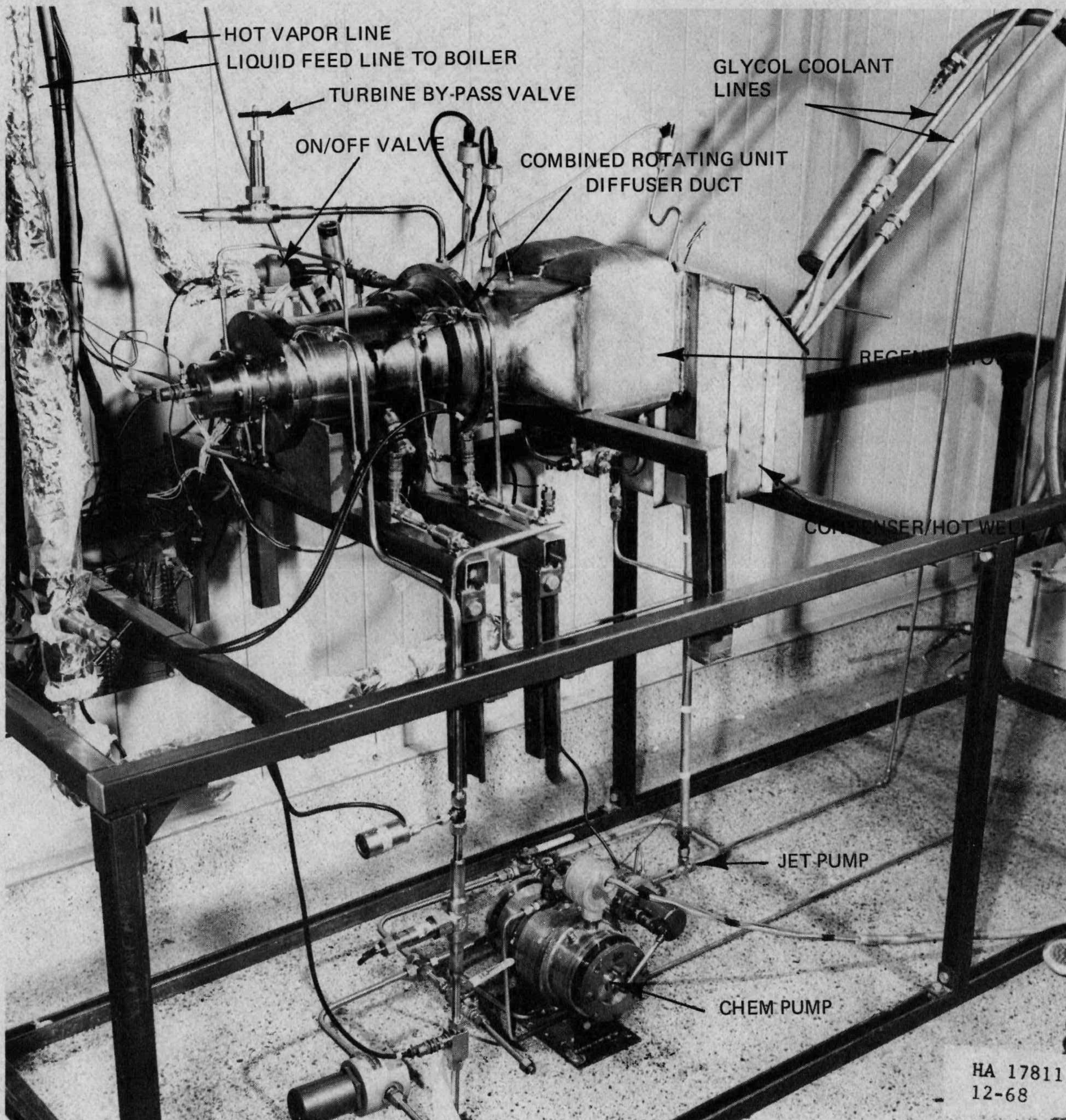


Figure 1 Sundstrand 6 KWe Power Conversion System

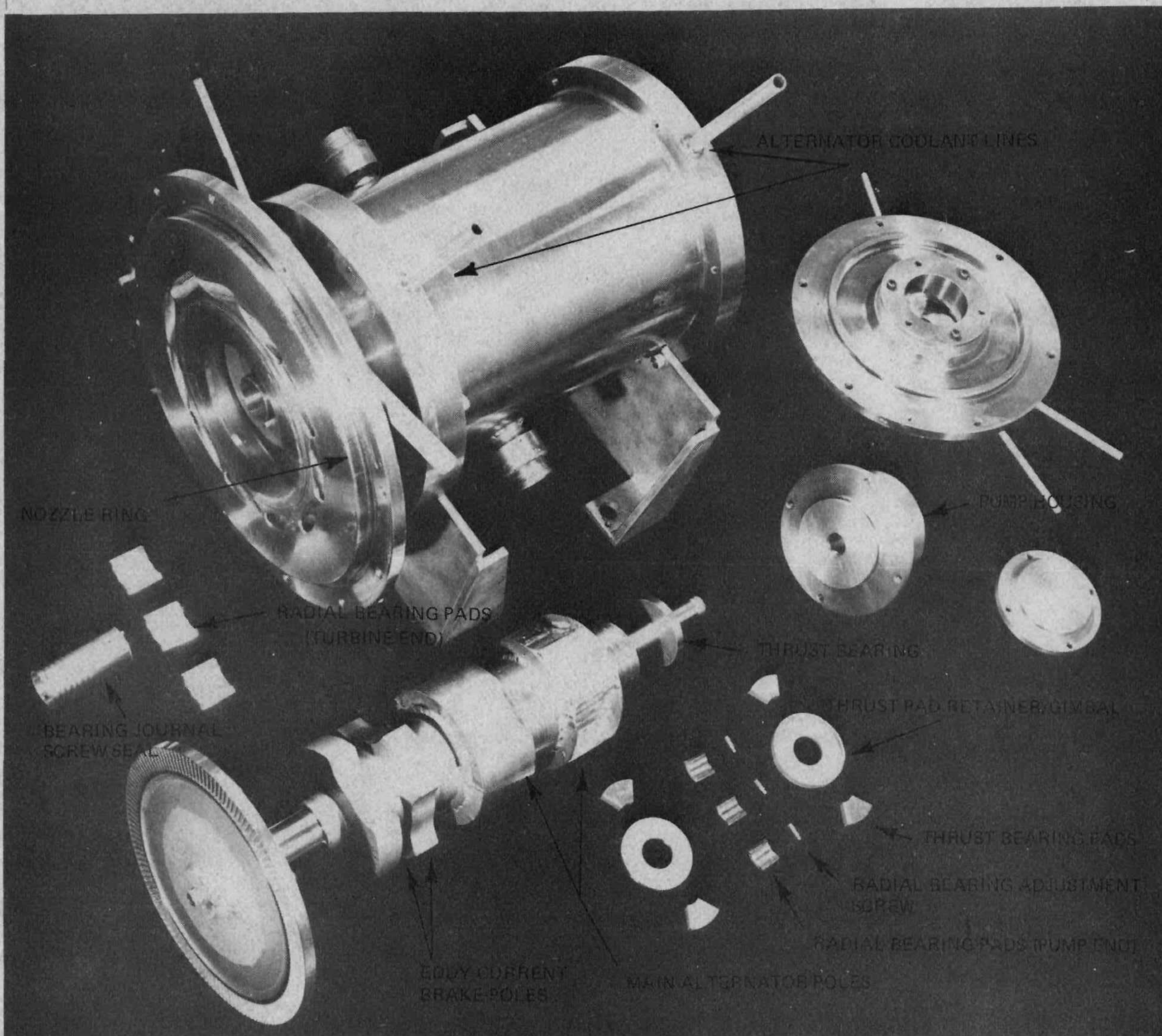


Figure 2 Sundstrand 6 KWe Combined Rotating Unit

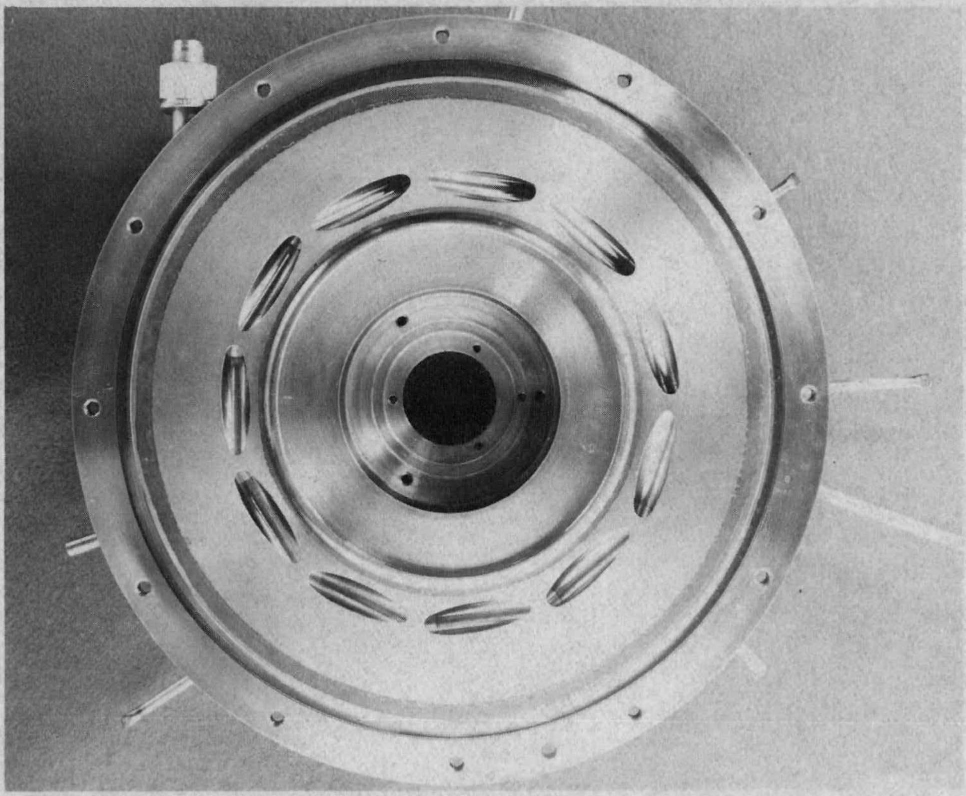


Figure 3   6 KWe Nozzle Ring

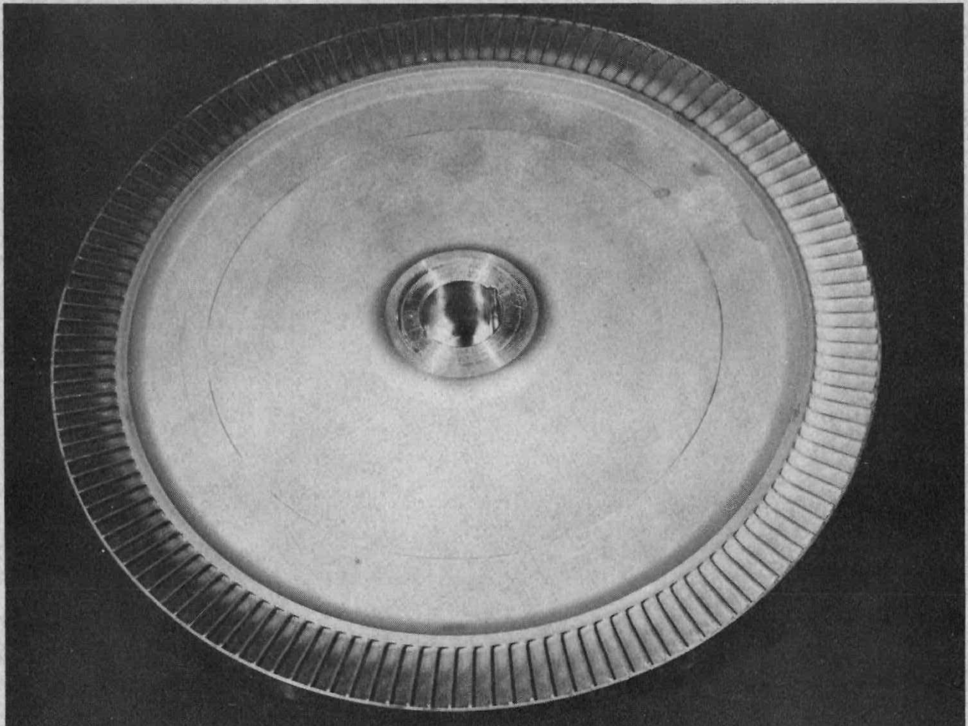


Figure 4   6 KWe Turbine

Table 1 Turbine Design Parameters

Number of nozzles	11
Arc of admission	100%
Nozzle type	Conical, convergent-divergent
Throat diameter	.090 inches
Nozzle angle	15 degrees
Inlet pressure	71.3 psia
Inlet temperature	1110°R
Mass flow	.164 lb/sec.
Turbine wheel specific speed	34.2
Turbine wheel specific diameter	1.74
Turbine diameter	7.1 inches
Blade height	.54 inches
Blade angle	25°
Rotor inlet relative mach no.	1.9
Rotor inlet Reynolds No.	1.1×10^4
Turbine efficiency at 650°F and 0.5 psia back pressure	72%

It generates 240/139 volts at 1200 Hz and is capable of producing 10 KWe. The output windings consist of a three phase power winding and an auxiliary winding for field and control power. At 6 KWe output it has an efficiency of 91.9%. A voltage regulator is incorporated in the control console for maintenance of the design conditions. Figure 5 presents a short summary of the system main alternator design.

The eddy current brake is a homopolar inductor alternator with no power windings. All the generated energy is dissipated as heat in the stator iron and rotor. It is capable of up to 2 KWe at 24,000 rpm. The speed controller adjusts the field current to this alternator, varying the work done and hence shaft power, to balance the turbine input torque. Figure 6 presents a short summary of the eddy current brake design.

Type:	Double stator homopolar inductor alternator with three phase power winding and one auxiliary winding for field and control power.
Rating:	Design - 10 KW, 20,000 rpm, 1000 Hz, 240/139 volts, .95 pf lag. 6 KWe Requirements - 6 KWe, 24,000 rpm, 1200 Hz
Efficiency:	89.4% at 10 KWe, 91.5% at 6 KWe
Cooling:	Provided by circulation of working fluid flow through stator cooling passages.

Figure 5 Main Alternator Design Summary

Purpose:	Turbine speed control
Type:	Double core homopolar eddy current dissipator
Rating:	Maximum of 2 KW at 24,000 rpm, 1200 Hz
Cooling:	Provided by circulation of working fluid flow through stator cooling passages

Figure 6 Eddy Current Brake Design Summary

Cooling of both alternators is achieved by passing the working fluid flow over the back of the stator iron through axial slots, prior to delivery of this flow to the regenerator inlet.

The alternator rotor showing the main generator poles (laminated) and the brake alternator poles is shown in Figure 7.

The alternator stator is shown in Figure 8.

2.1.3 Working Fluid Pump

The pump which provides system flow is mounted directly to the turbo-alternator shaft. It is a simple centrifugal pump consisting of radial slots cut into a rotating disc shown in Figure 9.

The housing incorporates an annular collector and a conical diffuser with a throat diameter of .100 inch. This pump has

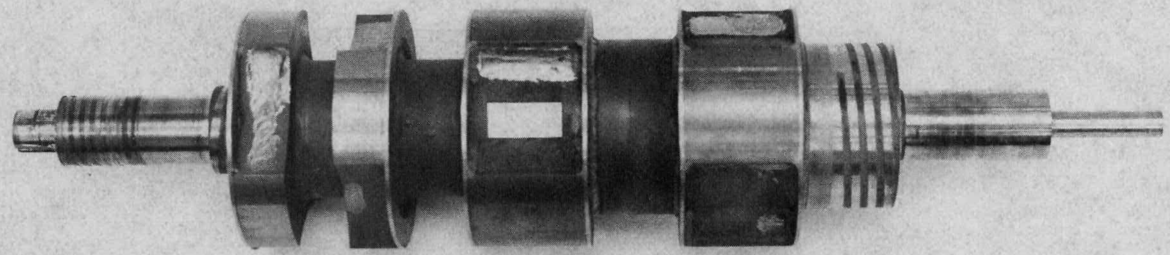


Figure 7 Alternator Rotor

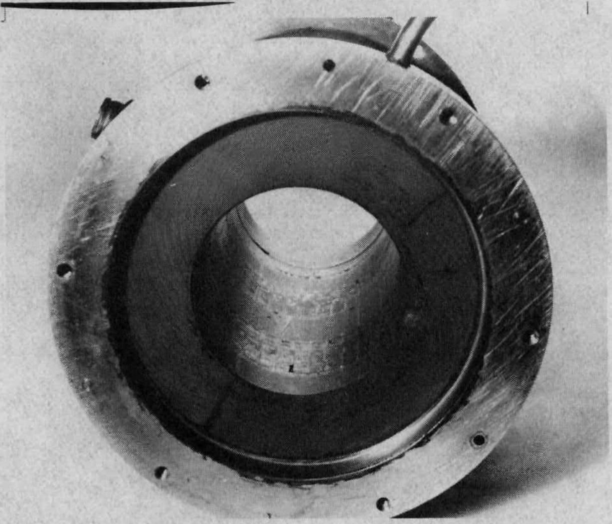


Figure 8 Alternator Stator

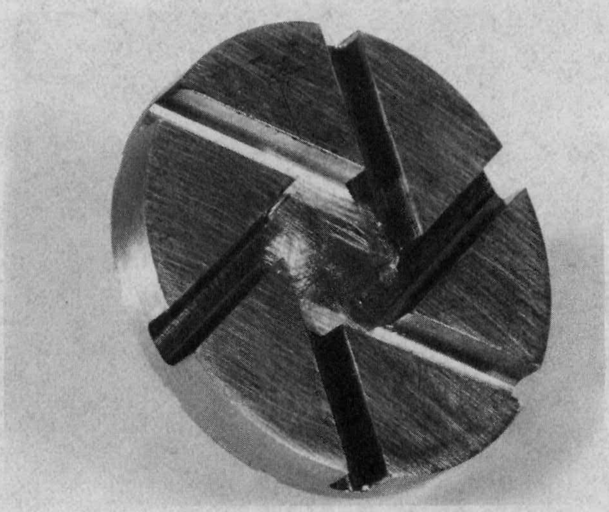


Figure 9 Centrifugal Pump Impeller

been tested on a pump test rig at Sundstrand and has demonstrated an efficiency of 32%.

2.1.4 Bearings and Seals

The radial bearings supporting the weight of the turbo-alternator rotor and unbalance forces are tilting pad fluid film bearings. These are of a three pad design in which the static load vector passes between two pads. The pads, shown in Figure 10, are free to tilt on spherically tipped adjustment screws. The bearing journals may be seen on the rotor photograph, Figure 11. The material combination is tool steel against tool steel, which has shown excellent results under start-stop conditions at Sundstrand.

Support of thrust loading, which is low in this machine, is accommodated by fully self-aligning tilting pad bearings. All bearing pads (both radial and axial) are individually lubricated by jets of working fluid and the bearing housing is actively scavenged to preclude high liquid churning losses.

Non-contacting Visco seals are provided around the bearing cavities to prevent leakage into the alternator cavity and turbine wheel housing.

2.2 ADDITIONAL SYSTEM COMPONENTS

The boiler, regenerator, condenser, and the facility and jet pumps are additional major system components needed to complete the system.

2.2.1 System Boiler

The boiler for the power system consists of a tube and shell heat exchanger in which the tube side fluid is the power system flow of Dowtherm and the shell side fluid is high pressure Dowtherm vapor which condenses on the boiler tube. The hot (700°F) high pressure (106 psia) Dowtherm vapor is supplied by a commercial, gas fired Dowtherm vaporizer. The boiler tube is 72 feet of 7/8 inch tubing arranged in 6, 12 foot passes inside a 4 inch standard pipe size steel pipe.

2.2.2 Regenerator

The regenerator assembly is shown in Figure 12 attached to the turbine exhaust housing diffuser and transition duct. The regenerator is made up of two modules, one of which is shown on Figure 13. The regenerator is a plate fin design with one vapor pass and 8 crossflow liquid passes arranged in an overall counter-flow configuration. The outer shell, headers, plates, and channels are of 347 stainless while the fins are 0.002 inch thick nickel foil with 0.233 inch high fins, 18.5 fins per inch on the vapor

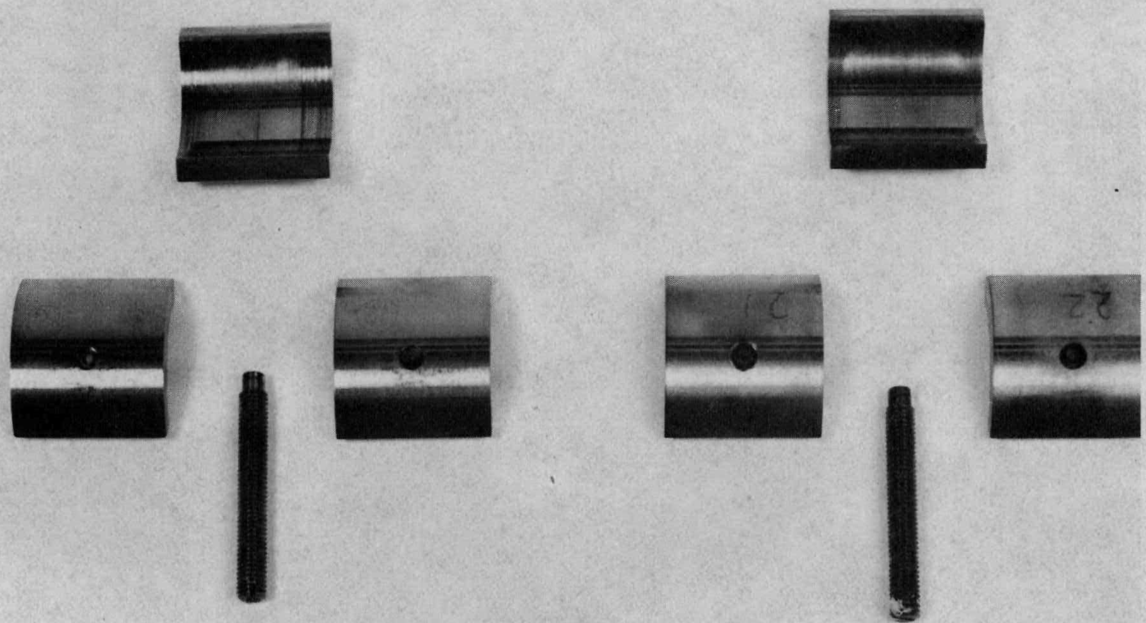


Figure 10 Tilting Pad Fluid Film Bearing

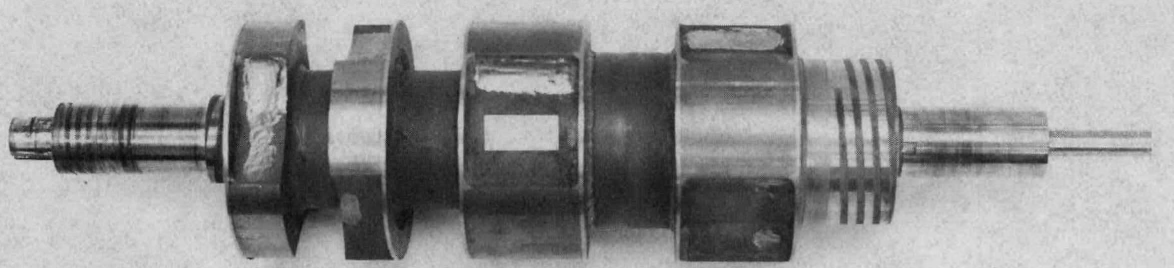


Figure 11 Alternator Rotor

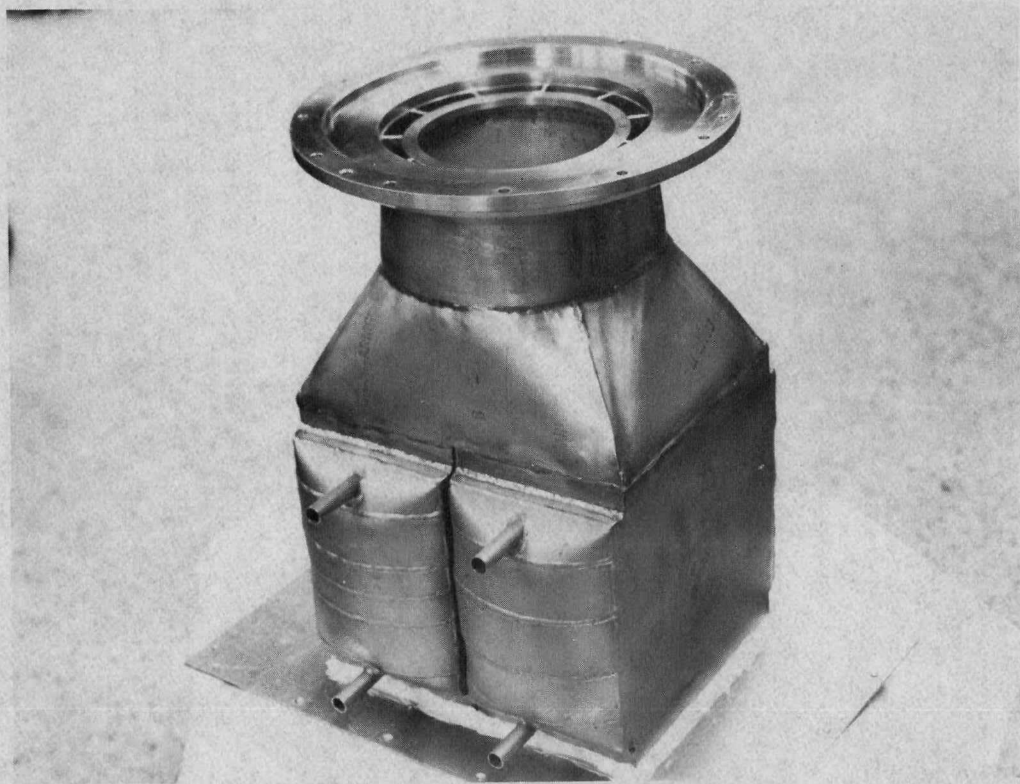


Figure 12 Regenerator and Turbine Exhaust Ducting

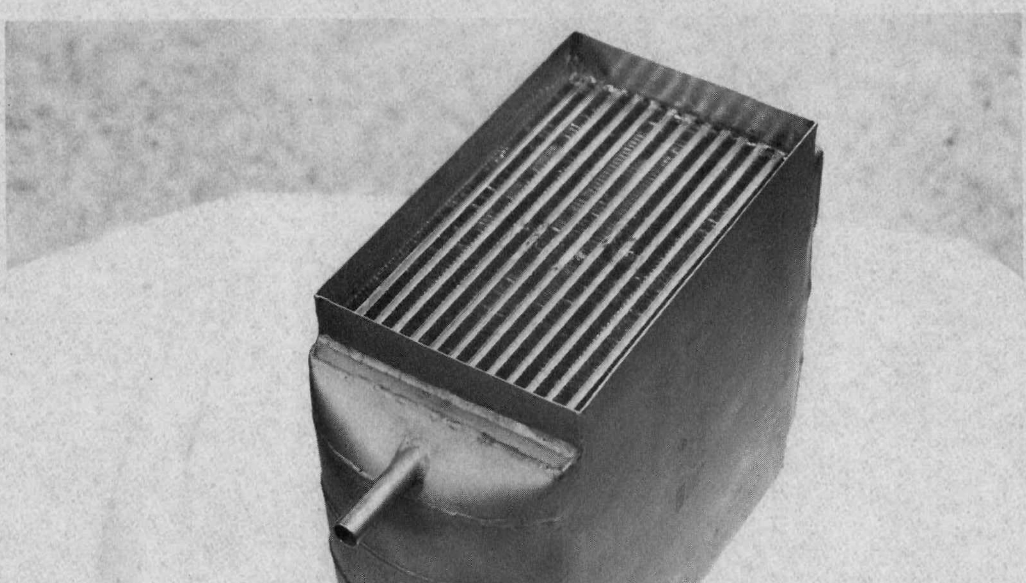


Figure 13 Regenerator Module

side and 0.145 inch fins, 19.5 fins per inch on the liquid side. The unit has a design effectiveness of 0.9 and has a pressure drop of 0.05 psi on the vapor side.

2.2.3 Condenser

The surface condenser is an ethylene glycol/water cooled tube in shell design. To minimize weld joints only 8 parallel coolant tube circuits were used. Arrangement of these tubes is shown in Figure 14.

These circuits consist of serpentine coils of 1/4 inch stainless steel tubing and were manufactured by Rudy Manufacturing Company, a Sundstrand subsidiary. The condenser design effectiveness is 0.45, and the vapor pressure drop is estimated at 0.02 psi, while the glycol pressure drop is 5 psi at 1460 lb/hr. The condenser housing and hotwell are fabricated from stainless steel sheet.

The condenser, shown in Figure 15, will be attached directly to the regenerator with the vapor passing over the tubes in a vertical direction, and collecting in the hotwell directly below the tubes.

2.2.4 Auxiliary Equipment

2.2.4.1 Facility Start Pump: The pump used for starting the system is a commercial, canned rotor, regenerative type centrifugal pump. This runs on pressurized, process fluid lubricated journal bearings and has demonstrated lifetimes in excess of 8000 hours in previous ORC testing at Sundstrand.

2.2.4.2 Jet Pumps: The jet pumps used for priming the facility start pump and the system pump have demonstrated their ability to meet predicted performance. They have a nozzle size of 0.074" and a diffuser throat size of 0.235". They have demonstrated operation at a primary/secondary flow ratio of 1.2 and provided pump inlet pressures up to 8.5 psia. The jet pump used for the bearing scavenge pump is existing and is somewhat oversized. It has a nozzle diameter of .062" and throat diameter of .135", and operates at primary/secondary flow ratios of approximately 4.0.

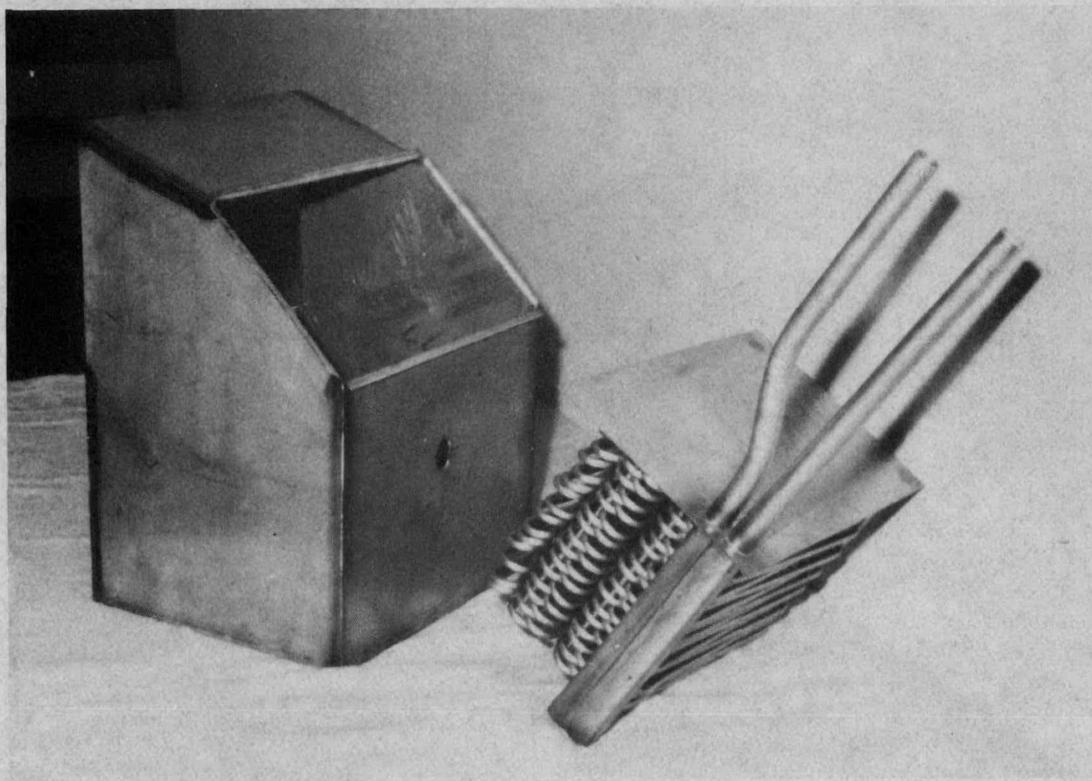


Figure 14 Glycol Tube Circuits

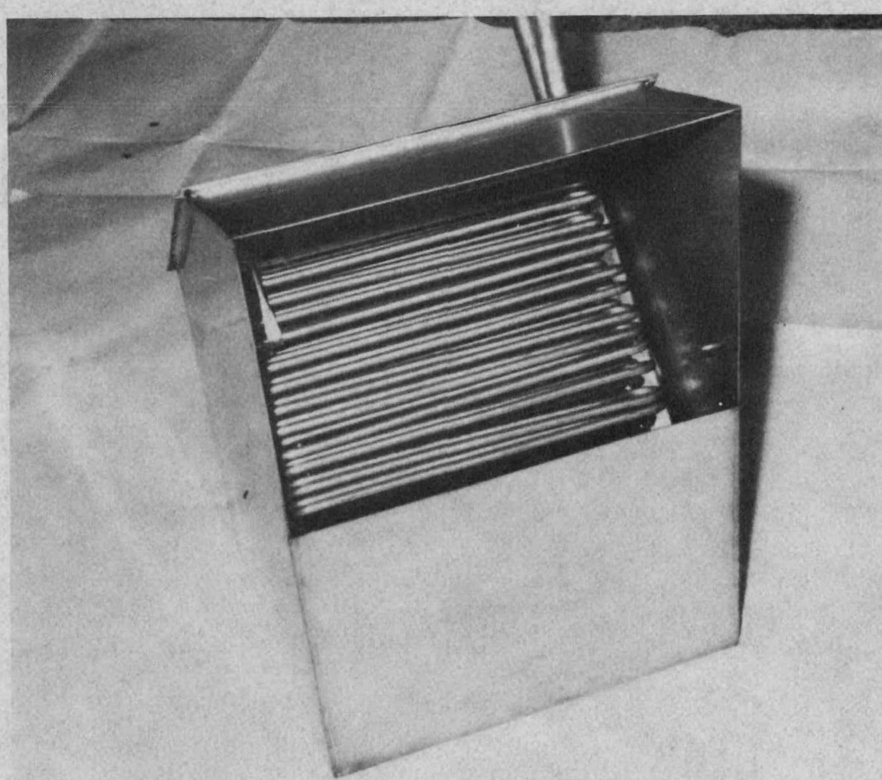


Figure 15 Condenser in Housing

3.0 SYSTEM OPERATION (SURFACE CONDENSER CONFIGURATION)

The following is a basic description of operation of the Sundstrand 6 KW organic Rankine cycle surface condenser configured system for the testing program. The sequence of operation is conveniently separated into four distinct functional loops that combine to form the complete system. Loop #1 is the primary loop with the essential power conversion components involved. Loop #2 is the turbine by-pass loop used for system start-up and shutdown. Loop #3 insures proper lube at all times to the bearing cavities. Loop #4 rejects the waste heat produced by the power conversion system and load bank. Each loop is briefly described with continual reference to Figure 16.

The primary loop (loop #1) represents the flow circuit which carries working fluid through the turbine to generate shaft power. Starting at the hot well (A) the working fluid flow enters the secondary side of a jet pump (B) with sufficient elevation head to prevent cavitation when it is accelerated into the pump throat by the high velocity primary flow. The flow from the hotwell passes through the diffuser in the jet pump and combines with the primary flow gaining sufficient static pressure to meet the net positive suction head requirements of the system feed pump (C).

The working fluid is then pressurized in the centrifugal system pump and passes through a one way check valve (D) which prevents back flow through the pump during start up operation. The flow then splits (E), with approximately half proceeding to the primary side of the system jet pump and the remaining proceeding to the alternator stator (F). Bearing flow is provided by splitting the flow at (H) when the pressure is sufficient for bearing requirements.

On entering the alternator stator housing the flow passes along axial slots and removes waste heat from the system generator and eddy current brake. On leaving the stator housing, the flow passes through a throttle valve (V_1), used to control mass flow into the vaporizer.

The liquid flow then passes through the liquid side of the regenerator in a counterflow direction and is warmed by the superheated turbine exhaust vapors. On leaving the regenerator, the flow passes through a check valve (G) which prevents back flow of hot vapor from the vaporizer into the regenerator on a system shutdown.

The liquid now passes into the vaporizer (H), and is vaporized and superheated. The vapor passes from the boiler through a low pressure actuated, shutdown valve (V_2) to the turbine nozzle where it is accelerated into the turbine blades and does work on the

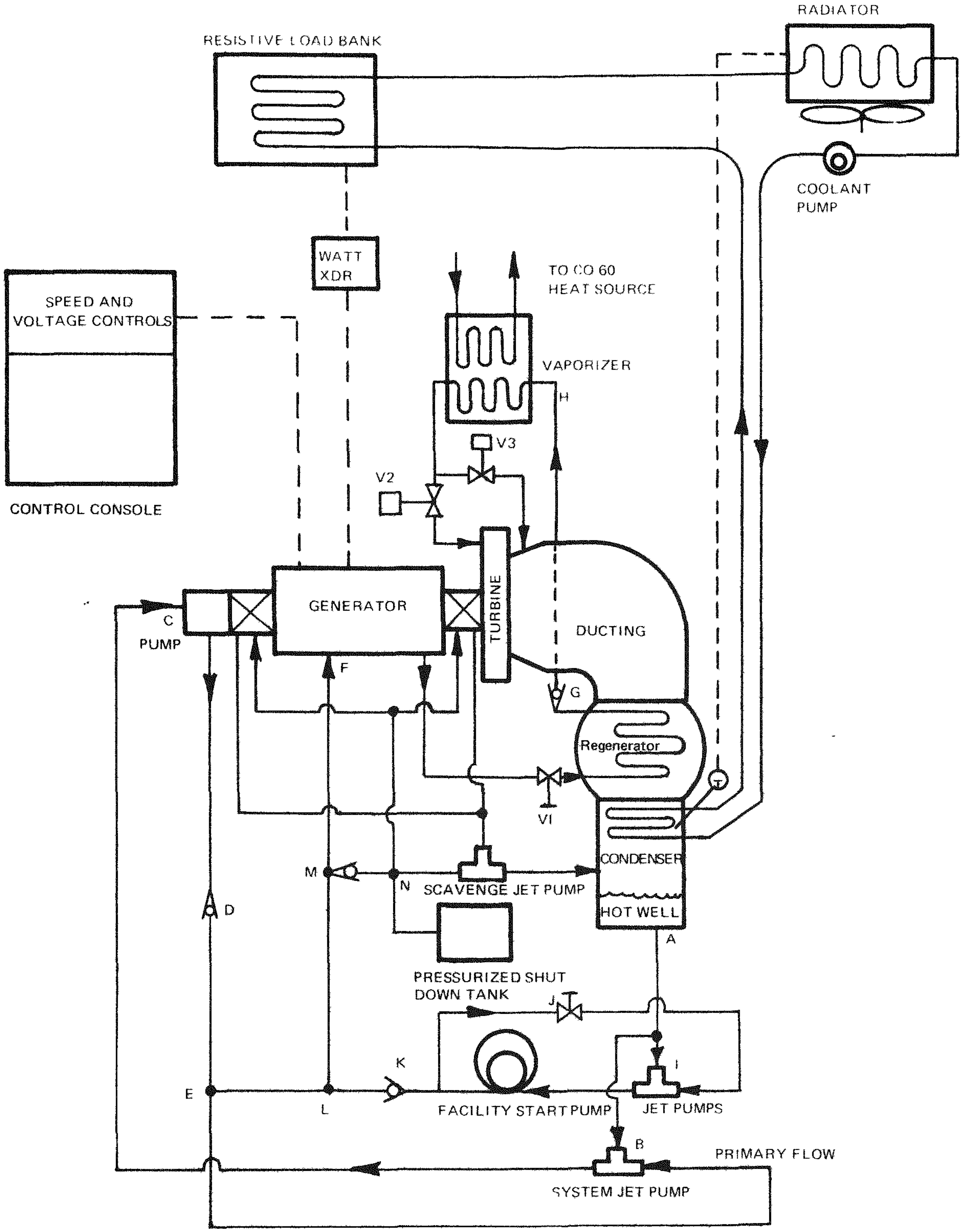


Figure 16 System Schematic — 650°F Turbine Inlet Temperature Surface Condenser

wheel. The high velocity exhaust flow passes through the annular diffuser where some static pressure is regained. The superheated vapor is then turned through 90° by the exhaust ducting and enters the regenerator giving up superheat to the cooler liquid.

The low pressure vapor then passes over the glycol cooled tube bundle in the condenser where it condenses and accumulates in the hotwell completing the cycle.

Loop #2 is a turbine bypass loop and is used for start-up and shutdown.

During start-up, low temperature liquid from the hotwell flows into a jet pump (I) similar to that in the primary loop and plumbed in parallel with it. This in turn induces the facility start pump. After pressurization, part of the flow is extracted (the amount being controlled by a manual throttle valve (J), to provide the primary flow to the jet pump, while the remainder of the flow passes through a check valve (K), and joins the primary loop at (L).

Valve (V_3) located just below the vaporizer in Figure 16 is used in loop #2 for system vapor turbine bypass during start-up and shutdown. Until the power conversion machinery is ready for rotation, the hot vapor will bypass the turbine and re-enter loop #2. When the power conversion machinery is ready for rotation, valve (V_3) is shut off allowing hot vapor to enter the turbine nozzle assembly.

For system shutdown, valve (V_3) is opened removing flow from the turbine nozzle assembly. The nitrogen pressurized shutdown tank is opened maintaining working fluid pressure during spindown of the power conversion machinery to insure bearing lubrication.

Loop #3 is the bearing system loop. At point (M) on the schematic, approximately 0.5 gpm working fluid flow is split from the main flow. At point (N) the flow splits again, the major portion going to the primary nozzle of the bearing scavenge jet pump while the remainder proceeds to the bearings. After lubricating and cooling the bearings, the drainage flow enters the suction side of the scavenge jet pump. The exit flow from this pump goes into the hotwell. During shutdown, loop #2 provides a pressurized reservoir for bearing and pump flow, while a check valve prevents leakage into the rest of the system.

Loop #4 is the system heat rejection loop containing a coolant pump, condenser coils, and a forced air radiator.

A 50/50 mixture of ethylene glycol and water is pressurized by the coolant pump and passed through the condenser coil where it absorbs the heat of the condensing system working fluid. It then enters the resistive load bank where it absorbs the heat generated by the load bank resistive heaters. The coolant is then cooled in the radiator by air flow developed by a fan blower. The temperature of the coolant is controlled by varying the air flow across the radiator with louvers controlled by a thermal sensor in the condenser (T). The coolant then returns to the low pressure side of the pump to repeat the cycle.

3.1 SURFACE CONDENSER PERFORMANCE TESTING SUMMARY (650°F)

The system was first assembled in December, 1968. The first three hours of initial operation were uneventful. The system was operated without the turbine driven pump, and power levels of up to 5.5 KWe were demonstrated.

After the initial runs many problems were encountered and resolved.

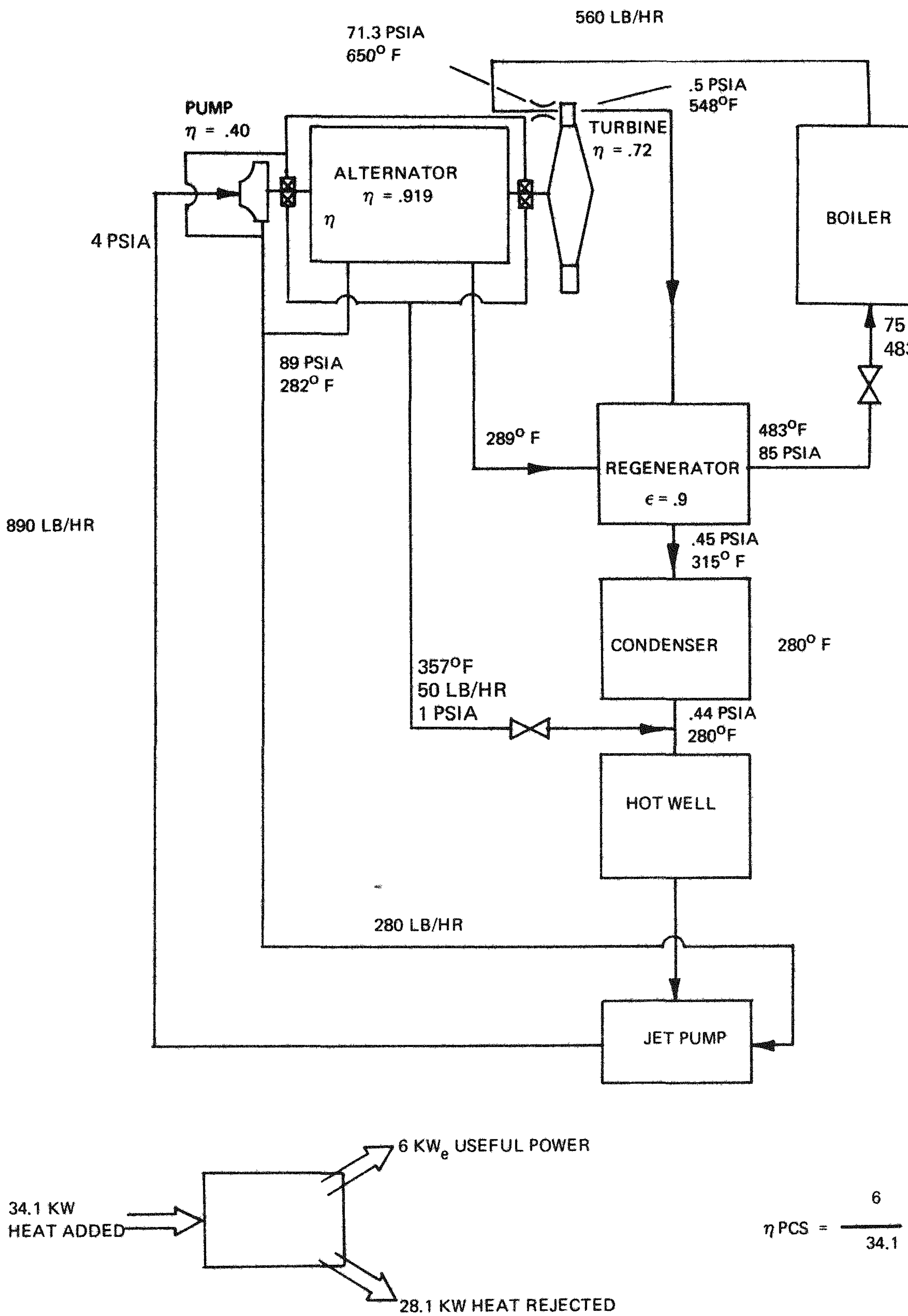
Difficulty was experienced with the alternator stator coil varnish, with the tilting pad bearings and pivots, with the method of turbine wheel/alternator rotor attachment, with the balancing of the rotating assembly and also with housing vibrations. It was not until November of 1969 that the system could be operated reliably.

The unit was then operated at various loads without the turbine driven pump impeller for over 100 hours to develop confidence that the machine was sound. At that time the turbine driven pump was added and performance data were taken for the machine with a 650°F turbine inlet and a surface condenser.

The incorporation of the turbine driven pump uncovered one more minor problem. The pump seal was inadequate to withstand the pump case pressure and leaked liquid into the pump end bearing cavity causing high churning losses. Rather then fix the problem on this build, it was decided to quantify the churning loss by measuring the pump leakage flow and its temperature rise through the bearing cavity.

Data were taken with turbine inlet temperatures between 660°F and 675°F, flow rates between 500 lb/hr and 530 lb/hr with a condenser pressure of 0.3 psia. The churning loss was about 1.6 KW in all cases. Assuming that with a modified seal or seal drain the churning loss would be available as additional alternator input power, the resulting electrical output (measured plus projected from churning loss measurements) varied between 5.67 KWe to 5.86 KWe. The resulting system efficiency (alternator useful output/heat added) ranged from 16.8% to 17.2%. Figure 17 shows the design point pressure and temperature schematic for the system which would yield an estimated PCS thermal efficiency of 17.6%.

The performance of all the components except the regenerators was as expected. The regenerator, due to a liquid side flow



$$\eta\,PCS = \frac{6}{34.1} = 17.6$$

Figure 17 Design Point Pressure and Temperature Schematic for 6 kwe Surface Condenser PCS

maldistribution, had a measured effectiveness of only 0.8 instead of 0.9. If this maldistribution were fixed and the regenerators performed as specified the projected system efficiency would have increased by about 1 point as shown in Figure 18.

Surface condenser operation was completed on February 3, 1970. A total of 141 hours operation was achieved with 140 start/stop cycles. The longest single run was 14 hours.

3.2 JET CONDENSER MODIFICATION TO 6 KW POWER CONVERSION SYSTEM

In a jet condenser subcooled liquid Dowtherm is injected through a central liquid nozzle coaxially with the vapor coming from the regenerator. The vapor condenses on the liquid jets and a portion of the injected liquid momentum is recovered in the diffuser section of the jet condenser. The subcooled liquid flow rate to the jet condenser is 10.5 gpm.

The jet condenser utilized was capable of stable operation (no flood out) with subcooled liquid outlet pressures up to 62% of the injector inlet liquid pressure. The pump that was used for system operation with the jet condenser is shown in Figure 19. It is significantly oversized for both the 650°F and 700°F turbine inlet jet condenser operation but was used due to its availability. This fact is of crucial importance when assessing the measured performance of the system with a jet condenser.

That is, the value of pump efficiency is crucial to the efficiency of a system utilizing a jet condenser due to the large amount of flow required by the jet condenser.

The system schematic, flows and state points are shown in Figure 20 for the 650°F turbine jet condenser system. Based on a pump efficiency of 80% and a jet condenser pressure recovery of 55%, the useful electrical output at the 560 lb/hr turbine design flow is 5.85 KWe. This power is slightly less than the 6 KWe output for the same mass flow with the surface condenser for the design point, due to a slightly higher pumping power requirement for the jet condenser because of the very large amount of flow associated with it.

This large amount of flow and relatively low pressure rise requirement for the jet condenser result in a relatively high pump specific speed which allows the attainment of the required high pump efficiency. If the pump is significantly less efficient than 80% or if its head rise is significantly greater than required by the system, appreciable reductions in CRU electrical output results as the pump shaft power requirements increase rapidly.

As mentioned previously, the pump used for jet condenser operation was oversized both in flow capacity and head rise for both the 650°F and 700°F operation. It was felt prudent to have pump head

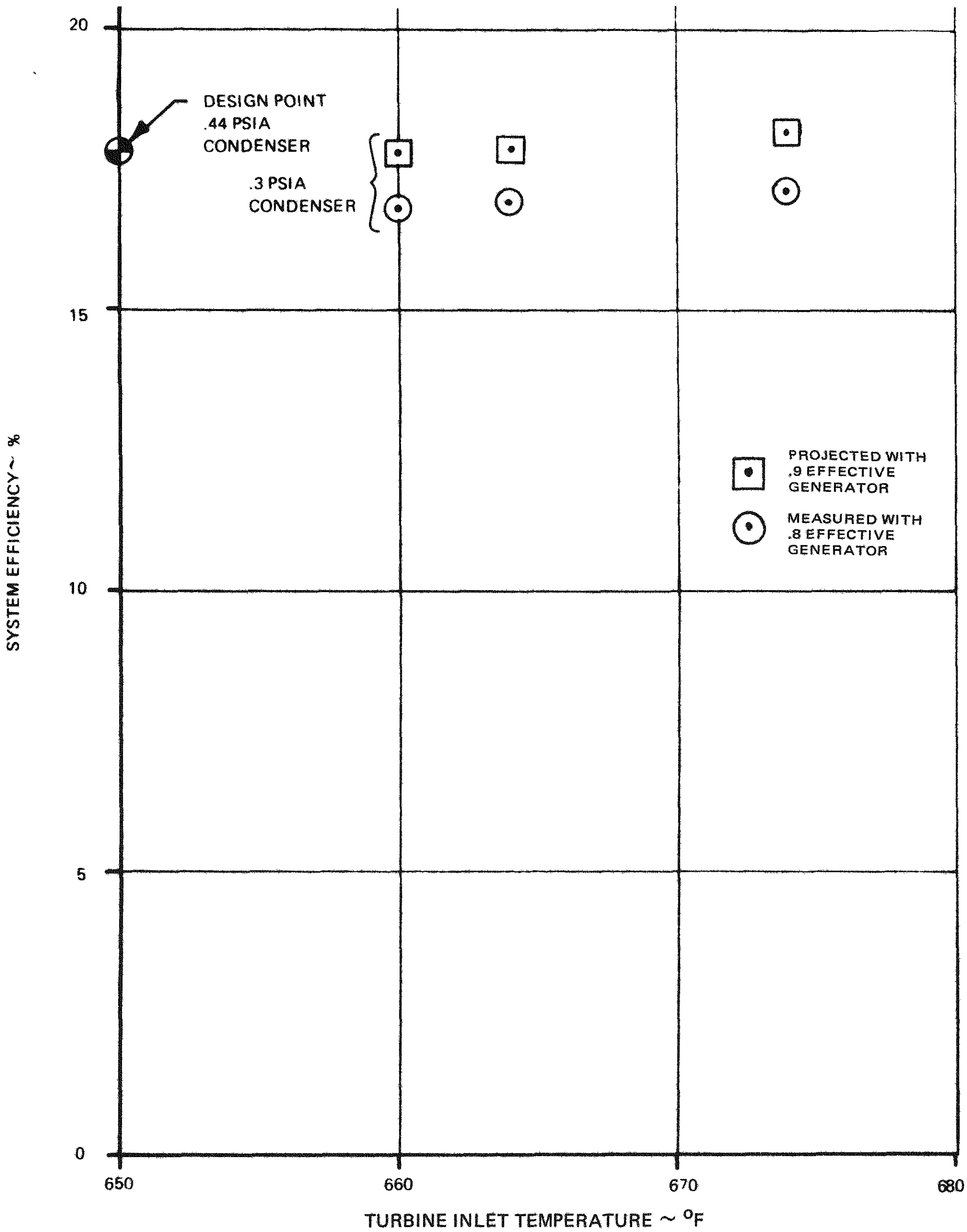


Figure 18 System Performance with Surface Condenser

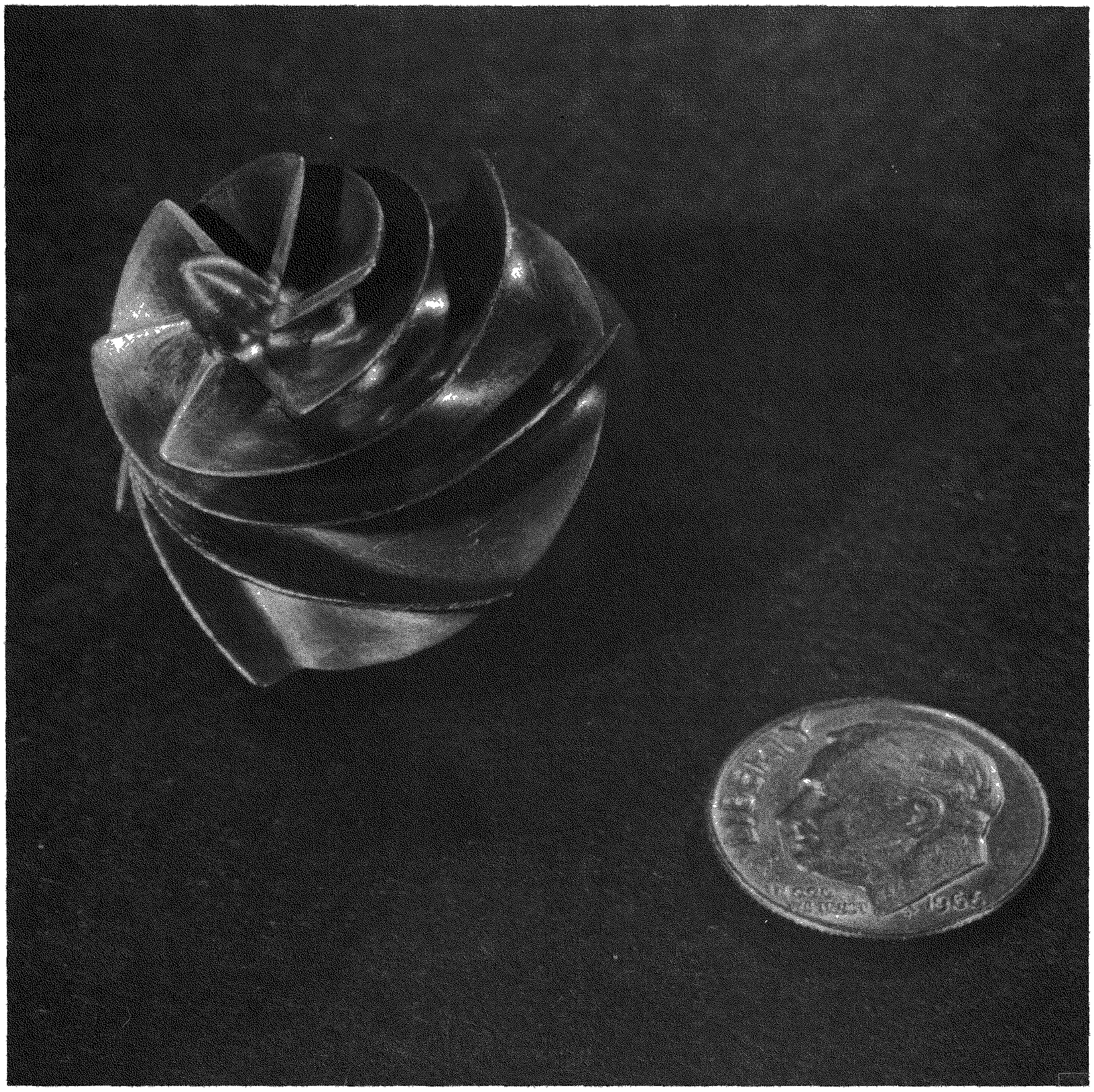
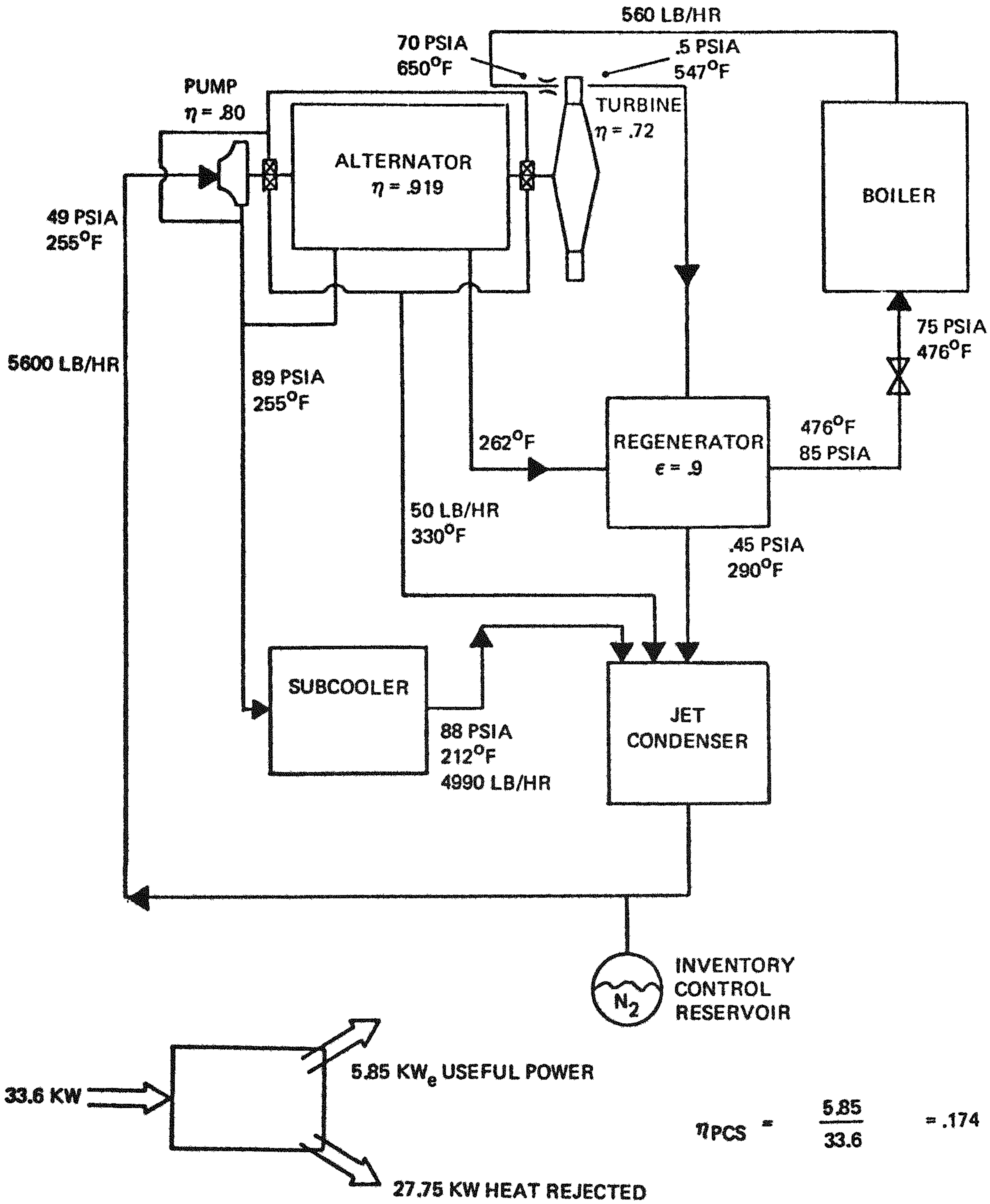


Figure 19 Turbine Driven System Pump for Jet Condenser Configuration PCS



$$\eta_{PCS} = \frac{5.85}{33.6} = .174$$

Figure 20 Design Point Pressure and Temperature Schematic for 6 KWe Jet Condenser PCS rise capability in excess of that required for use with a jet condenser with over 62% pressure recovery since the 62% was measured on a jet condenser test stand without any dynamic interactions between the inventory control system, the turbine driven pump, and the vaporizer.

The pump used was designed for a different application having a design speed of 20,000 rpm, a best efficiency point flow of 17.5 gpm with a pressure rise of about 60 psi and an efficiency of 80% at these conditions. At 24,000 rpm, the flow for best efficiency would be 21 gpm with a pressure rise of about 87 psi.

A new pump seal design was incorporated to prevent pump leakage into the bearing cavity. For jet condenser system operation the entire system was relocated in a new test cell. An electrically heated salt bath was used as the heat source with 130 feet of helically wound 7/8 inch tubing immersed in the salt bath serving as the vapor generator. The uninsulated system with the jet condenser is shown in Figure 21.

The system was built up for endurance operation with the jet condenser so that this build incorporated much fail safe emergency shutdown equipment characteristic of unattended test rigs. Most important of these were valving and plumbing between the vaporizer outlet and the turbine inlet, to shut off vapor flow, as well as between the vaporizer inlet and the inventory control reservoir to allow the vaporizer inventory to flow/boil off to the depressurized reservoir. This latter provision was to prevent significant fluid degradation during system shutdown due to the enormous heat capacity of the salt bath.

3.3 JET CONDENSER PERFORMANCE TESTING SUMMARY AT 650°F TURBINE INLET

Numerous data were taken both with and without the CRU pump installed in order to ascertain the pump shaft power requirement. Based on these measurements it was determined that .935 KW were required to drive the pump. Based on the delivered flow and 90 psi pump rise the pump had an efficiency of .48, this low value occurring due to operating the pump at 1/2 its best efficiency capacity.

If a pump that had a 40 psi rise with a best efficiency flow of about 11.5 gallons/minute were incorporated, the pump shaft power would be reduced to 260 watts based on a pump efficiency of 80% which is attainable at these specific speeds. Since the alternator efficiency was measured to be 92% efficient in the 5-6 KW range, the excess 675 watts for powering the oversized pump reduced the CRU electrical output by about 620 watts.

Performance data were taken with turbine flow rates between 504 to 542 lb/hr, inlet temperatures between 655°F and 662°F. The

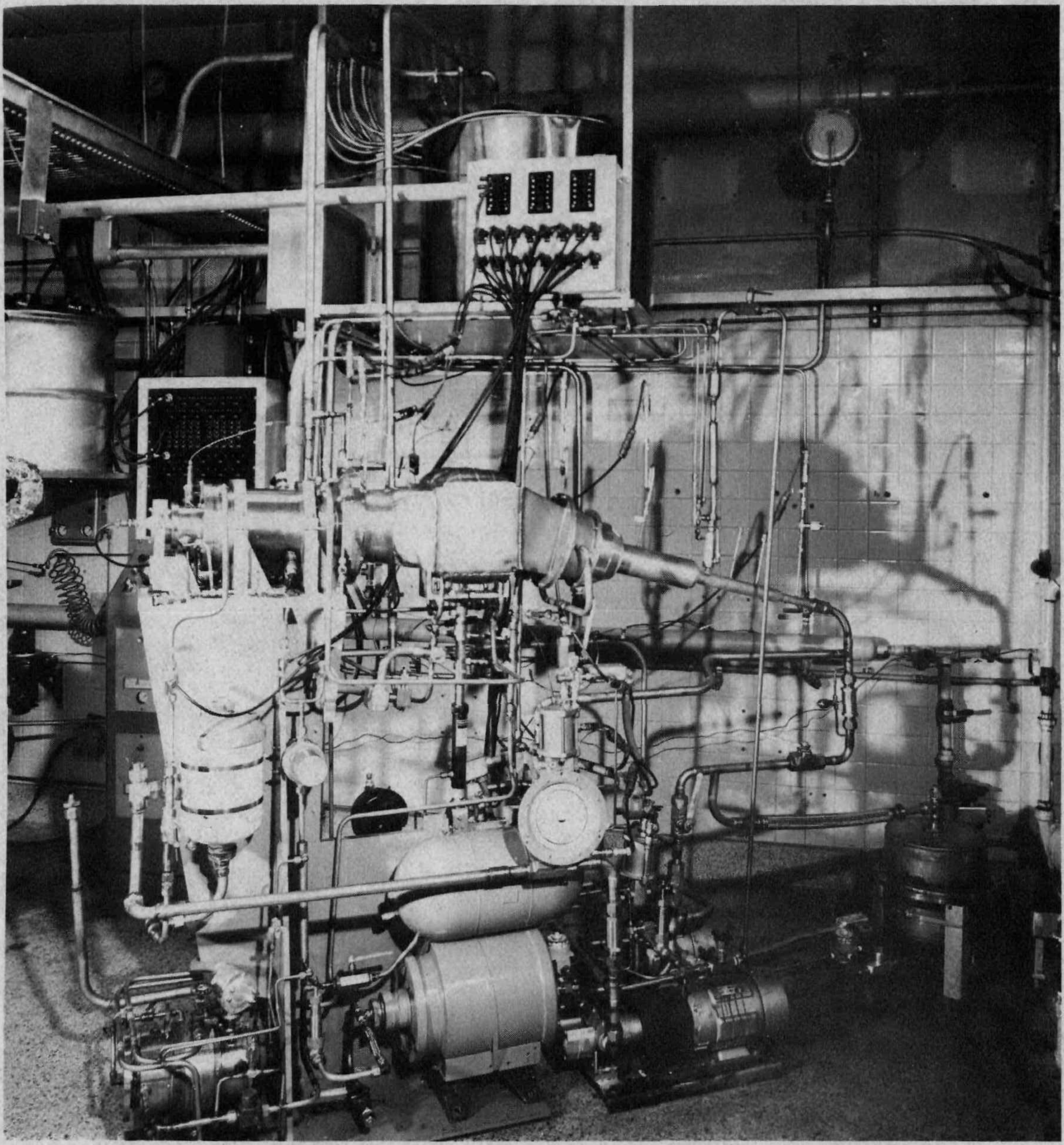


Figure 21 Uninsulated 6 KWe Jet Condenser Configuration PCS

measured electrical power ranged from 4.62 KWe to 4.98 KWe. Again the regenerators performed with an effectiveness of 0.8 rather than 0.9, in spite of an attempt to modify the regenerator manifolds to alleviate the flow maldistribution.

The resulting system efficiency as measured is shown in Figure 22. The measured efficiencies varied from 14.1% to 14.5%. These values would project to 16.8% and 17.2% if the regenerator effectiveness were 0.9 and if a properly sized pump were incorporated.

The 650°F turbine-jet condenser system testing was finished on August 5, 1970, with a total of 50 hours accumulated and 50 start-stop cycles.

3.4 SUMMARY OF JET CONDENSER ENDURANCE TEST WITH 700°F TURBINE INLET

A new nozzle ring capable of driving the turbine with 700°F saturated Dowtherm A vapor was incorporated into the CRU. Since this build was to run for endurance, all CRU parts were inspected prior to reinstallation.

The endurance unit was run with the 700°F nozzle ring for the first time on September 3, 1970, for checkout purposes. A total of 4.5 KWe was indicated at 86% full flow, which was the power anticipated at this flow rate with the oversized pump.

Since several of the amplifiers for safety shutdowns were found to be inoperative at this time, endurance operation could not be commenced.

During that weekend, with the system shut down, the salt bath temperature control malfunctioned and the salt bath overheated causing salt overflow. With no record of how hot the salt bath got or how long it was hot, there was considerable concern that the Dowtherm A working fluid had degraded significantly and formed deposits on the inside of the boiler tube. Without removal and destructive sectioning of the boiler tube the deposition could not be confirmed. Due to limited funding and schedule, the decision was reached to continue the test without teardown and inspection of the hardware.

The machine was placed on endurance operation on September 15, 1970, with performance data being recorded periodically on a digital data recorder. Some time was necessary to shake-down the automatic controls and the length of runs at this stage were relatively short.

It was found that after 100 hours, the output had rapidly dropped to about 3.5 KWe. Degradation products from the salt bath over-heat incident was believed responsible for the rapid output decay. It was found that as the number of running hours increased, the

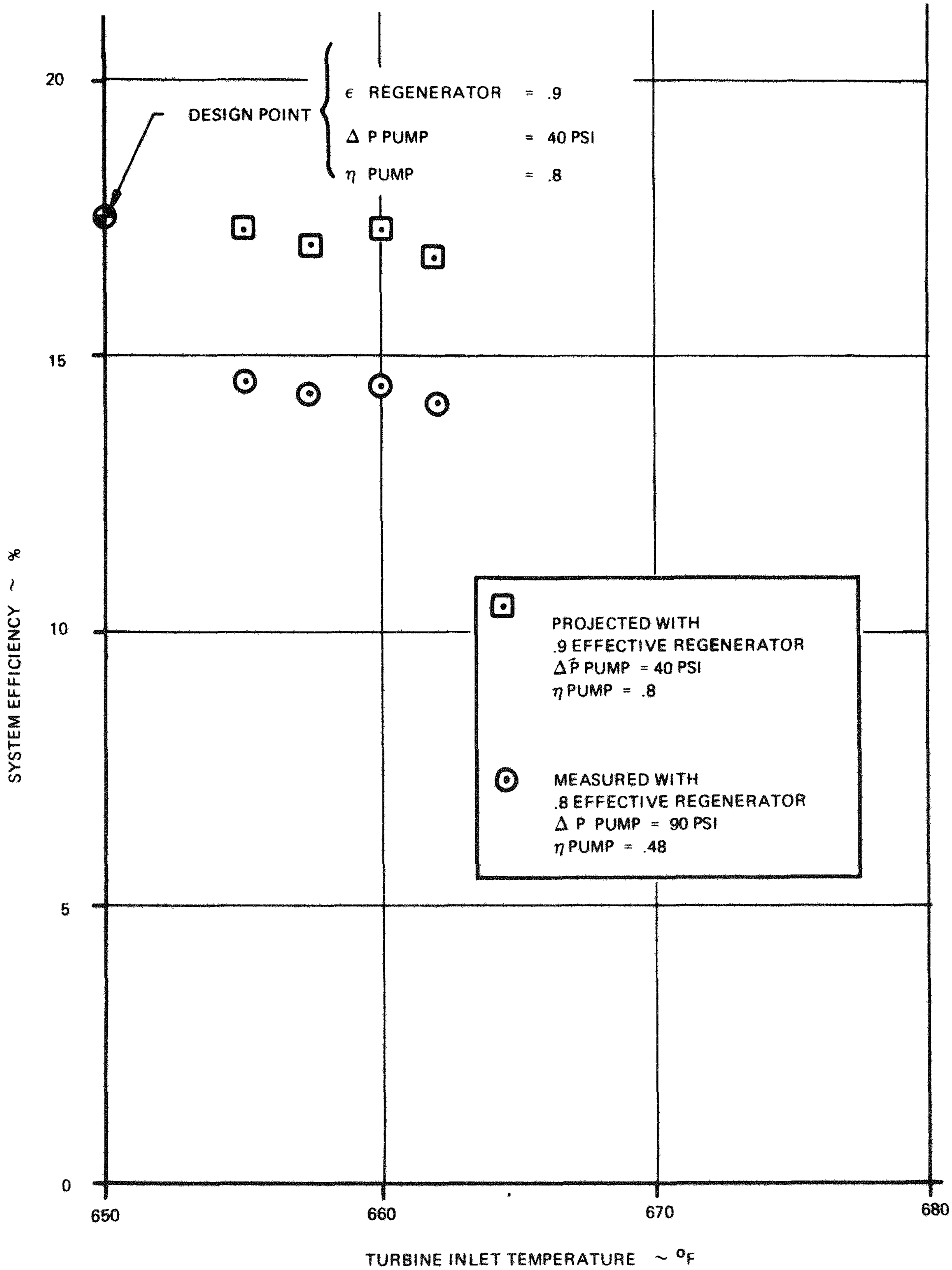


Figure 22 System Performance with Jet Condenser

power output slowly deteriorated but at a much slower rate than the initial drop. This latter change, coupled with an increase in turbine inlet pressure for the same flow, indicated a slow buildup of decomposition deposits in the nozzles.

Problems were encountered with the turbine inlet solenoid valve and it was decided to replace the valve by a larger high temperature valve which had already been tested and shown to be satisfactory for long life. This change occurred at 238 hours into the shakedown run.

Between the end of the shakedown run and the end of the endurance test, there were seventeen shutdowns, none due to any system component malfunction. The reasons for shutdown are listed in Table 2. The desired 2500 hours endurance testing was achieved.

The machine was run for a total of 3003 hours, and the system was shut down on April 3, 1971. The longest continuous run was 367 hours. The ratio of endurance operating hours to calendar hours is shown in Figure 23. During the last 500 hours, the unit was operating virtually continuously.

On disassembly, it was found that a black carbonaceous substance had been deposited in the exhaust ducting.

When the turbine wheel was removed, the reason for low power level output was immediately obvious. Of the eleven nozzles, three were completely blocked and another three almost blocked.

These nozzles were all adjacent to one another and were the first ones to see the gas stream from the boiler. As was feared, it appeared that a slug of viscous decomposition products was emitted from the boiler in one lump and blocked these nozzles at the same time, causing a sudden change in power output. The gradual decline from that point resulted from a slow deposition in all nozzles of material coming from the boiler. The decomposition probably occurred during the severe overtemperature at the beginning of the shakedown operation.

Table 2 - 2500 Hour Endurance Test Shutdown Listing

No. Shutdowns	Cause
4	Facility power failure.
8	Failure of a calrod in the electrically powered boiler.
2	Failure of N_2 supply.
3	Inadvertent shutdown due to instrumentation malfunction.

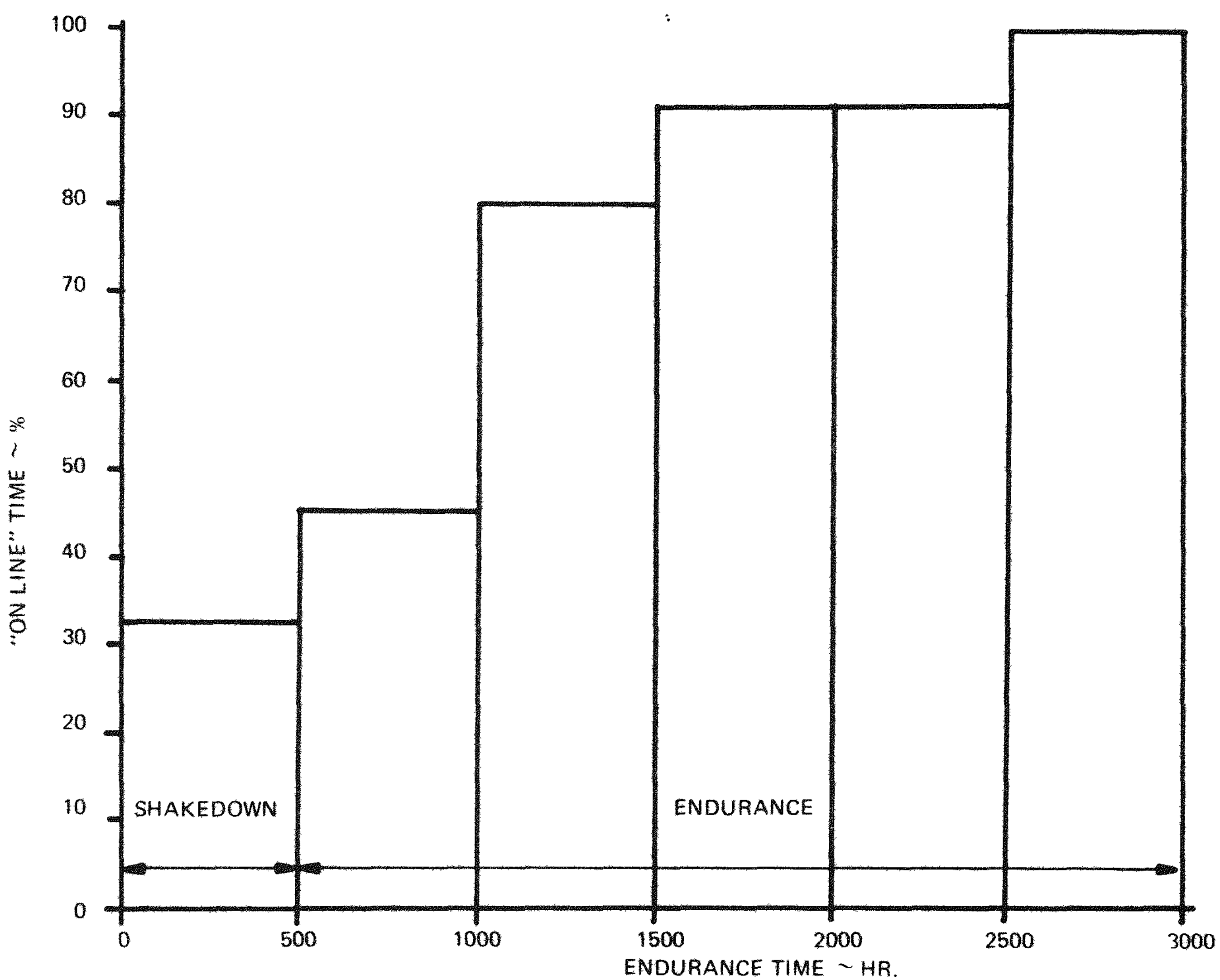


Figure 23 Endurance Time Vs. "On Line" Time Percentage for 6 KWe Test

3.5 CONDITION OF HARDWARE BEFORE AND AFTER SYSTEM OPERATION

3.5.1 Turbine Wheel

Surface condenser operation of the power system left the wheel in the condition shown in Figure 24 (inlet side) and Figure 25 (exit side) compared to the new wheel shown in Figure 26.

Preliminary testing with the jet condenser at 650°F resulted in little change in the wheel as may be seen from the close-up view of the turbine buckets shown in Figure 27 (inlet) and Figure 28 (exit).

After the endurance operation the wheel was greatly discolored with a film of carbon, but there were no heavy deposits on the blades as may be seen in Figures 29 (inlet) and Figure 30 (exit).

3.5.2 Nozzle Block

Before operation the nozzle block was spotlessly clear, as may be seen in Figure 31. As previously described, the nozzles were totally or partially blocked by decomposition products after the endurance operation, a general view being given of the nozzle block after testing in Figure 32.

Measurements were made of individual nozzle diameters before operation and after operation. The diameter remaining available for the passage of gas was also measured.

Numbering the nozzles one (1) through eleven (11), with number one being at the top but actually the last nozzle to be fed by the high pressure gas, the following results were obtained:

Nozzle No.	Before Test	After Test	Comment
1	.068	.063	
2	.068	--	Nozzle blocked
3	.068	--	Nozzle blocked
4	.068	--	Nozzle blocked
5	.068	.022	
6	.068	.045	

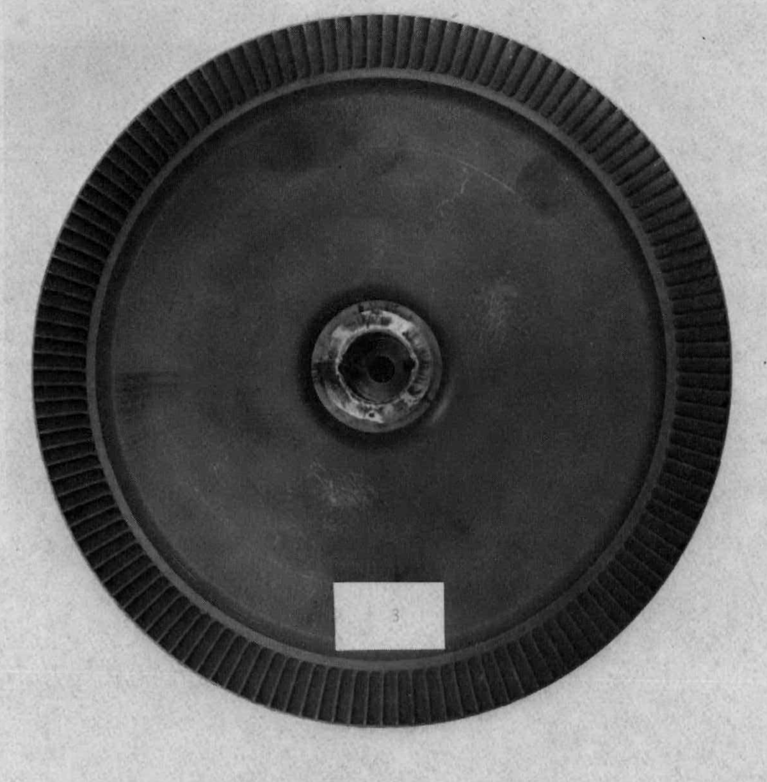


Figure 24 Turbine Wheel After Test (Inlet Side)

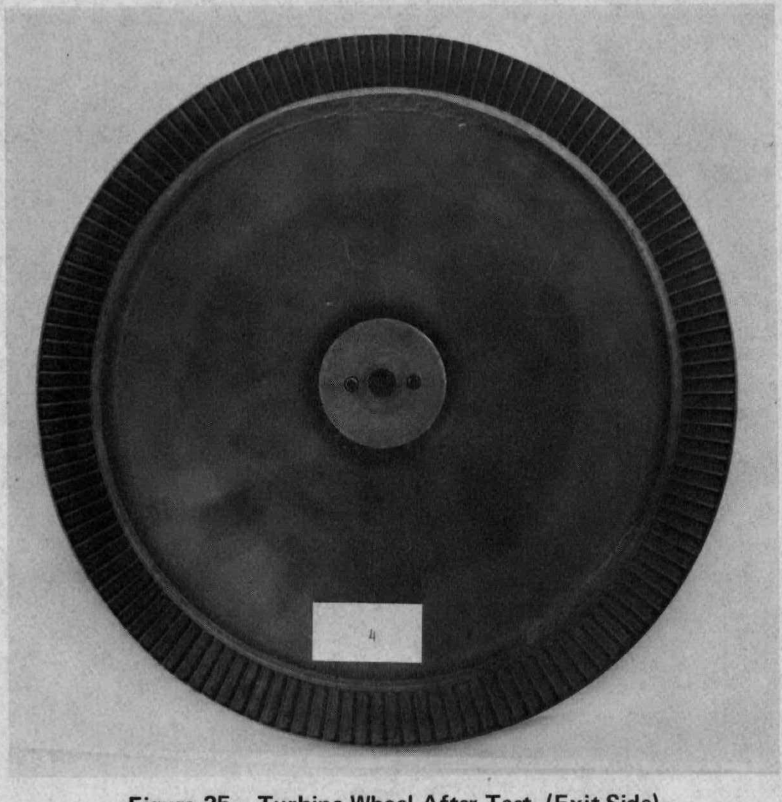


Figure 25 Turbine Wheel After Test (Exit Side)

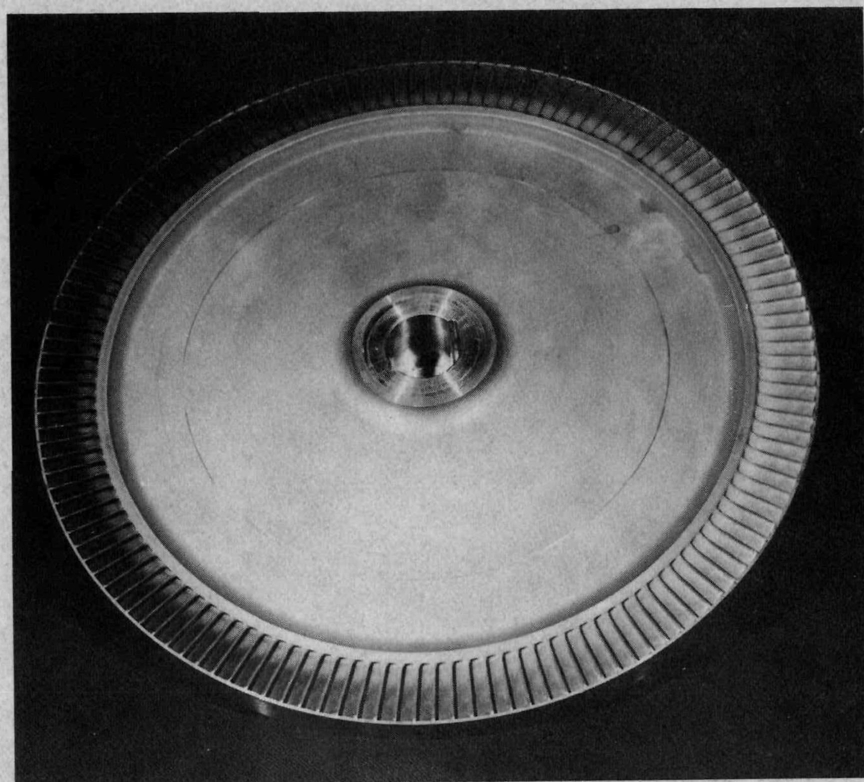


Figure 26 Turbine Wheel Before Test

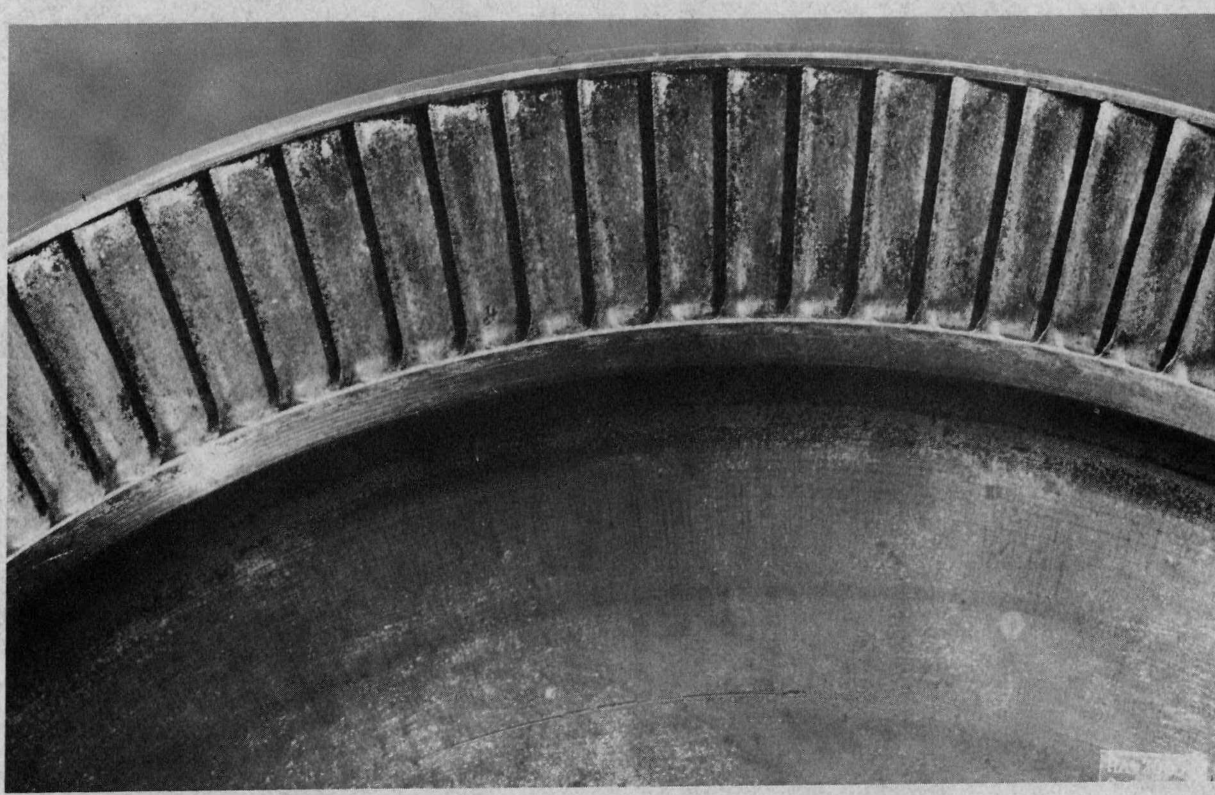


Figure 27 Turbine Buckets After Test (Inlet Side)

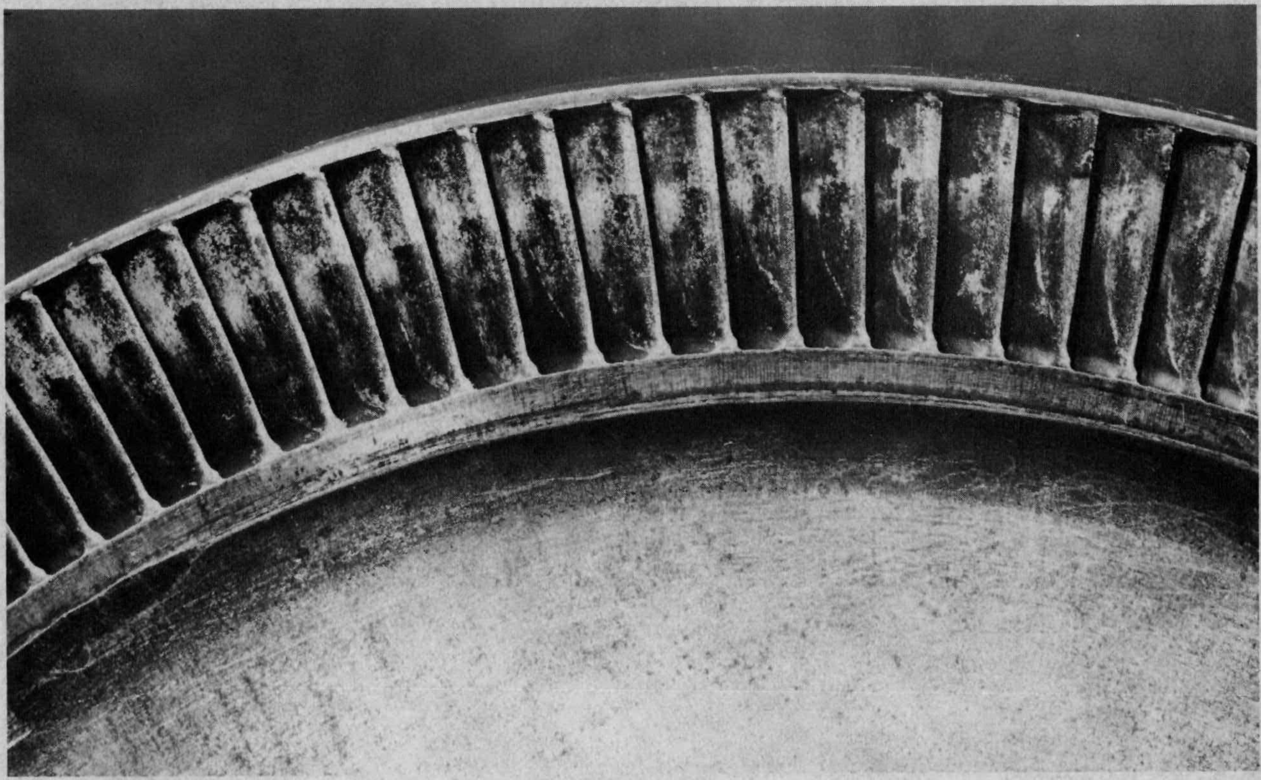


Figure 28 Turbine Buckets After Test (Exit Side)

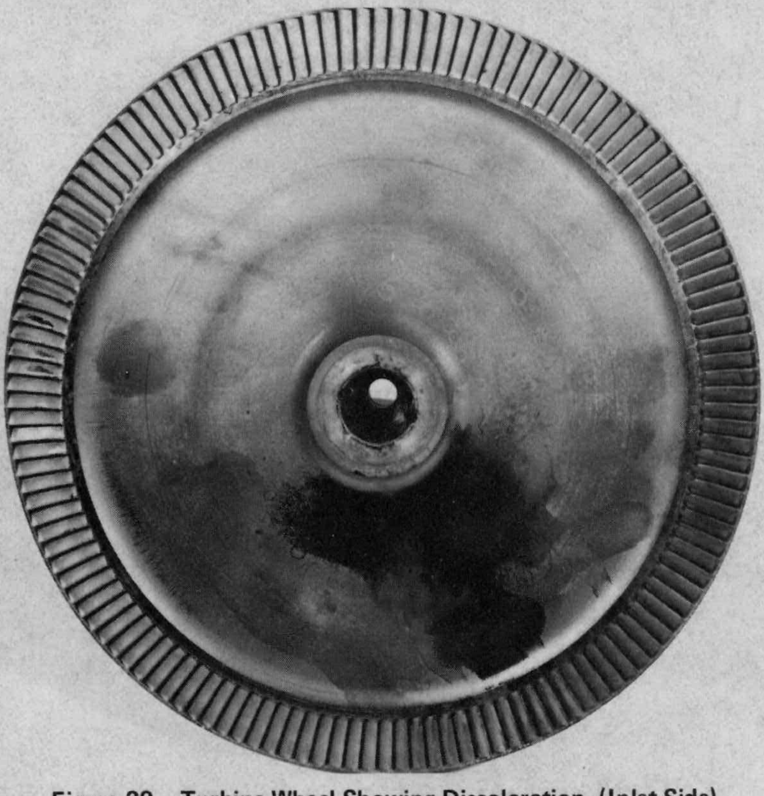


Figure 29 Turbine Wheel Showing Discoloration (Inlet Side)

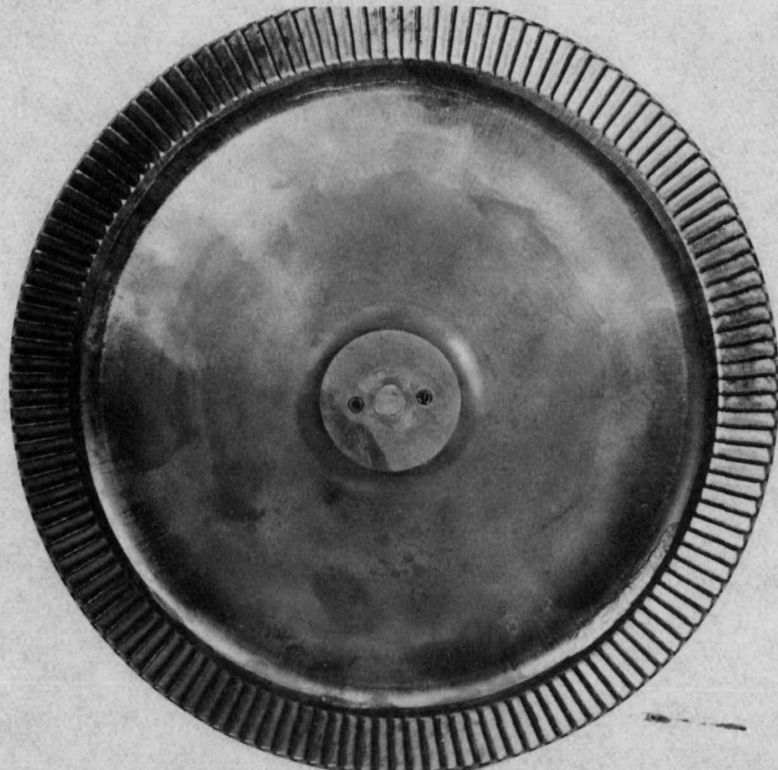


Figure 30 Turbine Wheel Showing Discoloration (Exit Side)

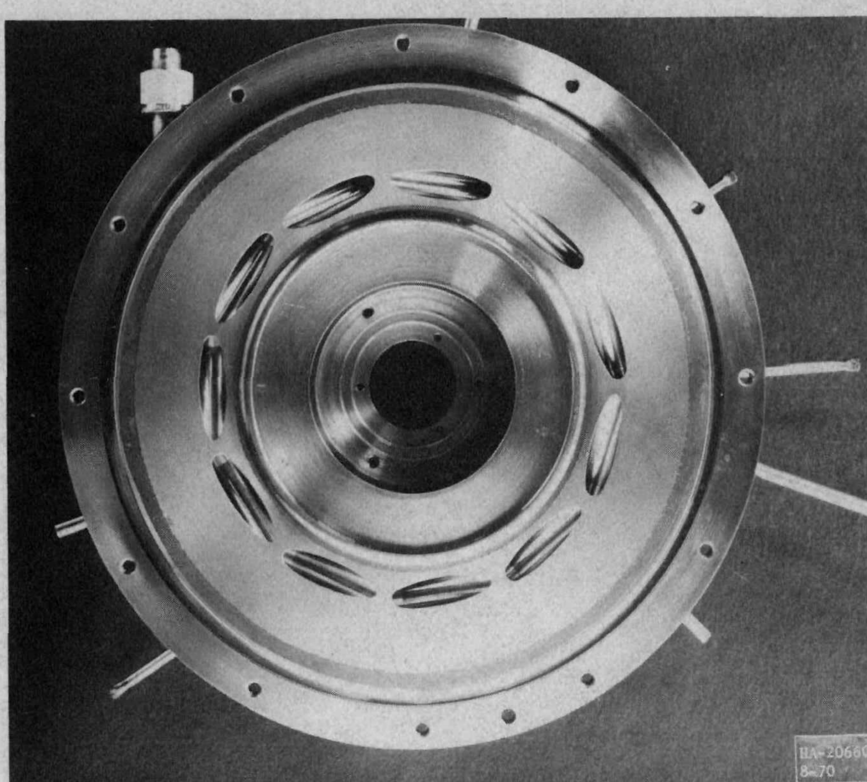


Figure 31 Nozzle Block Before Test

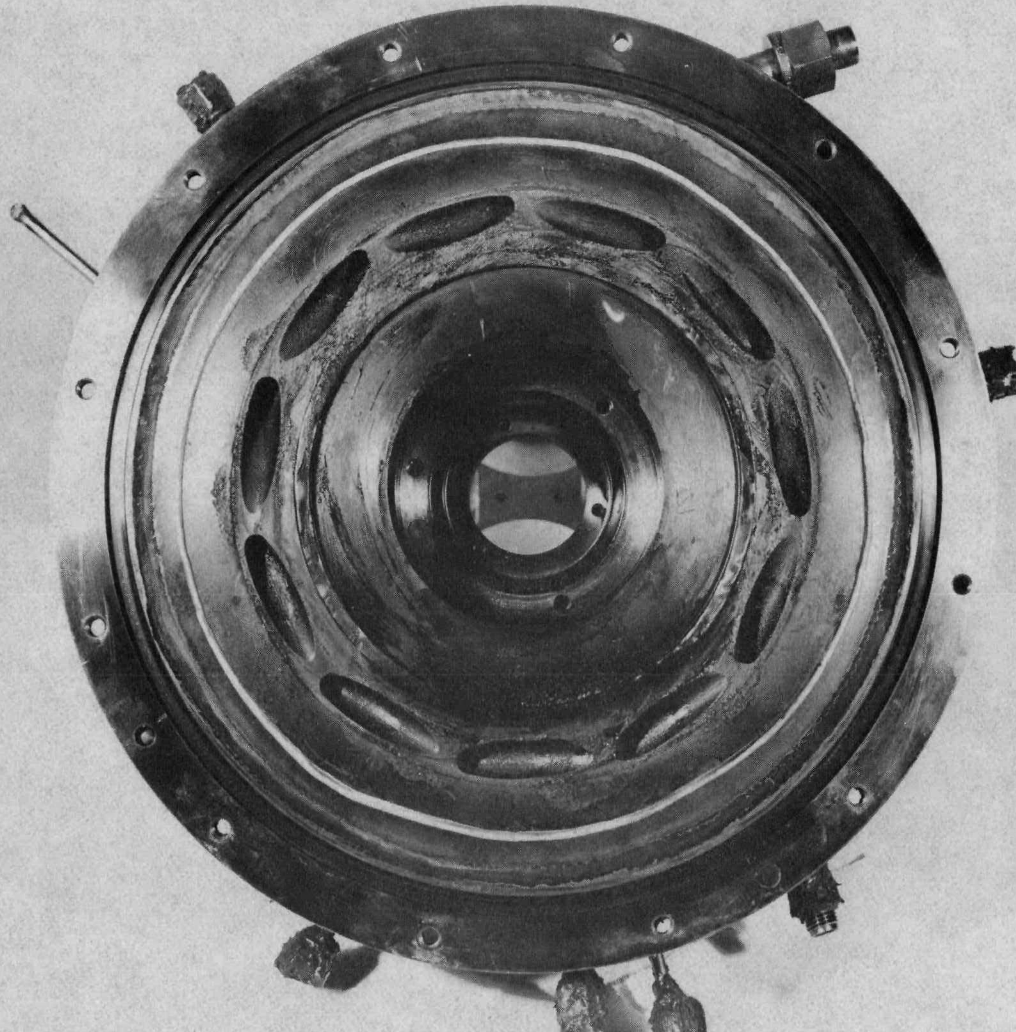


Figure 32 Nozzle Block After Test

Nozzle No.	Before Test	After Test	Comment
7	.068	.020	
8	.068	.050	
9	.068	.054	
10	.068	.052	
11	.068	.058	

A photograph of nozzle No. (3) is shown in Figure 33 before testing. Nozzles (1) and (2) are shown in Figure 34 after testing and nozzle (9) in Figure 35.

As can be seen the first three nozzles to see the gas stream (2-4) were completely blocked and from there through to the last one (i.e., number (1), they become progressively more open.

3.5.3 Screw Seals

Each screw seal liner showed signs of having been rubbed by the mating part of the shaft, Figures 36 and 37. Before insertion the liners were cleaned and thus all discoloration, especially noticeable on the turbine wheel screw seal resulted from the deposition of decomposition products during operation. Measurements were made of the internal diameters of each seal before and after testing.

	Before Testing	After Testing
Alternator pump end	2.550-2.551	2.4905-2.5505
Alternator turbine end	1.383-1.386	1.3809-1.3851
Turbine	1.109-1.111	1.1072-1.1093

3.5.4 Radial Bearings

3.5.4.1 Pump End

Comparison of the appearance of the wear surface of the tilting pads before and after endurance operation showed no change in appearance while wear was immeasurable, see Figures 38 and 39.

There was no sign of distress in the pivot socket of the pad, although there was some deposition of decomposition products around the edge of the socket.

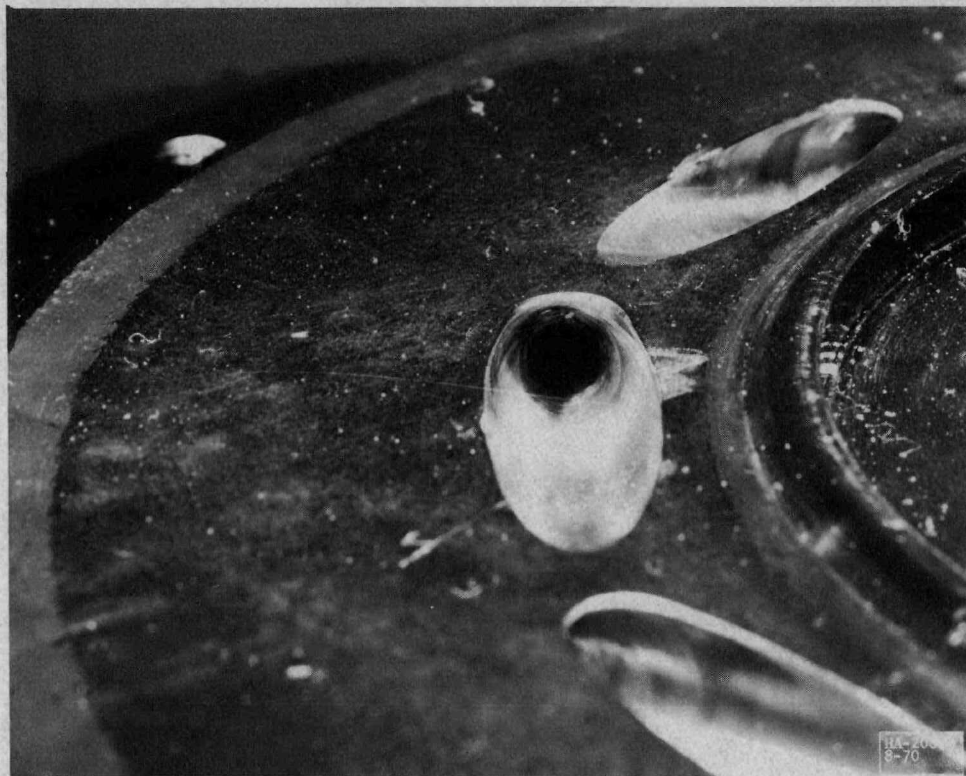


Figure 33 Nozzle Number 3 Before Testing

Figure 34 Nozzles Numbers 1 and 2 After Testing

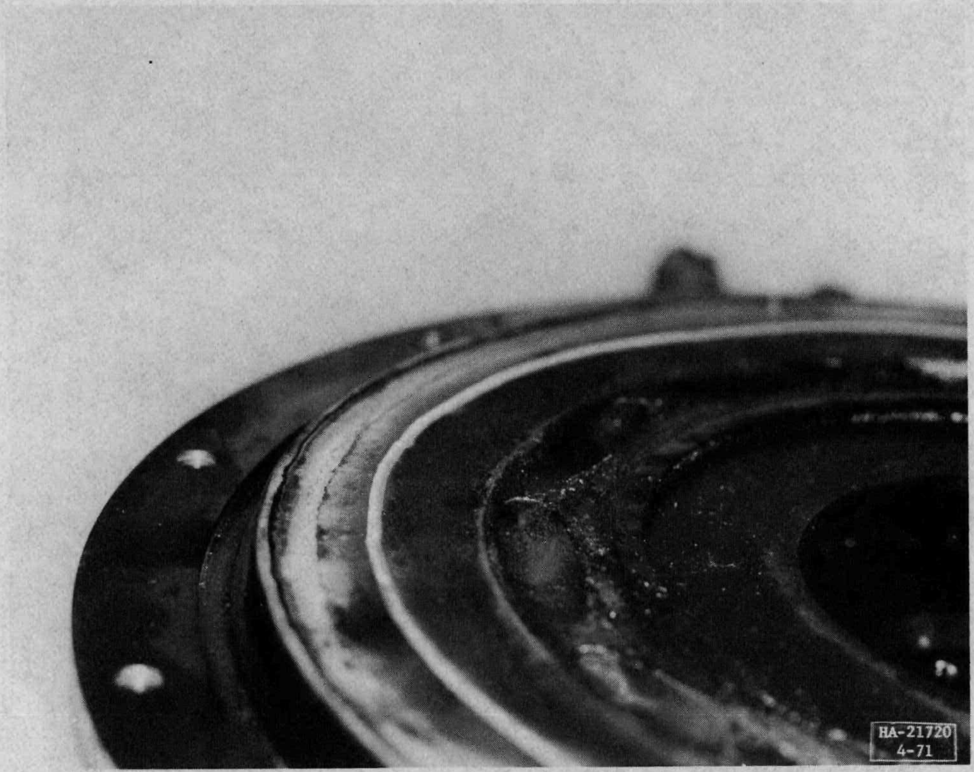


Figure 35   Nozzle Number 9 After Testing

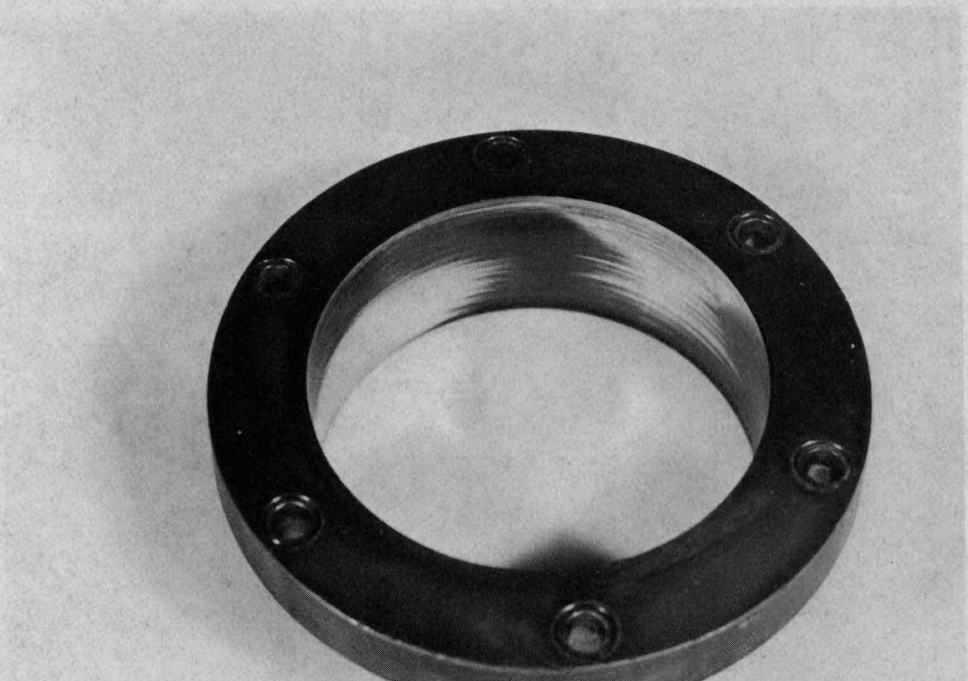


Figure 36   Screw Seal Liner After Test

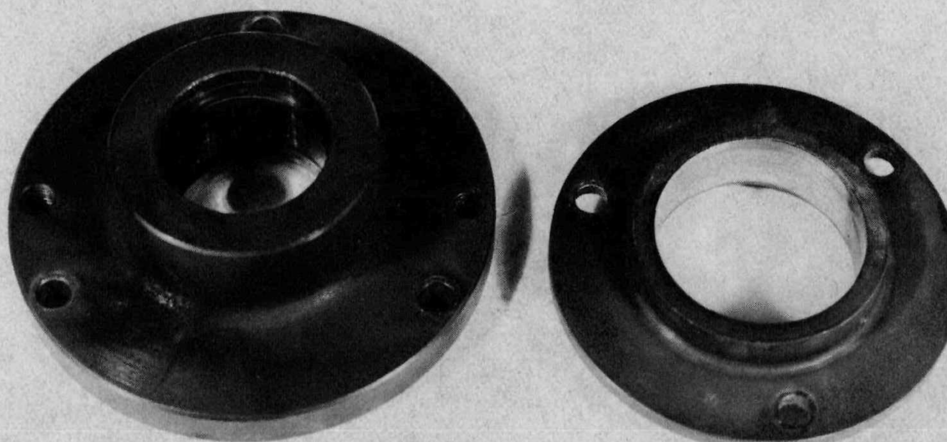


Figure 37 Screw Seal Liners After Test

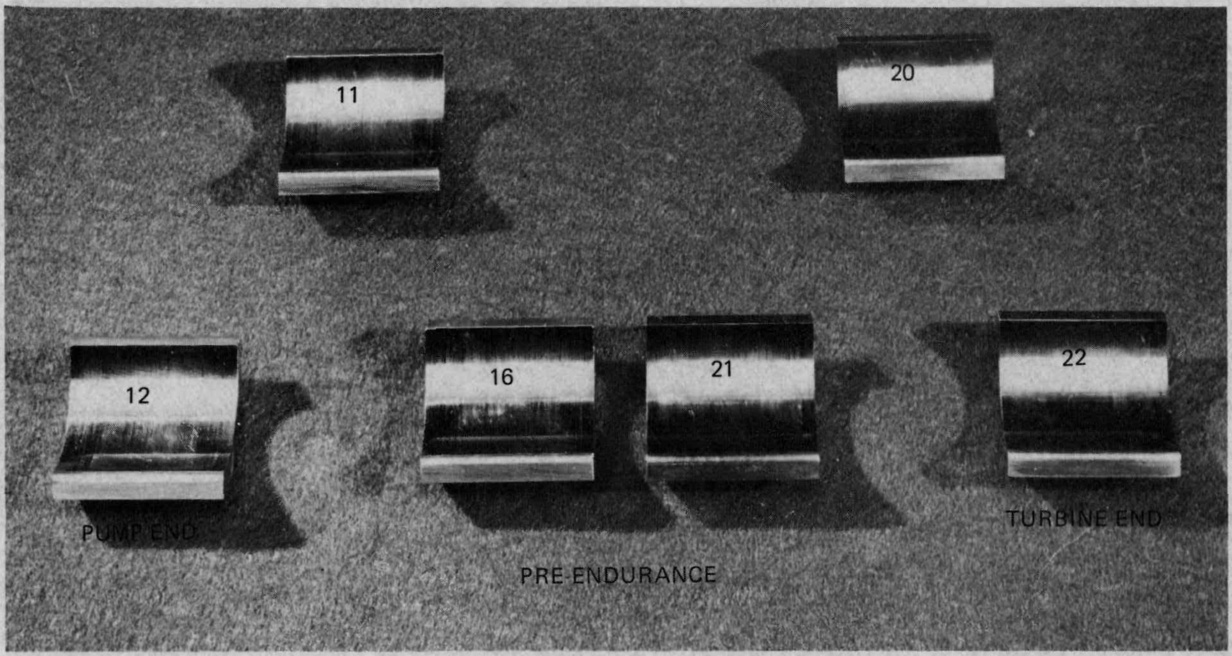


Figure 38 Tilting Pad Bearings Before Test

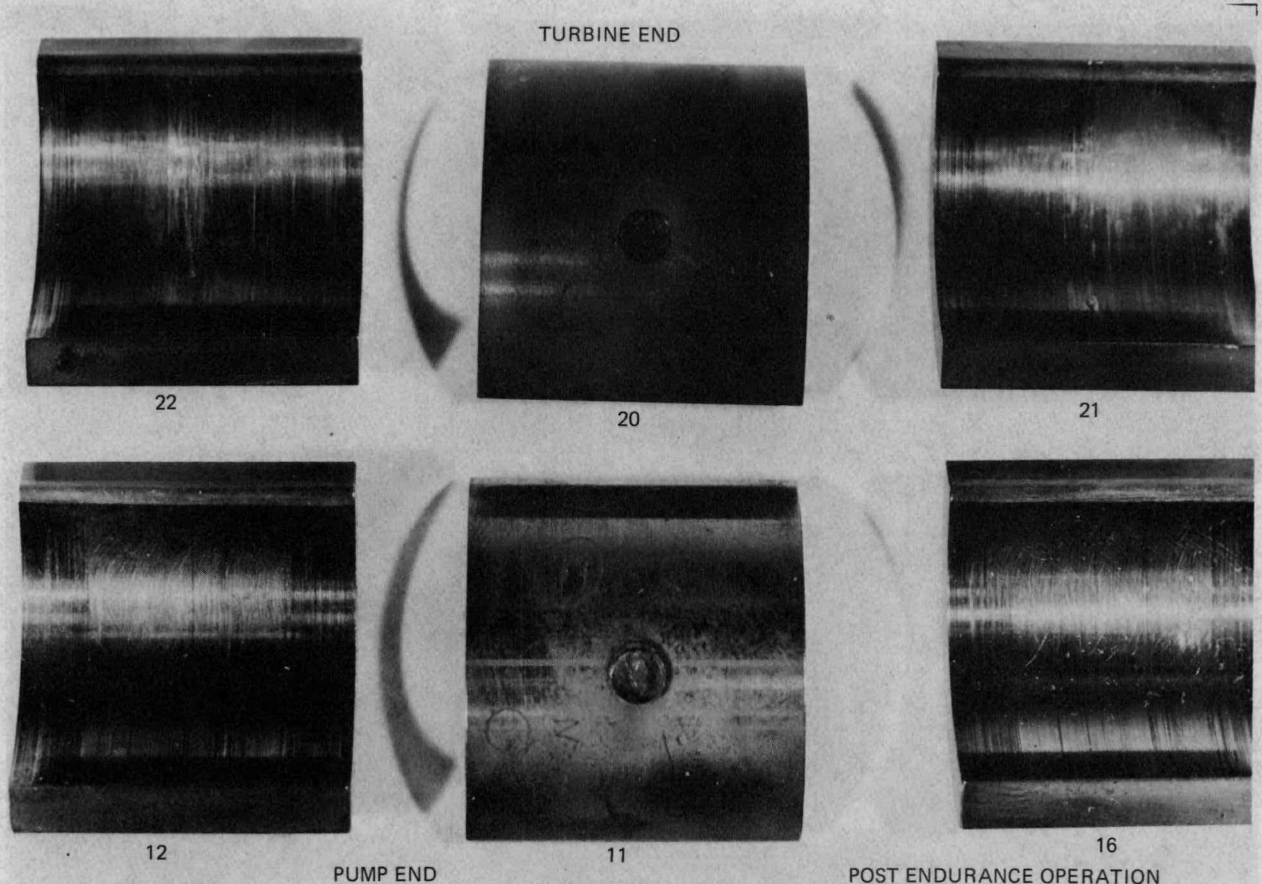


Figure 39 Tilting Pad Bearings After Test

The tip of the adjusting screws were lightly polished with crocus cloth before use. After endurance running there was evidence of very light polishing of the contact area but no sign of any severe fretting. In spite of this fact the bearing clearance had increased from the set up value of .0015 inches to .0045 inches, but the cause of the increase was not obvious. However, the overall length of the adjusting screws had decreased somewhat even though the tips showed no sign of deformation.

Pad Number	Before Test	After Test
11	1.5039	1.5038
12	1.5023	1.5020
16	1.5037	1.5017

The journal showed no signs of deterioration or measurable wear after endurance operation as may be seen from Figure 40 before operation and in Figure 41 after testing.

3.5.4.2 Turbine End

The wear surface of the pads at this end show more signs of distress than at the other end. A narrow band of the wear in the region of the adjusting screw may be seen in Figure 39. In general the pads were very discolored after endurance operation, presumably by decomposotion products which have been pumped through the screw seal. The bottom of each socket was badly discolored with some debris trapped in this location. The tips of the screws are polished in the contact area but no evidence of deformation exists. The bearing clearance had increased from .0015 inches to .0030 inches but measurements taken of the length of the adjusting screws actually showed no change or a very slight increase in length.

Pad Number	Before Test	After Test
20	1.5050	1.5059
21	1.5007	1.5005
22	1.5026	1.5029

The journal before testing had signs of scratching and wear at the edges only, see Figure 42, but after testing showed additional signs of distress in the center matching those found on the pads, as may be seen in Figure 43. However, measurements made with a micrometer showed no change in dimension.

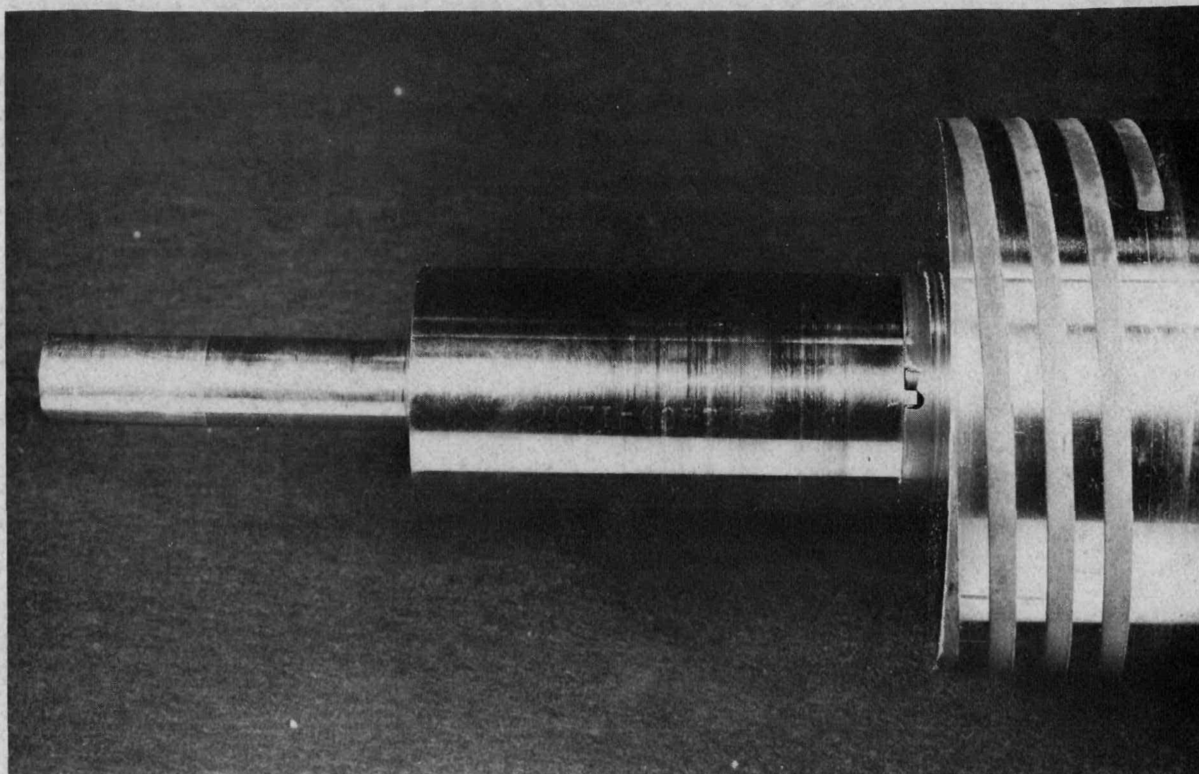


Figure 40 Journal Before Testing

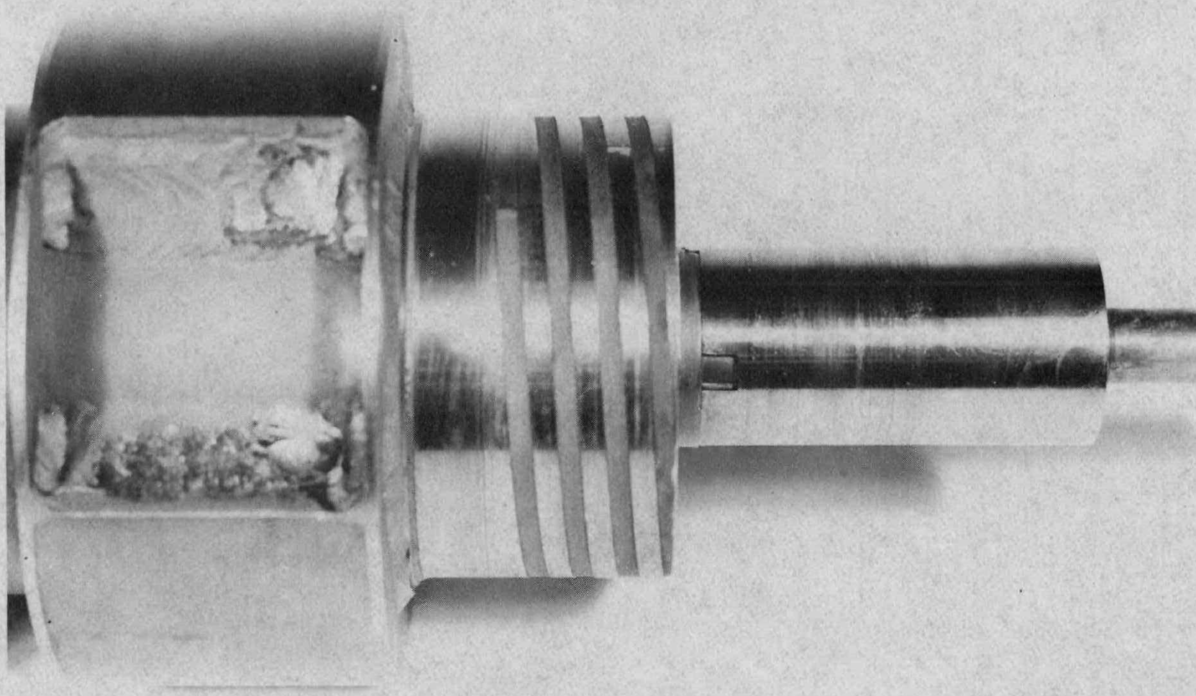


Figure 41 Journal After Testing

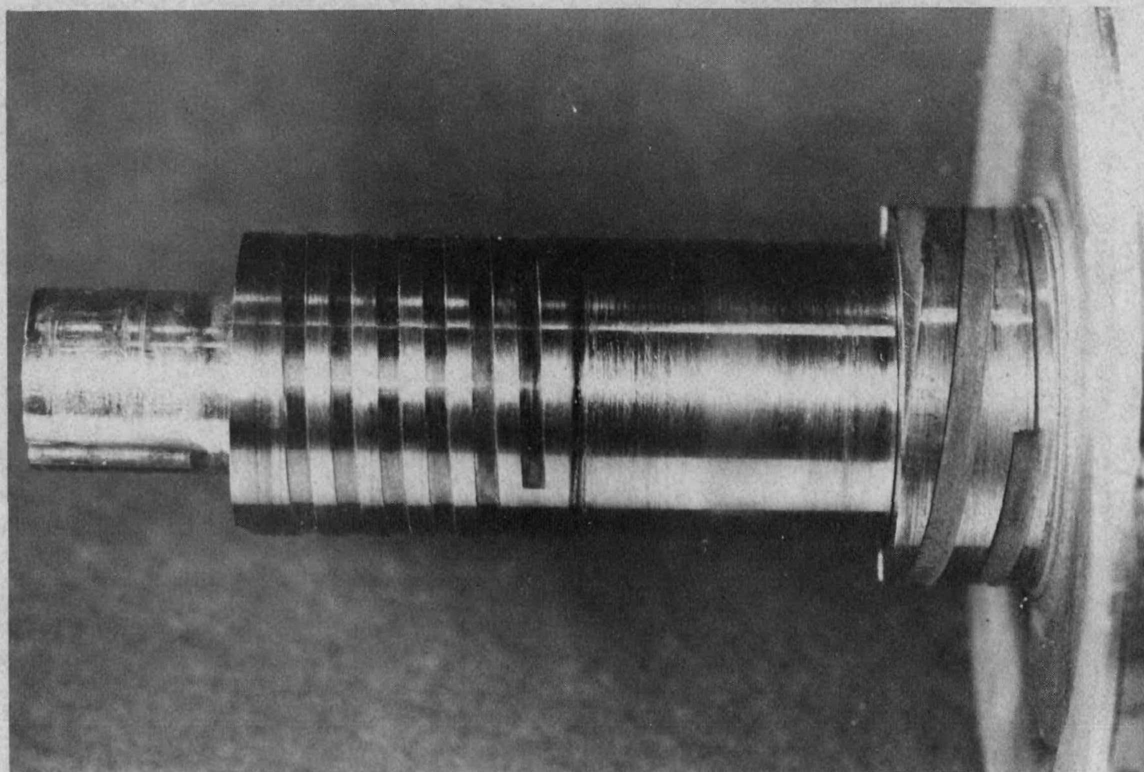


Figure 42 Journal with Scratching Before Testing

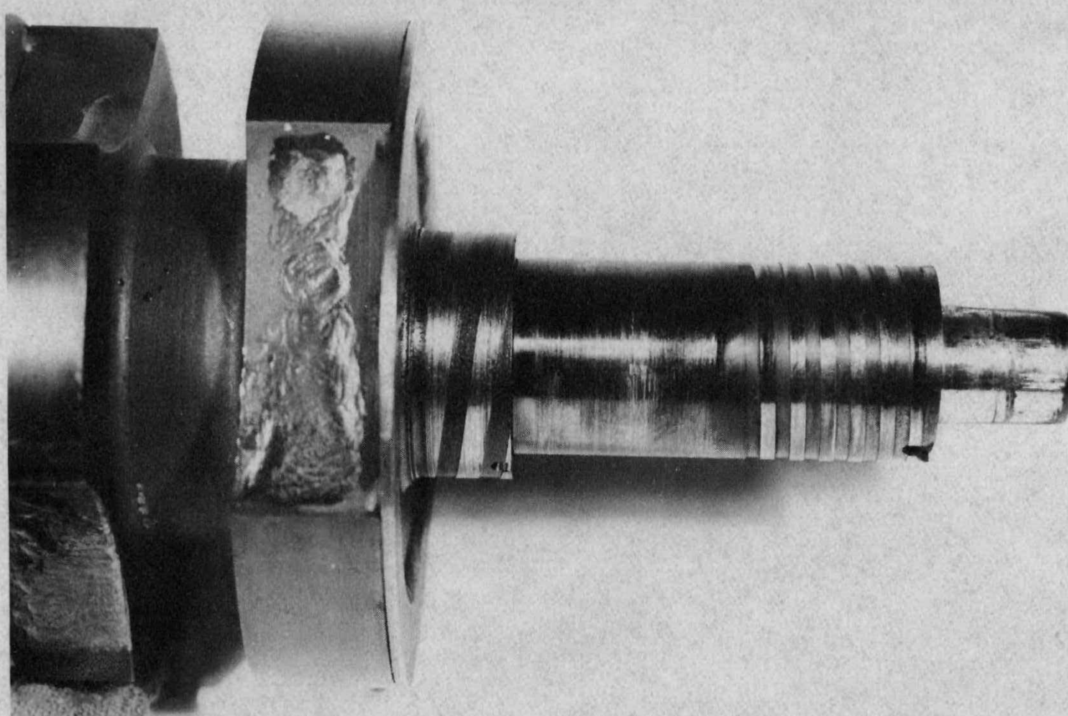


Figure 43 Journal Wear After Testing

3.5.5 Thrust Bearings

There was no detectable change in appearance of the thrust bearings during the testing of the system, as may be seen from Figure 44 before testing and Figure 45 after testing. Axial play was reduced from the setup value of .005 inches to a value of .0035 inches. Measurements were made of the pad and runner thicknesses before and after testing.

Pad	Before Test	After Test
Inboard Upper	.2523	.2528
Inboard Lower	.2523	.2524
Outboard Upper	.2479	.2482
Outboard Lower	.2493	.2497
Runner	.3049	.3049

3.5.6 Hydrostatic Face Seal

Very slight wear had occurred at the seal face but nothing serious - see Figures 46 (before) and 47 (after) testing. There was no evidence of any wear debris or decomposition products in the hydrostatic orifices. There was some black debris on the inside of the seal, probably resulting from the use of vacuum grease to lubricate the 'O' ring. The 'O' rings showed no sign of deterioration after their long immersion in Dowtherm.

3.5.7 Pump Impeller

The only apparent change in the impeller was some discoloration, Figures 48 (before) and 47 (after) testing. Scratches on the tips of the blades resulted from checkout testing before installation on the CRU.

3.5.8 Stator Housing

No change in the condition of the stator was obvious after endurance operation. A photograph is shown of the stator potting in Figure 49. The cracks in the Epoxylite appeared when the windings were first insulated.

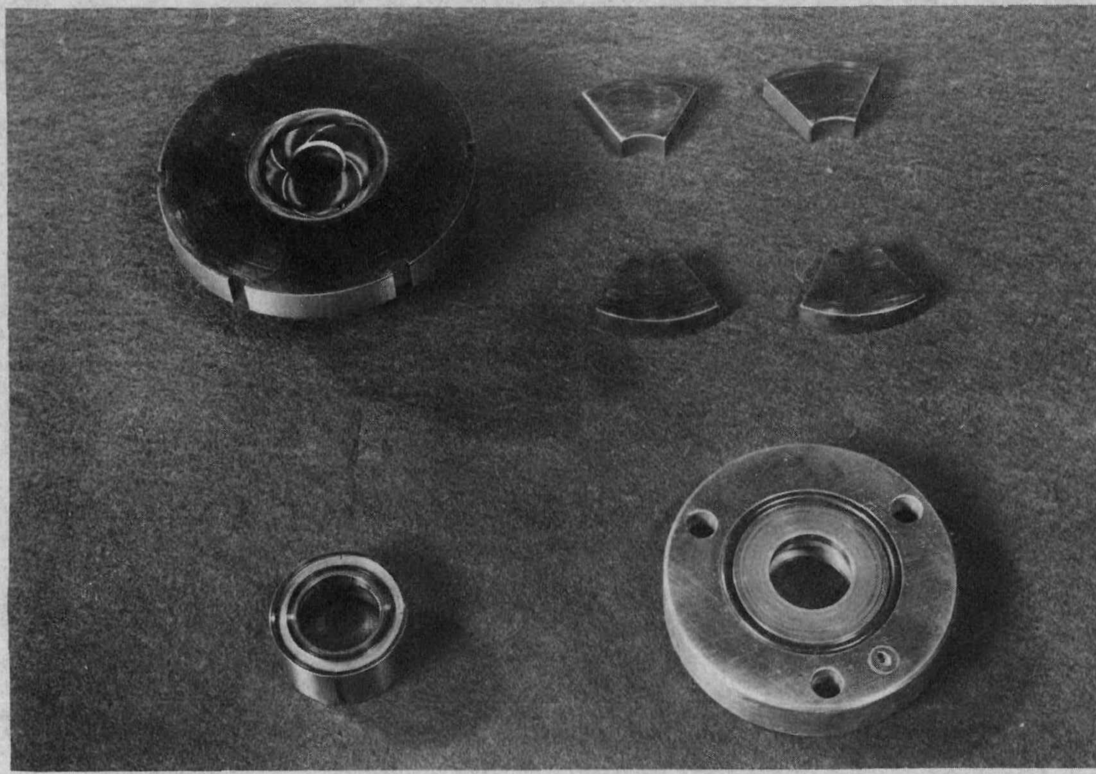


Figure 44 Thrust Bearings Before Testing

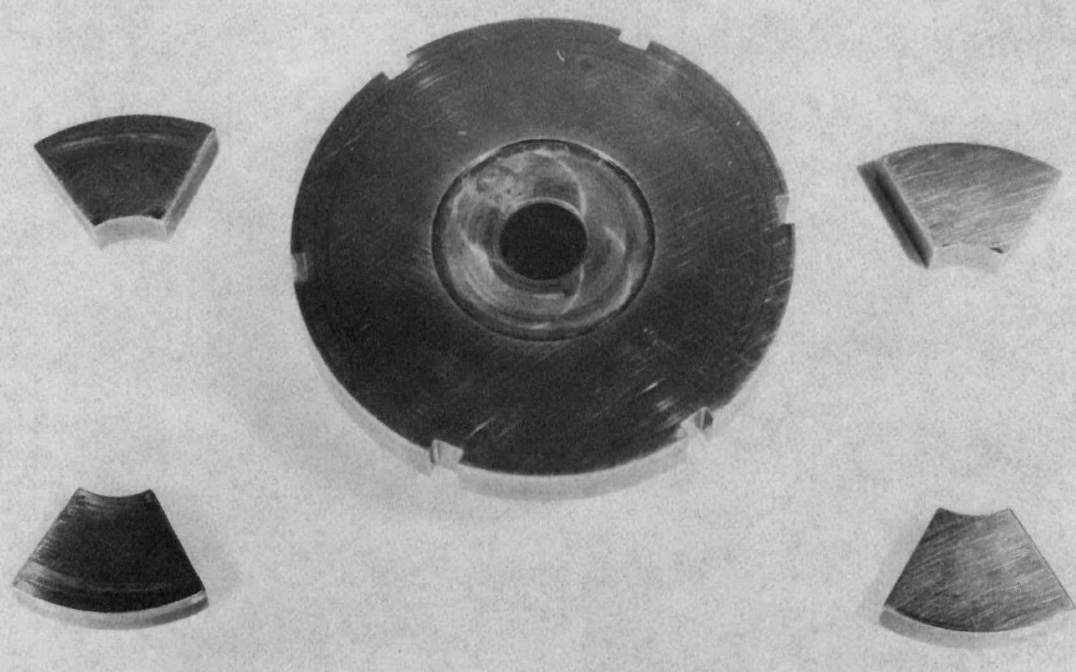


Figure 45 Thrust Bearings After Testing

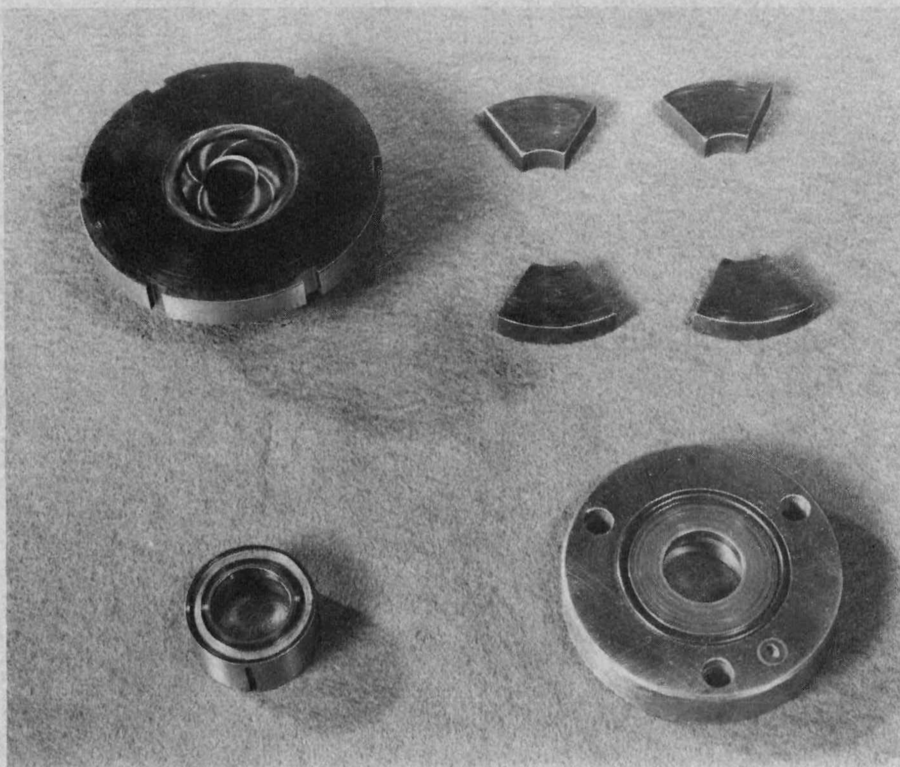


Figure 46 Face Seal Before Testing

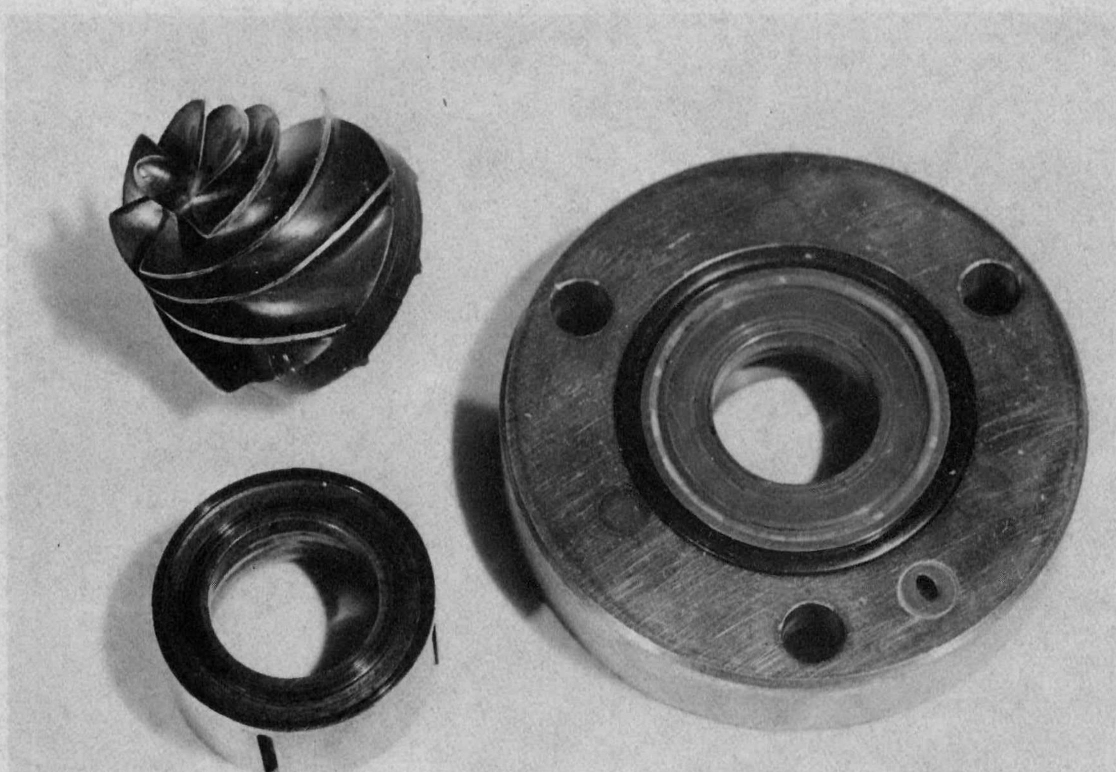


Figure 47 Face Seal After Testing

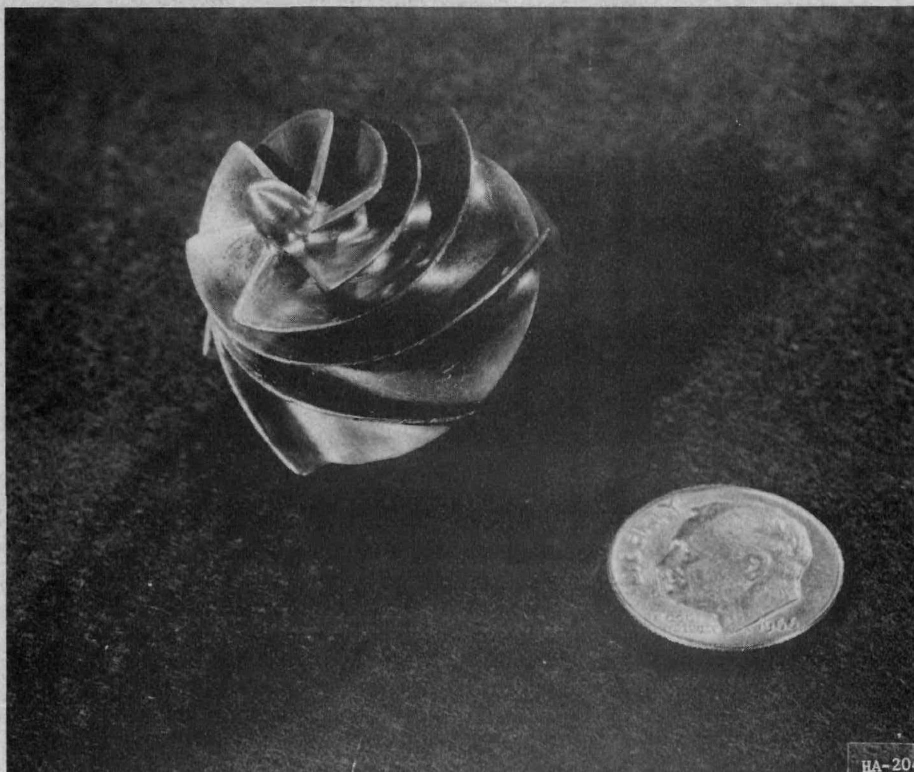


Figure 48 Pump Impeller Before Testing

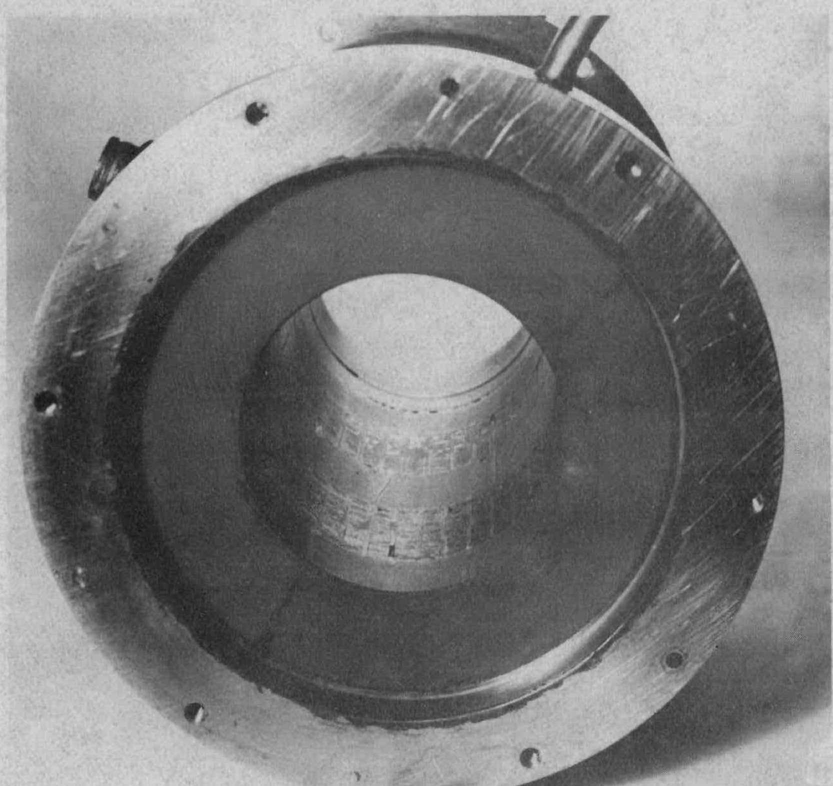


Figure 49 Stator Potting After Testing

APPENDIX A DETAILED SURFACE CONDENSER POWER CONVERSION SYSTEM HISTORY

The Power Conversion System was first assembled with the surface condenser configuration in December, 1968. After some trial runs a total of approximately three hours running was made at power levels up to 5.5 KWe. Subsequent to these three hours of operation, numerous problems arose, resulting in the inability to run for extended periods until November, 1969. A description of these problems and their solution is given in this section.

Initially, the stator windings, which had been coated with a varnish, had to be encapsulated in the stator housing with Epoxylite because of the apparent inability of the varnish to withstand the chemical attack of the eutectic working fluid.

The alternator stator windings had initially been potted with a silicone rubber compound which proved to swell into the air gap during alternator performance testing. As much rubber as could be removed without ruining the winding was removed at that time and the stator windings were then coated with the varnish which subsequently failed.

In addition, it appeared that the splash system of bearing lubrication employed for lubricating the thrust bearings was not satisfactory since scoring and scratching of their surfaces readily occurred during operation. It was decided to use a flooded cavity for the bearing housing which supports the thrust bearings, leaving the bearings at the turbine end to be lubricated as before, i.e., with a jet of liquid impinging between two pads. No further thrust bearing problems were then encountered.

However, after this difficulty several failures of the radial bearings at the pump end occurred. The original design had both bearings of 0.650 inches diameter, but a careful study of the critical speed showed that the overhung turbine configuration yielded a vibration problem, therefore the shaft size at that end was increased to 1.00 inches and a removable sleeve of AISI M-50 tool steel placed on top for ease of replacement.

The first two failures at the small pump end occurred when using flame plated tungsten carbide as the journal surface. This meant remachining and replating after a failure and it was therefore decided to go to a sleeve arrangement at that end also.

The next problem to arise was with the bearing pad adjusting screws which metallurgical inspection showed not to be as hard as designed resulting in distortion of the tip under load. In addition, it proved difficult to keep them locked in place. The combination of these two effects resulted in bearing clearances opening up well beyond design. New pins of larger diameter and spherical radius were built to reduce the contact Hertzian stresses. In addition, they were lengthened so as to extend out of housing allowing a more positive locking method to be employed.

On 21 April 1969, the Atomic Energy Commission began their support of the project. At this time fifteen runs had been made, with a total accumulated time of 228 minutes.

Three runs were then made to full speed at a low load, but each run proved noisy with rather high vibration levels. These were followed by two consecutive failures of the pump and bearings. It was suspected at this stage that large magnetic forces may have existed due to an eccentricity of the alternator rotor center of rotation to the magnetic center-line. Some experiments were performed at various field currents and locations of the rotor. However, no force larger than about 10 pounds could be measured in the worst case, and theoretically the bearing should be capable of withstanding this additional load.

The machine was reassembled and a test was performed with the alternator completely disconnected, in order that magnetic loads could be discounted. However, the unit again failed before reaching 24,000 rpm, proving that magnetic loading was not the cause of failure.

At this stage it was decided to order a new shaft with large bearings at both ends. In addition, it had been noted in previous runs that resonances appeared to occur in the generator output voltage. There was speculation that housing resonances may have been causing problems. Consequently, the alternator stator housing was vibrated through a vibration spectrum of 50 to 2000 Hertz to determine the values of the housing natural frequencies.

The largest resonance of the free housing was found to be at 240 Hertz with an amplification factor of approximately 75. It was felt that with the stator housing attached to the system this natural frequency could increase to correspond with the value at which failures tended to occur, i.e., 22,000 to 24,000 rpm. The problem appeared to be associated with the whole end cap which supports the bearings allowing large rocking amplitudes in the axial direction at resonance, which the pads could not accommodate. A new test stand was therefore built which would support the stator and end cap very firmly and prevent the large amplitudes.

The other approach at this stage was to attempt to reduce the exciting force, i.e., the rotor unbalance force, which caused high vibration and possibly failure. The rotor balance up to this time had been no better than .007 - .009 in. oz. The ability to give a better balance appeared to be restricted by the use of loose sleeves on the shaft and a turbine wheel held in place with dowel pins.

The ultimate solution to both these problems would be taper-locks and material selection which would prevent parts loosening from temperature rise or centrifugal forces. This, however, is a rather difficult machining operation with a subsequent long lead time. The use of an adhesive for the sleeves was therefore seriously considered and some compatibility tests performed to see how a Loctite product would withstand the eutectic. A measured shear strength of about 2000 psi and a good resistance to attack by the eutectic prompted the use of this adhesive. Using spring roll pins in place of dowel pins to hold the turbine wheel in place, and with this adhesive, a balance of .0015 in. oz. was obtained. In addition, material was removed from the wheel during this balance operation for the first time in case wheel unbalance should give a large couple at full speed (the rotor is balanced at about 2000 rpm).

The unit was prepared for running on July 24, 1969. Up to this time the unit had been run 23 times with a total accumulated time of 265 minutes and with 6 failures of the pump bearings. It was decided to run completely disconnected from the alternator and run the machine purely as a bearing test rig. Since there is no speed controller with the alternator disconnected, this test and all similar tests were made by spinning the machine up, unloaded, and closing the shut-off valve as the machine reached 24,000 rpm. A study of the coast-down curves allowed an immediate check to be made on bearing condition, since a decrease would indicate damage.

The first test was the quietest which had ever been made, with very low g levels. With all the following tests the machine got steadily noisier although the coast-down time increased up to as high as 7.8 minutes. The machine was disassembled once to try to determine the cause of the noise which mainly appeared to be a random rattle. Nothing significant was found. After seventeen such tests without a failure it was decided to apply a load. Four tests were made with loads up to 3.46 KWe but the noise level did not decrease at all. The machine was disassembled again for further inspection. A small rub of a brass screw seal liner had occurred but it was not felt that this was sufficient to cause all the noise.

The machine was reassembled and again run disconnected, again very noisily.

It was suggested that housing and test stand vibrations could still be the cause of the noise and that one way to lower their frequency without increasing amplitudes would be to add mass to the supports. This was done and the whole unit was then supported more directly from the floor instead of being supported on the rather flimsy mount previously used.

The next run was very quiet and it was apparent that this added mass had changed the system. The machine was then run loaded for 25 minutes. At this time an inspection was made of the bearings to make sure no damage had been done. The unit was then run again for 53 minutes with loads up to 5.3 KW. The machine was operating very smoothly at this time when it very suddenly failed. Inspection of the recordings from the run showed that the combination of bearing inlet temperature and condenser hotwell pressure allowed the lubricant in the flooded housing to boil. A pocket of vapor could have prevented liquid from lubricating the pad, resulting in failure.

On disassembly it was found that the shaft at the pump end was bent approximately .005 inches. This could have been the cause of the 800 Hertz (i.e., two per revolution) vibration which up to this time was a large component of the vibration. In addition, disassembly showed that the Epoxylite between the alternator main lamination stacks was bulging inwards. Apparently the remains of the original potting compound (a silicone rubber) were swelling under the influence of the eutectic and forcing the Epoxylite out. It was decided at this stage to wind a new alternator and to replace the old one at the earliest convenient opportunity.

To prevent further problems with boiling lubricant an orifice was placed in the drain line of the pump end bearing, raising the cavity pressure 1 psi above the condenser pressure. It was deemed unnecessary to do the same at the turbine end as this was still being lubricated by a jet directed onto the shaft with an unflooded cavity; any flashing of the liquid should not interfere with the liquid flow.

The machine was then run for three hours at 3.46 KWe. It ran fairly noisily at first but as it warmed up the audible noise level and the vibration level reduced considerably, with total vibration levels dropping from about 1.5 peak g's to 0.9 peak g's. The machine was run for these three hours in order to prove that the bearing system would work satisfactorily. The condensate temperature was kept extremely low and the power level was kept low so that the alternator would be very cool.

The machine was then run for an additional seven hours under the same conditions. The same noise characteristics were observed at start-up, but again after about 30 minutes of running the unit was very quiet and smooth.

At this stage it was decided to install the new stator windings as the new rotor had been received.

The new rotor employed two sets of three pad, pivoting-shoe radial bearings with a journal diameter of 1.1 inches. The bearing support housing at the pump end, which, in the past, has been very prone to bearing failure, was made more massive with the intention of raising the natural frequencies well above the operating range. Since the new stator was wound exactly as the first and the new rotor is the same as the old, it was not deemed necessary to repeat the calibration process.

The new machine was run for the first time on October 1, 1969 (Run #62), with the alternator electrically disconnected, i.e., a spin-up and run-down test was made. The first observation was that the run-down time was much shorter than with the previous machine and that the temperature rise across the bearings was much higher, leading to the conclusion of significantly higher power losses.

Since liquid drag in the alternator housing was suspected of being the cause of the problem, a scavenge jet pump was introduced to remove any liquid which might leak through the screw seals. However, this appeared to have little effect on the power loss.

Up to this time orifices in the bearing drains had been used to give flooded cavities for bearing lubrication. When these were removed the power losses decreased but the machine became quite noisy. Evidently liquid in the bearing housings was causing the high losses, but also damped out some of the vibrations.

Various lubrication configurations were employed to try to find the optimum method which would give quiet operation with low power losses. Attempts at a scavenged bearing housing, dependent upon splash for lubrication, were not very successful. Scratching of the loaded thrust bearings occurred during many of these runs.

Quite high bearing flows (approximately 0.1 gpm per bearing) had been run during all these tests, since it was felt that ample flow should be provided to keep the bearings cool. This proved to be a delusion since a reduction in flow (to .05 gpm) resulted in a drastic reduction in power loss.

It appeared impossible to make the machine run quietly while scavenging the bearing housing and depending upon splash lubrication. Apparently the unloaded thrust bearings were rattling considerably, while the loaded ones appeared to be very susceptible to scoring and metal pick-up, presumably due to poor lubrication.

Individual lube jets for the loaded thrust bearings were added and the flow rates adjusted to try to obtain successful operation. At this stage, during Run #92 on October 16, 1969, a failure of the pump end radial bearings occurred. Disassembly showed the presence of many wear chips in the internal, unloaded thrust bearing cavity.

These wear chips were non-magnetic and were suspected to have come from the chrome-plated thrust face. Apparently a chip had reached a radial bearing pad and caused failure. No damage occurred to the turbine end radial bearings.

Before reassembly, individual lube jets were provided for the radial bearings at the pump end and for the unloaded thrust bearings.

Further lubrication configurations were attempted until by Run #104, November 5, 1969, the final operating configuration was chosen and maintained for the remainder of the surface condenser testing. This employed individual lube jets for all bearing pads at the pump end. Separate scavenged drains for the inner and outer thrust bearing cavities enabled the outer thrust disc to run dry (except for that liquid required for lubrication) while a short drain stand-off in the inner cavity gave an annular ring of liquid of known size at the O.D., just sufficient to damp out the pad rattle.

With the final operating configuration established, full speed power losses could be calculated from the slope of the run-down curve at 24,000 rpm using the value for the mass inertia of the rotating portion of the machine.

Spin-down tests, see Figure A-1, with this configuration showed a total tare loss of approximately .450 KW of which 70 watts are calculated to be windage losses at a cavity pressure of 0.1 psia. Spin-down curves are shown at different bearing flow rates. These flow rates are higher than normal operating values (approximately .08 gpm total flow) since, at shutdown, bearing flow increases as the system flow goes to zero. The quoted value is based on extrapolation to the operating value. Thus the bearing power losses are somewhat higher than their calculated values, which were 125 watts per radial bearing, and approximately 40 watts for the thrust bearings (approximate because of lack of complete definition of thrust load).

At this stage it was decided to run the machine for a period of time to prove that the operating problems had been ironed out. The machine was therefore run until a total of 100 hours had been accumulated. This time was achieved at off-design thermodynamic conditions, and using a facility pump to provide system flow. Typical operating conditions during this period are given in Table A-1 taken from Run #108.

The operating behavior of the machine followed a reproducible pattern during the start-up mode. Having first reached full speed the CRU operated quietly, with vibration 'g' levels less than 1.0 (monitored at the pump housing). As it warmed up, however, it would start running more noisily with 'g' levels going as high as

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Table A-1 Summary of Operating Conditions During 100 Hour Operation

System Flow Rate	:	450 lb/hr
Turbine Inlet Temperature	:	672°F
Turbine Inlet Pressure	:	60 psia
Turbine Exit Pressure	:	0.4 psia
Condenser Hotwell Temperature	:	215°F
Regenerator Vapor Inlet Temperature	:	572°F
Regenerator Vapor Outlet Temperature	:	315°F
Regenerator Liquid Inlet Temperature	:	207°F
Regenerator Liquid Outlet Temperature	:	468°F
Total Electrical Power	:	4.23 kwe

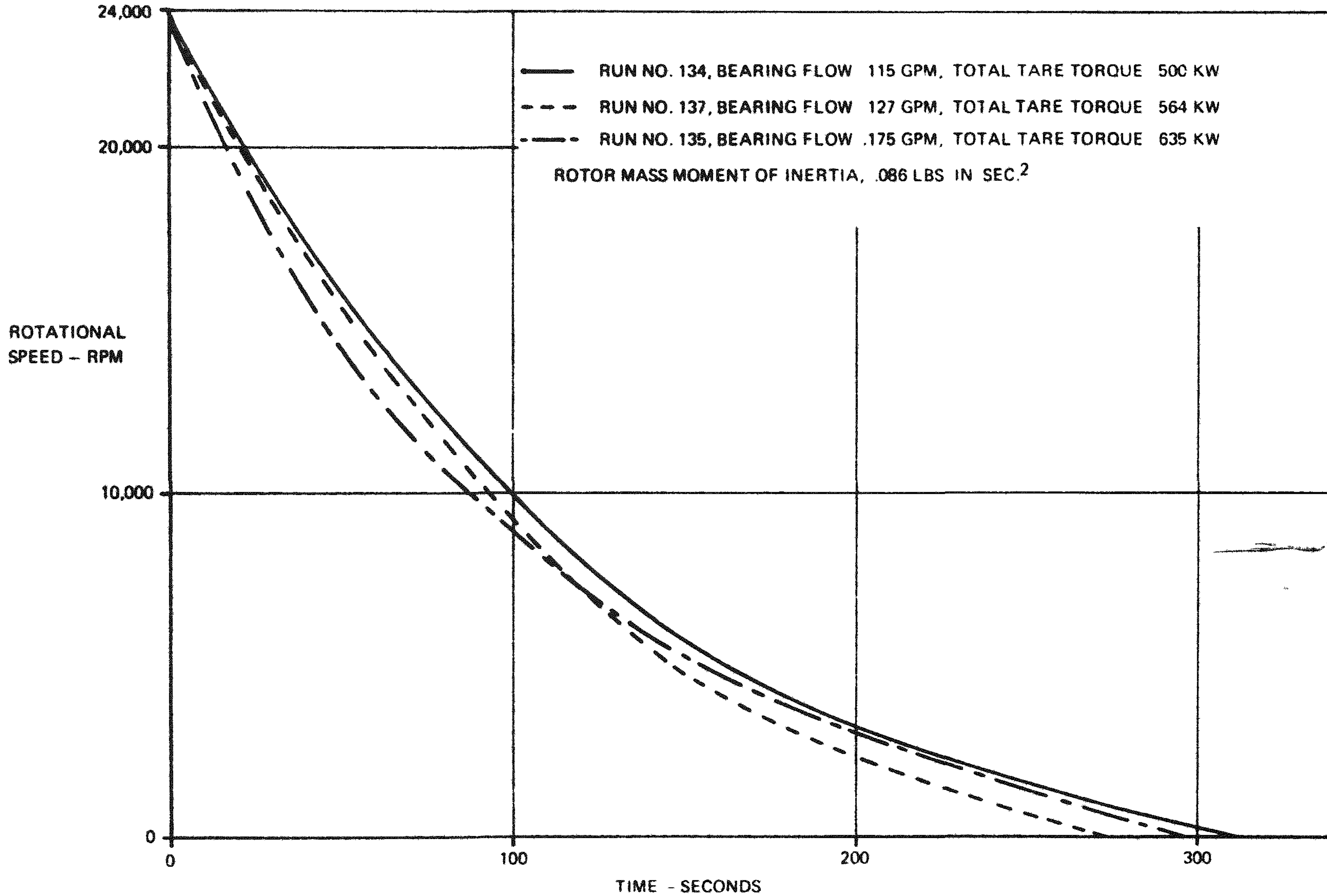

Figure A-1 Tare Torque Determination from Run-Down Curves

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3.0 g's. Finally when the hotwell was above about 200°F and the whole machine had warmed up it would quiet down considerably and run at vibration levels of less than 1 g again.

After the first 100 hours had been achieved, some test runs were made to try to determine the source of the noisy operation at start-up. Varying the back-pressure had no effect on the noise or vibration level. The noise level appeared to be a function of temperature level. If the hotwell temperature was allowed to come up slowly, the CRU would begin to quiet down above about 150-160°F and by 200°F it would be quiet. However, if the temperature was brought up very quickly to 200°F (by drastic reduction in glycol flow rate), it would still be noisy at 200°F, but after a period of time at that temperature it would quiet down.

It appears that the noise is associated with a critical speed. An increase in temperature has two effects on the bearing operation. The decrease in viscosity will cause an increase in bearing stiffness, which will follow hotwell temperature almost immediately. On the other hand, an increase in temperature will cause an increase in bearing clearance because of differential expansion rates. This will probably result in a decrease in bearing stiffness, but this effect will be more slow, since the housing will take some time to heat up.

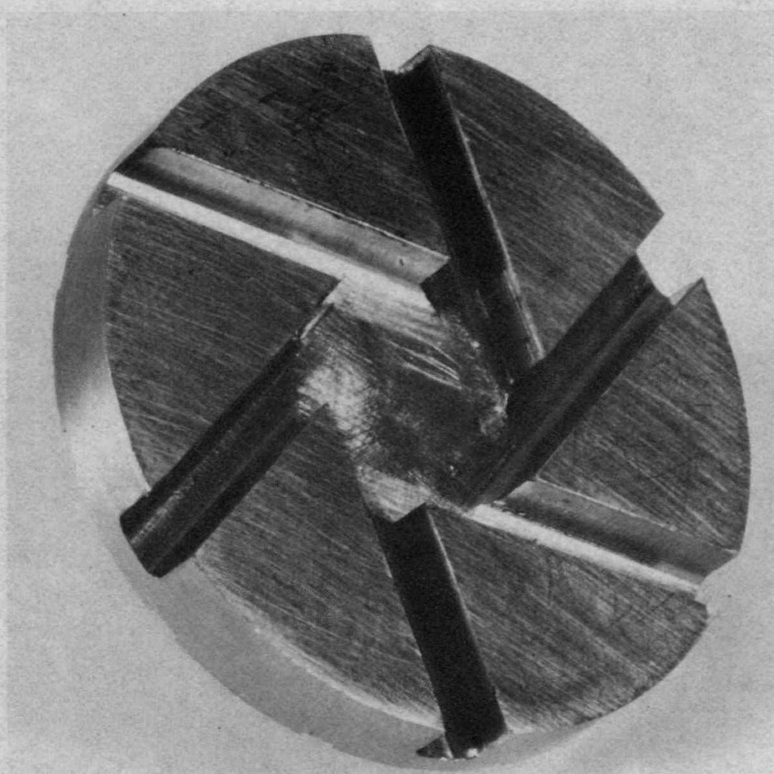
Therefore, it seems that at the constant operating speed the machine experiences a thermally induced critical speed resulting in noisy operation during that time.

Attempts were made at this stage to run at design operating conditions. The hotwell was brought up to design temperature and pressure, but the facility Chempump was incapable of delivering design flow rates. As the flow rate was increased the boiler inlet pressure decreased until the limiting stage was reached where there was insufficient pressure to drive more flow through the boiler.

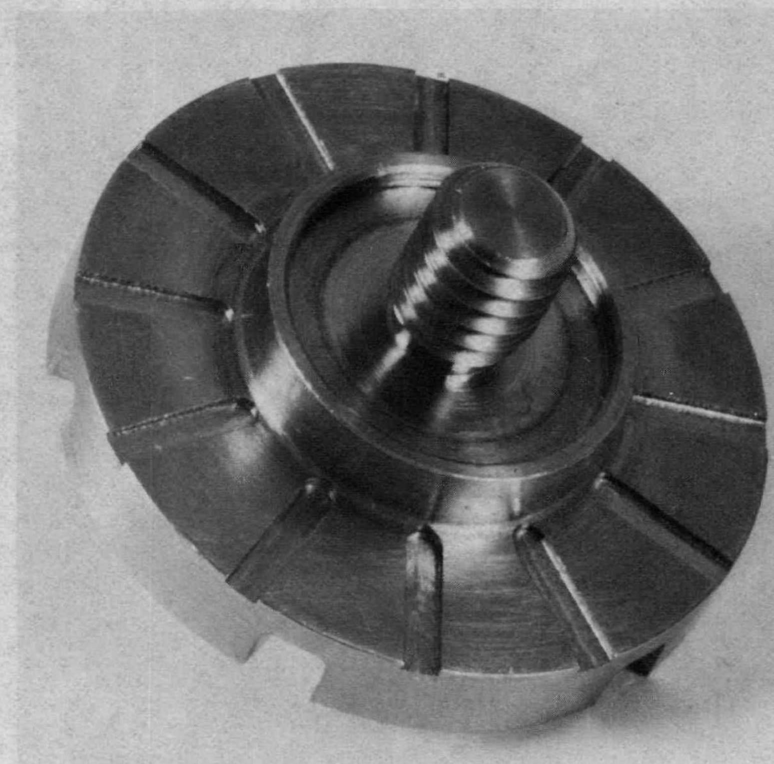
The pump impeller was put on the machine at this stage, with a back-side clearance of .016" and front-side clearance of .027". The impeller, Figure A-2, is a modified form of the Barske pump, using angled slots instead of radial vanes. There are radial slots on the back side of the pump to reduce pump leakage.

A jet pump was plumbed in parallel with the facility pump jet pump in order to induce the CRU pump. This could be driven by either the facility Chempump or the CRU pump.

The operating procedure for pump testing involved running on the facility pump until the system had attained quiet operation. The CRU pump outlet valve was then opened and primary flow introduced from the jet pump (driven at this stage by the facility pump).



IMPELLER FRONT SIDE



IMPELLER BACK SIDE

0 INCHES 10

Figure A-2 The 6 KWe PCS Pump Impeller

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The flow from the CRU pump was then increased, either by increasing the system flow rate or by valving out the facility pump. In this way head-flow characteristics could be obtained.

When the pump was started two things were immediately noticed. The vibration levels at the pump end increased considerably, at low flow rates going up to 3 g's, while for high flow rates reaching as high as 7 g's. This fact was attributed to hydraulic noise from the pump. The levels at the turbine end decreased, however, possibly due to damping of shaft vibrations by the liquid.

The available electrical power dropped considerably, by as much as 2 KWe. In addition the bearing outlet temperature increased slightly and the bearing cavity pressure increased from its normal level of about 0.6 - 0.7 psia to values as high as 1.5 psia. Apparently high leakage down the back of the pump and through the screw seal was occurring. This flow was leaking into the bearing cavity, where churning losses from the 2.5 inch diameter thrust runners were occurring.

The best that could be obtained from the CRU pump under these conditions was about 2.2 gpm at 90 psia. The CRU pump inlet pressure was as low as 2.8 psia (it was not felt to be safe to go lower) and the jet pump mass ratio was 1.3:1.0. It was felt that the jet pump should be able to perform better (it may not have had sufficient secondary pressure) and therefore it was replumbed in a lower position with lower pressure drop secondary lines. Pressure-flow characteristics for the pump are shown in Figure A-3.

At the same time the bearing inlet lines were plumbed directly to the facility pump, to provide better protection in the event of losing flow from the CRU pump. The new loop schematic is shown in Figure A-4 with a photograph in Figure A-5.

It was decided to try to reduce pump leakage by reducing the back-side clearance to .005". It was felt that allowing the front clearance to increase would not affect pump characteristics significantly. Inspection of Figure A-3 confirms this. Also a flow meter was inserted in the CRU pump primary so that a direct measurement of pump leakage could be obtained.

With the new plumbing the pump was used to deliver the entire system flow, i.e. with the facility pump delivering bearing flow only. A maximum system flow rate of .86 gpm, i.e. 422 lbs/hr., was obtained. At this condition the CRU pump was delivering 2.5 gpm at 85 psia with a jet pump mass ratio of 1.2:1. The additional flow was being used for the jet pump and the scavenge pump flow. The latter was not allowed for in the original design and is the reason for the inability to reach design system flow. By comparison of inlet and outlet flows to and from the CRU pump, it was determined that approximately 0.2 gpm was leaking into the bearing cavity. This combined with the increase in liquid temperature allowed a direct calculation to be made of the churning power losses.

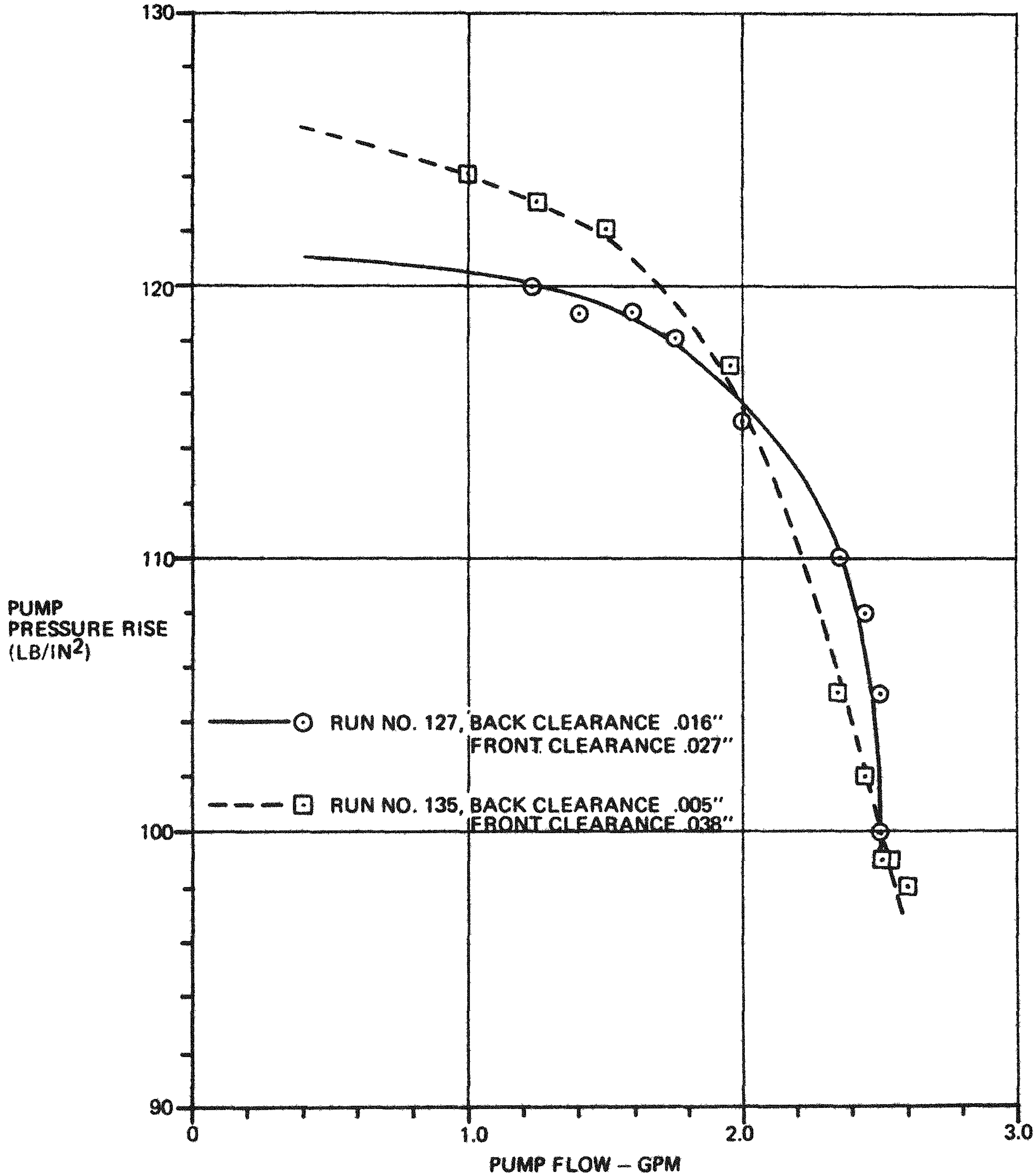

Figure A-3 CRU Pump Pressure Rise vs. Flow

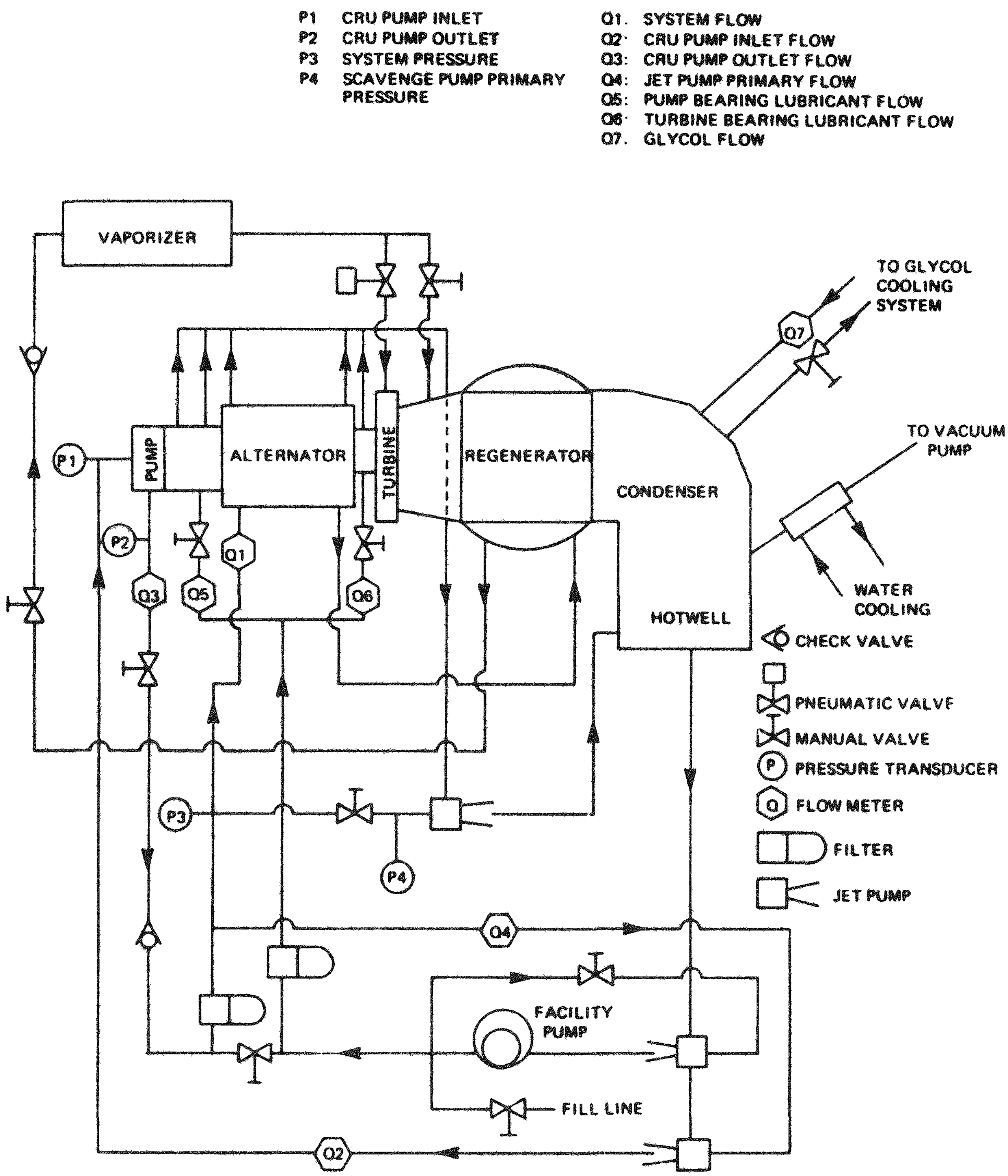

Figure A-4 6 KWe Power Conversion System Loop Schematic

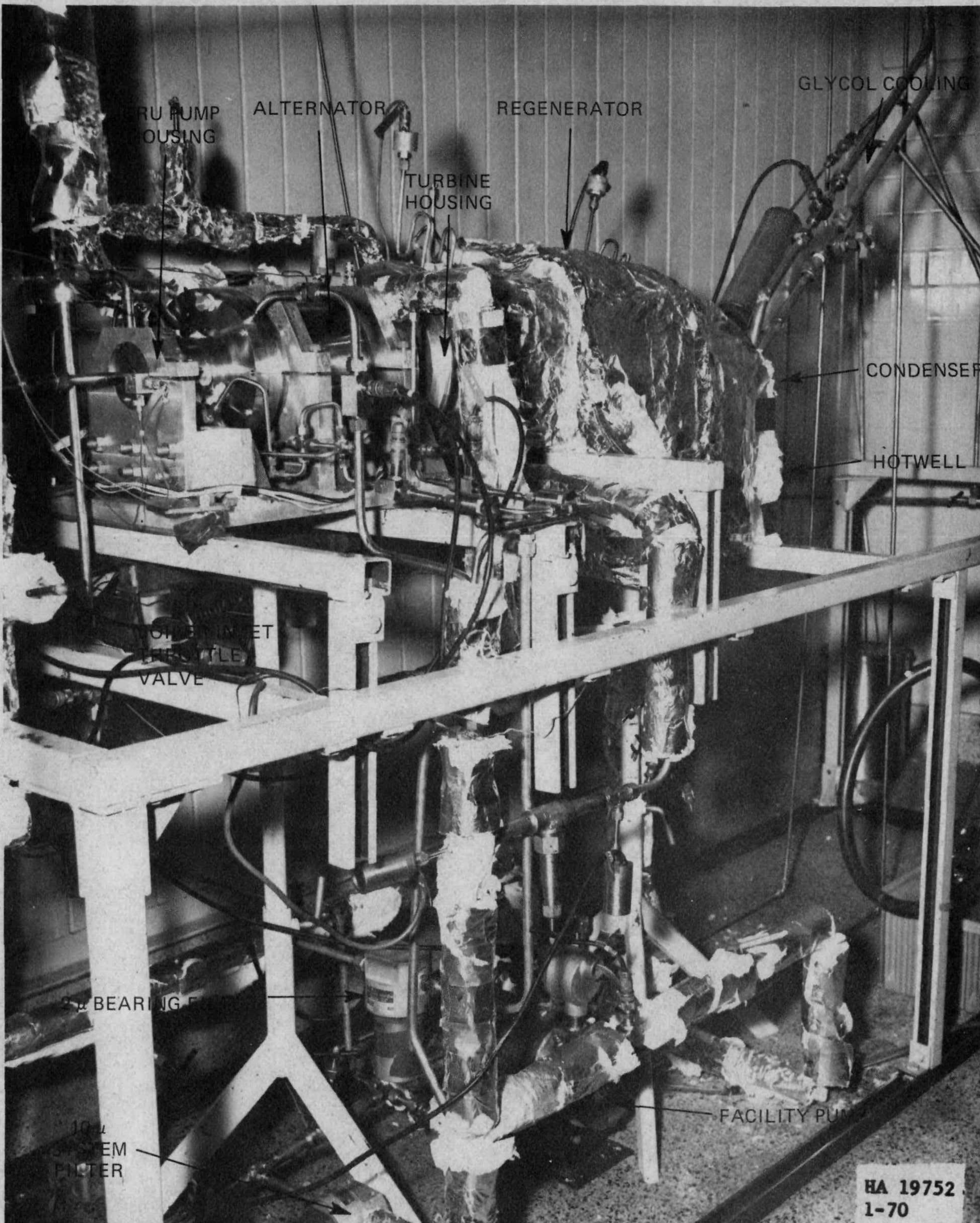

Figure A-5   6 KWe Power Conversion System Test Loop

After this test a run (#138) was made using both the facility Chempump and the CRU pump to deliver system flow rates around the design value (535 lb/hr), to obtain data points. It was decided at this stage that no attempt would be made to eliminate the pump leakage or to increase the pump flow, but that when the new pump impeller, designed for jet-condenser flow rates, was integrated into the system these problems would be solved.

Table A-2 presents a summary of the history of the machine in the surface condenser configuration.

Surface Condenser Performance Evaluation at 650°F

Five data points were taken during run #138 at different system flow rates around the design value. Two points were immediately discarded because of wet turbine flow, shown by the low turbine exit temperature.

These data were then reduced to give turbine and cycle efficiencies. The methods of obtaining these efficiencies are given in the following sections. In addition, the regenerator effectiveness was determined and is presented.

Determination of Turbine Wheel Efficiency

For this determination it was necessary to compare the actual shaft work done by the turbine wheel to the ideal turbine power based on the enthalpy drop and mass flow.

The determination of the shaft power is complicated by the additional churning losses resulting from the pump leakage. Once the valve of leakage was measured by comparison of the pump inlet and outlet flows, the total flow passing into the bearing cavity could be obtained. This, combined with the liquid temperature rise across the bearing cavity, gave a value for the total power loss in the cavity. The total bearing losses were determined by inspection of spin-down curves. These total losses were approximately .390 KW, while the calucalted loss for one radial bearing is 125 KW. If the assumption is made that the latter value is slightly low and a value of .150 KW is instead assigned to a radial bearing loss, then the pump end bearing losses become .24 KW. This loss was subtracted from the total loss measured by temperature rise and hence the value resulting from liquid churning was obtained.

The pump hydraulic work was obtained from the head rise and mass flow. An estimate of 40% was made for the pump efficiency based on past experience with similar pumps, which enables a value of equivalent shaft work to be determined.

The efficiency of the alternator (which includes rotor windage) at the operating power level was found from Figure A-6. This, combined with actual observed electrical power, gives the equivalent shaft power.

Table A-2 Summarized History of 6 KWe Power Conversion System - Surface Condenser Configuration

Total Hours On Old Rotor	:	19.6
Total Starts On Old Rotor	:	61
Total Hours On Modified Rotor	:	141.6
Total Starts On Modified Rotor	:	78
Total Hours On Existing Pump Bearings	:	138.5
Total Starts On Existing Pump Bearings	:	47
Total Hours On Existing Turbine Bearings	:	105.5
Total Starts On Existing Turbine Bearings	:	30
Total Hours With CRU Pump	:	35.9
Total Starts With CRU Pump	:	16

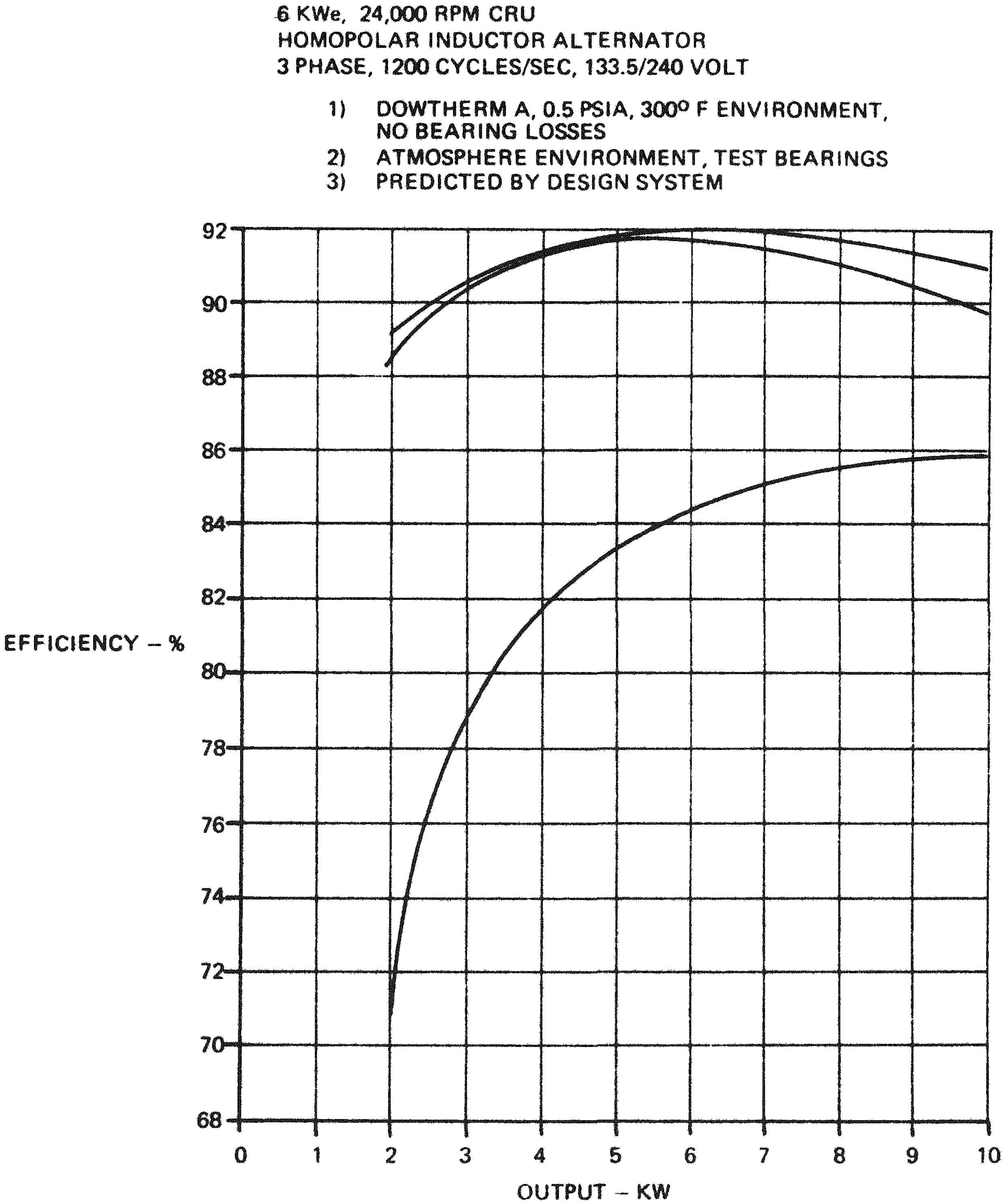

Figure A-6 6 KWe PCS Alternator Performance

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Thus the total shaft work done by the turbine, consisting of; bearing losses, chruning losses, dissipative alternator losses, hydraulic pump work and useful electrical power, may be obtained. These values were then compared to the ideal turbine power to give the turbine efficiency, as shown in Table A-3.

Determination of Cycle Efficiency

The cycle efficiency is based upon the electrical output and the heat added in the boiler. The heat rejected based on enthalpy change across the condenser would be complicated by the high flow rates used for the scavenge pump (which have been cooled in the facility pump) and by the hot liquid from the bearing drains entering the condenser.

Thus the Power Conversion System efficiency will be given by

$$\eta_{PCS} = \frac{P_{electrical}}{Q_{added}}$$

The existence of heat losses in the system make this value a lower limit estimate.

It was felt that the determination of cycle efficiency from these data points should make the assumption that pump leakage could be eliminated. In this case the additional power loss attributed to liquid churning would be directly available for useful electrical power. Therefore, if the churning losses are multiplied by the alternator efficiency at the approximate operating power level, a total value of useful electrical power can be obtained and used for calculating η_{PCS}.

On the other hand the CRU pump was not supplying the total flow during the test. Previous testing of the pump showed it to be capable of supplying about 0.80 gpm (with adequate scavenge pump flow) out of 1.02 gpm system flow. The difference between these two flows, plus the additional jet pump primary flow (with an estimated 1.2:1 mass ratio) amounts to approximately another 0.5 gpm required from the CRU pump. The pumping power required for this extra flow must be subtracted from the observed electrical output.

The heat added in the boiler was calculated from the enthalpy increase from the regenerator outlet to the turbine nozzle inlet and the system mass flow rate. The resulting estimates for cycle efficiency are shown in Table A-4.

In Figure A-7 a comparison is made of design state points and the state points which existed in data point number 5 of run #138.

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Table A-3 Determination of 6 KWe PCS Turbine Wheel Efficiency From Run No. 138

Data Point Number		:	1	4	5
Pump Leakage Flow	lb/hr	:	122.0	122.0	118.0
Pump Bearing Lubricant Flow	lb/hr	:	20.5	20.5	20.5
Total Bearing Cavity Flow	lb/hr	:	143.0	143.0	139.0
Bearing Cavity Temperature Rise	^{o}F	:	100.0	101.0	96.0
Total Bearing Cavity Power Loss	kw	:	1.84	1.84	1.70
Churning Power Loss	kw	:	1.60	1.60	1.46
Total Bearing Power Loss	kw	:	.39	.39	.39
Pump Flow Rate	lb/hr	:	1200.0	1250.0	1270.0
Pump Head Rise	ft	:	200.0	202.0	191.0
Pump Hydraulic Work	kw	:	.091	.092	.087
Pump Shaft Power (at 40% Efficiency)	kw	:	.226	.229	.216
Alternator Electrical Power	kwe	:	4.45	4.25	4.45
Alternator Shaft Power (at 91.5% Efficiency)	kw	:	4.85	4.65	4.85
Dissipative Alternator Shaft Power	kw:	:	.05	.05	.05
Total Shaft Power	kw	:	7.12	6.92	6.97
Turbine Inlet Pressure	psia	:	68.7	66.7	65.7
Turbine Inlet Temperature	^{o}F	:	660.0	664.0	674.0
Turbine Outlet Pressure	psia	:	.30	.29	.29
System Flow Rate	lb/hr	:	529.0	509.0	501.0
Ideal Turbine Power	kw	:	10.68	10.34	10.22
Turbine Wheel Efficiency		:	.67	.67	.68

Table A-4 Determination of 6 KWe Power Conversion System Cycle Efficiency

Data Point Number		:	1	4	5
Measured Electrical Power	kwe	:	4.45	4.25	4.45
Measured Churning Power Loss	kw	:	1.60	1.60	1.46
Additional Pump Power Required*	kw	:	0.062	0.049	0.040
Equivalent Alternator Power (at 92.0% Efficiency)	kwe	:	1.41	1.42	1.30
Net Available Electrical Power	kwe	:	5.86	5.67	5.75
Turbine Inlet Pressure	psia	:	68.7	66.7	65.7
Turbine Inlet Temperature	^{o}F	:	660.0	664.0	674.0
Turbine Inlet Enthalpy	BTU/lb	:	400.5	402.6	407.6
Regenerator Liquid Outlet Temp.	^{o}F	:	448.0	449.0	457.0
Regenerator Liquid Outlet Enthalpy	BTU/lb	:	175.0	175.7	180.0
Enthalpy Rise Across Boiler	BTU/lb	:	225.0	226.9	227.6
System Mass Flow Rate	lb/hr	:	529.0	509.0	501.0
Heat Added in Boiler	kw	:	34.9	33.6	33.4
Cycle Efficiency (η_{PCS})		:	.168	.169	.172

*Based on a pump efficiency of 40%

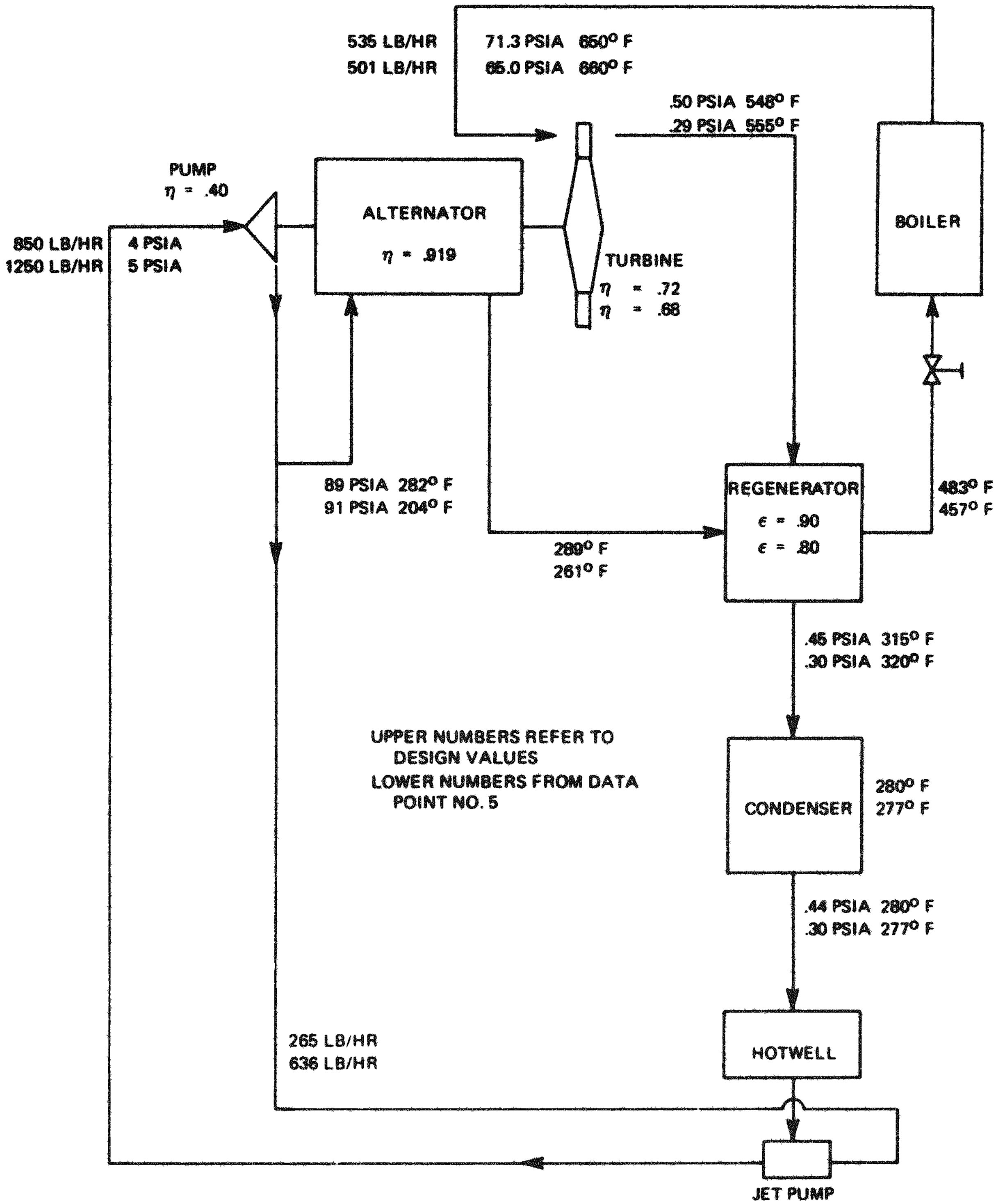


Figure A-7 Comparison Between Design Points and Values Obtained During Data Point No. 5, Run No. 138

Determination of Regenerator Effectiveness

Comparison of the heat removed from the vapor to the heat added to the liquid in the regenerators shows little heat loss to the surroundings, presumably because of good insulation.

The regenerator effectiveness is defined as

$$\epsilon = \frac{TRVI - TRVO}{TRVI - TRLI}$$

where TRVI is the vapor inlet temperature
TRVO is the vapor outlet temperature
and TRLI is the liquid inlet temperature.

Table A-5 gives the effectiveness values obtained during the testing. It can be seen that values of effectiveness of 0.80 were measured. This is lower than the design value of 0.90, but it was shown in previous tests on a spare regenerator that a gravity-induced flow maldistribution existed in the regenerator modules when oriented as in the test loop. The two regenerators to be used for future testing of the 6 KWe PCS have had flow restriction orifices added to increase their liquid side pressure drop which corrects the flow maldistribution.

If these regenerators had been used in the surface condenser testing, higher cycle efficiencies would have resulted. The values of cycle efficiency can be calculated based on the regenerator liquid outlet temperature which would have resulted with a .90 effective regenerator.

$$TRLO = TRLI + \epsilon \frac{C_{PV}}{C_{PL}} (TRVI - TRLI)$$

where ϵ = regenerator effectiveness
C_{PV} = vapor specific heat (0.41 btu/lb)
C_{PL} = liquid specific heat (0.48 btu/lb)
TRLO = regenerator liquid outlet temperature.

Using this new value of liquid outlet temperature, a new value of heat added in the boiler and hence cycle efficiency may be obtained. The results of this investigation are also shown in Table A-5.

Table A-5 Regenerator Effectiveness From Run No. 318, and the Effect of a 0.90 Effective Regenerator on Cycle Efficiency

Data Point Number		:	1	4	5
Regenerator Vapor Inlet Temp.	°F	:	534.0	543.0	553.0
Regenerator Vapor Inlet Pressure	psia	:	0.30	0.29	0.29
Regenerator Vapor Inlet Enthalpy	BTU/lb	:	345.8	349.9	354.4
Regenerator Vapor Outlet Temp.	°F	:	320.0	316.0	320.0
Regenerator Vapor Outlet Pressure	psia	:	0.30	0.30	0.30
Regenerator Vapor Outlet Enthalpy	BTU/lb	:	258.0	256.6	258.0
System Mass Flow Rate	lb/hr	:	529.0	509.0	501.0
Vapor Side Heat Loss	BTU/hr	:	46500*	48800	51000
Regenerator Liquid Inlet Temp.	°F	:	264.0	260.0	261.0
Regenerator Liquid Inlet Enthalpy	BTU/lb	:	86.3	84.5	85.0
Regenerator Liquid Outlet Temp.	°F	:	448.0	449.0	457.0
Regenerator Liquid Outlet Enthalpy	BTU/lb	:	175.0	175.7	180.0
Liquid Side Heat Gain	BTU/hr	:	47000*	48200	50200
Regenerator Effectiveness		:	.792	.802	.799
Regenerator Liquid Outlet Temp. with 0.90 Effective Regenerator	°F	:	471.0	477.0	485.0
Regenerator Liquid Outlet Enthalpy	BTU/lb	:	187.2	190.5	194.6
Turbine Inlet Enthalpy	BTU/lb	:	400.5	402.6	407.6
Heat Added in Boiler	kw	:	32.8	31.3	31.2
Useful Electrical Power	kw	:	5.82	5.62	5.70
Cycle Efficiency Using 0.90 Effective Regenerator		:	.177	.179	.182

*This discrepancy probably results from an erroneous thermocouple reading

APPENDIX B DETAILED JET CONDENSER CONFIGURED SYSTEM HISTORY AT 650° TURBINE INLET

After the 650°F turbine inlet temperature testing was completed, using a surface condenser, with which a cycle efficiency of 17% was achieved, the unit was disassembled and a new test stand was built, designed to allow unattended endurance operation.

An electrically heated salt bath boiler was designed for the jet condenser operation, since it was felt to be inherently more dependable. This boiler had a total capacity of 60 KW electrical input of which 15 KW was modulated by silicon controlled rectifiers and the other 45 KW consisted of three banks of constant power, which could be added one at a time. This allowed the turbine inlet temperature to be set at any predetermined level. The boiler tube itself consisted of a one-pass, helically coiled tube immersed in the molten salt.

The new test loop, shown in Figure B-1 and B-2 incorporated a jet condenser and associated plumbing.

The system reservoir was a tank which used pressurized nitrogen on one side of a rubber bladder to control the jet condenser recovery pressure. The jet condenser outlet commutes with the reservoir as shown in Figure B-2.

Positive displacement facility pumps were provided in the system for start-up and prelimianry check-out. A large, low pressure drop filter was incorporated to protect the jet condenser nozzles and the total system from any chips of foreign particulate matter, while a separate filter was employed for the bearing flow.

Where feasible the system plumbing was welded or brazed in order to reduce the possibility of air leaks. All the plumbing subassemblies were thoroughly cleaned and leak checked prior to assembly and the final unit was also leak checked. A gold 'o' ring seal was again used for the hot gas seal, since it was felt that this was the best solution available for use with the existing non-weldable flanges.

Two problem areas had arisen during the testing of the PCS with the surface condenser, which at the time of installation of the jet condenser had not been solved. Firstly, the tilting pad bearing pivots had shown signs of definite fretting erosion after 142 hours operation on the surface condenser. Also the visco seal employed to prevent overboard leakage from the centrifugal pump to the bearing cavity had been ineffective in preventing leakage and high churning losses. It was realized that the jet condenser version would generate higher pump pressures and consequently has a potentially higher leakage rate.

A hydrostatic face seal was designed, which depends on pump pressure to provide hydrostatic lift, allowing small axial clearances to be

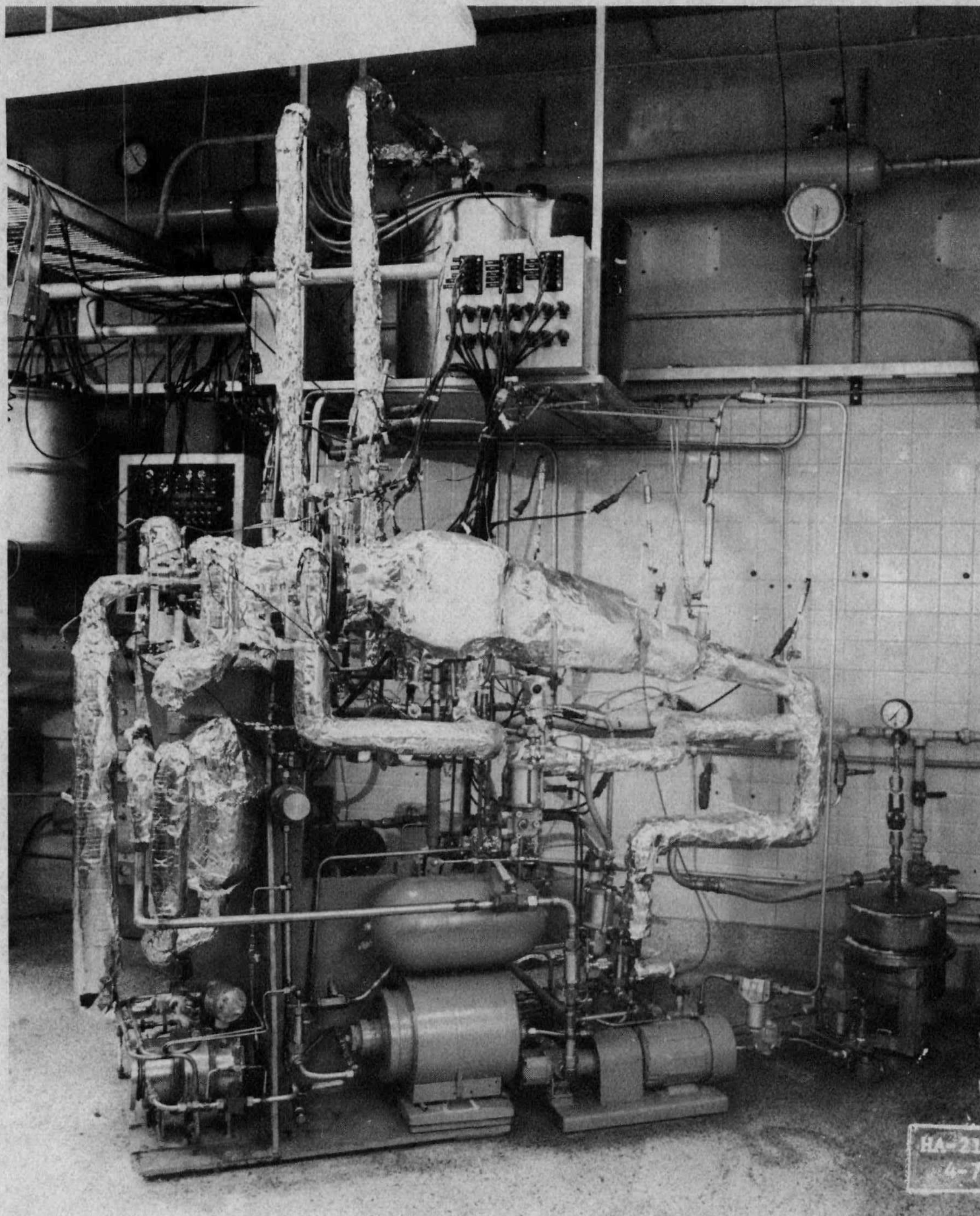
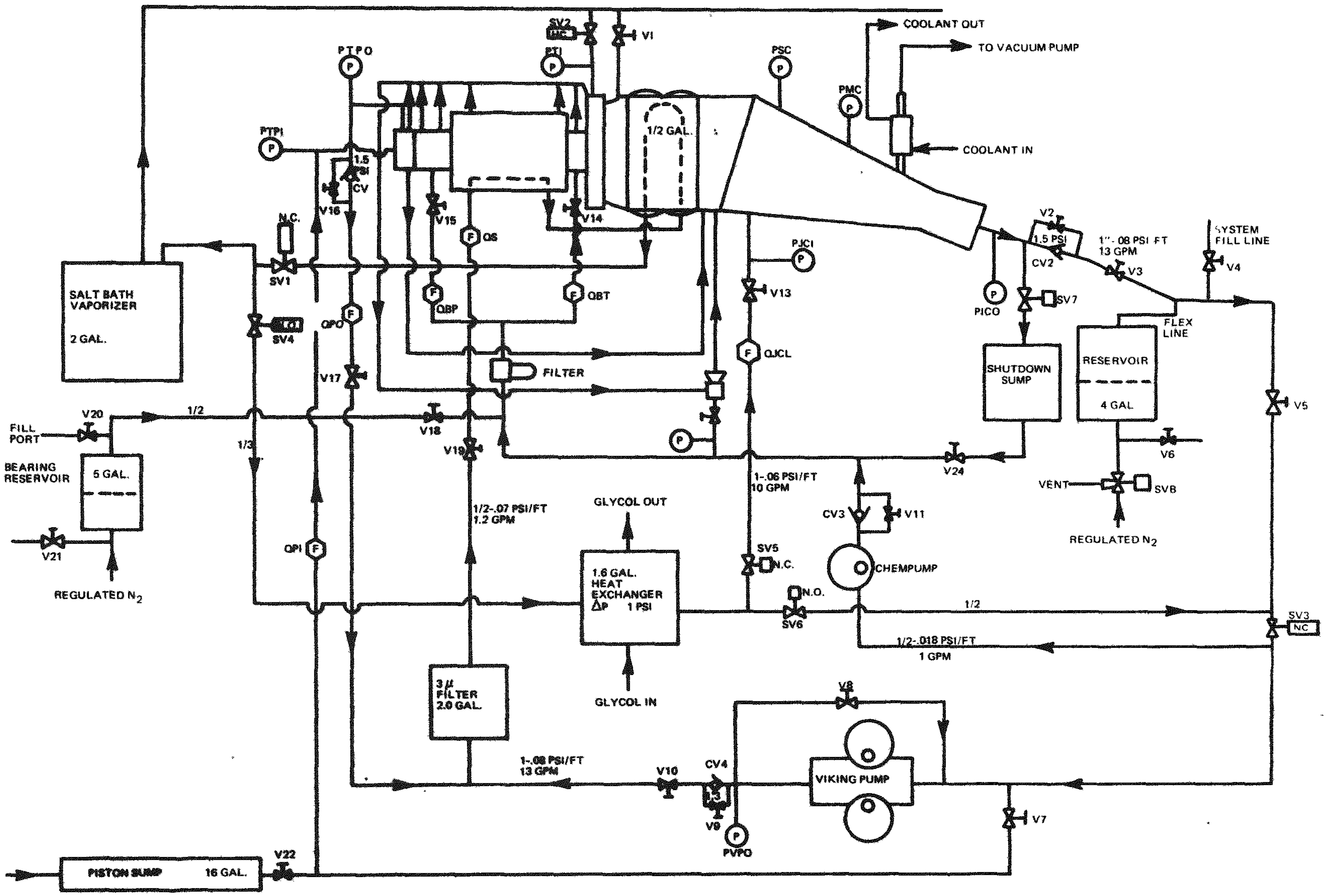


Figure B-1 Endurance Test Loop



P PRESSURE TRANSDUCER

FLOW METER

Z6PT – 1 TUBING – 1 GAL

OPERATING INVENTORY: 8 GAL.
RESERVOIR · 4 GAL.

Figure B-2 Test Loop Schematic for Endurance Operation maintained and a theoretically infinite life. The bearing pivots were coated with various dry film lubricants on the first assembly so that the coating with the most satisfactory wear performance could be applied in future builds.

The pump impeller originally intended for use on the PCS was a multivaned centrifugal pump cast in stainless steel from a brass impeller designed to run at 20,000 rpm. All attempts at this casting were unsuccessful and finally it was decided to employ the original brass impeller with slight modification. The impeller is shown in Figure B-3.

The test loop was designed to enable unattended operation with fail-safe emergency shut-down capability, Basically, certain important parameters, notably, bearing flow, rotational speed, bearing housing vibration, jet condenser recovery pressure and boiler outlet temperature were sensed by appropriate instrumentation. When the value of the parameter did not meet predetermined upper or lower limits an emergency shut-down was automatically called for.

When an emergency shutdown occurred, the turbine inlet and boiler inlet valves closed and a boiler vent valve opened, allowing the boiler inventory to be vented to the system reservoir which at the same time was vented to atmospheric pressure, thus preventing large liquid inventories from being at the high boiler temperature for extended periods of time. In addition, emergency bearing flow and scavenge pump flow was provided for three minutes, which was ample time for rotor spin-down, since the latter was greatly shortened by the addition of full electrical load (12 KW) on a shutdown. The liquid used for bearing lubrication and scavenge pump flow, and any liquid which remained in the jet condenser, was drained from the condenser into a sump through a valve which opened on shutdown. The jet condenser inlet valve was, of course, also closed and the boiler electrical power turned off at the same time. Emergency bearing flow was provided by a pressurized tank plumbed in parallel with the lube pump.

Instrumentation was added to the system to measure all important operating parameters and these values were recorded in digital format at preselected time intervals. In addition, a continuous strip recording was made of the more important parameters. The control console shown in Figure B-4 allowed different alternator loads to be selected, thus adjusting the load distribution between the main alternator and speed control dissipator alternator. Electrical boiler heater power could also be selected, depending on the system flow rate. The test stand assembly was completed in April 1970 using the 650°F nozzle block. Conversion to the high inlet temperature would merely require a change of nozzle block. Before rotation the jet condenser operation was checked out and liquid flowed through the lines to flush them thoroughly.

Initial checkout of the rotating hardware was performed without the pump or its seal, and system pressure was provided by facility

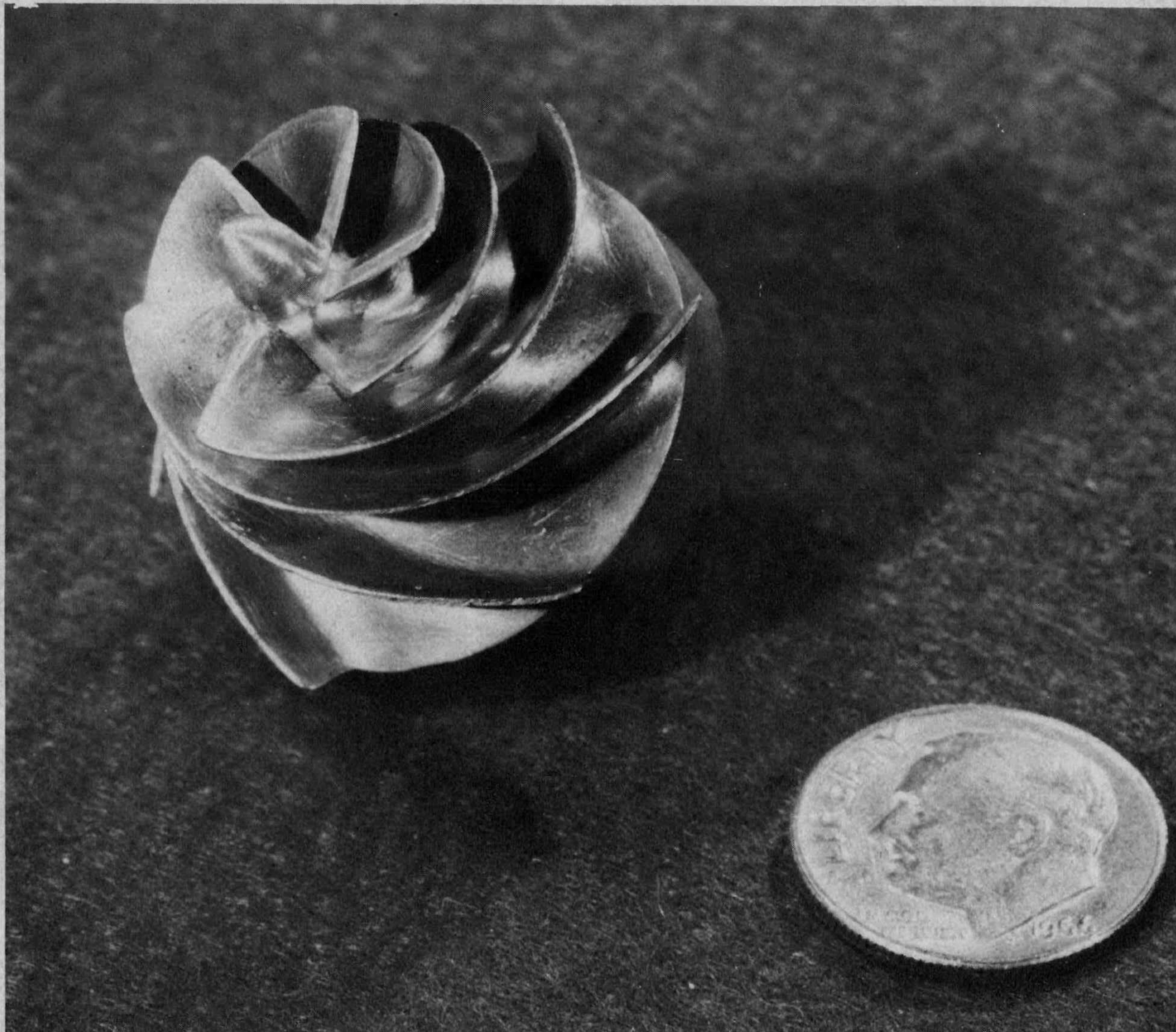


Figure B-3 Pump Impeller for Jet Condenser Configuration

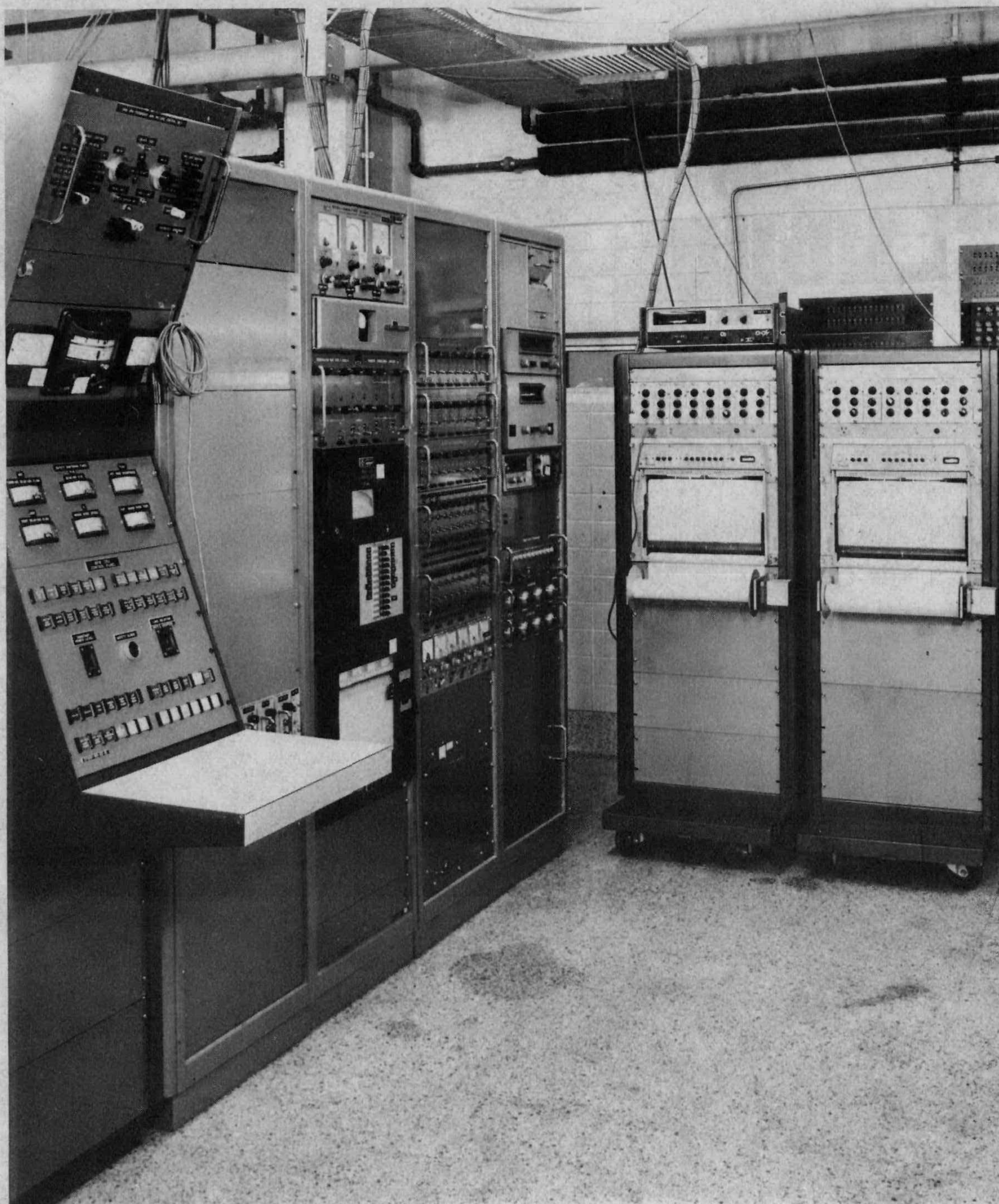


Figure B-4 Endurance Operation Control Console and Instrumentation pumps. The initial test runs were dynamically successful, but speed control and power output were both unobtainable. These problems were traced to blown fuses and reversed rotor polarity.

Many shakedown runs were then performed on the system. Unfortunately air inleakage caused jet condenser floodouts on many occasions before turbine rotation, but this was finally traced to leakage around the gold seal and the partial blocking of the vacuum pump line by scavenge pump flow when no vapor was being passed.

At this stage full speed operation could not be obtained even at high system flows. It was found that the alternator housing was funning flooded and that very high churning losses were occurring.

With these shakedown problems rectified the machine ran smoothly with low vibration levels and was capable of delivering approximately 5.75 KWe (without the integral pump). It was determined that the turbine back pressure at design flow was higher than design and it was theorized that this was due to incorrect axial location of the jet condenser liquid nozzles, causing the condenser vapor nozzle to choke and give a higher pressure level.

After about eleven hours operation the turbo-alternator was removed for installation of the pump and seal.

The bearing adjusting screws were inspected and of all the coatings the thin silver plating was far superior, with no signs of any wear at all. Consequently six pins were plated with .0004" of silver and installed for the new assembly.

The machine was reassembled on the test stand with the pump and seal and with a spacer to relocate the jet condenser nozzles.

At first start up, with the pump dead-headed, the vibration levels were very high, but as soon as the pump delivered flow and the facility pumps were turned off, the system ran at very low vibration levels, of the order of 2 g's peak.

However, a maximum of about 5 KWe only was available. This led to low system efficiencies of the order of 13%. The thermodynamic conditions indicated a high turbine efficiency, sufficient to give approximately 6 KWe with all the accountable losses. Also the regenerator effectiveness was measured to be the very low value of 71% even though they had been reworked to improve their value from 80% to their design value of 90%. However, there appeared to be poor flow distribution on the liquid side of the regenerators and therefore orifices were added in an attenpt to try to cure this problem. This brought the effectiveness up to about 80%, still much lower than the design value.

Inspection of the new seal at this stage showed no signs of damage on wear, and measurement of the drain temperature which carried away the seal leakage indicated very low leakage rates.

Tests were then performed in an attempt to find the reasons for the low power outlet. These tests exonerated the new seal and the alternator. Spin-down tests showed no evidence of liquid churning losses and so finally the conclusion was reached that the pump efficiency was lower (approximately 50%) than expected, resulting from operation at much lower flow rates than design. In addition the turbine back p!ressure was still slightly higher than design and much higher than it had been during surface condenser operation. Unfortunately, the effect of turbine back-pressure at the design flow rates was not easy to determine since the jet condenser would not operate at levels lower than about .5 psia under this condition. However, checking at lower flow rates showed that a drop of back pressure from about .5 to .3 psia gave an additional .5 KWe.

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APPENDIX C DETAILED HISTORY OF 6 KWe ENDURANCE TEST AT 700°F TURBINE INLET

At this stage the turbo-alternator was removed from the test stand for installation of the 700°F nozzle block. The boiler tube was recleaned at this stage to remove any deposits which may have built up due to air inleakage during previous testing. All important dimensions, such as those of bearins, seals, etc., were measured and recorded for checking after endurance operation. Photographs were also taken of the hardware prior to installation for comparison purposes.

The endurance unit was run with the 700°F nozzle ring for the first time on September 3, 1970, for checkout purposes. A total of 4.5 KWe was indicated at 86% full flow, which was the power anticipated at this flow rate with the oversized pump.

Since siveral of the amplifiers for safety shutdowns were found to be inoperative at this time, endurance operation could not be commenced.

During that weekend, with the system shut down, the salt bath temperature control malfunctioned and the salt bath overheated caused salt overflow. With no record of how hot the salt bath got or how long it was hot, there was considerable concern that the Dowtherm A working fluid had degraded significantly and formed deposits on the inside of the boiler tube. Without removal and destructive sectioning of the boiler tube the deposition could not be confirmed. Due to limited funding and schedule, the decision was reached to continue the test without teardown and inspection of the hardware.

The machine was placed on endurance operation on September 15, 1970, with performance data being recorded periodically on a digital data recorder. Some time was necessary to shake-down the automatic controls and the length of runs at this stage were relatively short.

It was found that after 100 hours, the output had rapidly dropped to about 3.5 KWe. Degradation products from the salt bath overheat incident was believed responsible for the rapid output decay. It was found that as the number of running hours increased, the power output slowly deteriorated but at a much slower rate than the initial drop. This latter change, coupled with an increase in turbine inlet pressure for the same flow, indicated a slow buildup to decomposition deposits in the nozzles.

Problems were encountered with the turbine inlet solenoid valve and it was decided to replace the valve by a larger high temperature valve which had already been tested and shown to be satisfactory for long life. This change occurred at 238 hours into the shakedown run.

During the reset of the endurance operation no problems with the actual system occurred. Shutdowns occurred but all resulted from facility failures, notably electrical boiler failure, power failure and nitrogen pressure failure.

The machine was run for a total of 3003 hours, with a down time for the whole period of 26.6%. The machine was started 56 times during this period which lasted approximately six months, and the longest continuous run was 367 hours.

After 3000 hours it was decided to dismantle the turboalternator in an attempt to determine the cause of the discrepancy in measured power levels and the reason for the low value of power output.

On disassembly it was found that extensive black carbonaceous deposits had been deposited in the exhaust ducting, presumably resulting from air inleakage during the turbine bypass start-up phase early in the endurance testing.

When the turbine wheel was removed the reason for low power level and the output discrepancy was immediately obvious. Of the eleven nozzles, three were completely blocked and another three almost blocked.

These nozzles were all adjacent to one another and were the first ones to see the gas stream from the boiler. It is postulated that a slug of viscous decomposition products were emitted from the boiler in one lump and blocked these nozzles at the same time, causing a sudden change in power output. The gradual decline from that point resulted from a slow deposition in all the nozzles of material coming from the boiler. The decomposition probably occurred during the severe overtemperature at the beginning of the endurance operation.

Inspection of the radial and thrust bearings and hydrostatic face seal showed their condition to be almost as at start-up. The bearing adjusting screws showed no serious wear, just a slight polished area at the contact point.

The bearings are shown in Figure C-1 after the completion of the endurance testing and the hydrostatic face seal and pump impeller is shown in Figure C-2 after 3030 hours operation.

Summarized below are the details of the bearings used in the endurance operation.

	Turbine Bearing	Pump Bearing
Total Hours	3164	3170
Total Starts	135	152
Total Hours on One Build	3011	3011

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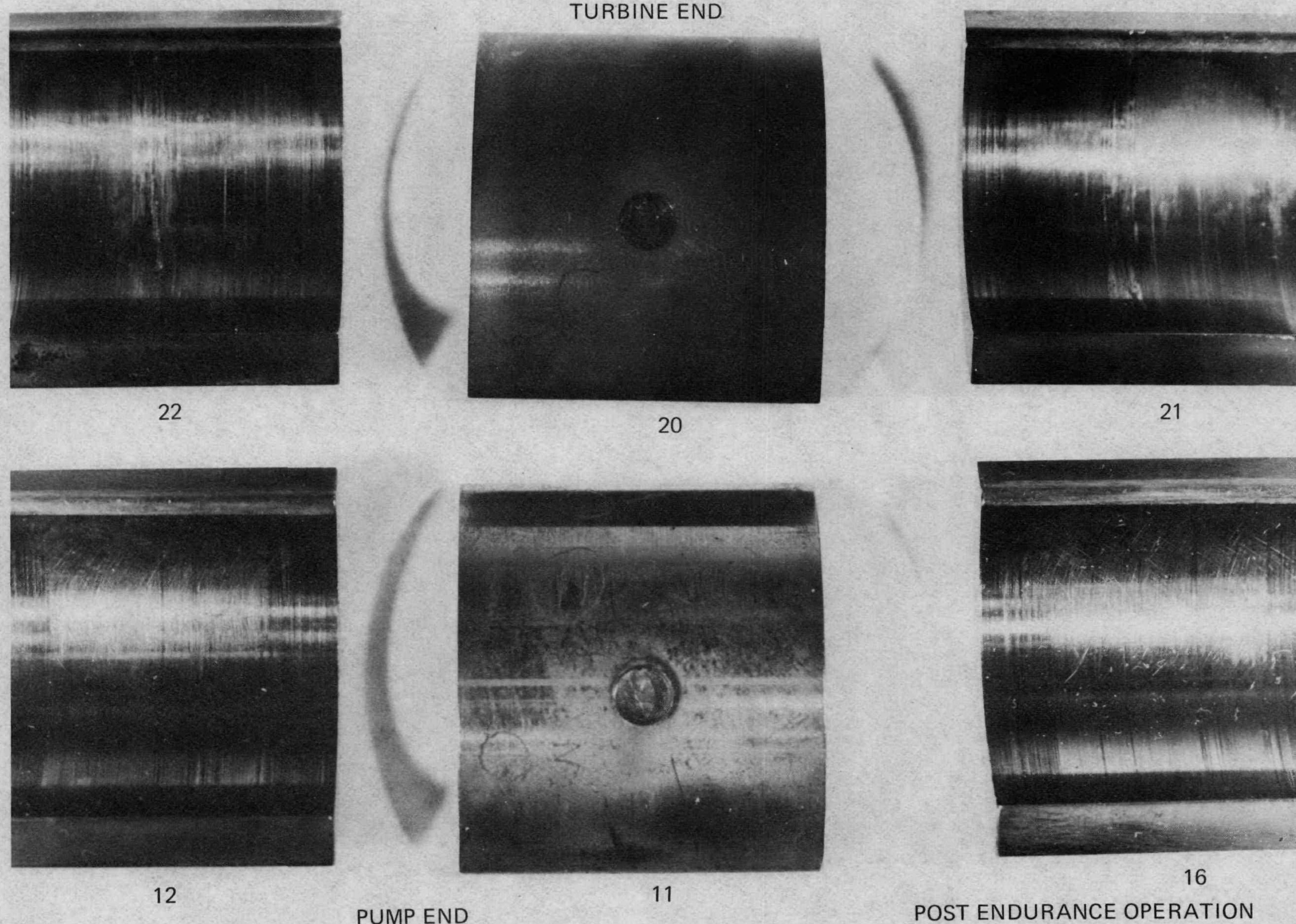

Figure C-1 Tilting Pad Bearings After Endurance Operation

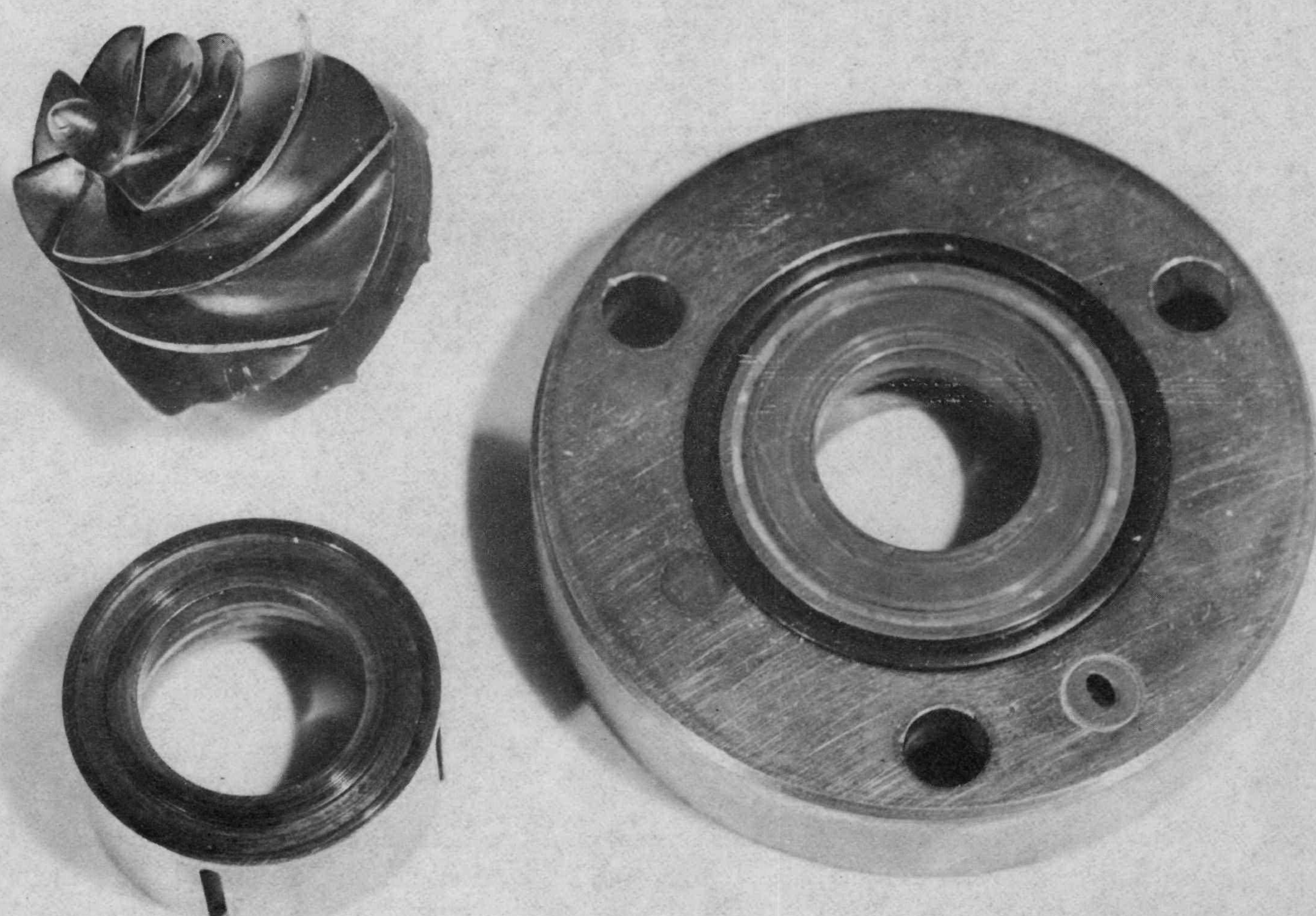

Figure C-2 Pump Impeller and Hydrostatic Face Seal After Endurance Operation