Argonne National Laboratory

OPERATIONAL TESTS OF EBWR VAPOR RECOVERY SYSTEM

by

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I. INTRODUCTION

Sargent and Lundy Engineers have recently conducted a detailed analysis of a direct-cycle boiling D_2O reactor power plant for the Atomic Energy Commission. The control of D_2O loss is a major factor in the economic analysis of such a plant.[1] The validity of the analysis of D_2O loss is strongly dependent upon the degree to which one can rely upon the capability and effectiveness of the vapor recovery system. Since the Experimental Boiling Water Reactor (EBWR) is uniquely equipped with a vapor recovery system, ANL was requested to conduct an operating test to contribute further data for D_2O plant evaluations. This report describes tests conducted on EBWR to evaluate the vapor recovery system.

The EBWR is a direct-cycle boiling water reactor. The steam that is formed in the reactor core is directly used to drive the power plant turbogenerator. The reactor is designed for operation with light water as the working fluid. However, since it is an experimental facility, the plant is designed to permit other operating modes, including operation with D_2O.[2]

In anticipation of the use of D_2O the original plant was equipped with a vapor recovery system. Upon introduction of D_2O, all primary plant seals would be vented or drained to the vapor recovery system. For operation with light water as the working fluid, only the major component seals which require a dry air barrier for their operation are included in the recovery system. These components are:

1. the turbine rotor shaft seal at the high-pressure end of the turbine;

2. the turbine rotor shaft seal at the low-pressure end of the turbine;

3. the seal groove in the turbine casing joint flange;

4. the seals on the turbine inlet valve lift rods;

5. the seals of the throttle valve stem and flange; and

6. the main feed pump drive shaft and flange seals.

II. OBJECTIVES OF TEST

System analysis groups have noted that a reliable prediction of D_2O losses is difficult because test reports present only examples conducted under ideal conditions, and thus provide no indication of losses resulting from abnormalities.[1] Partial objectives of the tests discussed here have been to make note of abnormalities and to determine the losses possible from equipment failure or operating errors. Gross losses resulting from such conditions cannot be accurately predicted since the time period of the abnormality cannot be known in advance. The sources and relative importance of malfunctions have been studied.

Additional objectives of this test were to:

1. study the operation and design of the vapor recovery system;

2. measure the seal air operating pressures, temperatures, and moisture content;

3. measure air flow through the seals, assess seal wear, and compare the measured flow through turbine labyrinths to theoretical flow;

4. evaluate the operation of the desiccator by measuring desiccator air flow, temperature, and moisture;

5. evaluate the design of the vapor recovery system with regard to its effectiveness and its equipment ratings;

6. assume direct-cycle D_2O operation of the reactor and from the test data determine the D_2O leakage that would be controlled by these seals;

7. determine the amount of contamination by atmospheric H_2O that would be introduced with makeup air for the seal system; and

8. recommend improvements to the installed system and for future systems.

III. DESCRIPTION OF VAPOR RECOVERY SYSTEM

The purposes of the EBWR vapor recovery systems are:

1. to reduce loss of the primary working fluid and radioactive gases to the containment atmosphere, and

2. to reduce contamination of the primary working fluid by air and atmospheric vapor in-leakage.

There are three principal operations of the vapor recovery system:

1. the collection of the effluent from all seals to a concentration system, by maintaining a vacuum in the collection gland at the seal and a greater vacuum in the concentration system;

2. the concentration of all reactor water leakage by means of condensers and desiccators and the return of the collected liquid to the turbine condenser hotwell; and

3. the provision of a positive pressure dry air barrier at major seals where the collection gland vacuum would otherwise draw in atmospheric air and consequently atmospheric vapor.

A schematic diagram of the fluid concentration and dry seal air supply equipment for the vapor recovery system is shown in Fig. 1.

The seal leakage is collected from the turbine, pumps, and valves together with air from the dry air barriers of the seals. This recovered air/vapor mixture is drawn into the vent condenser, where most of the vapor is condensed. Water that is already in the liquid form, such as feed pump leakage, is simply drained into the vent condenser sump. From the condenser, the recovered air passes over refrigeration coils to produce further condensation. The coils must be kept above 32°F to prevent frost; hence in the winter when cold cooling water is available, the coils are not loaded to capacity.

To remove all but 10 to 100 ppm of reactor water, the recovered air is passed through an activated alumina desiccator bed. To recover the reactor water collected by the alumina, two parallel-connected desiccators are used: one is reactivated while the other is on the line. The alumina is reactivated by blowing heated air through the bed, whereby the bed releases its moisture to the warm air. The moisture is condensed in a water-cooled heat exchanger and the dry air is then recycled.

The main blowers return the dry air to the dry air barrier glands of the seals. Since the dry air glands are at a positive pressure, some dry air is lost to the atmosphere. To replace this lost air, atmospheric air is drawn into the system. This makeup air is cooled and dried to reduce the introduction of atmospheric vapor.
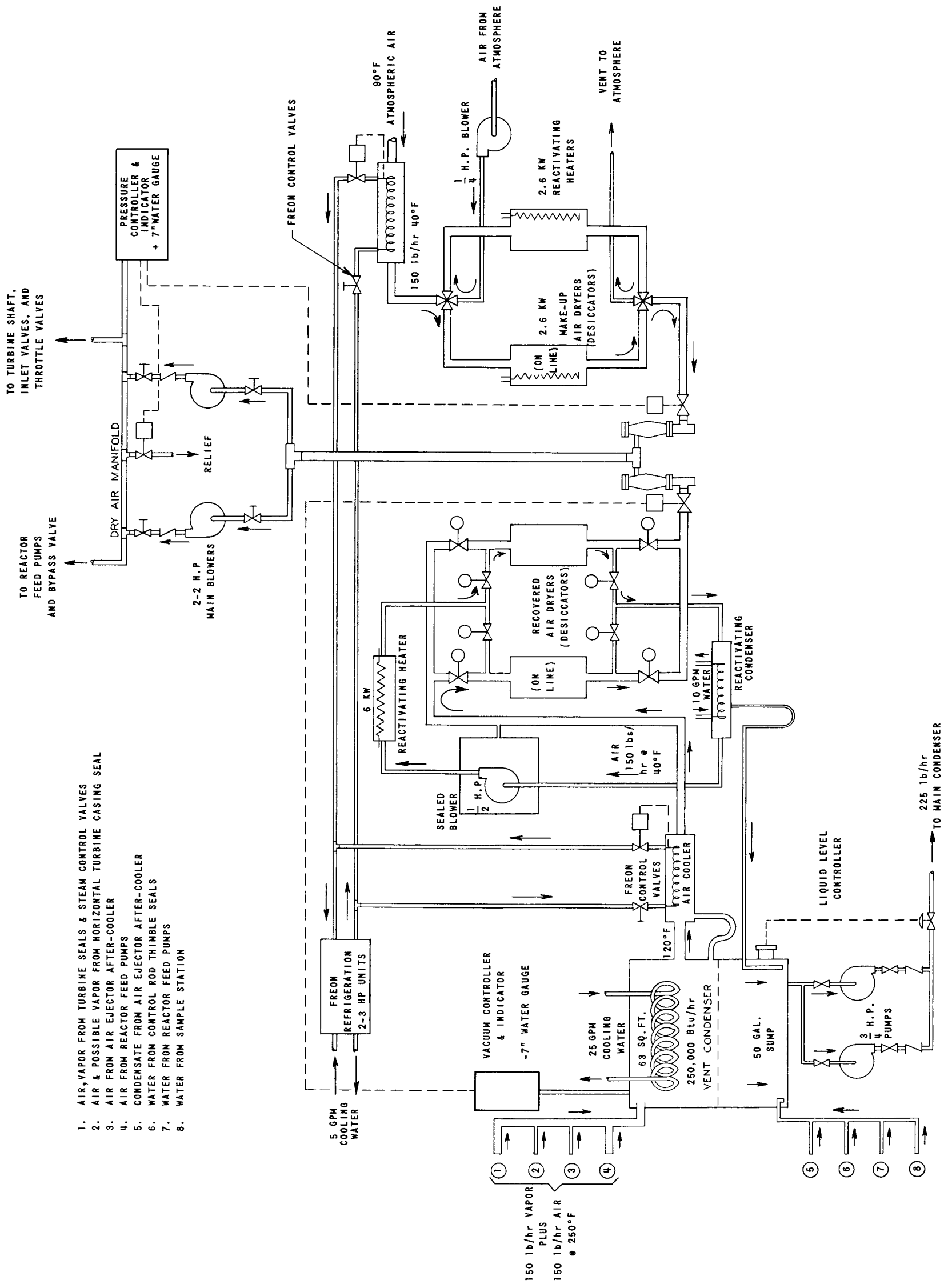
FIG. 1

FLUID CONCENTRATION AND SEAL AIR DRYING EQUIPMENT

TO TURBINE SHAFT,
INLET VALVES, AND
THROTTLE VALVES

TO REACTOR
FEED PUMPS
AND BYPASS VALVE

PRESSURE
CONTROLLER &
INDICATOR
+ 7"WATER GAUGE

DRY AIR MANIFOLD

RELIEF

2-2 H.P
MAIN BLOWERS

FREON CONTROL VALVES

90°F
ATMOSPHERIC AIR

AIR FROM
ATMOSPHERE

150 lb/hr 40°F

¼ H.P. BLOWER

2.6 KW
REACTIVATING
HEATERS

VENT TO
ATMOSPHERE

2.6 KW
MAKE-UP
AIR DRYERS
(DESICCATORS)

(ON LINE)

RECOVERED
AIR DRYERS
(DESICCATORS)

(ON LINE)

6 KW
REACTIVATING
HEATER

SEALED
BLOWER

½ H.P.

AIR
150 lbs/
hr @ 40°F

110 GPM
WATER

REACTIVATING
CONDENSER

FREON
CONTROL
VALVES

AIR COOLER

120°F

FREON
REFRIGERATION
2-3 HP UNITS

5 GPM
COOLING
WATER

VACUUM CONTROLLER
& INDICATOR
-7" WATER GAUGE

LIQUID LEVEL
CONTROLLER

225 lb/hr
TO MAIN CONDENSER

63 SQ.FT.

25 GPM COOLING WATER

250,000 Btu/hr
VENT CONDENSER

50 GAL.
SUMP

3/4 H.P.
PUMPS

1. AIR,VAPOR FROM TURBINE SEALS & STEAM CONTROL VALVES
2. AIR & POSSIBLE VAPOR FROM HORIZONTAL TURBINE CASING SEAL
3. AIR FROM AIR EJECTOR AFTER-COOLER
4. AIR FROM REACTOR FEED PUMPS
5. CONDENSATE FROM AIR EJECTOR AFTER-COOLER
6. WATER FROM CONTROL ROD THIMBLE SEALS
7. WATER FROM REACTOR FEED PUMPS
8. WATER FROM SAMPLE STATION

① ② ③ ④

⑤ ⑥ ⑦ ⑧

150 lb/hr VAPOR
PLUS
150 lb/hr AIR
@ 250°F

The vacuum in the vent condenser is regulated by an indicating pressure controller that operates a throttle valve at the recovered air desiccator exit. The controller has an automatic range of 0 to 10 in. water gauge vacuum, and is normally operated at 7 in. The pressure of air supplied to the seals is regulated by an indicating controller that operates a makeup air throttle valve and an overpressure relief valve. The controller automatic range is the same as above. Both controllers may be operated on "manual" to higher pressure up to the limits of the blower. The blower is capable of 30 in. of water differential head. With 150-lb/hr air flow, the design pressure drop through the heat exchangers and recovered air desiccators is 10 in. water gauge.

The result of proper operation of the system is that all seals have effectively zero leakage. The only primary vapor that is lost is that minute quantity entrained in the dry air lost from the dry air barrier.

IV. DESCRIPTION OF SEALS

A. Turbine Seal, High-pressure End

The turbine shaft seal at the high-pressure end of the turbine contains four essential chambers, as depicted in Figs. 2 and 3:

1. a dry air barrier chamber containing dry air which is maintained at a pressure slightly above atmospheric (the design pressure of this chamber is +6 in. water gauge);

2. an air/steam leakage collection chamber which is maintained at a vacuum slightly below atmospheric (the design vacuum is -6 in. water gauge);

3. a seal steam chamber containing primary system steam (the steam pressure is maintained at 2 psig by venting the excess to the condenser, as indicated in Fig. 3); and

4. a steam vent chamber that is vented directly to the eighth stage of the turbine.

The steam leaking from the high-pressure end of the turbine to the vent chamber is retarded by a 24-stage labyrinth seal. A 12-point labyrinth separates each of the four chambers from one another. The dry air in the barrier chamber is retarded from leaking to the atmosphere by a 6-stage labyrinth.

Allis-Chalmers measurements of the initial shop seal clearances are presented in Fig. 4. The dimensions are representative of the annular labyrinth orifices afforded.

B. Turbine Seal, Low-pressure End

The turbine low-pressure seal, shown in Fig. 3, is a three-chamber unit similar to the turbine high-pressure seal in all respects except for the omission of the 24-stage high-pressure inboard labyrinth seal section. The inward leakage of seal steam passes directly to the main condenser.

C. Turbine Horizontal Flange Joint Seal

At the time of the initial plant installation, a seal system composed of a vacuum leakoff channel and an external can-like air chamber was welded around the horizontal joint in the turbine casing to collect any leakage that might occur in this region. This is shown in Fig. 3. Running experience with this arrangement indicated that the leakage through the horizontal turbine joint was negligible, and in the course of servicing the
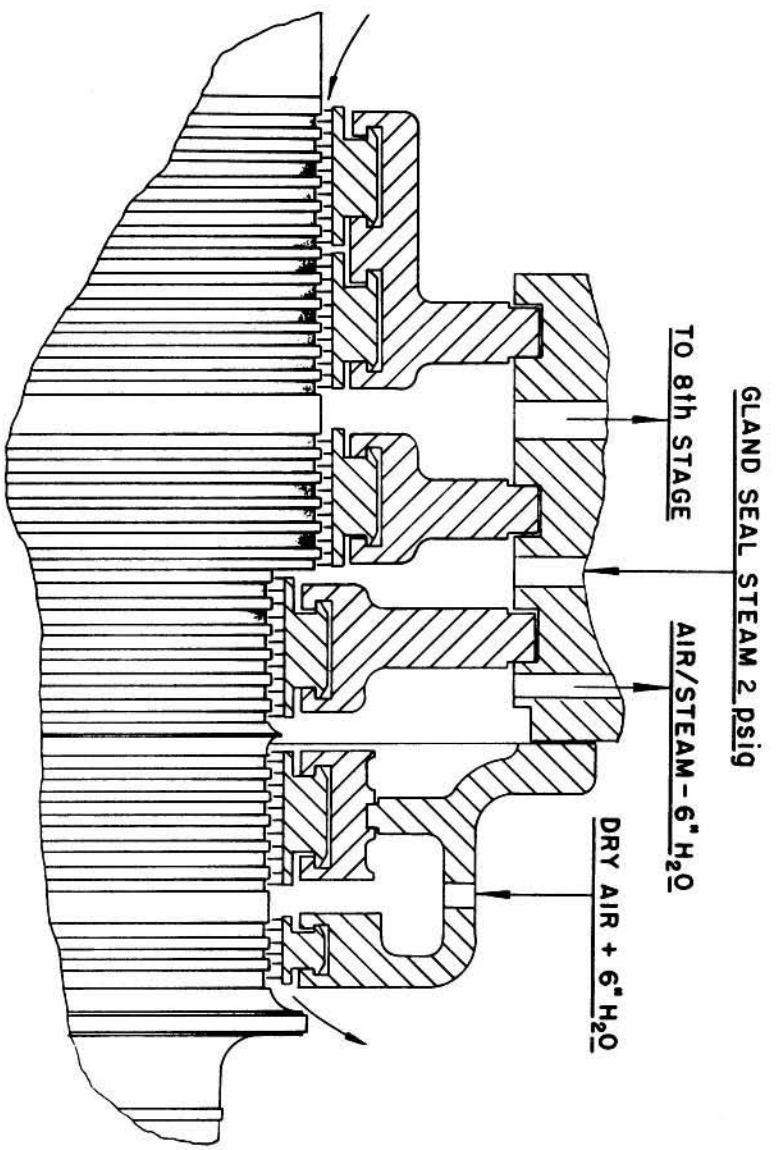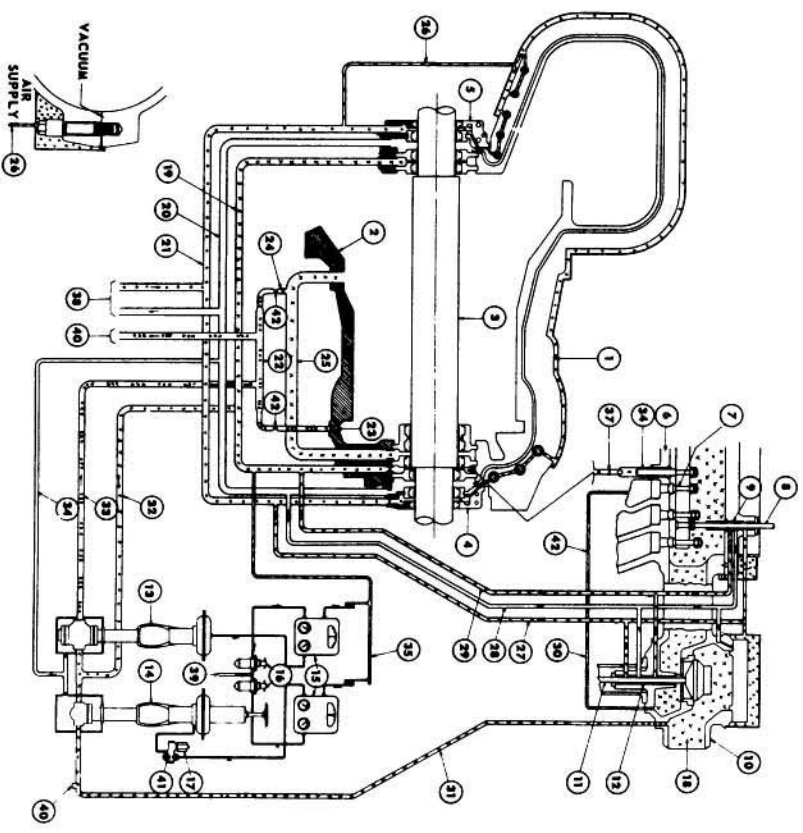
GLAND SEAL STEAM 2 psig
TO 8th STAGE
AIR/STEAM - 6" H₂O
DRY AIR + 6" H₂O

FIG. 2
EBWR TURBINE SHAFT SEAL (H.P. END)

VACUUM
AIR SUPPLY

FIG. 3
TURBINE SEALING SYSTEM

1. TURBINE CASING - HORIZONTAL JOINT
2. TURBINE CASING - VERTICAL SECTION
3. TURBINE SHAFT
4. STEAM END GLAND SEAL
5. EXHAUST END GLAND SEAL
6. INLET VALVE CHEST
7. INLET VALVE
8. INLET VALVE LIFT ROD
9. LIFT ROD PACKING
10. THROTTLE VALVE
11. THROTTLE VALVE STEM
12. THROTTLE VALVE STEM PACKING
13. RELIEF VALVE
14. MAKEUP VALVE
15. PRESSURE CONTROLLER
16. FILTER AND AIR PRESSURE REGULATOR
17. THREE-WAY SOLENOID VALVE
18. STEAM INLET
19. GLAND STEAM MANIFOLD
20. VACUUM MANIFOLD
21. AIR MANIFOLD
22. DRAIN MANIFOLD

23. STEAM END CASING DRAIN
24. LEAKOFF PIPE DRAIN
25. STEAM END GLAND CASING DRAIN
26. AIR TO TURBINE CASING SEALING COVER AND EQUALIZING PIPE
27. AIR TO THROTTLE VALVE SEALING COVER, VALVE CHEST SEALING COVER, THROTTLE VALVE STEM AND INLET VALVES LIFT RODS
28. VACUUM LINE FROM THROTTLE VALVE STEM AND INLET VALVES LIFT RODS
29. STEAM SEAL OR LEAKOFF TO THROTTLE VALVE AND INLET VALVE LIFT RODS
30. CONTINUOUS DRAIN FROM BELOW THROTTLE VALVE SEAT
31. GLAND STEAM SUPPLY FROM ABOVE THROTTLE VALVESEAT
32. GLAND STEAM MAKEUP OR RELIEF
33. EXHAUST FROM RELIEF VALVE
34. VACUUM LINE FROM RELIEF AND MAKEUP VALVE SEALING BOX
35. STEAM PRESSURE SENSING LINE
36. BALANCE ROD
37. BALANCE ROD LEAKOFF TO STEAM END GLAND
38. VAPOR RECOVERY SYSTEM
39. PLANT AIR CONNECTION
40. CONDENSER CONNECTION
41. ATMOSPHERIC EXHAUST
42. ORIFICE

13

14

ARGONNE NATIONAL LABORATORY
5000K$_\omega$-TURB. NO. 10648
ORIGINAL SHOP GLAND CLEARANCES-RADIAL

LEFT SIDE	RIGHT SIDE		
.005	.007		6 POINTS
.008	.008		12 POINTS
.003	.008	NO. 1	12 POINTS
.005	.008		11 POINTS
.003	.008		12 POINTS
.003	.010		12 POINTS

HIGH PRESSURE

.008	.007		11 POINTS
.010	.006	NO. 2	11 POINTS
.004	.005		11 POINTS
.005	.005		6 POINTS

LOW PRESSURE

FIG. 4
TURBINE SEALS
ORIGINAL SHOP GLAND CLEARANCES

turbine the welded can forming the dry air chamber was removed. At the present time, the air supply to the seal is closed but vacuum is still applied to the circumferential leakoff groove. The flow conditions and the possiblity of restrictions within the groove are currently unknown and any in-leakage that may be collected from this system is being vented to the vapor recovery system via the turbine shaft collection chambers.

D. Throttle Valve Seal

As indicated in Fig. 5, the throttle valve seal arrangement is of a less complex design than that of the turbine seals and consists of a grooved sleeve located between the packing gland and leakoff bushing. The sleeve contains three grooves which act as dry air, vacuum, and seal steam chambers. The seal so formed operates by the same principle as the high and low-pressure turbine seals. Interchamber flow is retarded by the close sleeve fit. Air leakage to atmosphere is virtually prevented by the valve stem packing.

E. Inlet Valve Seal

The turbine inlet valve is actuated by two lift rods. A general arrangement of the valve is shown in Fig. 3. Sealing of each lift rod is accomplished by the same method as that employed in the turbine throttle valve.

F. Bypass Valve Seals

The turbine bypass valve is shown in Fig. 6. It is similar to the throttle valve seal.

G. Feed Pump Seals

The pump seals are shown in Fig. 7. The shaft seal consists of four chambers:

FIG. 5
TURBINE TRIP THROTTLE VALVE

TO CONDENSER

STEAM FROM REACTOR

RECOVERED AIR
& STEAM

SEAL STEAM

DRY AIR SUPPLY

DRY AIR LEAKAGE

FIG. 6
BY-PASS VALVE SEAL

DRY AIR LEAKAGE

DRY AIR SUPPLY

MECHANICAL SEAL

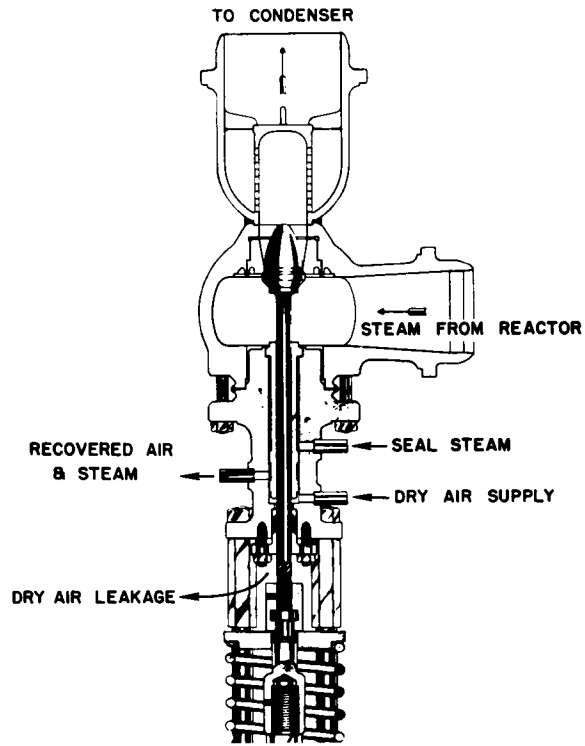RECOVERED AIR

SEAL WATER
100 PSIG.

PUMP DISCHARGE
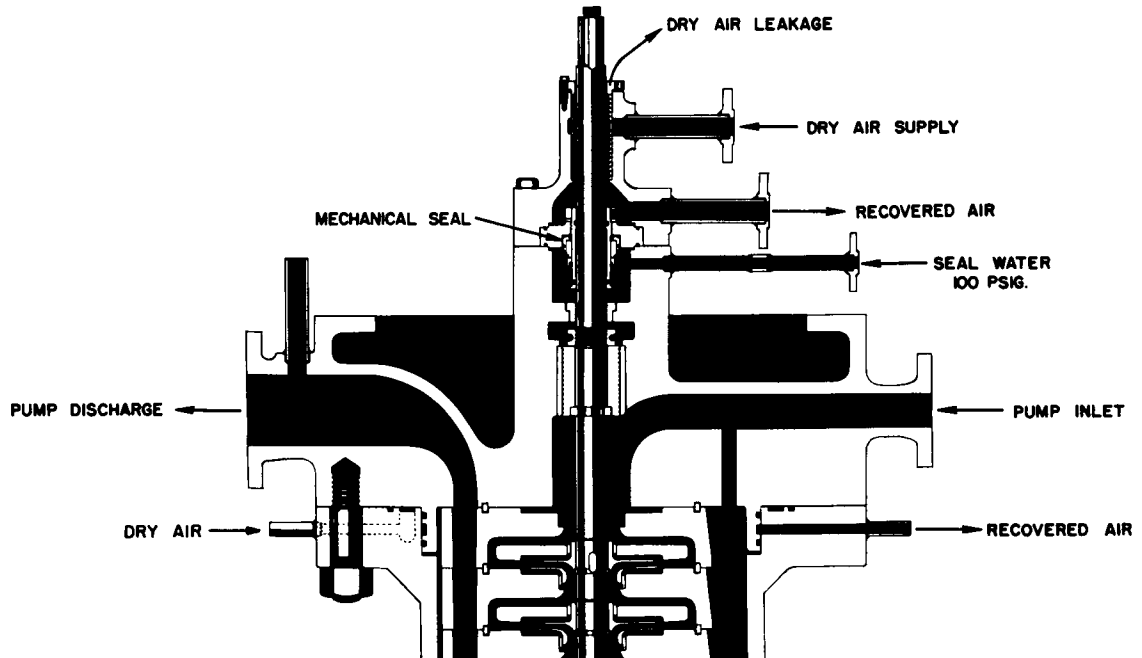
PUMP INLET

DRY AIR

RECOVERED AIR

FIG. 7
CROSS-SECTIONAL VIEW OF EBWR FEEDWATER PUMP SEAL

1. The innermost chamber is connected to the main condenser vacuum. The 900-psig water that leaks up the drive shaft is thus returned to the pump inlet.

2. The seal water chamber is connected to the 100-psig discharge of the first stage of the pump. A serrated bushing separates this chamber from the condenser vent chamber.

3. The air/water leakage collection chamber is maintained at a vacuum slightly below atmospheric by the vapor recovery system. A rotary mechanical seal, comprised of a Stellited ring operating against a graphite face, prevents the 100-psig seal water from entering this chamber.

4. The dry air barrier is similar to those of the turbine shaft and valve stems. Labyrinths are used to retard air flow to the collection chamber and to atmosphere.

V. DESCRIPTION OF TEST APPARATUS

A. Flow Diagram

The flow diagram of Fig. 8 illustrates the location of test equipment in the vapor recovery system. The arrangement is designed to permit performance of all operational tests on any individual seal or group of seals without detrimental interference to the operation of the remaining parts of the vapor recovery system. The test system provides all the instrumentation required to accomplish the objectives listed in Section II of this report. The preliminary arrangement of apparatus was designed and the equipment sized by Sargent and Lundy Engineers. The final installed arrangement was carried out by ANL.

B. Pressures

The recovery system pressures and vacuums were all measured by standard "U"-tube manometers using fluorescein-tinted water. The seal steam pressure was measured by a bourdon tube pressure gauge. The pressure taps are indicated by the letter "P" in Fig. 8.

C. Temperatures

During the phases of testing, temperatures were measured at the points labeled "T." The thermocouples were of No. 20 iron and con-stantan wire with silver-soldered junctions. Thermocouple output was recorded on two Brown multipoint recorders. All couples were calibrated with an ice bath after installation.

Thermometers were used on the: (1) dry air line manifold immediately up-stream of the turbine rotometers; (2) feed pump rotometer; (3) turbine seal return line; (4) recovered air rotometer manifold; and (5) exit line from the makeup air desiccators.

Temperature Recorders:

Brown Potentiometer
Model 157X64P16-X-41
Strip Chart, 16 point, Multipoint Recorder
Range: 0 to 100°F with I-C couples

Brown Potentiometer
Model 157X72-P8-(11)(111)16-A4A8
Strip Chart, 4 point, Multipoint Recorder
Range: 0 to 500°F with I-C couples

FIG. 8
SEAL TEST FLOW DIAGRAM
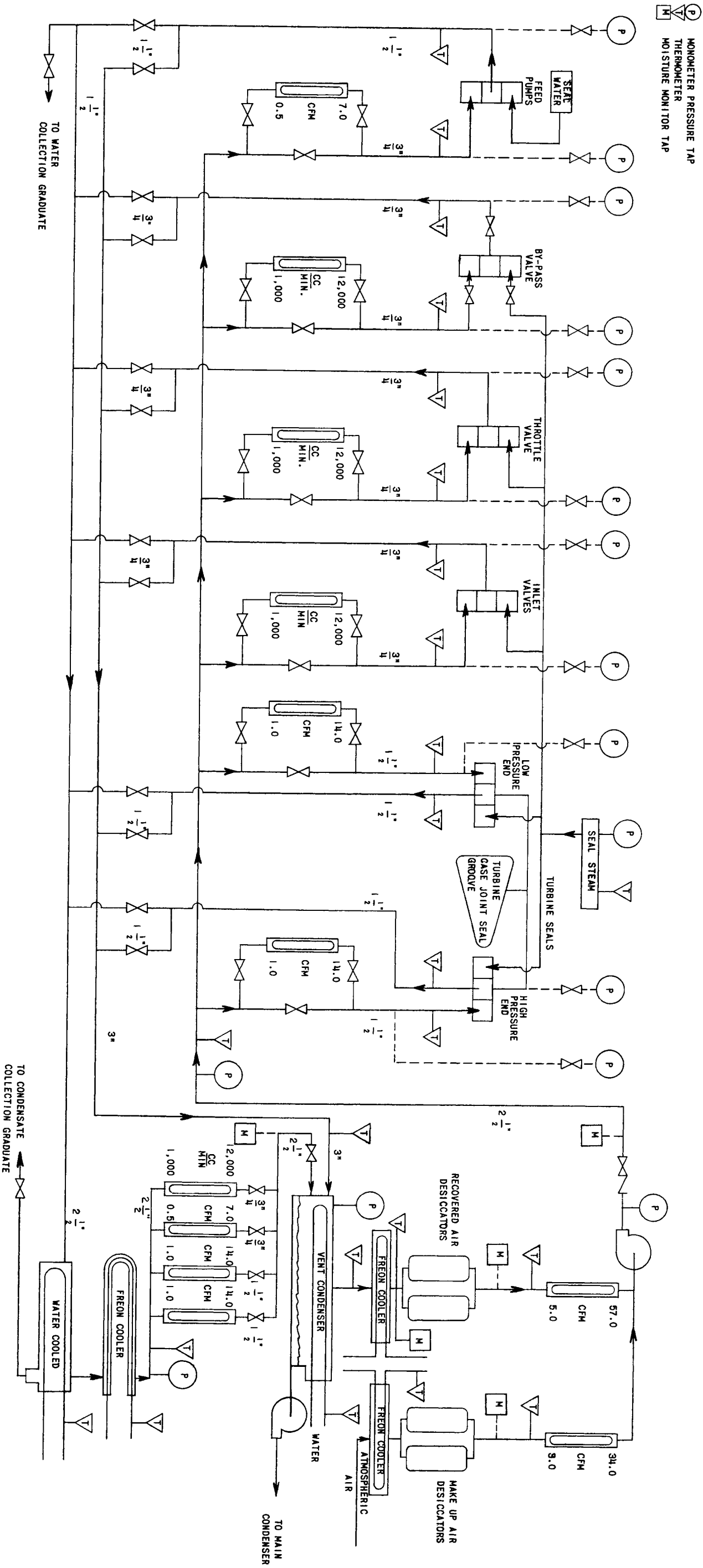
- LEGEND -

MONOMETER PRESSURE TAP
THERMOMETER
MOISTURE MONITOR TAP

19

Thermometers:

American Model No. 268 with 5 lb scale and $\frac{1}{2}$ in. NPT Class B Sockets. For the cold lines, (1)(2) and (4), 20°F to 120°F range was used. For the hot lines, (3) and (5), 100°F to 400°F range was used.

D. Moisture

An extensive effort was made to obtain accurate measurements of the moisture content of the dry air supplied to the seals. Since it was known from previous tests that there was a variation in the moisture content of the air leaving the desiccators, a recording moisture monitor was used to plot the air quality at the following points:

1. dry air outlet of the recovered air desiccators;

2. dry air outlet of the makeup air desiccators; and

3. the mixture of recovered and makeup air at the outlet of the main blowers (seal air supply).

An Alnor dew point instrument was used to make an order-of-magnitude spot check for the Consolidated instrument. Spot checks of air quality were also taken at the recovered air desiccator inlet and at the rotometer manifold of the test apparatus to verify air moisture saturation presumptions.

Recording Hygrometer:

Consolidated Electrodynamics
Portable Moisture Monitor, Type 26-302
Range: Highest - 1 to 20,000 ppm by volume
           Lowest - 1 to 10 ppm by volume
Sample Flow: 20 cc/min
Operating Principle: Electrolysis

The instrument output signal of 0-100 mv was recorded on a Brown, single-point, 0-100 mv, potentiometric strip chart recorder. The 100-mv corresponded to the full scale of the monitor range in use.

Dew Point Indicator:

Illinois Testing Laboratories, Inc.
"Alnor Dew Pointer" - Type 7000 U
Dew Point Range: Room temp. to minus 80°F
Sample: Representative
Principle: Condensation upon adiabatic expansion

E. <u>Air Flow</u>

Special effort was made to assure accurate flow measurements and to introduce very little line pressure drop in so doing. Rotometers were installed in the dry air supply lines to each seal and a bank of four meters were installed after the Freon air cooler of the test apparatus to measure recovered air flow. Meter ranges and locations are indicated in Fig. 8. Figure 9 shows the installed arrangement of the rotometers under the turbine. All flow tubes were calibrated and engraved to an accuracy of 1% of full scale for 32°F, 14.696 psia air.

Rotometers:

Fischer & Porter Company
Flowrator Model 10A2735A
Flow Tubes - Borosilicate Glass
Floats: Stainless Steel (spool type in CFM meters and balls in cc/minute meters)

Valves:

Crane No. 430 Gate Valves
Brass body and trim
Double wedge disk
Rising stem with back seat and separate packing gland provisions for leak tightness.

F. <u>Steam Flow</u>

The flow rates of steam effluent from the seals was measured by condensing the vapor and timing the condensate flow to a two-liter graduate. Figure 10 shows the test vapor recovery equipment as installed in the plant. Figure 11 is a top view of the valves installed under the turbine to direct recovered air/steam to the test apparatus.

Condenser:

Patterson-Kelly Company
Heat exchanger Model DF6-36
Tube Bundle: 4-pass "U" tube, with 7.6 sq ft of $\frac{3}{4}$-in. OD 18 ga copper tubes
Shell-to-tube design pressure: 125 psig - water in tubes

Air Cooler:

Brown Fin Tube Company
Hairpin Shell Heat Exchanger - Model 017-000-010 counter flow
Tube: "U" tube, 10 sq ft of $1\frac{1}{2}$-in. steel tube
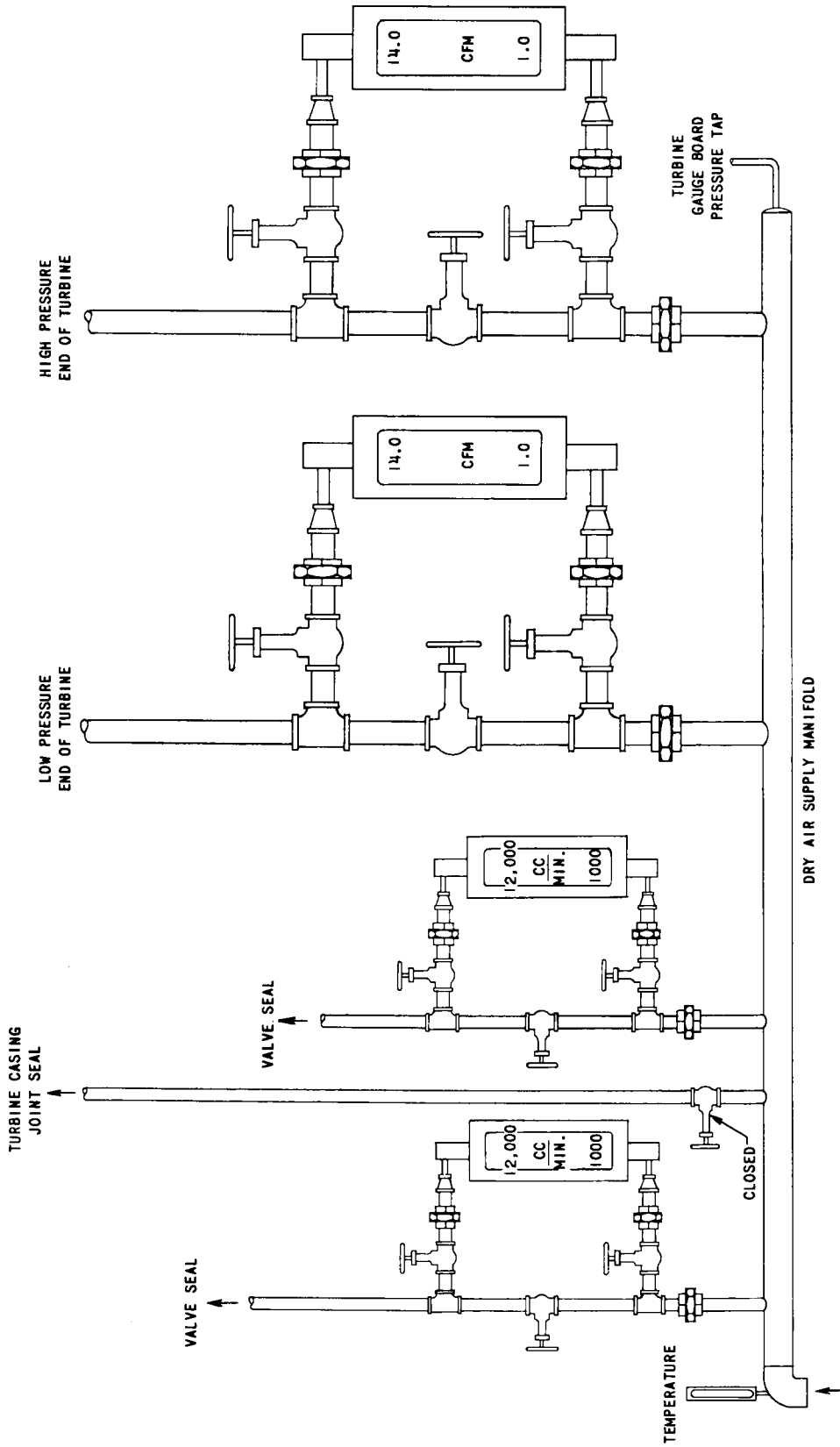Shell-and-tube design pressure: 500 psig - Freon 12 in. tube
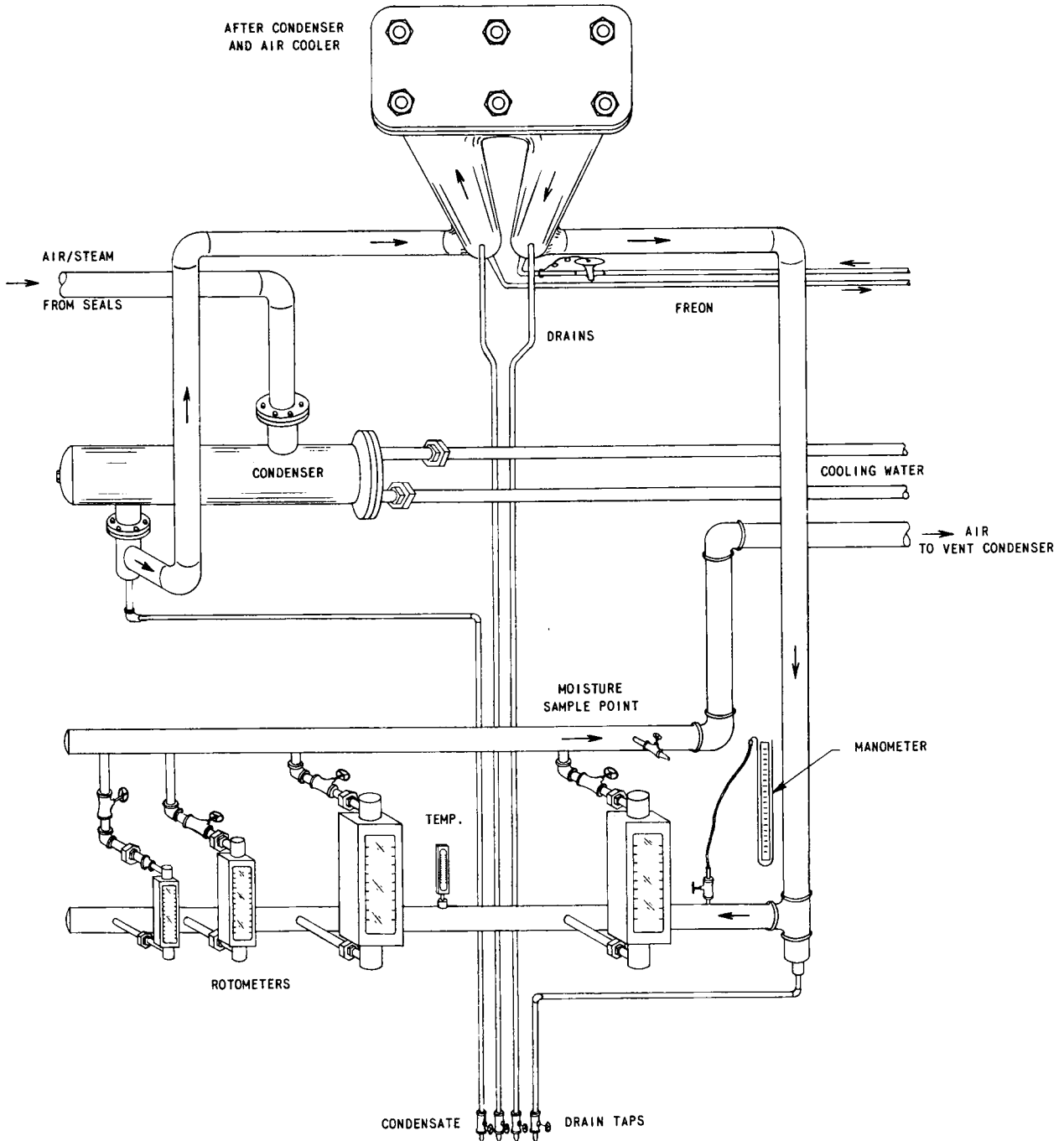
TURBINE SEAL DRY AIR SUPPLY ROTOMETERS

FIG. 9

AFTER CONDENSER
AND AIR COOLER

AIR/STEAM

FROM SEALS

FREON

DRAINS

CONDENSER

COOLING WATER

AIR
TO VENT CONDENSER

MOISTURE
SAMPLE POINT

MANOMETER

TEMP.

ROTOMETERS

CONDENSATE   DRAIN TAPS

FIG.10
TEST RECOVERY APPARATUS

LOW PRESSURE
END OF TURBINE

TO TEST
APPARATUS

INLET
VALVE

TO VAPOR
RECOVERY
CONDENSER

THROTTLE
VALVE

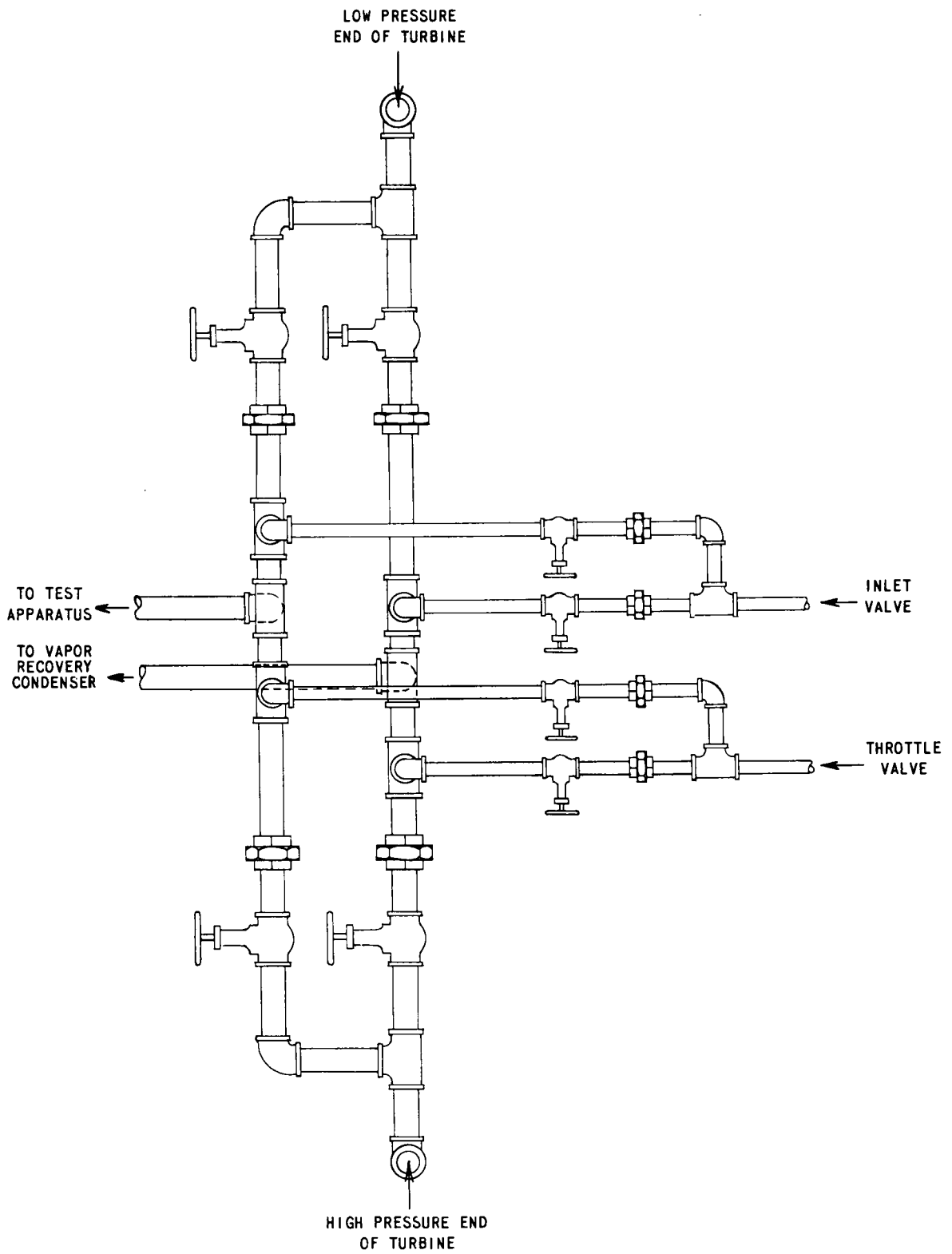HIGH PRESSURE END
OF TURBINE

FIG.II
TURBINE AIR/STEAM LINES FROM SEALS

VI. TEST PROCEDURE

A. General Discussion

The tests were divided into two phases. The objectives of the first-phase tests were to measure pressures, temperatures and air moisture under normal operating conditions in order to determine the

1. average operating condition,

2. normal operating fluctuations,

3. variances from design parameters, and

4. control data needed for the second-phase tests.

The objectives of the second-phase tests were to measure system air flow and water (vapor and liquid) flow with seal pressures regulated

1. to design values, i.e., to 6 in. W.G.,

2. to the average operation values determined in the first phase, and,

3. in 1 in. W. G. increments from 1 in. W.G. to 10 in. W. G., to determine seal characteristics.

B. Steam Plant Conditions during Tests

The steam plant conditions during test were those normally required to operate the turbine at 3600 rpm with about one Mw of electrical load. Steam plant parameters pertinent to these tests were:

1. Turbine speed   3600 rpm

2. Turbine power $\sim$1000 kw

3. Reactor steam flow $\sim$20,000 lb/hr

4. Reactor pressure   600 psig

5. Main condenser vacuum $2\frac{1}{2}$ in. Hg

6. Seal steam pressure   2 psig

7. Turbine operating time, before test   8900 hr

C. Specific Procedure for First-phase Tests

With the vapor recovery system in operation and all recovered air and vapor being directed to the vent condenser, the following procedure

was carried out:

1. Pressures were read and recorded at 2-hr intervals for 72 hr at

   a. all seal dry air chambers,

   b. all seal collection chambers (vacuum),

   c. main blower outlet and inlet,

   d. vent condenser, and

   e. seal steam header.

2. Temperatures were automatically recorded at 10-min intervals for 72 hr at those points required to determine the temperatures of

   a. dry air supplied to seals,

   b. the steam-air mixture returning from the seals,

   c. the vent condenser air at inlet, air at outlet and cooling water,

   d. the air cooler Freon and air at outlet, and

   e. the makeup air in and out of desiccators.

3. Moisture content of the dry air was automatically sampled and continuously recorded for 24 hr at each of the following points:

   a. outlet of the main system blower (dry air to seals);

   b. outlet of the recovered air desiccators; and

   c. outlet of the makeup air desiccators.

   The air leaving the "recovered air" Freon air cooler was sampled periodically.

4. Upon termination of the 72-hr test period, all data was analyzed and averaged.

D. Specific Procedure for Second-phase Tests

1. With the turbine and vapor recovery system operating normally, test apparatus was introduced to prepare the recovery system for flow tests as follows:

a. Shifted all dry air rotometers into the dry air lines by means of the bypass valves provided.

b. Increased dry air supply pressure to 10 in. of H_2O (or higher when required). The valves at the dry air rotometer inlets were used to obtain the proper normal dry air operating pressure in the seal gland.

c. Closed valve on dry air supply to turbine casing joint seal.

d. Started blower and coolants in test fluid recovery loop.

e. Selected seal to be tested and selected the proper recovered air rotometer to achieve as near a full-scale reading as possible.

f. Directed the recovered air and steam from the vacuum gland of the seal being tested to the test fluid recovery loop by means of the valves provided.

g. Regulated the pressure in the seal vacuum gland to the desired pressure (Section VI-A) by use of the valve on the outlet of the recovered air rotometer.

h. Recorded the following data:

   (1) moisture, pressure and temperature of the dry air supply,

   (2) dry air rotometer flow, pressure and temperature,

   (3) recovered air rotometer flow, pressure and temperature,

   (4) time required to fill 2-liter graduate with recovered water for flow rate determination, and

   (5) makeup and recovered air flow moistures, pressures and temperatures.

2. Using the above procedures, each seal was tested individually. When the turbine shaft seals were tested, the vacuum gland pressures at the high and low end were regulated to the same pressure to prevent interflow between seals via the casing joint groove.

3. Both turbine shaft seals were tested simultaneously to check the individual seal tests.

4. All seals were tested simultaneously by venting all vacuum glands to the test apparatus. Individual seal pressures were regulated to operating average values by using valves to throttle individual flows, and increasing main supply pressure to achieve required head.

VII. TEST RESULTS AND OBSERVATIONS

A. Seal Pressures

As part of the first-phase tests, the seal air pressures were recorded during the first four days of operation. The high, low, and average readings are listed in Table 1.

Table 1

DATA SUMMARY

Seal Pressures	High Pressure, in. W.G.	Low Pressure, in. W.G.	Average of 30 Readings, in. W.G.
A. Blower Pressure			
1. Outlet	8.0	6.5	7.1
2. Inlet	-24.0	-21.5	-23.0
B. Dry Air Glands of Seals:			
1. Turbine High-pressure End	7.3	5.8	6.6
2. Turbine Low-pressure End	7.0	5.8	6.4
3. Feed Pumps Seals	4.0	2.7	3.1
4. Throttle Valve Stem	7.8	5.9	7.0
5. Inlet Valve Stem	7.6	6.5	7.0
6. Bypass Valve Stem	3.7	2.7	3.1
C. Vent Condenser Vacuum	-7.0	-6.5	-6.9
D. Recovered Air Glands of Seals:			
1. Turbine High-pressure End	-5.0	-3.7	-4.3
2. Turbine Low-pressure End	-4.6	-4.2	-4.4
3. Feed Pump Seals	-6.9	-6.4	-6.9
4. Throttle Valve Stem	-5.2	-4.5	-4.7
5. Inlet Valve Stem	1.6	-0.5	-1.0
6. Bypass Valve Stem	-7.0	-4.5	-5.7
E. Seal Steam	2 psi	2 psi	2 psi

Figure 12 is a plot of the turbine and pump seal pressures. As can be seen, variations of blower speed cause the supply air pressure to deviate with an amplitude of one-inch water gauge. The gland pressures followed the fluctuation of the supply. As the gauges were not read simultaneously, the plot does not give an accurate indication of pressure drops in the line. The line loss between the blower and dry air gland of the turbine seals appears to be on the order of $\frac{1}{2}$ in. W.G.; however, there is a relatively large pressure drop between the blower and the feed pump seal. The bypass valve seal is supplied by the same line and exhibits a similar drop. The throttle and inlet valve seals operate very close to blower pressure. These seals are supplied by $\frac{3}{4}$- in. pipe taps off the main line supply to the turbine seals. This indicates that the line drop to the turbine seals occurs in the pipe branches leading to the individual seals.

The pressure drop in the recovered air line between the turbine glands and the vent condenser tank was 2.5 in. W.G., as a result of the large volume of seal steam returning with the air. The line drop in the $\frac{3}{4}$-in. recovered air line from the inlet valve seal was 3.5 in. W.G. more than that of the adjacent throttle valve seal. Since the lines are similar, the pressure drop difference is attributed to higher seal steam flow rates from the inlet valve stem seal. It was observed that the inlet valve seal was operating improperly early in the test. Seal steam condensate seeping into the dry air gland manometer gave further evidence of seal wear.

Improper drainage of the bypass valve seal vacuum line caused line condensate to drain into the manometer tubing. When the tap valve was closed, this condensate was forced to return to the vent condenser. However, this is not a desirable design characteristic since it could lead to back leakage via the dry air gland. The line should be corrected to drain to the vent condenser.

B.   System Temperatures

High, low and average temperatures are recorded in Table 2. The dry air supplied to the seals was observed to be heated enroute to the seals by the pipe walls. The pipes receive heat both from the atmospheric air and by conduction from the seal housings. The steam seal housings operated at temperatures approaching seal steam temperature (336°F). The temperature of the air in the seal was not amenable to direct measurement, but from inlet conditions, seal temperature and observed flow rates, it would appear to be of the order of 200°F in the turbine and bypass valve seals, 80°F in the pump seals, and 300°F in the inlet and throttle valve seals.

The steam-air mixture averaged 197°F at the inlet to the vent condenser. The recovered air temperatures were in the expected range considering cooling water and Freon temperatures.
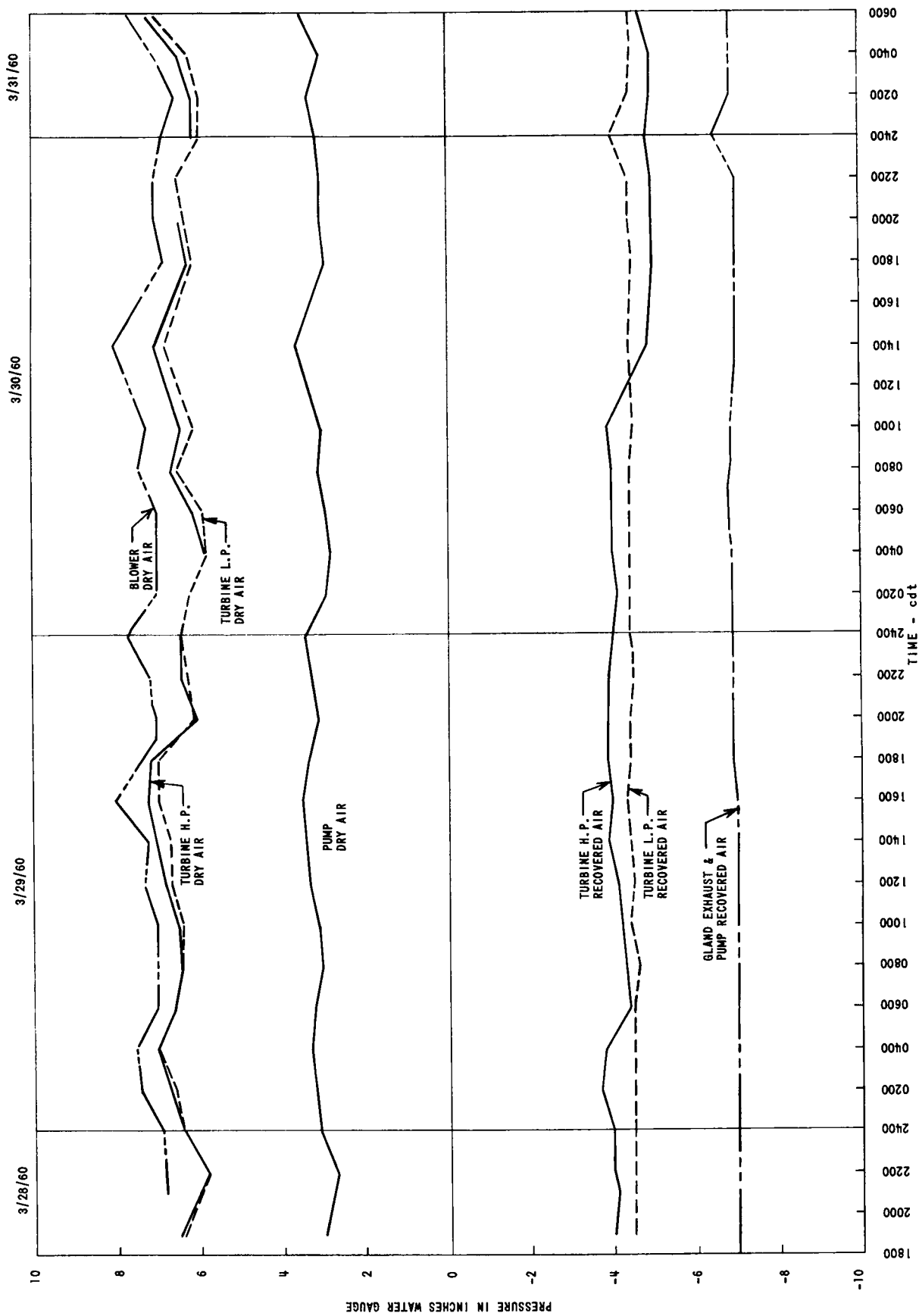
FIG. 12

SEAL CHAMBER PRESSURES

31

Table 2

SYSTEM TEMPERATURES

	Temperatures, °F		
	High	Low	Average
A. Dry Air Supplied to Seals			
1. At blower discharge	103	77	83
2. Turbine seals, throttle and inlet valve seals	100	91	95
3. Feed pump and bypass valve seals	82	77	80
B. Steam-Air Mixture Returning from Seals			
1. Turbine high-pressure end	210	197	200
2. Turbine low-pressure end	193	168	181
3. Feed pump seals	70	70	70
4. Throttle valve stem	322	306	315
5. Inlet valve stem	318	302	310
6. Bypass valve stem	202	174	200
C. Recovered Air-Vapor Mixture			
1. Steam-air entering vent condenser	200	192	197
2. Vent condenser cooling water	76	43	64
3. Recovered air entering Freon cooler	98	75	84
4. Freon temperature	51	46	47
5. Recovered air entering desiccator	71	63	67
6. Recovered air leaving desiccator			~75
D. Makeup Air			
1. Air entering Freon cooler (ambient)	~90	~80	~88
2. Freon temperature	45	32	35
3. Air entering makeup desiccator	60	42	55
4. Air leaving makeup desiccator	150	82	118

Makeup air was drawn from the atmosphere. Ambient temperatures in the containment shell vary between 80°F and 90°F during plant operation. The 35°F Freon cooled the air to 55°F before entering the makeup desiccators. A plot of the temperatures leaving the makeup desiccators indicated that these desiccators were not cooling by radiation as per design. Figure 13 illustrates a typical plot of air temperature at the exit of the makeup desiccators. The effect of this is discussed further under "Moisture."
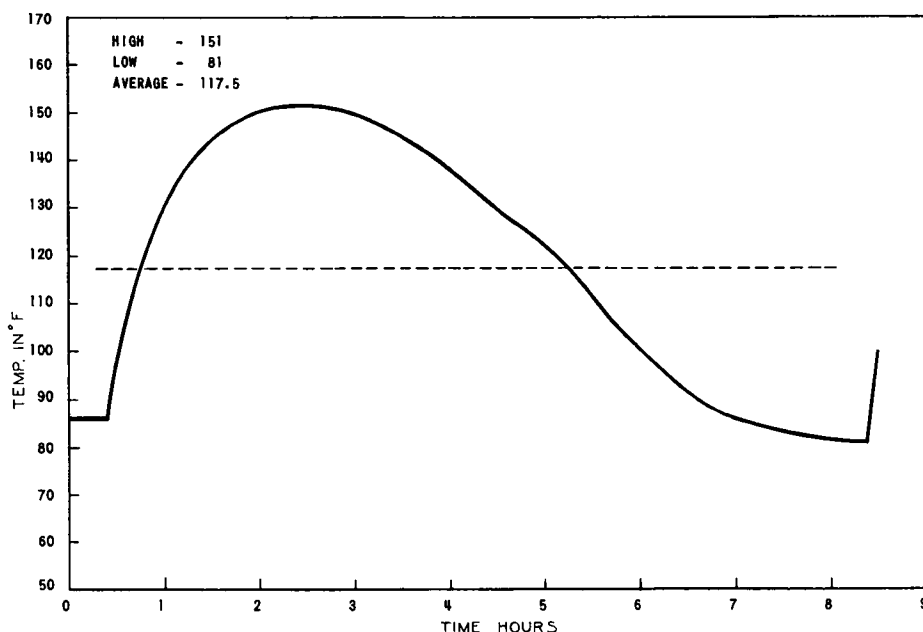

FIG. 13
TYPICAL PLOT OF AIR TEMPERATURE AFTER MAKEUP AIR DESICCATOR

C. Air Moisture

The moisture content of the air at points of design interest are tabulated in Table 3. Water vapor is indicated both in volumetric parts per million and by the ratio of "lb of water" per "million cubic ft of dry air at 60°F."

The "consolidated" moisture monitor was used to record the moisture in air from: (1) exit of makeup desiccator; (2) exit of recovered air desiccator; and (3) blower discharge. A composite plot of a typical 8-hr cycle is illustrated in Fig. 14. The major contribution of moisture to the "dry" air supplied to the seals is from the makeup air side, in the form of atmospheric H_2O. Only a very small per cent of reactor water is contributed from the desiccators of the recovered air side. It is apparent that the makeup side is less effective than the recovered air side.

Table 3

AIR MOISTURE

|	Water Vapor, ppm by volume			Water per 1×10^6 ft^3 of dry air at 60°F, lb
	High	Low	Avg	
A. Recovered Air Moisture				
1. Air entering vent condenser				183,000
2. Air entering Freon cooler	50,000	30,000	40,000	2,000
3. Air entering desiccator	22,000	19,000	20,000	1,000
4. Air leaving desiccator	30	20	25	1.3
B. Makeup Air Moisture				
1. Air entering Freon cooler			~20,000	1,000
2. Air entering desiccator	16,000	9,000	12,000	610
3. Air leaving desiccator (see Fig. 14)	1,700	40	490	25
C. Air to seals (mixture of A and B)	820	30	230	12
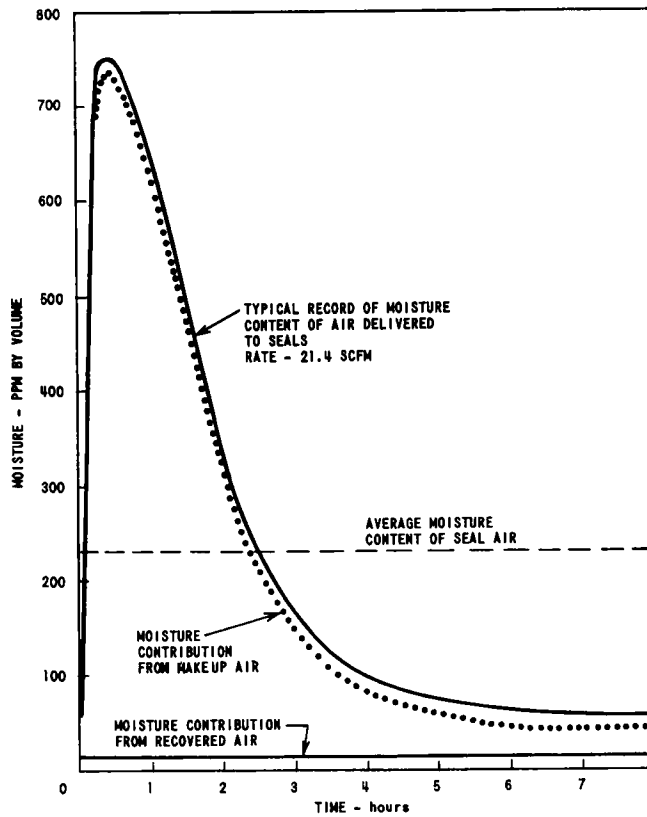
FIG. 14
DRY AIR BARRIER MOISTURE

The desiccator systems cycle from the spent alumina beds to the recharged fresh beds on an 8-hr period. During the period, the spent recovered air bed is regenerated by blowing air through a closed regeneration loop. Air from a sealed blower is heated electrically and forced through the bed. The heated bed gives up its stored moisture to the air. After leaving the bed, the air is chilled in a water-cooled heat exchanger and the entrained water vapor is condensed on the tubes. The condensate drains to the vent condenser tank and the air is drawn to the blower for recycle. After four hours the heaters are de-energized, and the blower continues to circulate, cooling the bed for the remaining four hours.

During the same 8-hr period, the spent makeup air desiccator is regenerated as follows: the makeup bed is heated by electrical heaters in the alumina bed. A blower forces atmospheric air through the bed. The bed releases its moisture to the air and the air returns to the atmosphere. Both the blower and the heaters are de-energized after four hours. For the remaining four hours the bed is left to cool by external air convection and radiation.

During the cooling period of the regeneration cycle, the recovered air bed has been cooled to less than 80°F, but the makeup air bed has only cooled to 200°F. Consequently, when the regenerated beds are shifted into operation, the makeup bed is less effective than the recovered air bed until the cool makeup air lowers the bed temperature.

The similarity of Figs. 13 and 14 readily indicates the relationship of bed effectiveness to bed temperature. In analyzing Figs. 13 and 14, the slow response of the instruments must be taken into account. The shift of the peak to the right of zero time indicates the magnitude of the lag.

The air temperatures were recorded by insulated thermocouples on the surface of the air lines. The moisture monitor drew its sample at a rate of 20 cc/min through 4 ft of $\frac{1}{4}$-in. stainless steel tubing.

During the test period, the readings of the Consolidated instrument were checked by means of an "Alnor Dew Pointer." The "Dew Pointer" was also used to sample ambient air and air at the Freon cooler exits. Moisture measurements of air leaving the Freon coolers indicated that this air is saturated. Moisture at these points is therefore indicated by the air temperature records.

D. Air Flow

The data of Table 4 and Figs. 15, 16 and 17 are a summary of more than 80 individual seal tests. The tests were conducted at intervals over a two-month period to observe the influence of general plant parameters.

Table 4

AIR FLOW

A. With Design Pressures at Seals
(i.e., +6 in. W.G. at dry gland and -6 in. W.G. at recovery gland)

1. Air Flow to Seals

	Measured Flow (scfm)	ANL-5607 Design Flow (scfm)
a. Turbine, high-pressure end	9.2	7.5
b. Turbine, low-pressure end	9.0	7.5
c. Feed pump seals	3.3	6.2
d. Throttle valve, inlet valve and bypass valve	0.1	0.7
	21.6	21.9

2. Air Flow Recovered

a. Turbine, high-pressure end	5.5	3.7
b. Turbine, low-pressure end	5.0	3.7
c. Feed pump seals	2.0	3.1
d. Throttle valve, inlet valve and bypass valve	~ 0.08	0.5
	12.6	11.0

3. Air Lost to Containment Atmosphere
(also equals makeup flow)

a. Turbine, high-pressure end	3.7	3.8
b. Turbine, low-pressure end	4.0	3.8
c. Feed pump seals	1.3	3.1
d. Throttle valve, inlet valve and bypass valve	~ 0.02	0.2
	9.0	10.9

B. With "Average" Measured Operating Pressures in Seal Glands

1. Air Flow to Seals

	Measured Flow (scfm)
a. Turbine, high-pressure end (+6.6 in. pressure & -4.3 in. vacuum)	9.2
b. Turbine, low-pressure end (+6.4 in. pressure & -4.4 in. vacuum)	8.9
c. Feed pump seals (+3.1 in. pressure & -6.9 in. vacuum)	2.7
	20.8

Table 4 (Cont'd.)

	Measured Flow (scfm)
2. Air Flow Recovered	
a. Turbine, high-pressure end (10.9 in. H_2O pressure drop through seal)	5.2
b. Turbine, low-pressure end (10.8 in. H_2O drop)	4.7
c. Feed pump seals (10.0 in. H_2O drop)	1.8
	11.7
3. Air Lost to Containment Atmosphere	
a. Turbine, high-pressure end (6.6-in. drop)	4.0
b. Turbine, low-pressure end (6.4-in. drop)	4.2
c. Feed pump seals (3.1-in. drop)	0.9
	9.1

4. % Lost = 9.1/20.8 = 43.7%

   % Recovered = 11.7/20.8 = 56.3%

LEGEND
O SEAL AIR LEAKAGE 5-4-60
X SEAL AIR RECOVERED 5-4-60
SCFM AT 32°F. AND 29.92" HG.
● SEAL AIR LEAKAGE - OTHER TESTS
⊠ SEAL AIR RECOVERED - OTHER TESTS

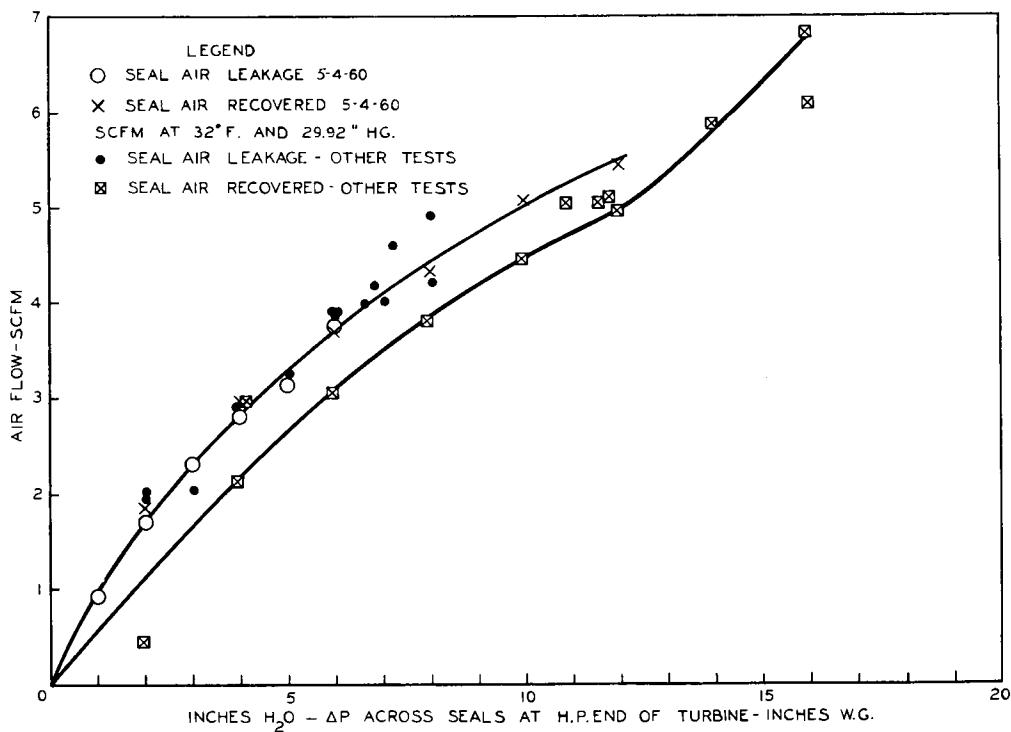INCHES H_2O — ΔP ACROSS SEALS AT H.P. END OF TURBINE - INCHES W.G.

FIG. 15
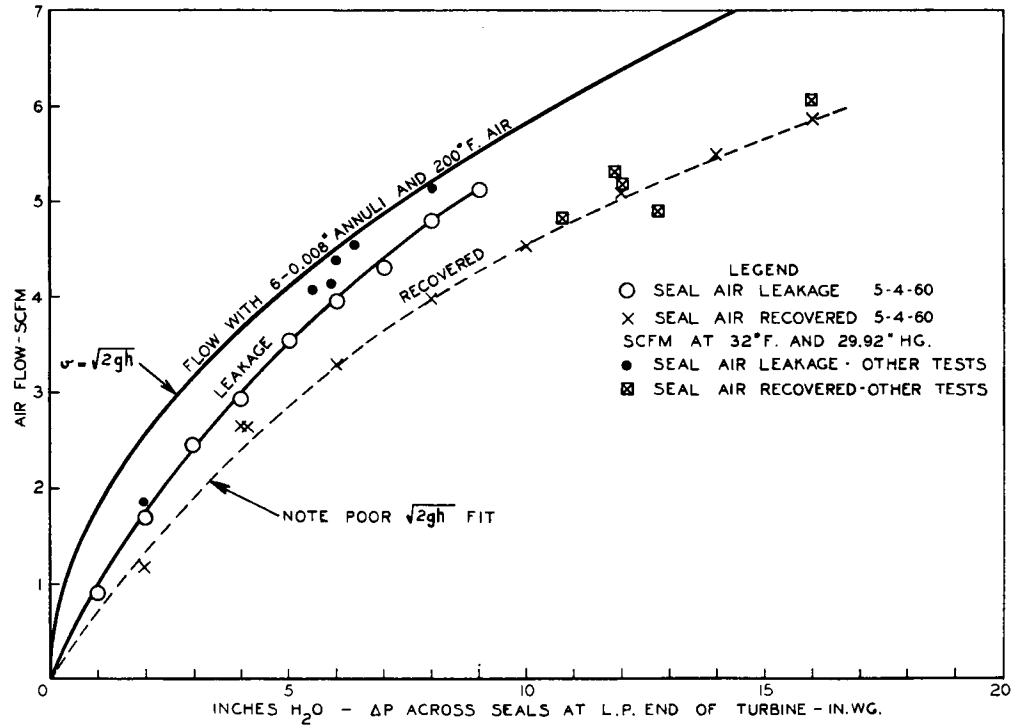SEAL TEST ON HIGH-PRESSURE END OF EBWR TURBINE

FIG. 16
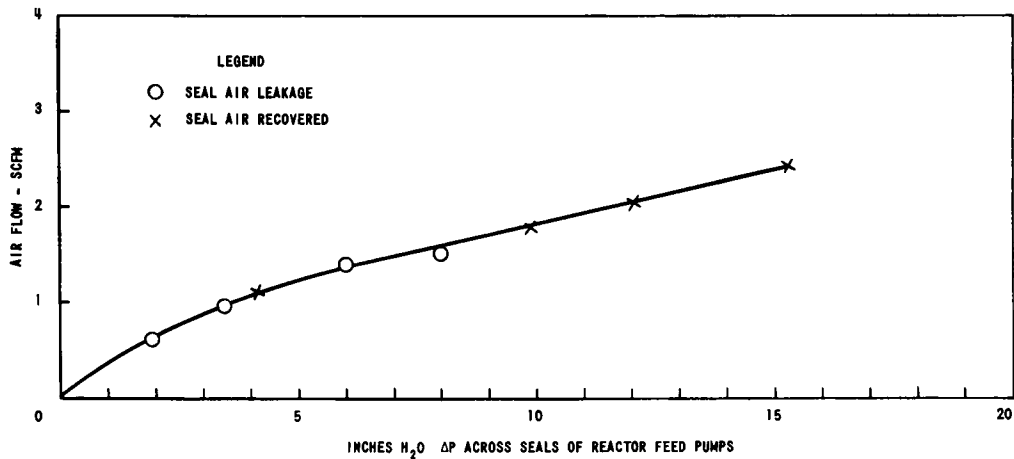SEAL TEST ON LOW-PRESSURE END OF EBWR TURBINE

FIG. 17
SEAL TEST OF FEED PUMP SEAL

Turbine seal flow rates were not found to be significantly affected by turbine load; however, the effect of turbine shaft rotation is observable. The turbine inlet valve stem seal flows were slightly affected by stem position. It was found that simultaneous flow readings were necessary to obtain good data correlation of flow rates to observed pressure drops.

1. Flow Rates with "Design" Seal Pressures

All supply air was directed through the seal rotometers, and the pressure at each seal was adjusted to +6 in. water gauge. To measure the recovered air flow of the individual seal, the recovered air was directed to the test recovery system and the seal vacuum gland pressure was adjusted to -6 in. water gauge.

Table 4-A lists the flow of air supplied to each seal, the air recovered, and the dry air lost to the reactor containment atmosphere. The measured flows listed have been corrected to standard conditions (32°F at 29.92 in. Hg). The design flow rates listed are those presented.[2]

2. Flow Rates with "Operating Average" Seal Pressures

The operating seal pressures measured during the first-phase tests were averaged to obtain seal pressures for "operating" flow rate measurements. The seal pressures were adjusted to the operating averages and the individual seals were tested. The observed flow rates are tabulated in Table 4-B. Data on valve seals are not tabulated because air flows were always much less than 0.1 scfm.

3. Seal Characteristics

The characteristics of the turbine seals are presented in Figs. 15 and 16. Feed pump seal characteristics are plotted in Fig. 17. The plots were obtained by varying seal pressures in 1 in. water gauge increments from a) 1 in. W.G. to 9 in. W.G. in the dry air gland, and b) -1 in. W.G. to -9 in. W.G. in the recovered air gland. The pressure drop through the six-stage labyrinth to atmosphere was related to leakage flow. The pressure drop through the 12-stage labyrinth was related to recovered air flow.

The seal at the high-pressure end of the turbine has coincident curves for both 6-stage and 12-stage labyrinths. Figure 4 shows the original shop clearances in the labyrinths. Simple flow theory indicates that with such clearances the coincidence of curves is not unreasonable.

The curves of the low-pressure end seal, Fig. 16, are separated, as would be expected from the similarity of clearances.

4. Labyrinth Seal, Flow Model, and Derivation of Correlation Equation

Experimental turbine seal flow rates could not be correlated by simple "orifices in series" flow equations (see Fig. 18). An attempt has been made to establish a more realistic flow model (see Fig. 19) and derive these from a flow equation as follows:
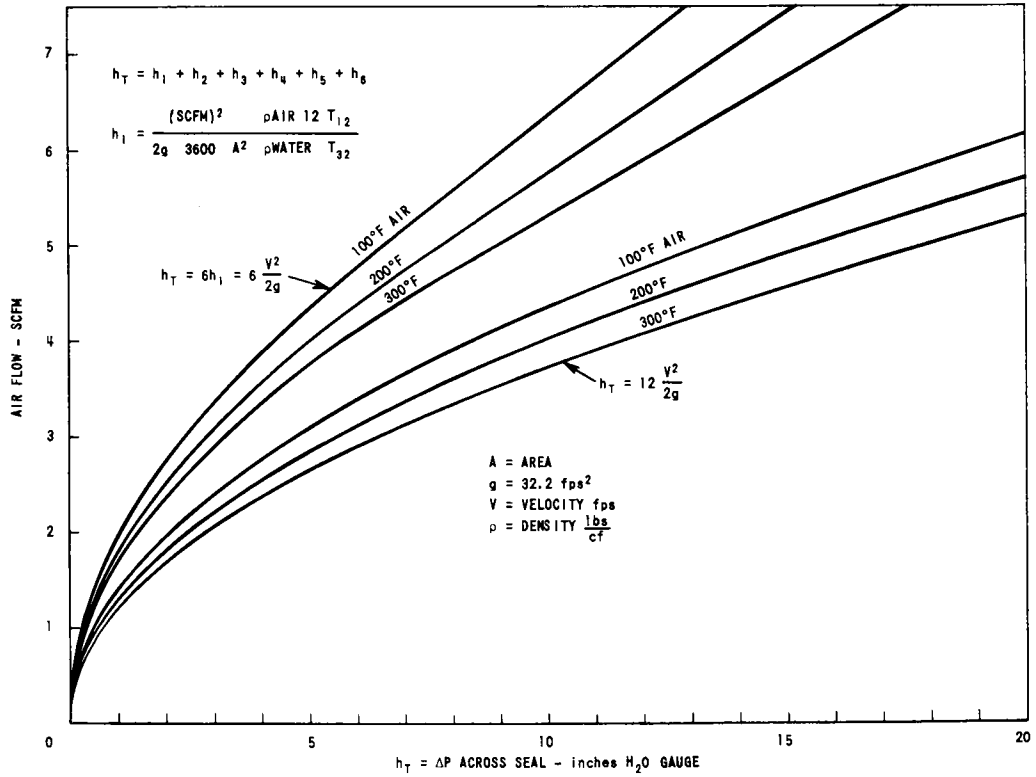
$$h_T = h_1 + h_2 + h_3 + h_4 + h_5 + h_6$$

$$h_1 = \frac{(SCFM)^2}{2g} \frac{\rho AIR}{3600} \frac{12}{A^2} \frac{T_{12}}{\rho WATER} \frac{T_{12}}{T_{32}}$$

AIR FLOW – SCFM

100°F AIR
200°F
300°F

$h_T = 6h_1 = 6 \frac{V^2}{2g}$

100°F AIR
200°F
300°F

$h_T = 12 \frac{V^2}{2g}$

A = AREA
g = 32.2 fps²
V = VELOCITY fps
ρ = DENSITY $\frac{lbs}{cf}$

$h_T = \Delta P$ ACROSS SEAL – inches H_2O GAUGE

FIG. 18
IDEAL FLOW THROUGH 6-STAGE AND 12-STAGE
LABYRINTH AIR SEALS WITH .008" ANNULAR ORIFICES

g = 32.2 ft/sec²
h = HEAD LOSS THROUGH ORIFICE
V = AVERAGE AIR VELOCITY IN ORIFICE
V_a = AXIAL COMPONENT OF AIR VELOCITY IN THE ANNULUS
V_t = TANGENTIAL COMPONENT OF AIR VELOCITY IN
THE ANNULUS

$\frac{1}{16}$"

$\frac{5}{16}$"

$\frac{1}{16}$"

125 FPS

$\frac{1}{16}$"

$\frac{3}{16}$"

.006 ± .002

V_a

h_a

122 FPS

SHAFT SURFACE SPEED

V_t

V

h_t

h

8.00" DIA.

7.75" DIA.
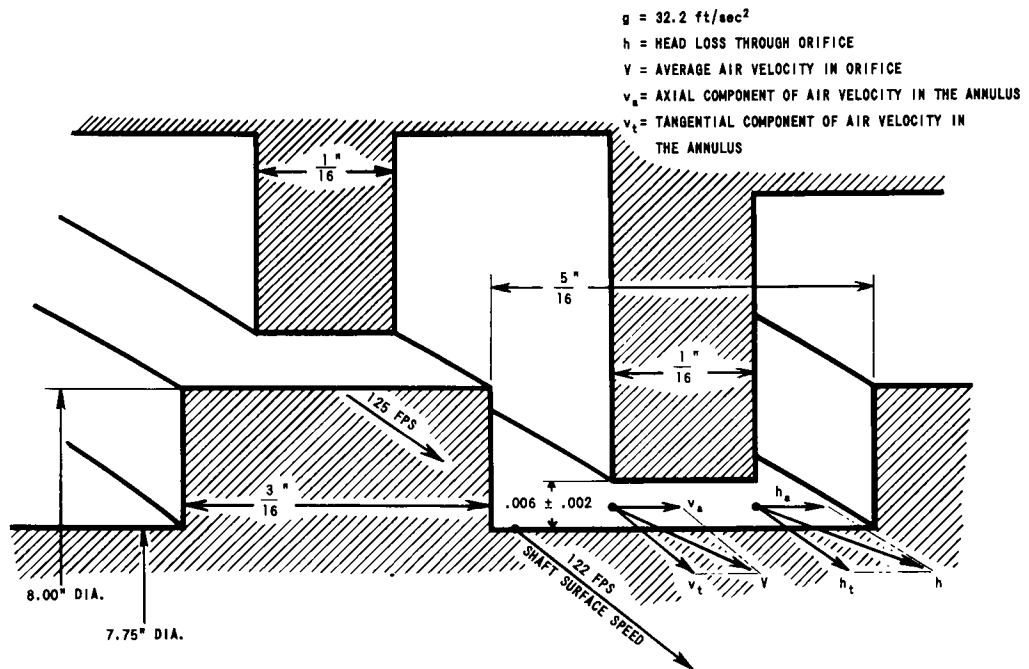
FIG. 19
LABYRINTH SEAL, FLOW MODEL

$$(1) \quad V = \sqrt{v_t^2 + v_a^2}$$

$$(2) \quad h = \sqrt{h_t^2 + h_a^2}$$

$$(3) \quad h = V^2/2g$$

$$(4) \quad \sqrt{h_t^2 + h_a^2} = (1/2g)(v_t^2 + v_a^2)$$

$$(5) \quad h_t^2 + h_a^2 = \frac{1}{(2g)^2} (v_t^4 + 2 \, v_t^2 \, v_a^2 + v_a^4)$$

$$(6) \quad h_t^2 + h_a^2 = \frac{v_t^4}{4g^2} + \frac{2 \, v_t^2 \, v_a^2}{4g^2} + \frac{v_a^4}{4g^2}$$

But for practical purposes, the axial inlet velocity before the annulus is zero, the axial velocity after the annulus is zero, and none of the axial velocity head in the annulus is recovered. Since the tangential velocity is induced by shaft rotation it is not unreasonable to assume that it is near constant, both in and out of the orifice.

$$(7) \quad h_t^2 = \frac{v_t^4}{4g^2} \cong \text{ a constant.}$$

Then the head loss is equal to

$$(8) \quad \Delta h_a^2 = \frac{v_a^4}{4g^2} + \frac{2 \, v_t^2 \, v_a^2}{4g^2}$$

or

$$(9) \quad \Delta h_a = \frac{v_a}{2g} \sqrt{v_a^2 + 2 \, v_t^2} \quad .$$

A good fit is achieved with $v_t = (1/4)$(shaft surface speed)(see Fig. 20).

The equation shown in the figure:

$$h_1 = \frac{v_a}{2g}\left(v_a^2 + 2v_t^2\right)^{\frac{1}{2}}$$

h = PRESSURE DROP FOR ONE STAGE
h_T = TOTAL PRESSURE DROP IN SEAL
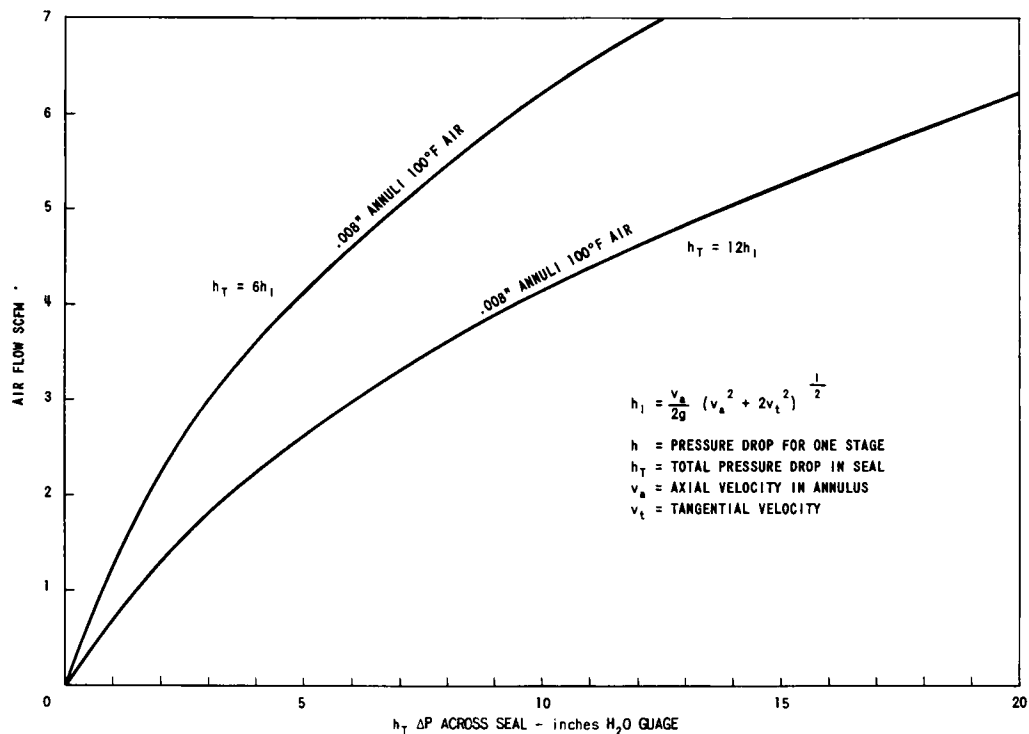v_a = AXIAL VELOCITY IN ANNULUS
v_t = TANGENTIAL VELOCITY

FIG. 20
CORRELATION FOR TURBINE ROTATING LABYRINTH SEALS
"TWO COMPONENT FLOW" MODEL

E. Seal Steam and Seal Water Flows

The seal steam flow entering the recovered air vacuum glands remained essentially constant throughout the test. Both high-pressure and low-pressure end flow rates were equal. Seal steam pressure was automatically controlled to a constant 2 psi and the labyrinths on both ends have 12 stages; hence, the equal flow rates observed infer equal labyrinth clearances at both ends. The flow rates to the recovery system were higher than the design estimates, probably indicating labyrinth wear. This wear was noted at the 3,900-hr turbine seal inspection. The turbine has operated 5,000 hr since that time. The values of Table 5 are the averages of 30 "individual" seal measurements, and checked by 15 "multi" seal tests. During testing, deviations from this average were observed to be caused by holdup in the water seal loops of the drainage lines, and by the measurement technique. From the data, the actual flow is judged to be very stable. The seal steam temperature averaged 336°F at 16 psia. The heat rate of the vent condenser at 120 lb/hr is 125,000 Btu/hr, which is one-half its rated load.

The seal water flow of the feed pumps is very close to design expectations, indicating proper seal operation. Flow rate was relatively small and leakage collection extended into hourly periods.

Table 5

SEAL STEAM AND SEAL WATER FLOWS*

	Measurement, lb/min	Design, lb/min
A. Turbine, high-pressure end	1.07	0.55
B. Turbine, low-pressure end	1.08	0.55
C. Feed pumps	0.04	0.043
D. Throttle, inlet and bypass valves	0.3	0.13
	2.49	1.27

* These flows are proportional to the pressure of the seal
steam (or water). Seal steam pressure was 2 psig during
all tests. The small variations in seal air gland vacuum
therefore had negligible effect upon the seal steam flow
rate. Further, the flows are not leakages, since they are
all collected in the vapor recovery system.

Flow rates from valve seals were greater than design expecta-
tions; however, the low flow rates and line holdup precluded a high degree
of accuracy in these measurements. No rearrangements were made to
improve accuracy since flows were a negligible per cent of total.

F. Leakage and Contamination

From the above data, the primary system moisture out-leakage
rate and the atmospheric air moisture in-leakage rates were calculated.
To illustrate the leak rates that would occur, it has been assumed that the
primary system is filled with D_2O. The air flows assumed for these calcu-
lations were 11.7 scfm recovered, 9.1 scfm lost, and 9.1 scfm makeup.
The total flow supplied to seals is 72,000 lb/month.

1. With "best" vapor recovery system operation:
    a. D_2O in air to seals 14.1 ppm by volume
    b. H_2O in air to seals 11.5 ppm by volume
    c. D_2O leakage to containment shell 0.27 lb D_2O/month
    d. H_2O contamination of primary D_2O 0.29 lb H_2O/month

This condition could have been achieved if the makeup air desiccators were regenerated by a system similar to that of the recovered air desiccators. It was assumed that the makeup air desiccators were as effective as those of the recovered air.

2. With the actual observed normal operation:

a. D_2O in air to seals	14.1 ppm
b. H_2O in air to seals	134 ppm
c. D_2O leakage to containment shell	0.27 lb D_2O/month
d. H_2O contamination of D_2O	5.5 lb H_2O/month

3. With observed abnormal operation:

a. D_2O in air to seals	3540 ppm
b. H_2O in air to seals	216 ppm
c. D_2O leakage to containment	69 lb D_2O/month
d. H_2O contamination of D_2O	5.5 lb H_2O/month

This considers the makeup desiccators operating normally and the recovered air desiccators inoperative, as observed in Section VII-G.

G. Observed Abnormalities

1. Failure of Recovered Air Reactivation Loop Blower

During the first week of testing it was observed that the dry air supplied to the seals was clearly higher in moisture content than specified by design. The dew point of the main dry air supply to seals was +27°F to +35°F, and averaged +33°F (6300 ppm by volume) over a 24-hr period. The difficulty was traced to a motor burnout of the subject blower. Note that this blower is a sealed unit without rpm indication or power alarm. The only indication of failure was the low temperature of the desiccant bed being reactivated. As a consequence of the blower failure, the alumina desiccant became saturated with moisture. Upon repair of the blower, one week of normal 8-hr cycle reactivation was required to dry the beds.

2. Valve and Pump Seals

The inlet valve and bypass valve seal chamber pressures did not correspond to design. However, since sufficient pressure was available for an effective dry air barrier, this abnormality did not result in abnormal leakage. The feed pump seal and bypass valve seal low

pressures are attributed to a pipe restriction. The low pressure of the inlet valve recovered air/steam chamber is attributed to a high seal steam flow rate caused by seal wear. None of these abnormalities were corrected during the test period.

3. Hydrostatic Testing

During hydrostatic tests on the reactor vessel (which is normal practice after reactor pressure vessel modifications), it was observed that water was forced into the dry air lines through the throttle valve and bypass valve seals. This occurred because (a) the vapor recovery system was normally not operating at such time and (b) the bypass valve vacuum line was not properly pitched to drain the chamber to the vent condenser tank. This occurrence was partially overcome by operating the vapor recovery system during hydrostatic tests. The line pitch was not corrected during the test period.

4. Makeup Air Desiccator Cooling

The design of the reactivation loop for the makeup air desiccators led to a prediction that the refreshed bed would cool to operating temperature by radiation. It was observed that radiation did not completely cool the bed. The effects of this are presented in Section VII-C. An improved cycle could be achieved by providing post reactivation cooling of these beds.

5. Seal Air Pressure Controller

During the test it was found that the seal dry air pressure controller could not be operated in the "automatic" position without excessive hunting. Satisfactory operation was obtained by use of the "manual" setting allowing manual regulation of the makeup air control valve.

VIII. CONCLUSIONS

The vapor recovery system has operated in reasonable accord with design expectations. All associated seal leakages were collected and recovered. Effective dry air barriers were maintained at all system seals. Air and vapor flows were found to be in close accord with design values. All this was accomplished in spite of the fact that some valve seal pressures differed considerably from design and the makeup desiccators were not cooled fully after regeneration. Even with such a severe abnormal condition as a complete failure of the recovered air desiccators, the system was able to hold leakage to a tolerable level.

It was found that the heat exchange and desiccation equipment is capable of handling greater leak rates than experienced under normal operation. This results from the system design to recover vapor from air ejectors during condenser-hogging operations and from the assumed design parameter of 100°F vent condenser cooling water.

Extrapolating operating experience to a nonexperimental power plant, it is concluded that substitution of two half-capacity units would be preferable to the one installed. This would give the overcapacity needed for hogging and possible seal failure plus the ability to make recovery system repairs with the plant in operation.

From measured air flows it appears that the turbine shaft air labyrinths have very little wear. The turbine seal steam flows were twice the design rate, indicating labyrinth wear of the order of 0.005 in. This steam labyrinth wear was noted, but not measured, at the last turbine overhaul. Inlet valve seal pressures indicate seal wear sufficient to warrant replacement. The feed pump seal appears to be in excellent condition.

Maintenance of "dry" seal air is the major factor required for low D_2O leak rates. With good dry air control, seal labyrinth tolerances could be relaxed. The recovery system is capable of maintaining the D_2O leakage from its associated seals to less than one pound per month. It could also be made capable of holding the atmospheric vapor in-leakage to less than one pound per month. With system redesign to "heat-less" type desiccators, leakage could be reduced to the order of one pound per year. It should be noted that these low leak rates do not represent total system leakages, since all seals are not connected there-to and there are other leakages not amenable to recovery system application.[2]

IX. RECOMMENDATIONS

The existing system should be modified to (1) drain air lines properly; (2) remove obstruction in the feed pump dry air line; (3) change make-up air desiccator regeneration to cool bed properly; and (4) insert an accumulator in the dry air manifold to permit utilization of the automatic pressure controller.

It is believed that more simpler vapor recovery could be achieved with a pressurized desiccation system. A flow diagram for a pressurized system is presented in Fig. 21. Flow rates are based upon the results of this test. It is assumed that short-term overloads would be handled by a standby system. The flow in the figure is similar to the installed system except that the air is compressed to 100 psig. The desiccators are regenerated by expanding a small portion of the 100-psi dry air through the idle bed to atmosphere.

Pressurized desiccators would eliminate the reactivation heaters, blowers, and air coolers, and could be operated without a Freon system. In addition, the resultant quality of the air supply could be as low as 1 ppm.
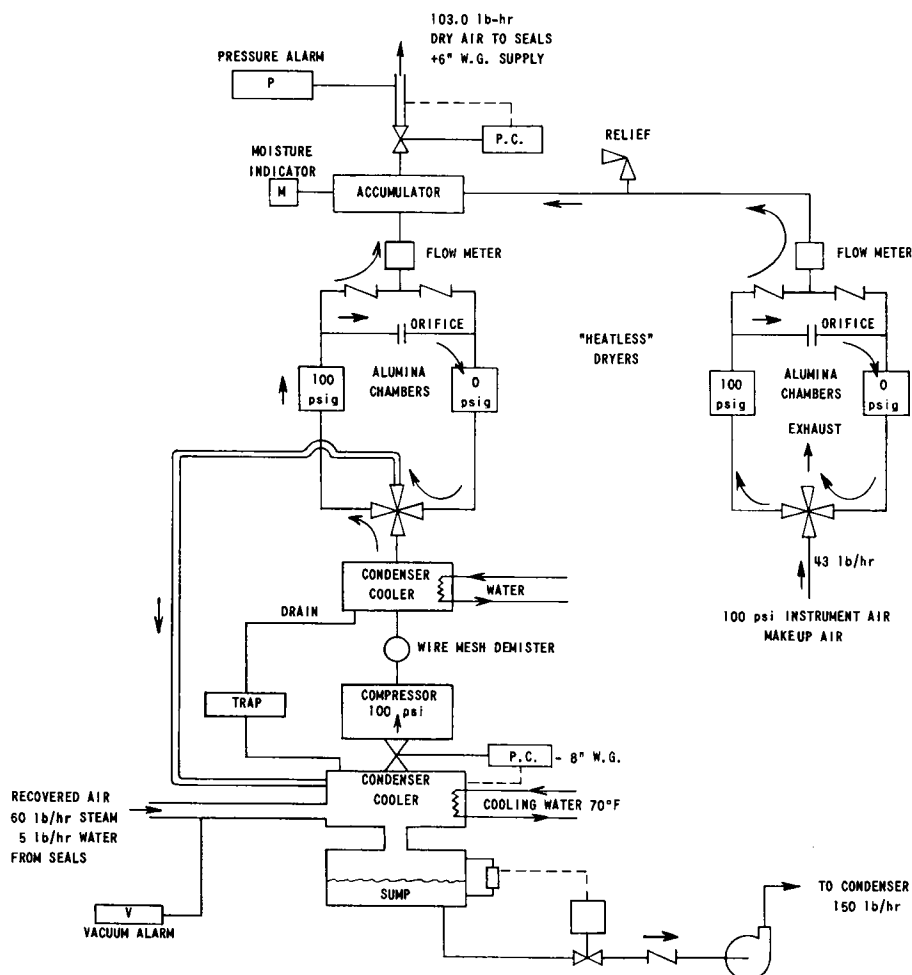
FIG. 21

VAPOR RECOVERY WITH PRESSURIZED DESICCATORS

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