Title
DIRECT CONTACT HEAT EXCHANGER 10 kW POWER LOOP. SECTION 1: EXECUTIVE SUMMARY. SECTION 2: TEST SERIES NO. 1. SECTION 3; TEST SERIES NO. 2

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DIRECT CONTACT HEAT EXCHANGER 10 kW POWER LOOP

SECTION 1: EXECUTIVE SUMMARY
SECTION 2: TEST SERIES NO. 1
SECTION 3: TEST SERIES NO. 2

Barber-Nichols Engineering Co.

July 1979

Prepared for the U.S. Department of Energy
under Contract W-7405-ENG-48

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FOREWORD

The extraction of thermal energy for electric power generation from moderate (350°F) temperature geothermal brines, such as those found in East Mesa, California, appears feasible. In exploiting these moderate temperature resources, the use of a direct-contact process in a conventional Rankine cycle with a turbine-generator seems like a very promising configuration.

While the direct contact approach, in which the brine and hydrocarbon working fluid come into physical contact, has initial economic advantages (e.g., lower capital cost of heat exchanger equipment), it also has some potential operating disadvantages such as CO_2 buildup, turbine scaling and erosion, and hydrocarbon loss. In order to identify and evaluate these problem areas, a 10 kW Power Loop was set up and run at East Mesa, California, under DOE funding. This report includes the results and conclusions of Test Series 1 and 2 performed between December 1976 and July 1978, as well as the Executive Summary of the total testing effort. It has been reprinted directly from copy provided by Barber-Nichols Engineering Co.

This work has been supported by the Division of Geothermal Energy, U. S. Department of Energy.

SECTION 1

DIRECT CONTACT HEAT EXCHANGER
10 kW POWER LOOP – EXECUTIVE SUMMARY

Prepared for:

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1.0 INTRODUCTION

In December of 1976 a program was initiated to couple a turbo-generator and controls to a direct contact heat exchanger system utilizing geothermal brine to vaporize isobutane. The purpose of the program was to identify and evaluate problem areas in utilizing such a system for electrical power generation.

The turbo-generator consisted of an axial flow, partial admission turbine, a gearbox, and a 60 kw generator. These components along with controls and safety features were shipped to the East Mesa Geothermal Component Test Facility near Holtville, California.

Turbine calibration tests were completed on pure isobutane vapor using a conventional tube-in-shell heat exchanger and then a 500 hour endurance test was completed with the direct contact heat exchanger (DCHX).

A number of significant problems were identified during the 500 hour endurance run. Larger than anticipated amounts of brine carryover caused scale deposits to occur in the turbine rotor and nozzles reducing turbine output power. Condenser pressures were much higher than predicted for an isobutane/water vapor mixture. Isobutane state points entering the turbine were not always as predicted. The isobutane flow rate as measured by the liquid rotameter did not give a true representation of the actual flow rate through the turbine nozzles.

As a result, in March and April, 1978, a second phase of testing was conducted to answer the questions identified during the earlier tests conducted in July and August, 1977. The purpose of these tests was to obtain improved nozzle flow calibration by using a flow orifice in the isobutane vapor line to the turbine, to obtain improved turbine performance calibration data at subcritical and supercritical condition by using a calorimeter to determine isobutane state points at the turbine inlet, and to evaluate turbine scaling problems associated with the DCHX by removing entrained moisture from the turbine vapor flow in a demister vessel.

To achieve the above objectives another series of calibration tests and a 200 hour endurance run were performed after modifying the test loop to add the demister vessel, the flow orifice and a calorimeter. In addition, a new nozzle was designed and fabricated to take into account the higher condenser pressure encountered during the earlier testing.

At the completion of the second series of tests, turbine scaling still remained as a significant problem. Although the demister removed 50 to 75 percent of the entrained moisture from the turbine vapor flow, the gradual reduction of output power observed during the first test series was still evident.

To evaluate brine carryover, a third series of tests was performed.

During these tests the turbogenerator was replaced by a turbine nozzle identical to the one used for the second series of tests. The nozzle was installed in the flow tube which had been used to measure turbine vapor flow during the second series of tests. A water trap and conductivity cells were also installed in the demister to monitor the amount and salinity of water collected.

The direct contact heat exchanger was modified with a second level controller and the tangential brine inlet line was placed just at the free surface level inside the DCHX in order to centrifuge out the brine entrainment. A 100 hour test with this configuration resulted in no scale formation in the nozzle.

This section contains only the most significant results obtained during the last three test series. For further information consult Sections 2 and 3 and LBL-8558 (Ref. 5).

2.0 POWER MODULE

2.1 DESIGN PARAMETERS

2.1.1 Turbine

During the design phase it was anticipated that the turbo-generator would be operated with two basic direct contact heat exchanger systems. The first, and the only one which was tested, was a subcritical system fabricated and tested by DSS engineers. The second was a supercritical system from Occidental Research Corporation (ref. 1 and 2).

Based on discussions with DSS and Occidental, the system operating parameters were defined as shown in Table 1. The anticipated water carryover fraction shown in the table was calculated assuming saturated water vapor and real gas properties of the isobutane at the discharge of the vaporizer. Turbine available energy was calculated based on an isentropic expansion of the vapor mixture. Turbine design conditions were then determined for the respective cycles.

Table 1. System Operating Parameters

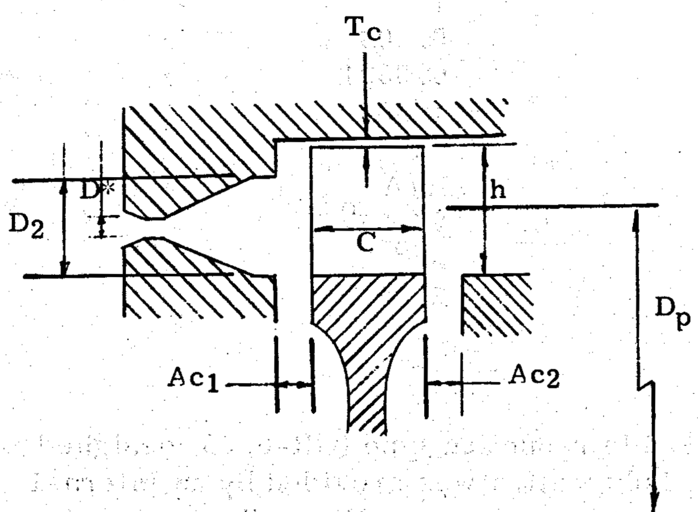
	Occidental	Dss
Isobutane flow rate, lb/hr	2000	2650
Water Vapor flow rate, lb/hr	50	32
Combined flow rates, lb/hr	2050	2682
Max. total pressure, psia	667	315
Max. temperature, ^{o}F	300	220
Condensing temperature ^{o}F	110	110
Total Condensing pressure, psia	83.2	83.2
Turbine available energy, BTU/lb	36.2	23.7

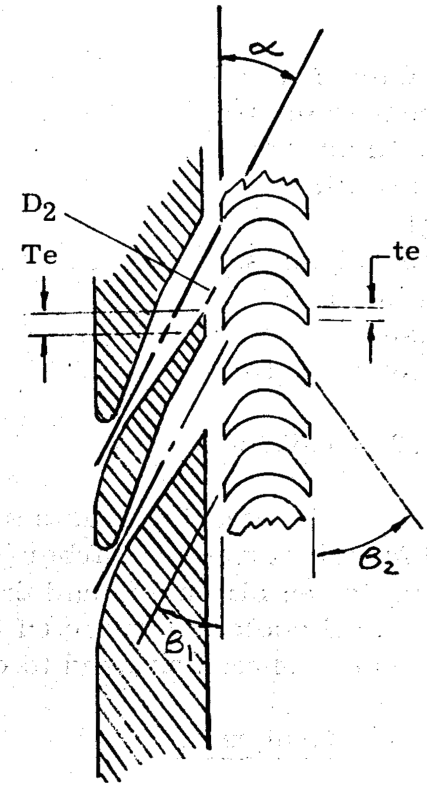
The turbine selected for this application was a partial admission impulse design incorporating a single rotor. Tables II and III give the detailed

TABLE II

TURBINE DESIGN PARAMETERS FOR

DIRECT CONTACT POWER LOOP TURBINE





1.	Pitch diameter, D_p	5.37 in.
2.	Blade chord, C	0.32 in.
3.	Blade height, h	.404 in.
4.	No. of blades, Z	102
5.	Blade entrance angle, β_1	32°
6.	Blade exit angle, β_2	32°
7.	No. of nozzles, N	*
8.	Trailing rotor edge width, te	.010 in.
9.	Throat diameter, D*	*
10.	Exit diameter, D_2	*
11.	Nozzle area ratio, Ar	*
12.	Nozzle angle, α	*
13.	Edge thickness between nozzles, Te	*
14.	Arc of admission, ϵ_n	*
15.	Tip clearance, T_c	.015 in.
16.	Axial clearance between nozzle & wheel, A_{c1}	.020 in.
17.	Axial clearance between wheel & exhaust, A_{c2}	.030 in.
18.	Arc of exhaust, $\epsilon_{exh.}$	*
19.	Desigh pressure ratio, $PR_{des.}$	*

*NOZZLE DESIGN PARAMETERS

		Sub.	Super.
7.	No.-nozzles	2	1
9.	Throat dia.	.229 in	.199 in
10.	Exit dia.	.286 in	.337 in
11.	Area ratio	1.567	2.87
12.	Angle	16°	16°
13.	Edge thk.	.010 max	N/A
14.	Admission	45°	26.1°
18.	Exhaust	55°	55°
19.	Pres. ratio	3.79	7.21

design parameters for the turbine rotor and the nozzles. The nozzle parameters shown in Table II are for the nozzle block used during the series 1 tests. Since during the series 1 testing the turbine exhaust pressure was approximately 10 to 20 psi higher than anticipated, a new subcritical nozzle was designed for the series 2 testing. Table III lists the design parameters for the series 2 subcritical nozzle.

Table III Series 2 Nozzle Design Parameters

	Subcritical (Single Nozzles)
Number of Nozzles	1
Throat Diameter	0.306
Exit Diameter	0.3521
Area Ratio	1.306
Angle	16°
Edge Thickness	N/A
Admission	27.34°
Exhaust	55°
Pressure Ratio	3.32

2.1.2 Gearbox

The gearbox was a standard double reduction type (GR=6.05) designed and manufactured by Barber-Nichols. Lubrication was provided by an internal chain driven oil pump, and the oil temperature was controlled with a water cooled oil cooler mounted on the gearbox housing. A mechanical-seal on the high speed shaft was used to contain the isobutane in the turbine exhaust housing.

2.1.3 Generator

The generator was a standard 60 kw "KATO" 3 phase synchronous machine. The generator power package used a speed control consisting of a Woodward governor, an SCR power controller and a load bank consisting of two electric space heaters. The governor was an electronic unit with isochronous capability that measured shaft speed and provided a voltage output that increased or decreased as the speed attempted to move from the setpoint. The output voltage from the governor modulated a power controller which adjusted the load on the generator by controlling power fed to the parasitic load bank. The speed set point on the governor could be changed allowing turbine performance to be measured over a range of speeds. This variable speed feature allowed the one turbine design to meet the different levels of the subcritical and supercritical cycles.

2.2 PREDICTED PERFORMANCE

Based on the design parameters of Tables I, II, and III, the predicted variation of turbine efficiency with speed is shown in Figure 1. Because of the allowable test speed range of the alternator as mentioned above, the turbine

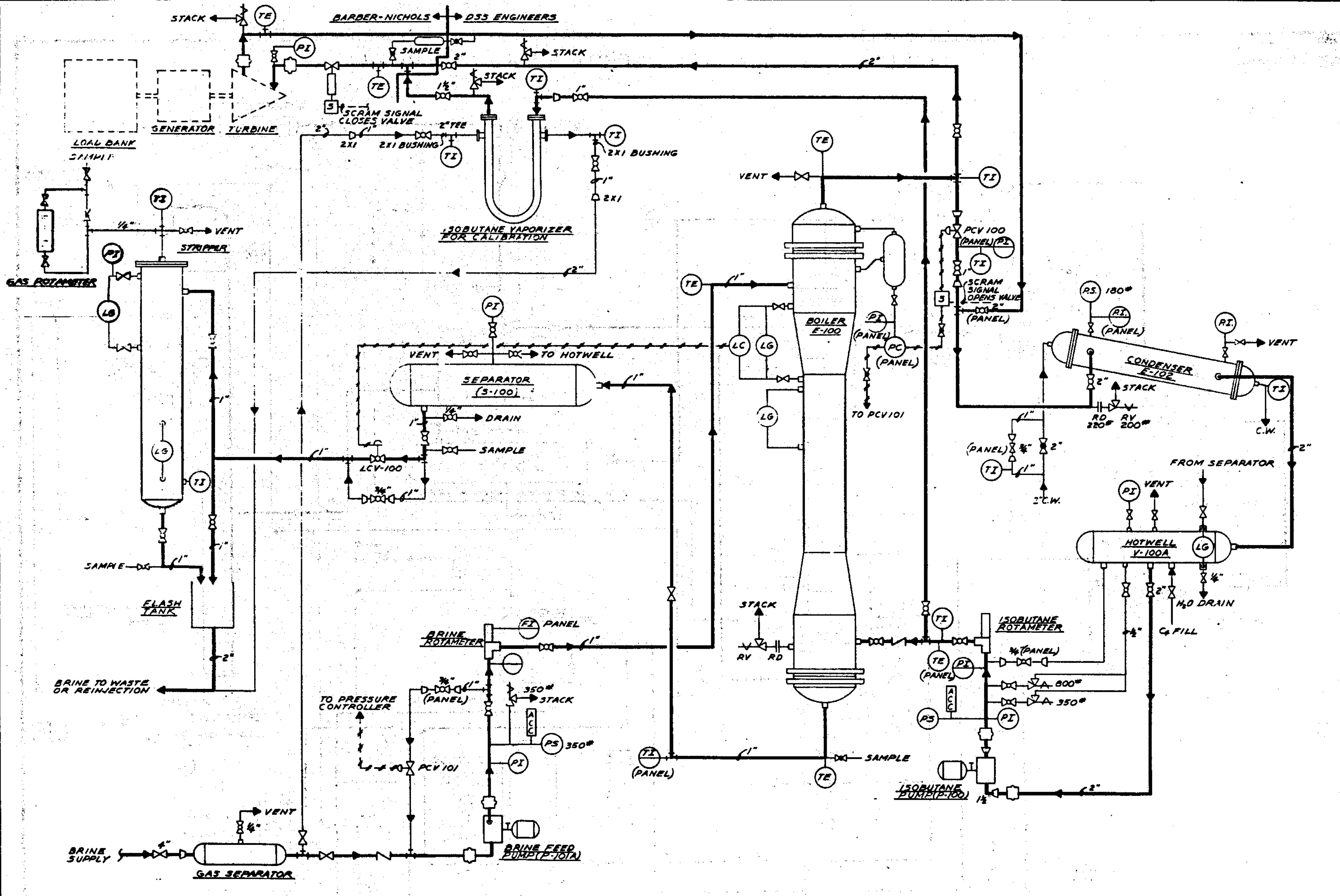


Figure 2. P & I diagram for DCHX power plant.

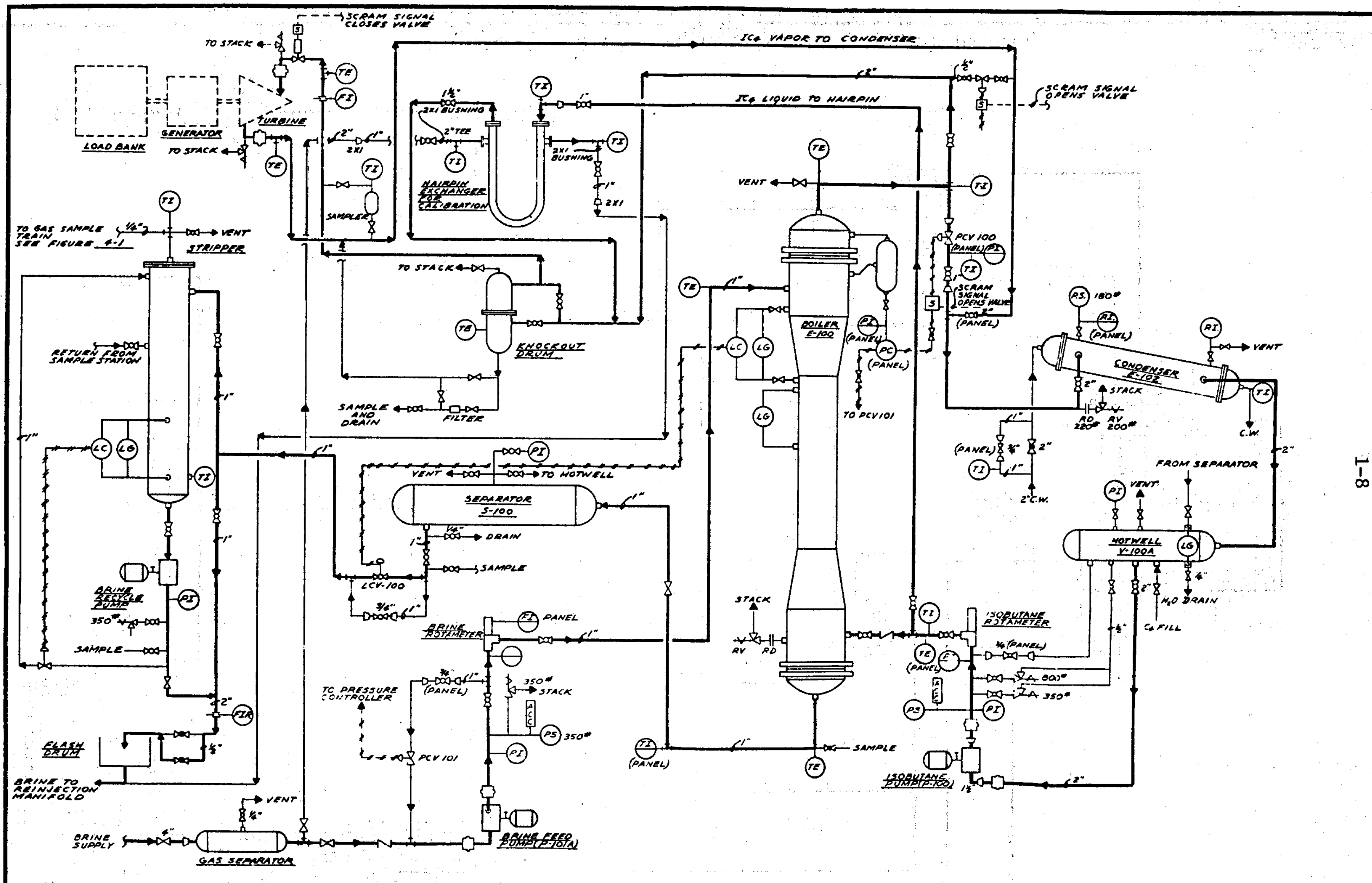


Figure 3. P & I diagram for DCHX power plant.

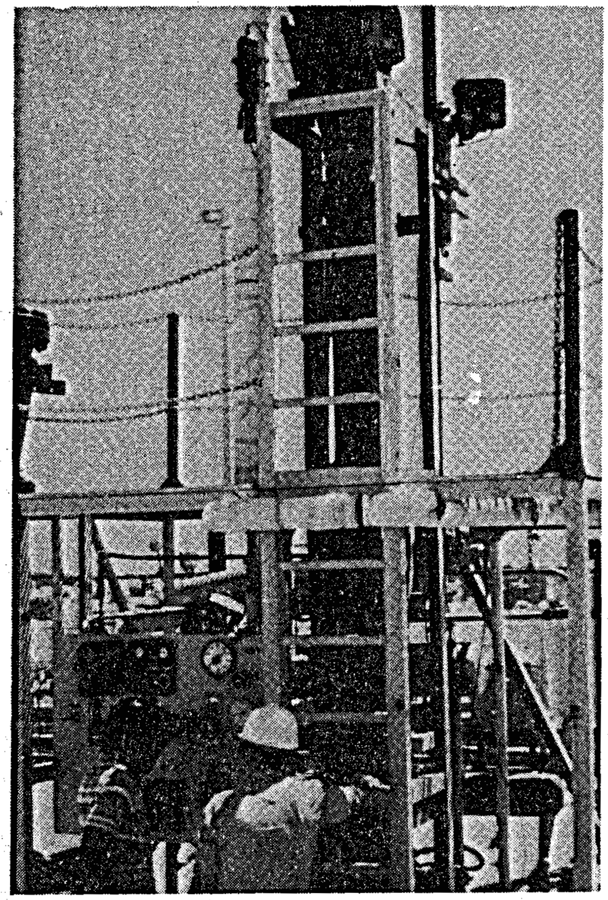


Figure 4. Direct contact heat exchanger.

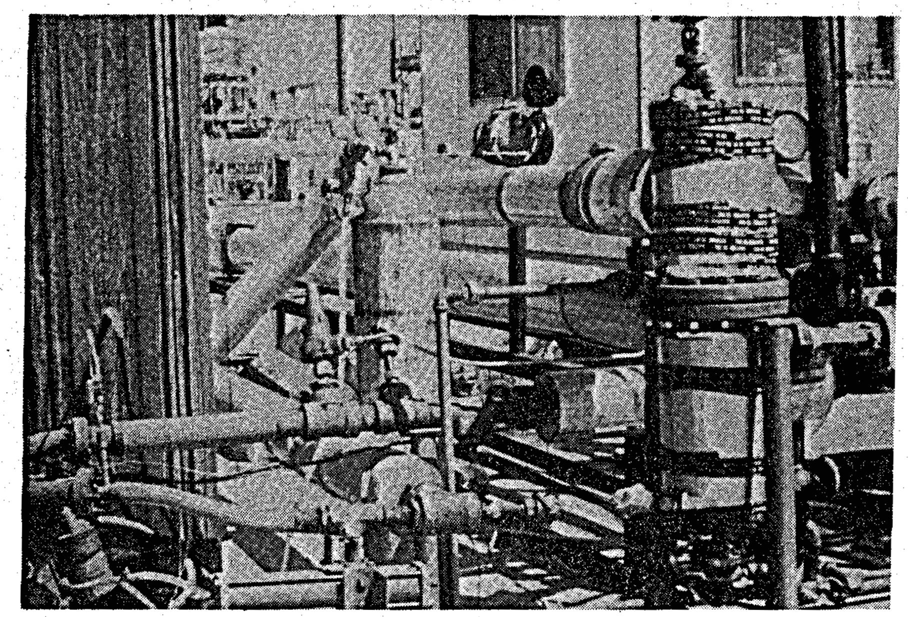


Figure 6. Demister vessel.

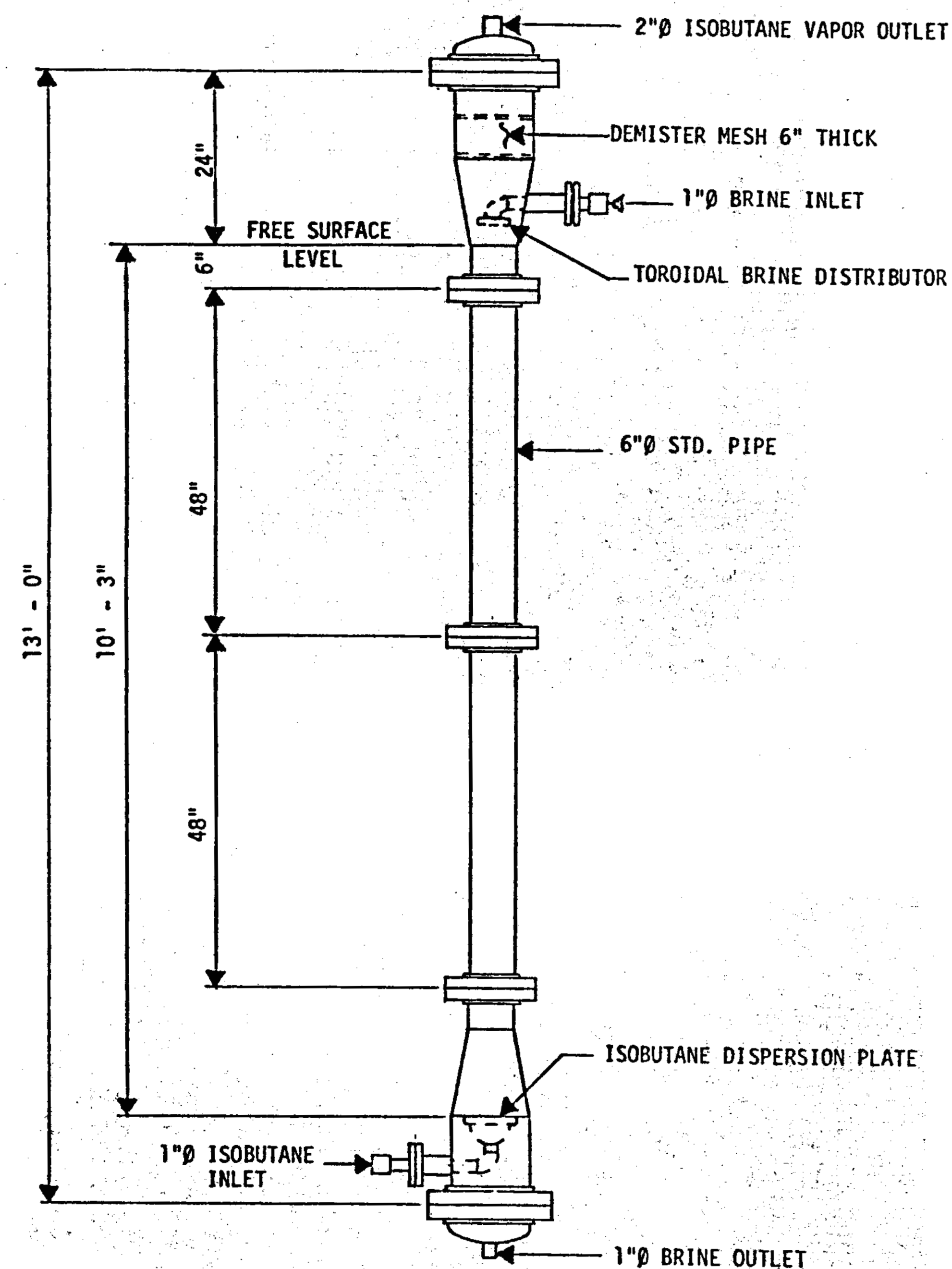


Figure 5. Direct contact heat exchanger.

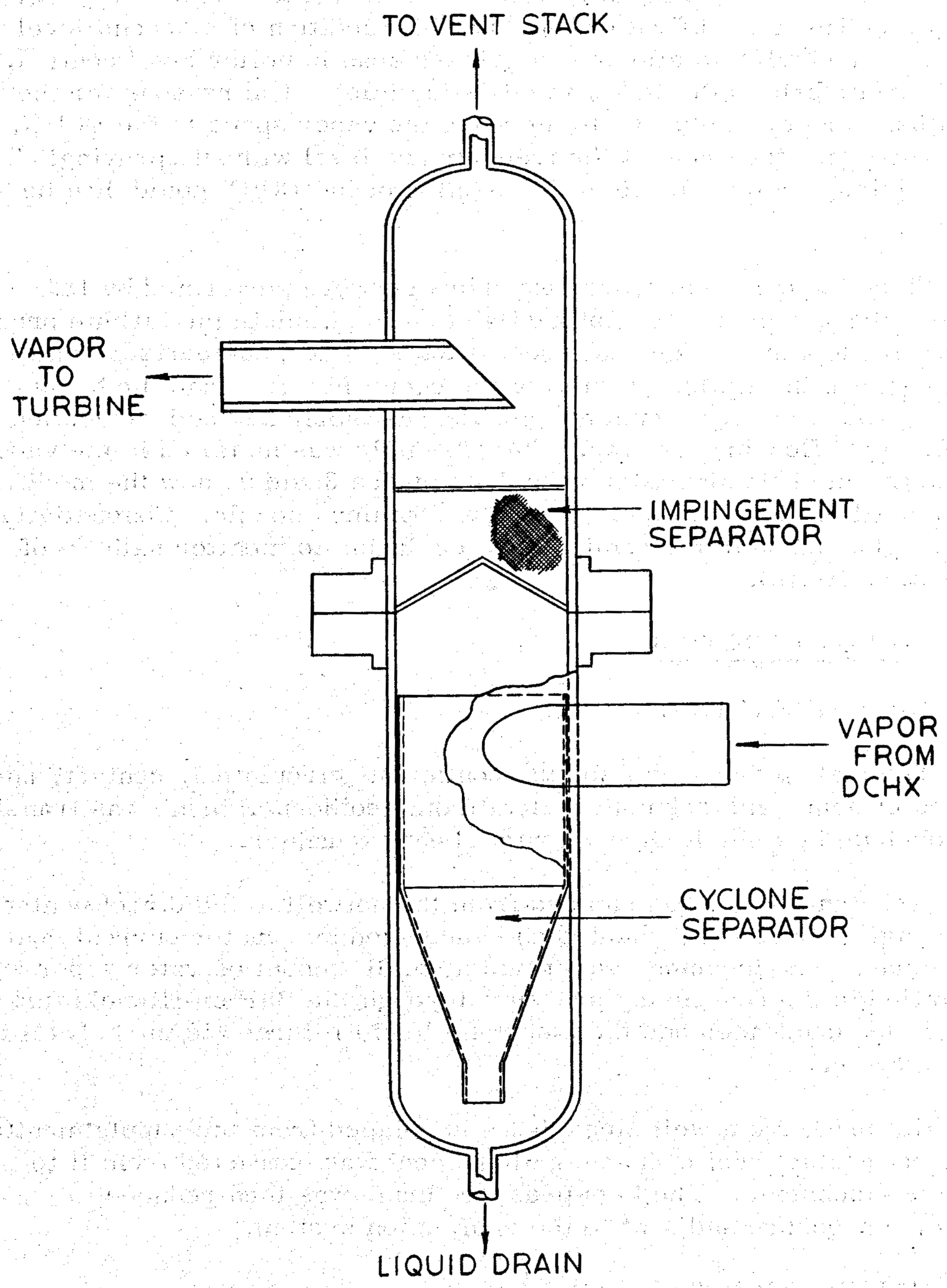


Figure 7. Demister vessel.

The test loop was again modified to perform the brine carryover experiments. These modifications included the addition of a second level controller to the DCHX to allow for a greater span in boiler level control, and an alternate brine inlet below the existing inlet. The reasons for the modification were two fold: 1) to increase the vapor space in the DCHX, and 2) to introduce the brine at the free surface level without spraying. The effect of minimizing turbulence in this section of the DCHX could then be evaluated.

Since the brine carryover experiments were performed by DSS only, the turbo-generator was not available. To simulate the turbine pressure drop and provide a media for scale deposition so that a comparison could be made between the scaling results of the endurance runs and the brine carryover experiments, a "dummy" nozzle was fabricated and assembled into a "Daniels" flow orifice tube. The assembly was installed in the vapor line downstream of the demister vessel. Figures 8 and 9 show the modifications to install the level controller and the "dummy" nozzle. Conductivity cells were also added to the demister water drains to monitor salinity of the blowdown effluent.

4.0 FIELD OPERATIONS

4.1 LOOP OPERATION

The test unit provided for the continuous circulation, contact, and separation of brine and isobutane. Heat from geothermal brine was transferred to the isobutane in a single direct contact heat exchanger.

Isobutane liquid was pumped from the hotwell to the direct contact heat exchanger where it was heated and vaporized by counter current contact with the brine. The isobutane vapor and a small amount of water vapor was then throttled to the condensing pressure through the Barber-Nichols turbine. The vapor was condensed and the isobutane liquid returned to the hotwell to repeat the cycle.

Hot brine from well Mesa 6-2 was pumped from the supply manifold to the direct contact heat exchanger where heat was extracted from it to vaporize the isobutane. The high pressure brine was then reduced to atmospheric pressure and sent to the reinjection system.

4.2 CALIBRATION TESTS

The primary goal of the calibration tests was to obtain turbine and cycle performance data with pure isobutane for use in evaluating results of the direct contact heat exchanger tests.

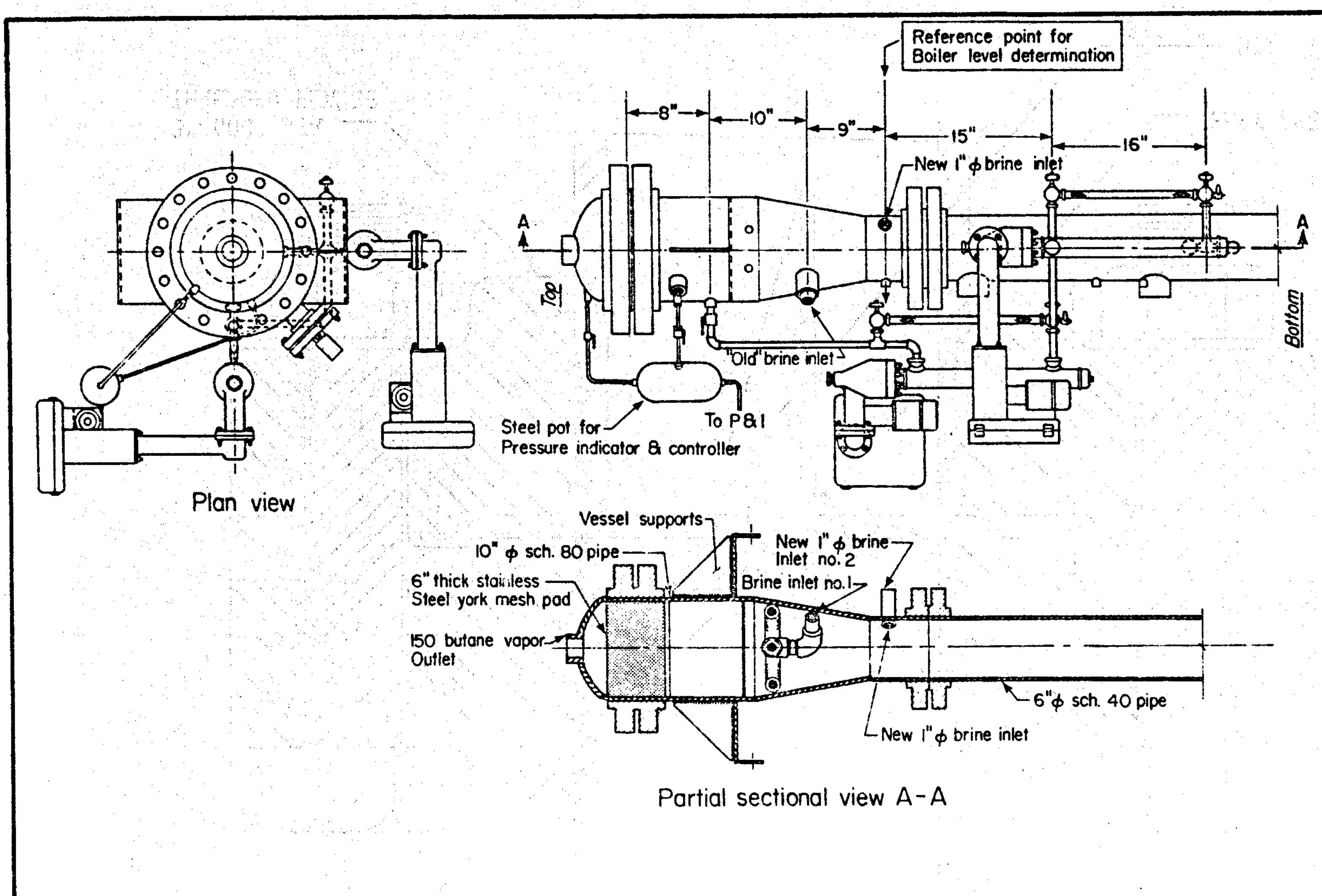


Figure 8. Dual level controller installation.

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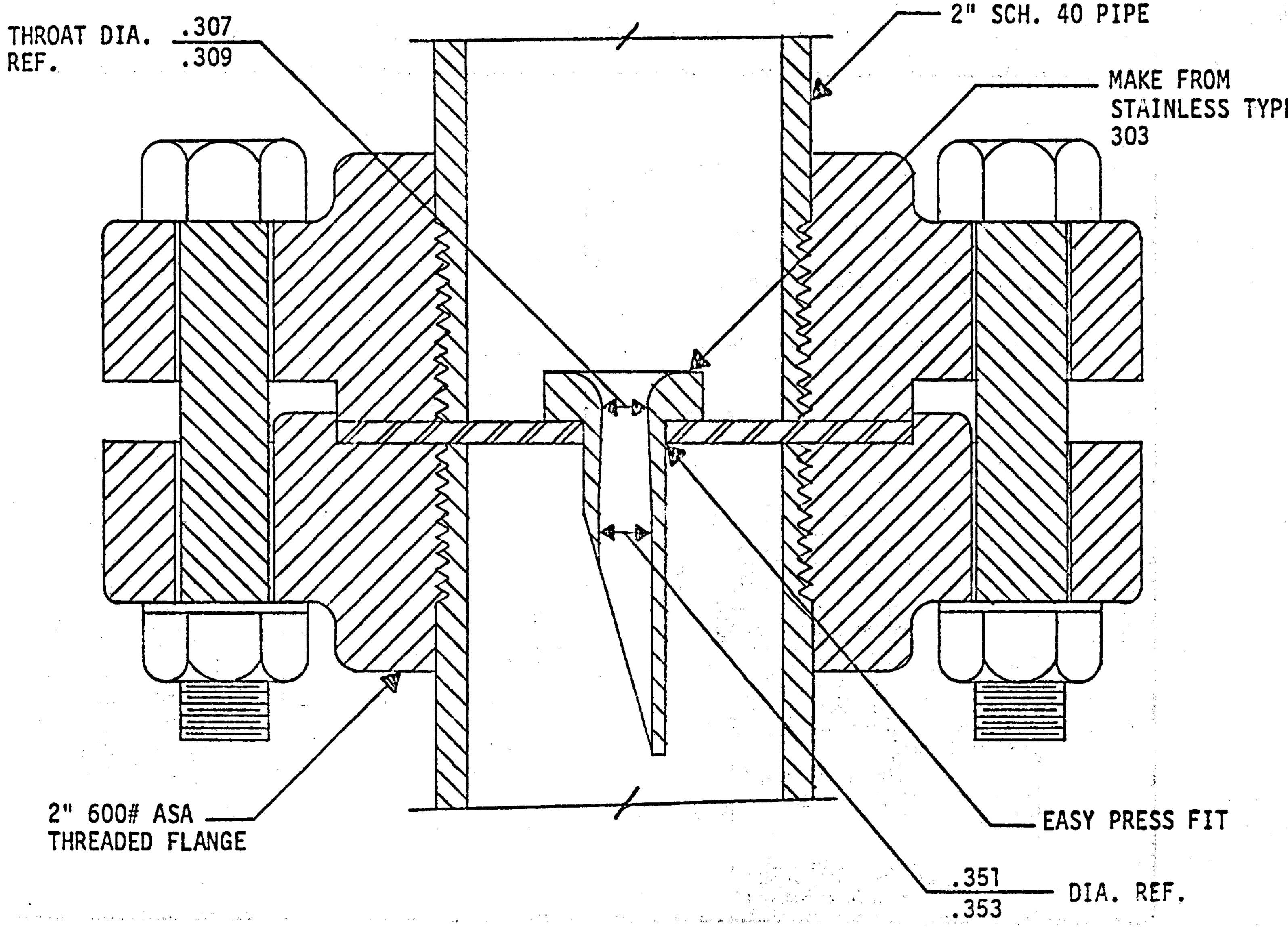

Figure 9. Dummy turbine nozzle assembly.

The calibration loop consisted of the direct contact heat exchanger loop with the DCHX replaced by a conventional tube-in-shell heat exchanger to keep the brine and isobutane separated. Subcritical and supercritical tests were run with both subcritical nozzles and with the supercritical nozzle. Data was obtained by varying degrees of superheat, isobutane flow rate, brine flow rate, and turbine pressure ratio. Several turbine speeds were run to identify and measure peak power output. Turbine output power and efficiency and nozzle flow coefficient were correlated with various conditions run. The effect of turbine pressure ratio on overall efficiency was also evaluated for each of the subcritical nozzles.

4.3 ENDURANCE RUNS

A total of three endurance runs were performed. The series 1 endurance run in July and August 1977 had the following goals 1) to obtain operating experience on a power producing loop utilizing an organic working fluid (isobutane) with some entrained geothermal brine, 2) to evaluate cycle efficiency, 3) to accumulate sufficient run time to identify and/or evaluate potential problems, and 4) to identify loop or fluid degradation.

The series 2 endurance runs were performed in March and April 1978. The primary goal of these endurance runs was to observe any difference in scaling of the turbine nozzle block, rotor, and exhaust housing with the moisture removed from the isobutane/water vapor in a demister vessel.

Operating parameters for the endurance runs were as shown in Table IV:

Table IV. Operating Parameters For Endurance Runs

	Series 1	Series 2
Brine Inlet Temp to DCHX	330±5^{o}F	322±3^{o}F
IC_4 Outlet Temp to Turbine	245±5^{o}F	230±5^{o}F
DCHX Operating Pressure	300±10psig	300±10psig
Brine Flow Rate	5.4±0.2 gpm	5.4±0.3 gpm
IC_4 Flow Rate	9.5±0.3 gpm	9.5±0.4 gpm
Turbine Speed	2750±5 rpm	2750±5 rpm
IC_4 Level in Hotwell	greater than 1/2 full	greater than 1/2 full
Condenser Pressure	95±10 psig	65±10 psig

These parameters resulted in:

Turbine Inlet temp	232±3°F	222±3°F
Turbine Inlet Pressure	300±10psig	300±20 psig
Turbine Exhaust Pressure	95±10 psig	65±10 psig

4.4 BRINE CARRYOVER EXPERIMENTS

Eighteen carryover tests were run in June, 1978. Tests were run with brine flowing into the DCHX through the "new" brine inlet. Then comparative tests were run for brine introduced through the "old" inlet. During these tests data were collected for various DCHX levels while holding the brine and isobutane flow rates constant for each set of levels observed.

5.0 TEST RESULTS

5.1 TURBINE CALIBRATION - CONVENTIONAL HEAT EXCHANGER

5.1.1 Subcritical Calibration

The subcritical calibration tests were performed to provide turbine performance data and nozzle calibration data on pure isobutane for later correlation with flow rates and performance of the direct contact heat exchanger. The calibration tests were performed with the hairpin heat exchanger.

Subcritical calibration tests were performed with two different nozzles (area ratios of 1.57 and 1.30) to evaluate the effect of area ratio on turbine performance. Figure 10 is a comparison of variation of turbine efficiency with pressure ratio for the two subcritical nozzles.

5.1.2 Supercritical Calibration

Calibration tests were performed with the supercritical nozzle utilizing the hairpin heat exchanger. The tests were designed to obtain performance data expanding through the saturated vapor dome and to use the calorimeter data to verify inlet enthalpy and turbine efficiency obtained from the test data. Turbine output horsepower is shown versus inlet enthalpy in figure 11. As may be seen, the variation in output horsepower with inlet enthalpy reasonably follows the predicted variation over the range of enthalpy examined.

5.2 ENDURANCE TESTS

Three separate endurance tests were performed. A 500 hour test was performed in July and August 1977 and a 200 hour and a 40 hour test were performed in March and April, 1978. Throughout the endurance tests the direct contact heat exchanger maintained the same level of performance. The

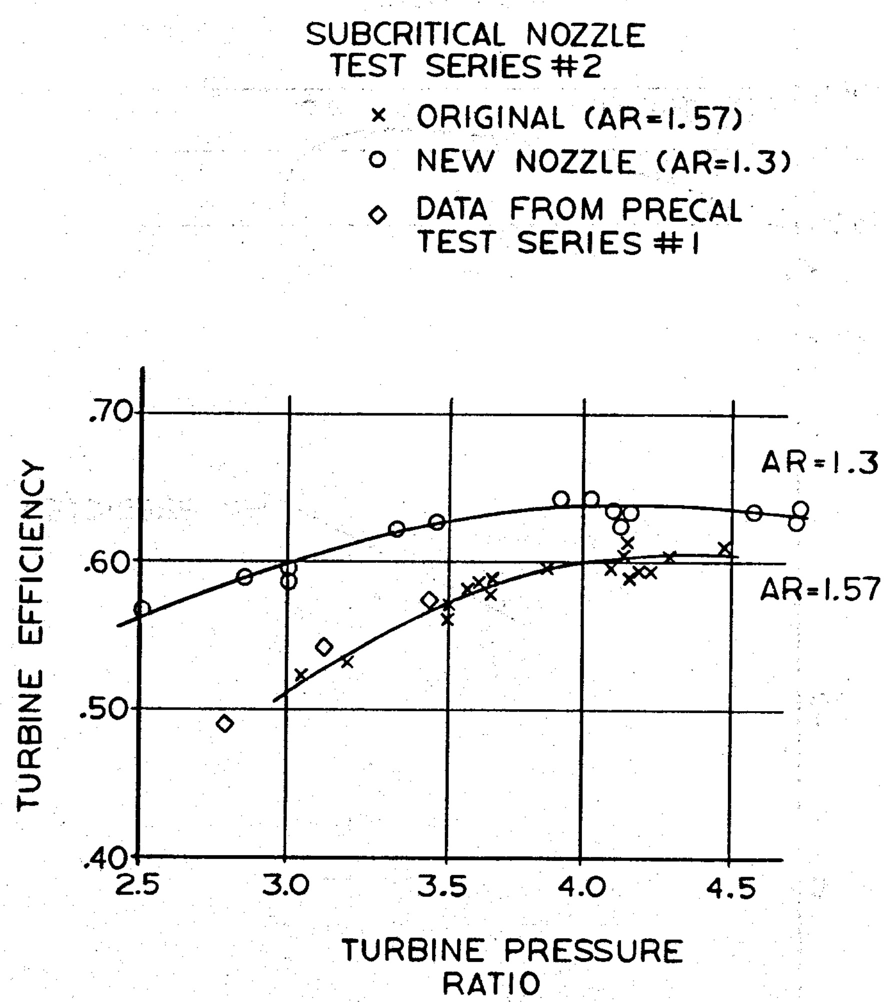

Figure 10. Pure IC_4 calibration test (using hairpin heat exchanger).

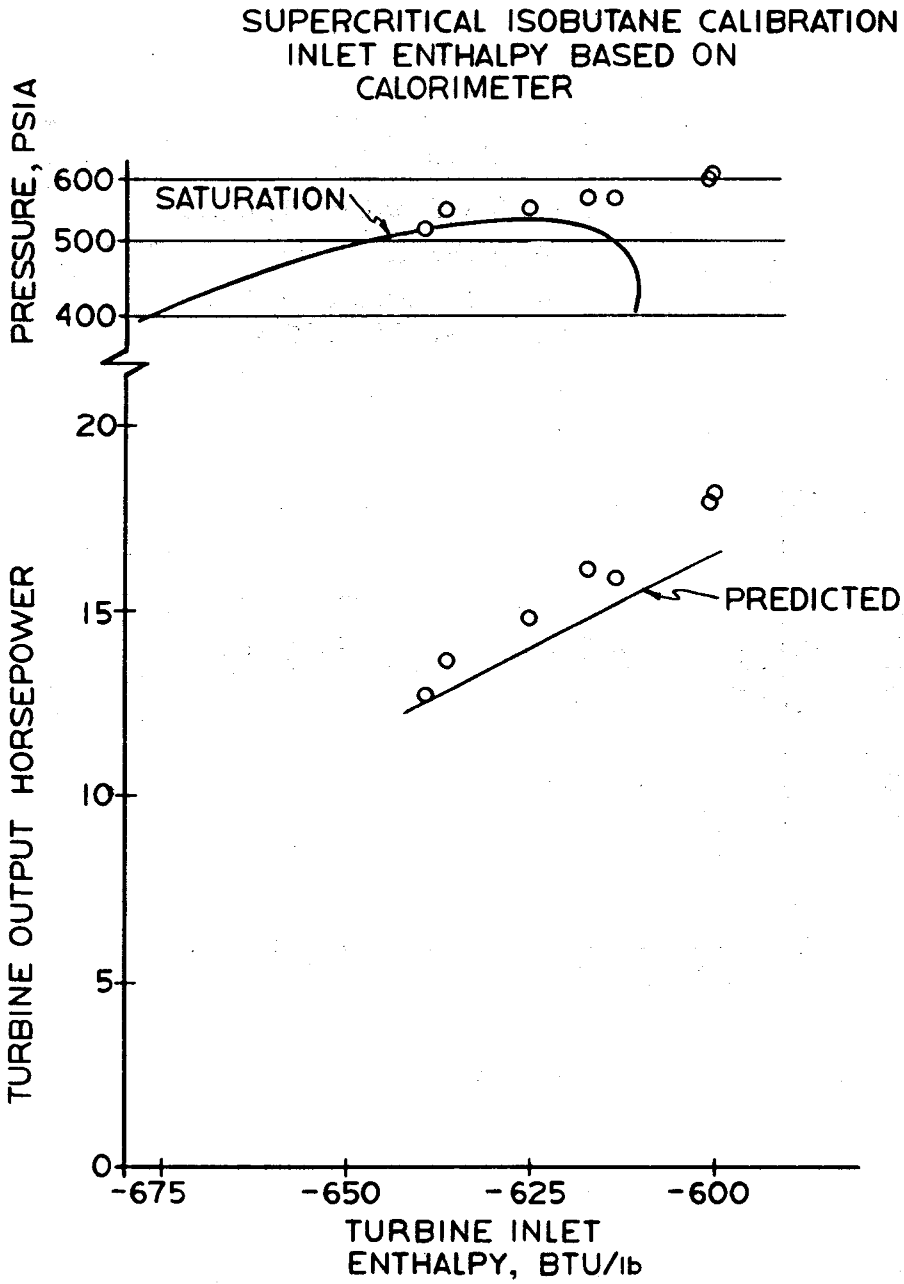

Figure 11. Turbine output vs. inlet enthalpy.

capacity to transfer heat was not affected by corrosion or scaling properties of the geothermal brine. Figure 12 shows a typical plot of temperature versus amount of heat added to the isobutane during the endurance runs. Any variations in the DCHX heat transfer performance were caused by variations in incoming brine temperature, brine flow rate, and isobutane flow rate.

Scaling properties of the brine did affect the performance of the turbine. Figure 13 illustrates the effect on turbine efficiency as a function of time. The turbine efficiency gradually changed from 58% at the beginning of the 200 hour endurance run to 41% at the end of the run. At the conclusion of the 200 hour endurance run, the nozzle and rotor were descaled and a 40 hour endurance test was performed. During the 40 hour endurance run the turbine efficiency averaged 54%. At the end of the 40 hour endurance run there was a thin layer of scale (approximately 0.002 inch) on the divergent section of the nozzle.

5.3 TEARDOWN INSPECTION

Teardown inspections of the turbine nozzle and rotor were conducted at various points during the endurance runs. Figures 14, 15 and 16 show the areas of scale buildup on the nozzle divergent section, rotor inlet and exhaust housing at the end of the 200 hour endurance run.

5.4 BRINE CARRYOVER EXPERIMENTS

As a result of the brine carryover experiments, no definite correlation between DCHX operating level and the amount or salinity of the water content of the isobutane vapor stream are apparent. The only significant result is that the high knockout drum water blowdown ratio measured during some of the test runs were accompanied by the highest measured salinity.

At the end of the testing period the "dummy" nozzle was removed and examined for scaling. It is estimated that the nozzle was intermittently in service for a total of 120 hours. The surfaces of the nozzle were found to be free of any significant scale deposits. Only a fine powdery coating which was later analyzed as carbon was evident.

5.5 ISOBUTANE LOSS MEASUREMENT

Attempts were made to measure the amount of isobutane lost due to entrainment in the brine returning to the reinjection pond. However, because a substantial amount of isobutane was lost through numerous leaks in the system due to worn pump packings, leaking pipe fittings, blown burst disks, leaking relief valves, etc., any estimate of isobutane lost to entrainment in the brine is purely speculative.

6.0 CONCLUSIONS

1) Stable, long-term operation of the direct contact heat exchanger coupled with the turbine was achieved in the test program.

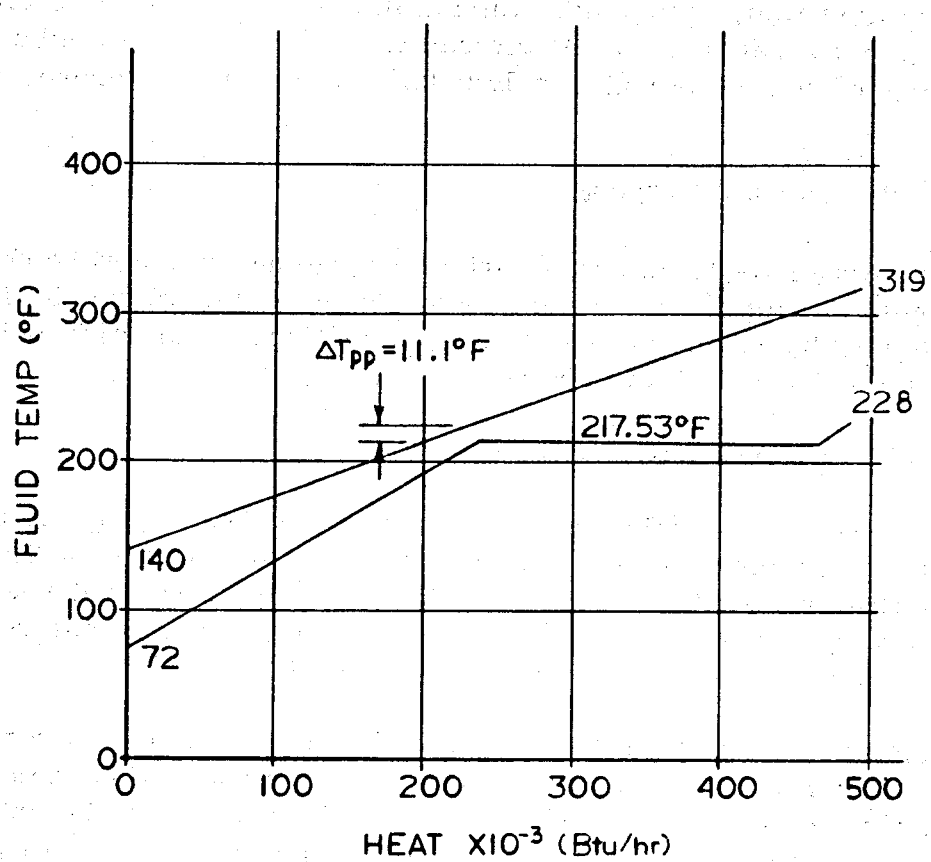

Figure 12. Boiler pinch-point calculation.

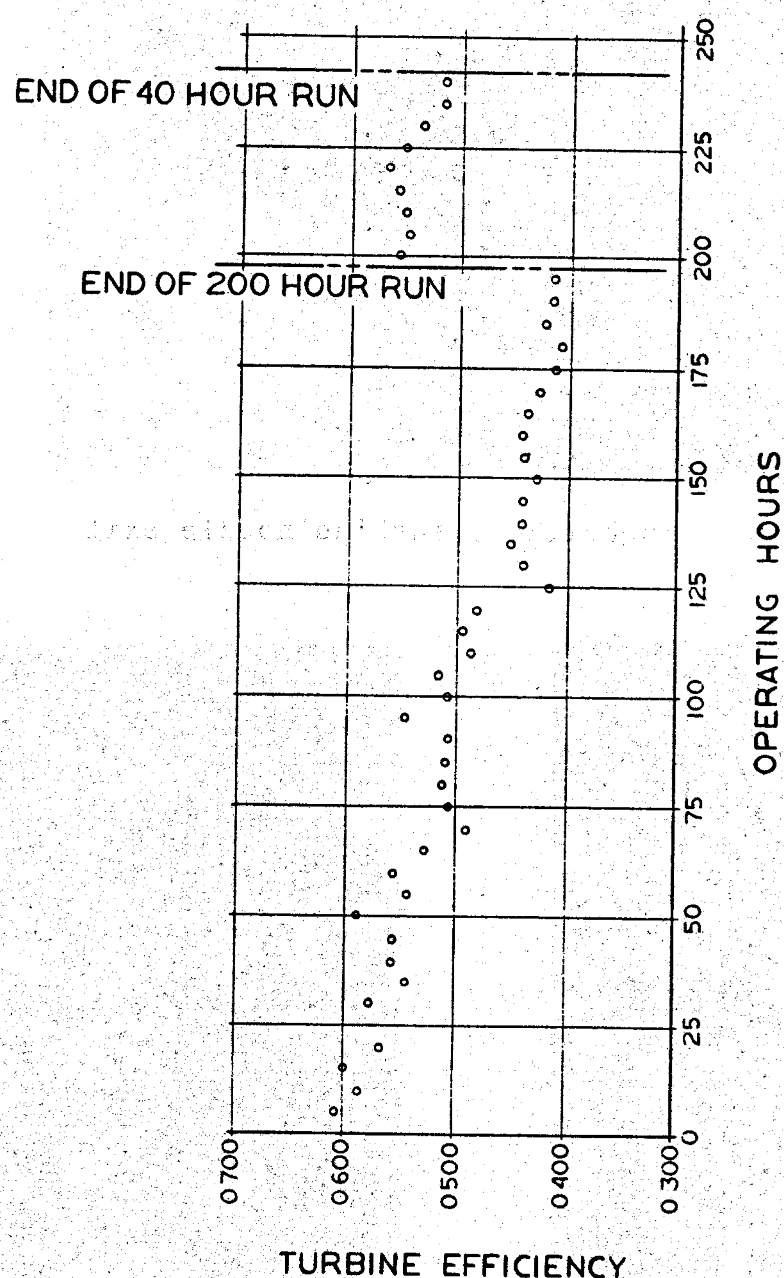

Figure 13. Turbine efficiency vs. operating hours.

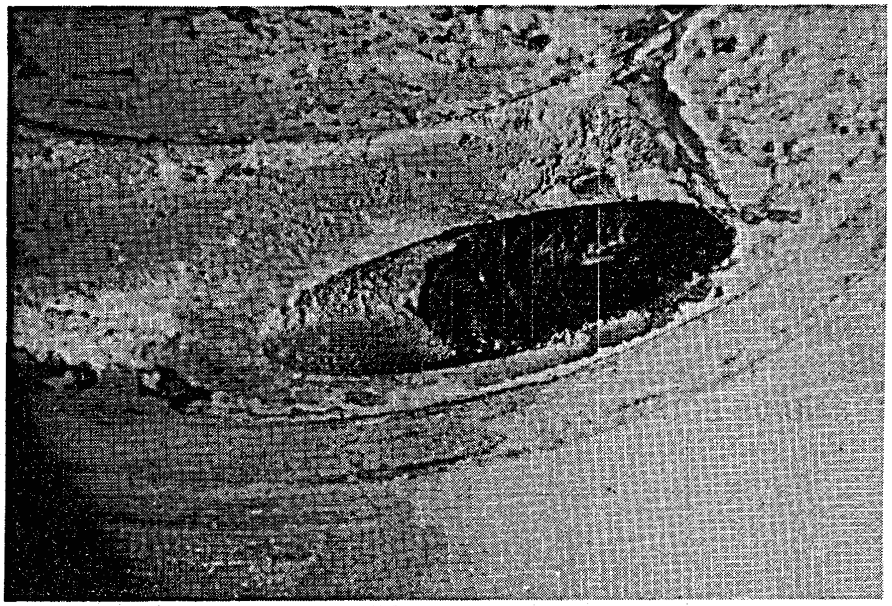

Figure 14. Subcritical turbine nozzle exit.

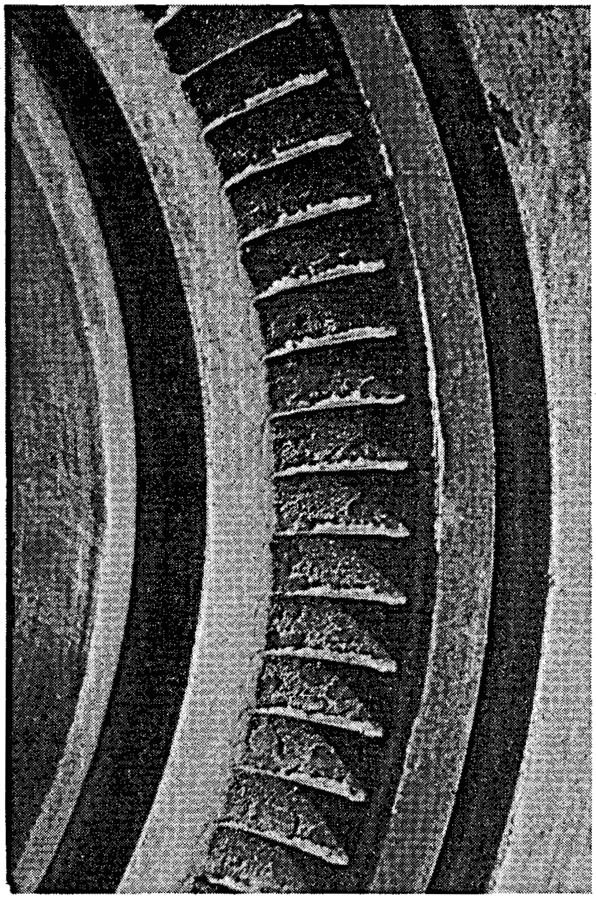

Fig. 15. Rotor inlet blade leading edge.

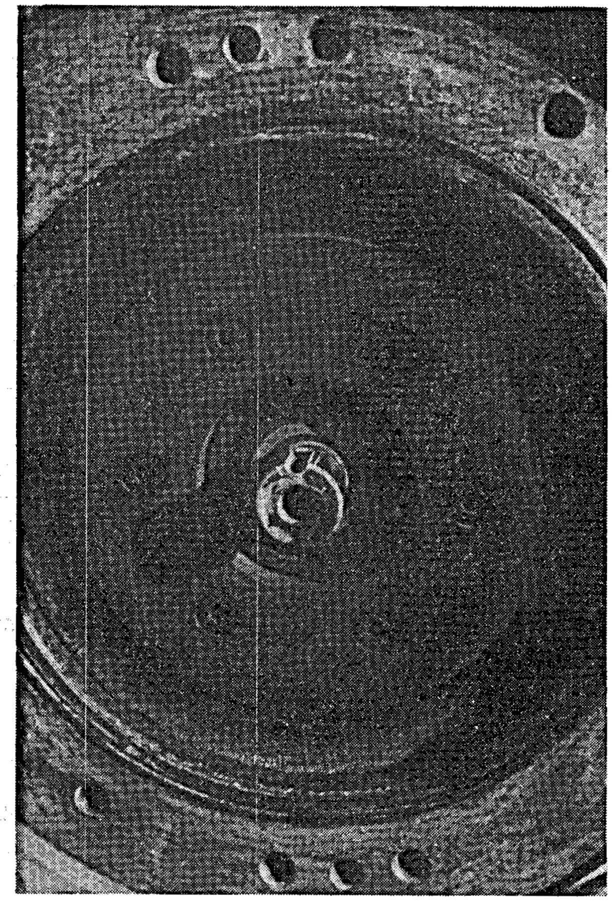

Fig. 16. Turbine exhaust housing.

2) Turbine efficiency measured during the testing verified performance predictions techniques and design methods for partial admission axial flow hydrocarbon turbines. The influence of entrained water in the working fluid vapor flow on turbine performance during direct contact heat exchanger testing was shown to be reasonably predictable.

3) Supercritical test results showed that turbine performance could be predicted with reasonable accuracy, however, turbine and cycle efficiency were reduced slightly by expanding inside the isobutane saturated vapor dome as compared to expanding to a superheat condition at the turbine exhaust.

4) Overall performance of the supercritical cycle was not as good as the subcritical cycle because of the additional pump work.

5) Nozzle scaling was found to occur during endurance operation with the DCHX. Although the demister removed large liquid droplets, some entrained mineral content was carried to the turbine resulting in a silica scale formation. No significant scaling was observed in the DCHX or in the connecting pipes.

6) When the DCHX brine inlet was lowered to the internal liquid interface, the scaling problem appeared to be significantly reduced.

7.0 REFERENCES

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SECTION 2

DIRECT CONTACT HEAT EXCHANGER
10 kW POWER LOOP TEST SERIES NO. 1

Prepared for:

Contract No. 2590502
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LIST OF FIGURES AND TABLES

1.0 INTRODUCTION

Extraction of thermal energy from geothermal brines has recently received significant attention. The present program was initiated in December of 1976 to couple a turbo-generator and controls to a direct contact heat exchange system to identify and evaluate problem areas in utilizing such a system for electrical power generation.

The turbo-generator consists of an axial flow, partial admission turbine, a gearbox, and a 60 KW, 3600 rpm, KATO generator. These components, along with controls and safety features, were skid mounted prior to shipment to the East Mesa Component Test Facility near Holtville, California.

The turbine was calibrated using a R. W. Holland shell and tube heat exchanger on pure isobutane at subcritical and supercritical turbine inlet cycle conditions. An endurance run of 500 hours was then completed using an existing direct contact heat exchanger designed and operated by DSS Engineers of Ft. Lauderdale, Florida. At the conclusion of this testing the turbine, connecting pipes, and a portion of the direct contact heat exchanger were disassembled and inspected for evidence of scaling or erosion in the test hardware.

This report presents the results obtained during these tests and is submitted to fulfill the work defined in LBL Purchase Order No. 2590502.

2.0 SUMMARY

The geothermal power loop test program successfully demonstrated the production of electricity from a geothermal source using isobutane in a direct contact heat exchanger. The tests were completed at the East Mesa Component Test Facility utilizing brine from well Mesa 6-2.

The test program involved laboratory check out tests on the skid mounted turbo-generator. During these tests the gearbox power loss was determined, the controls were checked out and the subcritical

nozzles were calibrated on air. The skid mounted assembly was then shipped to the East Mesa facility and installed adjacent to the DSS direct contact heat exchanger test loop. A conventional R. W. Holland shell and tube heat exchanger was also installed in parallel to the direct contact heat exchanger to allow calibration of the turbo-generator on pure isobutane.

Calibration tests were completed for subcritical and supercritical turbine inlet cycle conditions. Subcritical tests were completed at three flow rates and at degrees of superheat from about 10^{o}F to 60^{o}F. A number of turbine speeds were run at each test condition to identify and measure the peak output power. The nozzle flow coefficient calculated from the subcritical data was in all cases greater than unity. In general the nozzle flow coefficient decreased toward its laboratory measured value of 0.962 as the nozzle inlet pressure and apparent superheat increased to dry out the iosbutane. The maximum turbine output power (17.0 hp) also approached the predicted value of 17.6 hp at increased superheat as the nozzle flow coefficient approached its laboratory measured value. The measured turbine efficiency for this data was 57% compared to a design value of 59% for the subcritical cycle conditions.

The turbine contained two different nozzle configurations designed for subcritical and supercritical cycle conditions. The subcritical configuration consisted of two nozzles with an arc of admission of 45^{o}, and the supercritical configuration was a single nozzle with an arc of admission of 26.1^{o}.

An interesting result obtained with the supercritical calibration occurred as the turbine inlet conditions were varied. Considering the saturated vapor dome on a P-H diagram, as the turbine inlet condition moves to the left above the dome an isentropic turbine expansion results in greater turbine exit wetness, and reduced available energy compared to expansion into the superheat region. One would expect a significant reduction in output power under these conditions. The actual turbine output power (14.3 hp corresponding to a turbine efficiency of 52%) agreed with the predicted output at the right-most extremity of the dome. Moving to the left above the dome the turbine output remained higher than predicted. The reasoning behind this behavior is not clear, although compressibility effects and fluid properties just above the dome are difficult to predict. Additional measurement of total heat input to the cycle is needed to improve evaluation of this effect.

Turbine calibration tests with the direct contact heat exchanger were performed at the start and the end of the 500-hour endurance run. As with the previous calibration data the measured performance was below the predicted performance and approached the predicted as the amount of superheat increased. Measured nozzle flow coefficient operating with the DCHX was in the range of 1.11.

During the 500-hour test the measured turbine efficiency gradually changed from an average value of 49% at the beginning of the run to 39% at the end of the endurance run. The turbine horsepower changed from 10.75 hp to 7 hp. The loss in efficiency was due to a scale build up in the turbine nozzles, particularly in the area of the supersonic expansion cone. Analysis of the turbine nozzle scale showed primarily a silicate constituent, which would be expected from the brine from well Mesa 6-2. This is considered further evidence of a liquid carryover occurring from the DCHX.

The primary test components of the direct contact power loop performed quite well during the 500-hour endurance test. The heat exchanger loop was originally configured for short term testing and contained some components not suitable for continuous operation. Problems were consequently encountered with isobutane and brine pump packings and certain fluid loop controls. The turbine shaft seal began leaking about 341 hours into the endurance run and was replaced. Otherwise, the turbo-generator and controls performed entirely as expected during the endurance test. No problem with heat transfer or stability attributed to the direct contact heat exchanger was observed during the test program.

Brine samples taken at the outlet of the direct contact heat exchanger were analyzed by the chemistry laboratory at the East Mesa Test Facility. Samples were evaluated to measure the concentration of isobutane in the brine and to estimate the isobutane loss.

Results of these tests showed an average of 87 ppm isobutane fraction in the latter stages of the endurance run. This quantity amounts to only 25% of the equilibrium solubility (based on isobutane solubility) reported in Ref. 1 expected for the brine from well Mesa 6-2. The equilibrium solubility was taken for a brine having a salinity of 0% by weight TDS. An attempt was made to verify the above figure by measuring the quantities of isobutane added to the loop during the endurance run. Significant leakage occurring in the pump packing and turbine seal, however, invalidated the measure of isobutane loss fraction in this manner.

Following the endurance run the turbine, connecting pipes, and a portion of the direct contact heat exchanger were disassembled and inspected for evidence of scaling or erosion in the test hardware. The only scale deposition noted was that mentioned previously as being in the turbine nozzles. The remainder of the loop was in good condition and gave no evidence of distress which would prevent continued operation of the test loop.

3.0 POWER EQUIPMENT MODULE

A partial list of equipment contained in the power equipment module is as follows:

1. Turbine, housings and connecting piping
2. Gearbox
3. Torque meter and readout
4. Generator
5. Speed control
6. Pressure, temperature and speed instrumentation
7. Safety controls and interlocks
8. Air cooled resistive load bank

Items 3 through 7 were located in a pressurized enclosure to circumvent any possibility of an explosive isobutane-air mixture in the vicinity of the generator. Item 8 was remotely located a safe distance from isobutane sources. Discussion of the safety features contained in the power equipment module is contained in Section 3.3 of this report. This equipment was skid mounted for ease in transportation and was installed next to the direct contact heat exchanger test loop at the East Mesa Facility.

Each of these primary components are discussed in turn in the following sections.

3.1 DESIGN PARAMETERS

3.1.1 Turbine

During the design phase it was anticipated that the turbogenerator would be operated with two basic direct contact heat exchange systems. The first, and the subject of this report, was a subcritical system fabricated and tested by DSS Engineers. The second was a supercritical system from Occidental Research Corporation. Ref. 2, 3.

Based on discussions with DSS and Occidental, the system operating parameters were defined as shown in the following table. The anticipated water carry over fraction shown in the table was calculated assuming saturated water vapor and real gas properties of the isobutane at the discharge of the vaporizer. Turbine available energy was calculated based on an isentropic expansion of the vapor mixture. Turbine design conditions were then determined for the respective cycles.

TABLE I

	Occidental (Supercritical)	DSS (Subcritical)
Isobutane flow rate, lb/hr	2000	2650
Water vapor flow, lb/hr	50	32
Combined flow rate, lb/hr	2050	2682
Max. total pressure, psia	667	315
Max. temperature, ^{o}F	300	220
Condensing temp., ^{o}F	110	110
Total condensing pressure, psia	83.2	83.2
Turbine available energy, Btu/lb	36.2	23.7

An axial flow turbine with a wide latitude of flow through-put and power output was selected and designed specifically for this application. Based on the fluid conditions above a partial admission turbine design would yield reasonable efficiency and would allow excellent growth potential or application to different resources.

The partial admission impulse design incorporated a single rotor and a nozzle block with two nozzle geometries. One nozzle configuration was designed for the subcritical cycle. The desired nozzle configuration is selected by rotating the nozzle block so that the inlet flange of the desired nozzles lines up with the inlet piping. The inlet piping contains a short length of flexible metal hose to reduce piping loads on the turbine gearbox.

A summary of detailed turbine design parameters is shown in Table II. The nozzle throat and exit areas were calculated using the gas properties for isobutane presented in "Fluid Thermodynamic Properties for Light Petroleum Systems" by Kenneth E. Starling (Ref. 4). The analytical approach for the detailed nozzle design involved calculating the mass averaged density and velocity of the isobutane (IC4) and water mixture assuming an isentropic expansion from the specified inlet conditions. The nozzle throat pressure was determined to maximize the product of density and velocity which, for a given mass flow rate, uniquely determines a nozzle throat area. The nozzle throat pressures for the subcritical and supercritical cycles are estimated to be 212 psia and 410 psia, respectively. The above design approach has been verified in a number of similar organic fluid turbine nozzles previously designed and tested by Barber-Nichols.

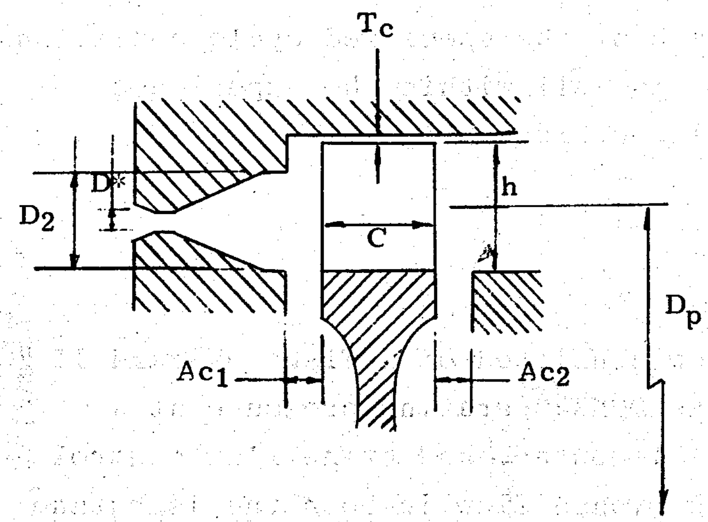
3.1.2 Gearbox

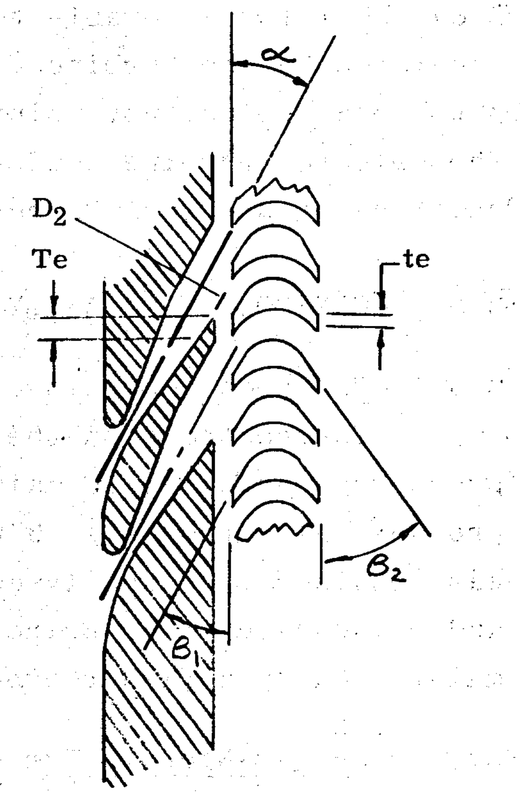
A standard double reduction Barber-Nichols gearbox (GR=6.05) was used in the power equipment module. The gearbox used rolling element antifriction ball bearings to minimize the parasitic power losses of the gearbox and to allow improved evaluation of the turbine output power. Lubrication was provided by an internal chain driven lube pump, and oil temperature was controlled by a water cooled heat exchanger mounted on the gearbox housing.

A face seal on the high speed shaft was used to seal the isobutane contained in the turbine exhaust housing. A nominal design face loading of 9.7 lbs. was estimated at a turbine exhaust pressure of 83 psia. The seal was designed to increase the face loading to about 11.9 lbs. at 110 psia.

TABLE II

TURBINE DESIGN PARAMETERS FOR DIRECT CONTACT POWER LOOP TURBINE





1.	Pitch diameter, D_p	5.37 in.
2.	Blade chord, C	0.32 in.
3.	Blade height, h	.404 in.
4.	No. of blades, Z	102
5.	Blade entrance angle, β_1	32°
6.	Blade exit angle, β_2	32°
7.	No. of nozzles, N	*
8.	Trailing rotor edge width, te	.010 in.
9.	Throat diameter, D^*	*
10.	Exit diameter, D_2	*
11.	Nozzle area ratio, Ar	*
12.	Nozzle angle, α	*
13.	Edge thickness between nozzles, Te	*
14.	Arc of admission, ϵ_n	*
15.	Tip clearance, T_c	.015 in.
16.	Axial clearance between nozzle & wheel, A_{c1}	.020 in.
17.	Axial clearance between wheel & exhaust, A_{c2}	.030 in.
18.	Arc of exhaust, $\epsilon_{exh.}$	*
19.	Desigh pressure ratio, $PR_{des.}$	*

*NOZZLE DESIGN PARAMETERS

		Sub.	Super.
7.	No.-nozzles	2	1
9.	Throat dia.	.229 in	.199 in
10.	Exit dia.	.286 in	.337 in
11.	Area ratio	1.567	2.87
12.	Angle	16°	16°
13.	Edge thk.	.010 max	N/A
14.	Admission	45°	26.1°
18.	Exhaust	55°	55°
19.	Pres. ratio	3.79	7.21

3.2 PREDICTED PERFORMANCE

The predicted variation of turbine efficiency with speed is shown in Figure 1. Efficiency curves are presented for pure isobutane and IC4/water mixture at supercritical and subcritical conditions. Because of the allowable test speed range of the alternator (see Section 3.3) the turbine can be tested above and below the anticipated peak efficiency point for each of the specified cycle conditions. The selected turbine configuration is well within the experience envelope of previous Barber-Nichols turbine designs.

3.3 CONTROLS AND SAFETY

3.3.1 DSS Loop Controls

The controls on the DSS loop consisted of a Fisher Wizard II pressure controller to maintain the DCHX operating pressure at a pre-set value (normally 340 psi). Isobutane and brine flow control was obtained by manually adjusting bypass flow in both the isobutane and brine circuits. Brine level in the column was automatically maintained by a Fisher type 2500-2493 level controller.

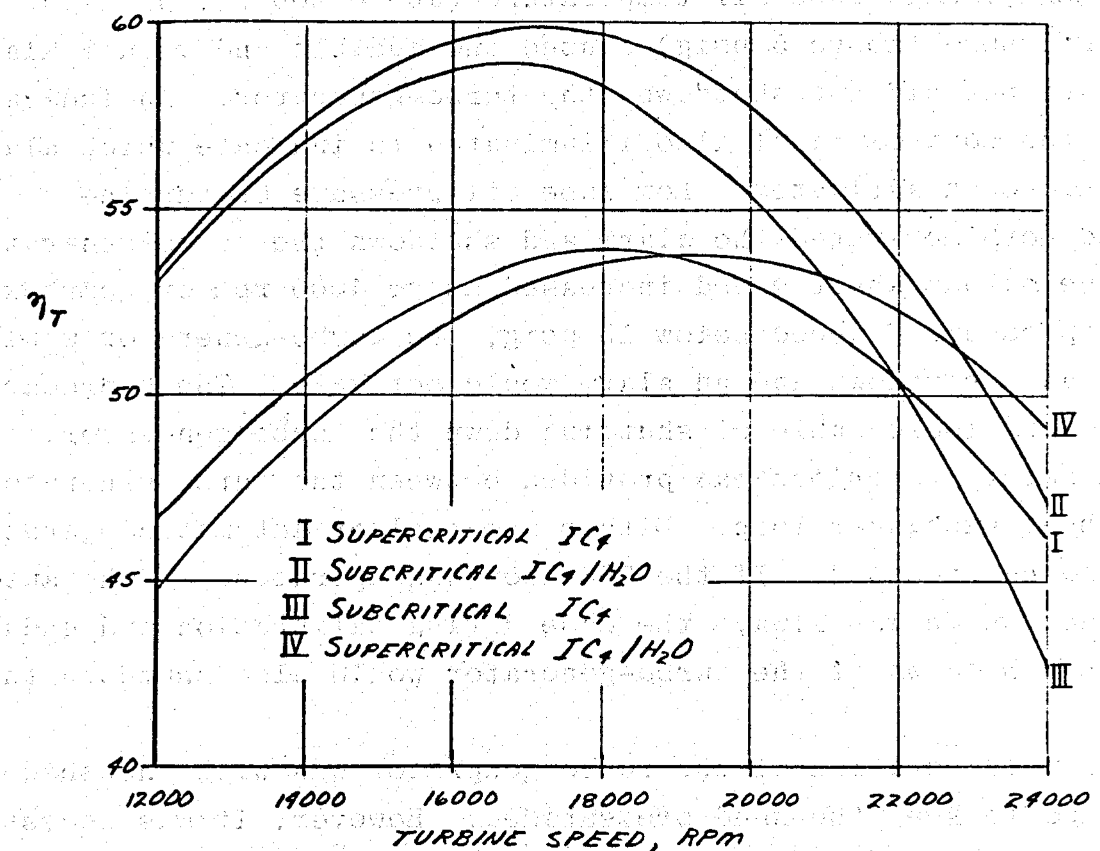
3.3.2 Barber-Nichols Turbo-Generator Controls

The controls for the turbo-generator consisted of a variable load speed control, overspeed switch, a gearbox lube oil high temperature switch, a lube oil low pressure switch, a shed high and low pressure switch, a gearbox high pressure switch, a turbine inlet shutoff valve, visual and audible alarms, and ancillary switches, relays and indicator lights.

The lube oil temperature switch and the overspeed switch were mounted on the turbo-gearbox and were, therefore, explosion proof. The warning alarm light was also explosion proof.

Since the generator and its speed control were not explosion proof they were mounted inside a pressurized enclosure to meet safety requirements. To reduce component cost, all other electrical components were also mounted inside the shed.

The control system featured warning alarms and an automatic shutdown mode. A high isobutane level outside the shed (greater



	Supercritical		Subcritical	
	IC_4/H_2O	IC_4	IC_4/H_2O	IC_4
Inlet Pressure, psia	667	600	315	300
Inlet Temperature, °R	760	760	680	680
Exhaust Pressure	83.2	83.2	83.2	83.2
Δ H', Btu/lb	36.2	33.3	23.7	22.1
Flow Rate, lb/hr	2050	2000	2682	2650
Turbine Design, hp	15.5	13.6	14.9	13.5

Figure 1. Variation of turbine efficiency with speed.

than 50% LEL), high lube oil temperature (above 200°F), or high gearbox pressure (above 5 psig) caused the audible and visual alarms to activate but did not shutdown the turbo-generator. An indicator light on the control panel also illuminated to indicate which alarm circuit had been activated. Low lube oil pressure or turbine overspeed both activated the alarm and shutdown the turbo-generator. If turbine output shaft speed increased above 4000 rpm or gearbox lube oil pressure dropped below 15 psig, the turbo-generator would automatically shutdown and an alarm would activate. The hydrocarbon detector was also capable of shutting down the turbo-generator.

A safety interlock was provided between the turbo-generator and the heat exchanger loop. Either a manual or automatic operating mode could be selected. If the DSS loop was operating in the automatic mode, which was always the case during calibration and endurance runs, shutdown of the turbo-generator would also shutdown the DSS loop.

Normal operation of the turbo-generator was with the shed door closed to keep the shed pressurized. However, it was necessary to open the door periodically to add water to the shed cooler. To prevent the audible alarm from sounding when the shed overpressure dropped, a key operated switch was available to turn off the alarm buzzer. If any other fault occurred while the shed door was open, a relay contact across the key switch would close to reactivate the buzzer. The buzzer could then be shut off again by depressing a push button. The alarm light would stay on in both cases until the fault was corrected and the reset button was depressed. Figure 2 shows the control system schematic and Appendix A contains a more detailed description of the control logic.

In addition to the turbo-generator controls discussed above, the generator power package used a speed control consisting of a Woodward model 2301-8271-347 governor, a Vectrol VPAC 5106-480-35E-LSER power controller and a load bank consisting of two Singer 34702T electric space heaters. The governor was an electronic unit with isochronous capability that measured shaft speed and provided a voltage output that increased or decreased as the speed attempted to move from the set point. The output voltage was used to modulate

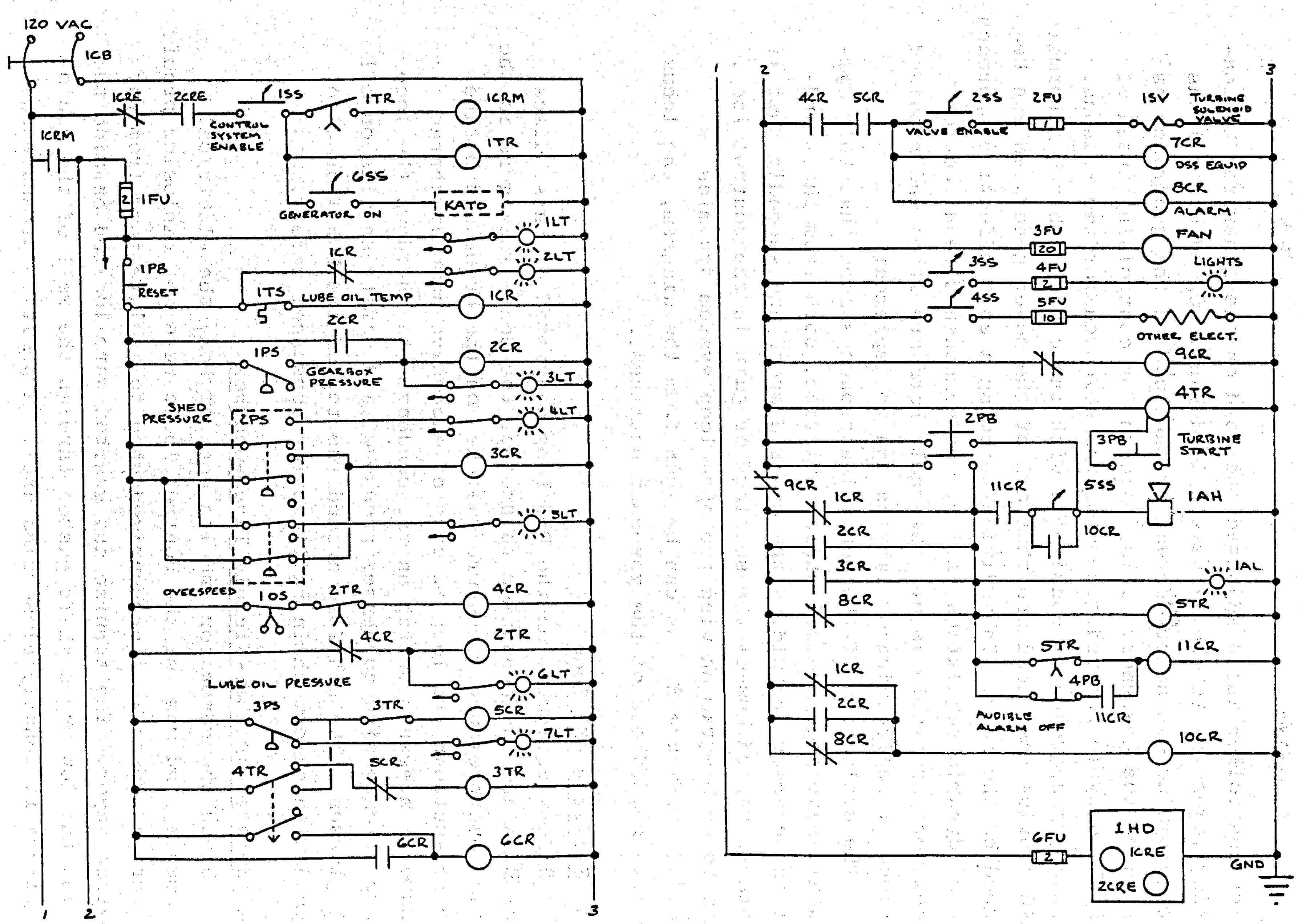


Figure 2. Control system schematic.

a power controller which adjusted the load on the generator by controlling power fed to the parasitic load bank. Speed control was achieved by increasing or decreasing the load on the generator as dictated by the governor. This approach for controlling speed of the Rankine cycle powered generator was selected for two reasons. This loading approach is consistent with the concept that geothermal should provide base load and that all of the power generated can be absorbed by the grid. In addition to being a cost effective approach for good dynamic regulation, all of the components required for control were available as off-the-shelf hardware items and required a minimum of engineering or fabrication to implement.

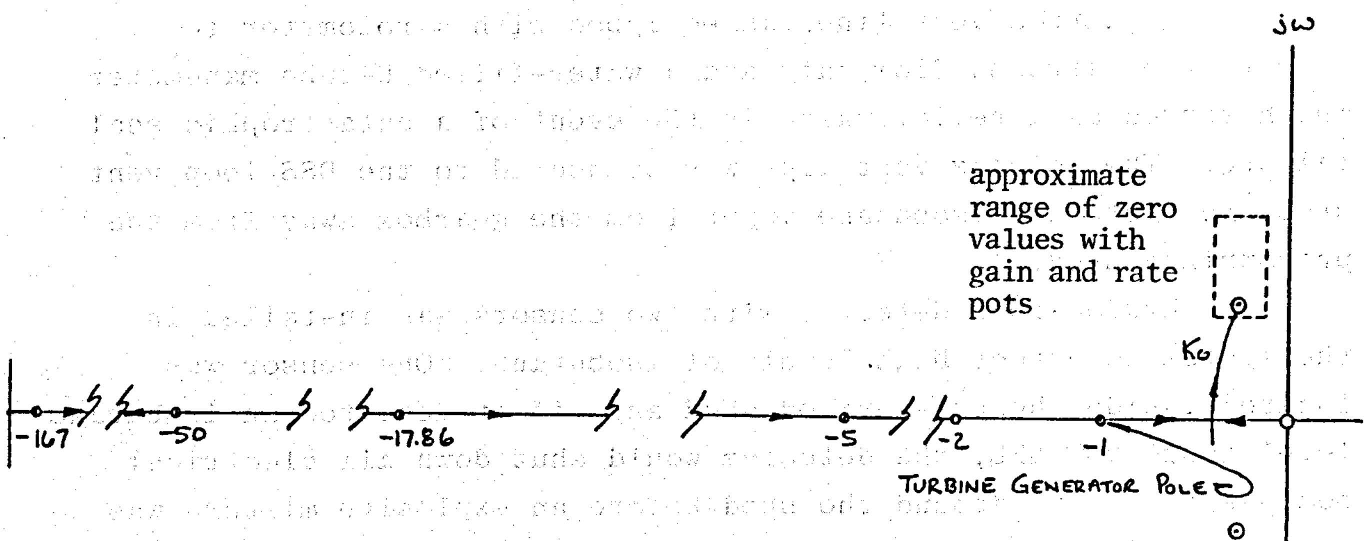
The evaluation of the control approach showed that this system would have good control characteristics, both from the standpoint of accuracy in holding the desired speed and secondly that it was stable regardless of the settings or gain in the control loop. A Root Locus Diagram showing the open loop system dynamics is presented in Figure 3. As can be seen from the diagram, the locus is stable regardless of the governor settings.

3.3.3 Safety - DSS Loop

Safety equipment on the DSS Loop consisted of burst discs on the DCHX and the condenser to protect the heat exchangers in the event of overpressure, pressure switches on the brine and butane pumps to shutdown the loop in case of excessive pump discharge pressure, a pressure switch on the condenser to shutdown the loop in case of excessive condenser pressure, and safety shields on all sight glasses. The platform around the top of the DCHX was also equipped with safety chain railings for personnel safety.

3.3.4 Safety - Barber-Nichols Turbo-Gearbox

Since the generator and its control box, the electrical instruments, and most of the control logic were not explosion proof, they were enclosed in a pressurized shed to isolate them from any isobutane vapor. The turbine and gearbox were located outside the shed and the low speed shaft passed through the wall of the enclosure with a reasonably close clearance. A positive pressure was



$$Gs = K_o \frac{(.2S+1)(K_2 + K_1S + K_3S^2)}{S(.056S+1)(.04\beta S+1)(.0065S+1)(.823S+1)}$$

K_1, K_2, K_3 and β change with the gain and rate setting pots on the woodward governor

Figure 3. Stability plot of 60 kW turbine generator.

maintained inside the enclosure to keep isobutane vapor from entering. Since the turbo-gearbox was outside the pressurized shed and there was the possibility of isobutane vapor leaking through the turbine shaft face seal into the gearbox, the gearbox was continuously purged with commercial dry nitrogen at the rate of 2 standard cubic feet per hour. This flow rate was selected to keep the gearbox internal pressure below 5 psig.

The gearbox vent line was equipped with a rotometer to measure the nitrogen flow rate and a water-filled U-tube manometer which served as a relief valve in the event of a catastrophic seal failure. The gearbox vent line was connected to the DSS loop vent stack to route any isobutane vapor from the gearbox away from the pressurized shed.

A hydrocarbon detector with two sensors was installed in the system to detect high levels of isobutane. One sensor was located inside the pressurized shed and, if it detected an isobutane level above 20% LEL, the detector would shut down all electrical components in and around the shed before an explosive mixture was attained. The second sensor was located outside the shed next to the door to detect high isobutane levels outside the shed and warn personnel not to open the shed door if the isobutane concentration was high enough (above 50% LEL) to present a hazard. The hydrocarbon detector was equipped with a meter which read the highest isobutane level which was being sensed and with visual and audible alarms independent of the control system alarms to warn of a high isobutane level. The detector was also equipped with a failure relay to shutdown all electrical equipment if the detector either lost power or malfunctioned and with a steady audible alarm to warn of a malfunction. The high isobutane warning alarm was a pulsating alarm to distinguish it from the failure alarm.

3.4 LABORATORY TESTS

3.4.1 Gearbox Losses

For the gearbox loss test, the gearbox was driven by a hydraulic motor connected to its output shaft through a Morse gearbox and a Lebow torque meter (see Figure 4).

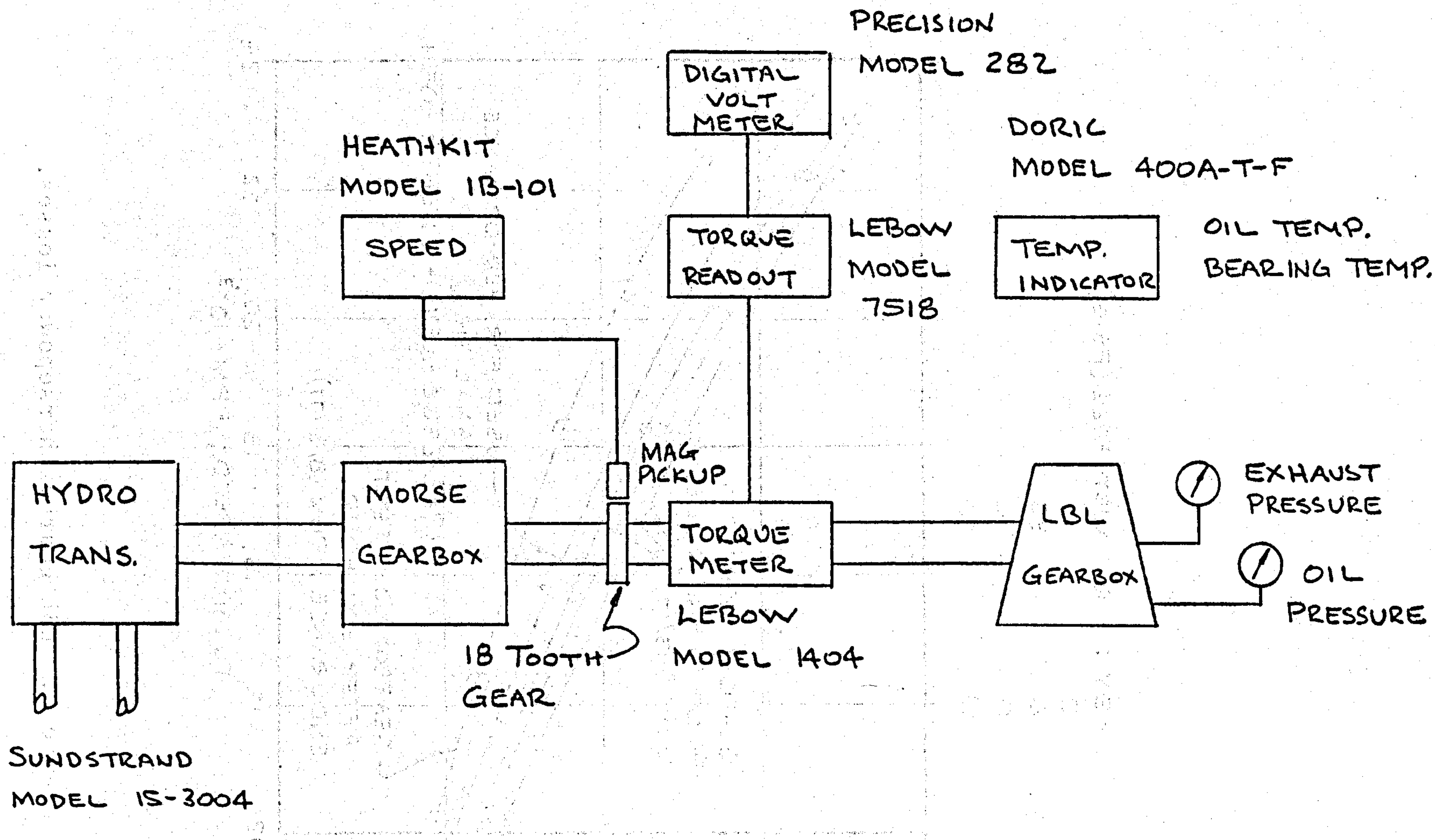


Figure 4. Lab test setup for gearbox losses.

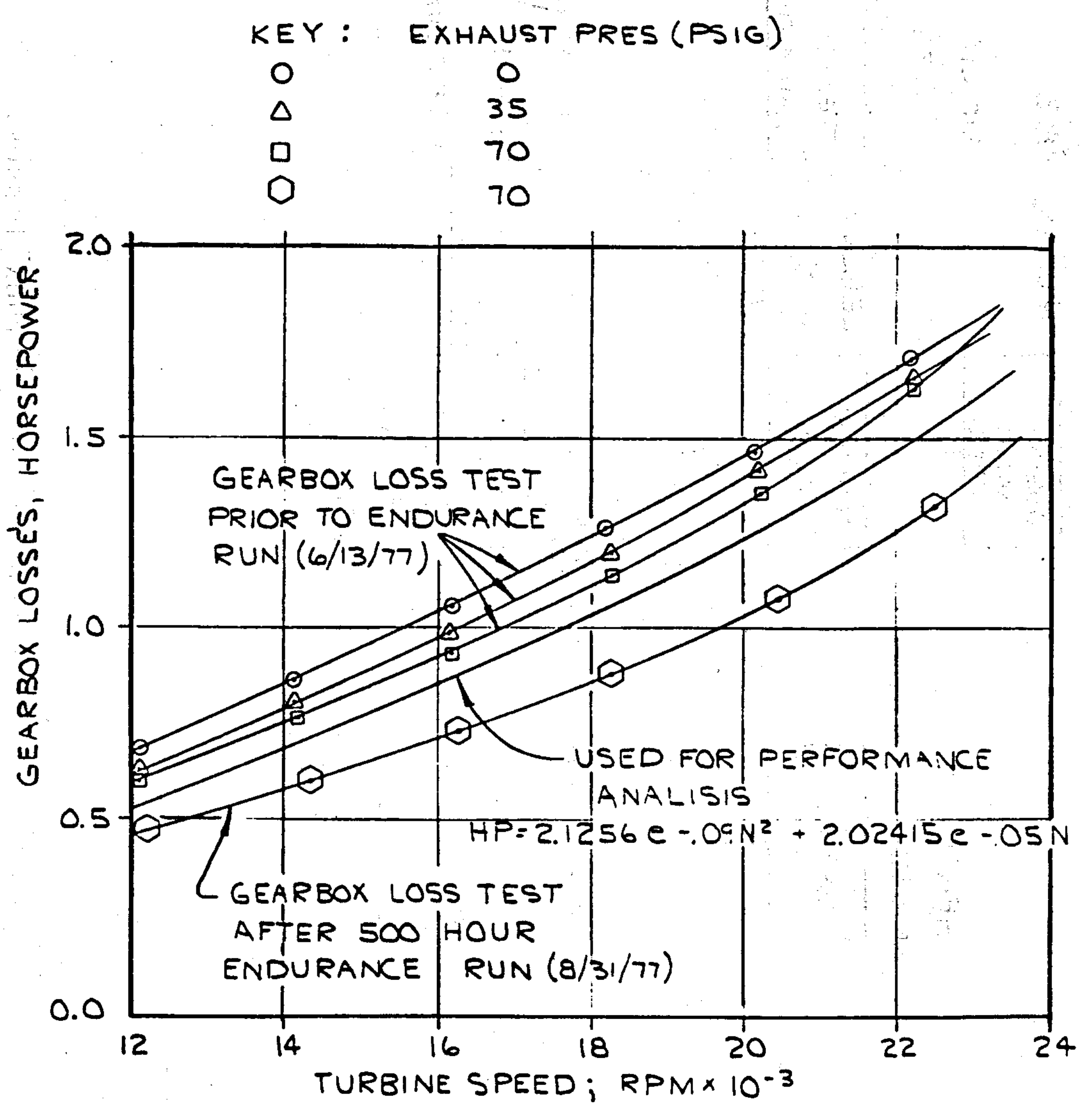


Figure 5. LBL gearbox horsepower losses.

The gearbox was driven over a range of output speeds and the lube oil temperature and turbine exhaust pressure were varied. The exhaust pressure was varied from 0 to 70 psig by pressurizing the turbine housing with dry air. The oil temperature was varied from $160^{o}F$ to $100^{o}F$ by cooling the oil in an oil-to-water heat exchanger. The hydraulic motor speed was varied to change the gearbox output shaft speed from 2000 to 3600 rpm. This corresponded to a turbine input shaft speed variation of 12,000 to 21,800 rpm. Gearbox loss tests were conducted both with and without the turbine rotor installed.

In addition to the above mentioned parameters, bearing temperature, lube oil pressure, exhaust temperature and output shaft torque were also measured. These parameters were used to calculate gearbox horsepower loss as a function of oil temperature, exhaust pressure and turbine shaft rpm. These data are presented graphically in Figure 5.

3.4.2 Air Tests

To verify proper operation of the complete system and to calibrate the turbine nozzle flow coefficient, the system was tested at Barber-Nichols. Air was used as the working fluid to minimize problems during this initial operation of the turbine. Figure 6 is a simplified plumbing schematic of the air test rig. Air was supplied from a screw type air compressor. The air was compressed, cooled, and the oil removed before the flow was measured using a sharp edge orifice. The compressed air then flowed through the turbine run valve and into the nozzles designed for the subcritical cycle conditions. The turbine pressure ratio was varied by adjusting the valve in the turbine exhaust duct.

The air flow data was reduced using the relations in the ASME Power Test Code, Chapter 4, Flow Measurement (1959 edition). The theoretical air flow through the nozzles was calculated using the ideal gas relations for flow through a choked nozzle. That calculation used the measured nozzle throat area and included the effect of water vapor, but neglected the effect of any entrained oil vapor or mist. The nozzle flow coefficient (or discharge coefficient) was then calculated by dividing the measured air flow by the air flow

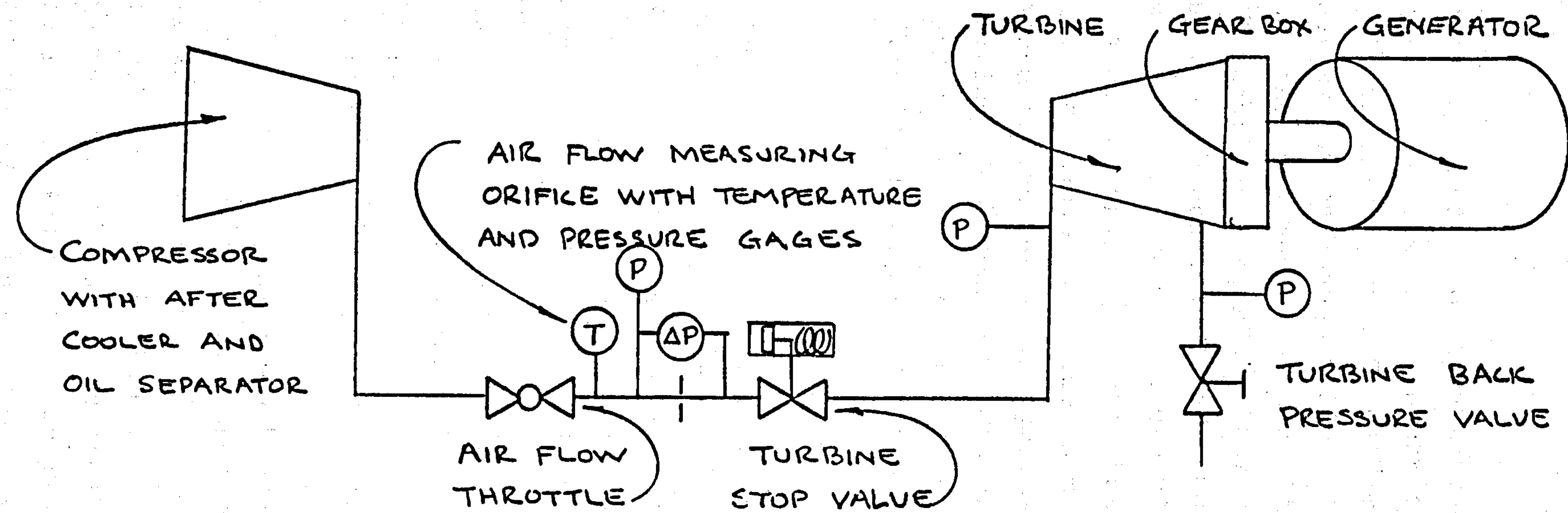


Figure 6. Turbine air test schematic.

predicted from the ideal gas relations. It should be pointed out that over the range of air pressures (120 psig maximum) and temperatures ($83^{\circ}F$ to $142^{\circ}F$) observed during the tests, air behaves like an ideal gas. However, during the isobutane tests, the turbine inlet conditions were near the vapor dome and real gas properties had to be used. The average measured subcritical nozzle flow coefficient was 0.962 which, based on our experience for similar nozzle designs, is very reasonable. Proper operation of the turbine controls and safety equipment was also verified during the air tests.

During the air tests it was determined that the control system would perform better if the generator exitation was changed from generator produced power to external line power.

4.0 LOOP CONFIGURATION

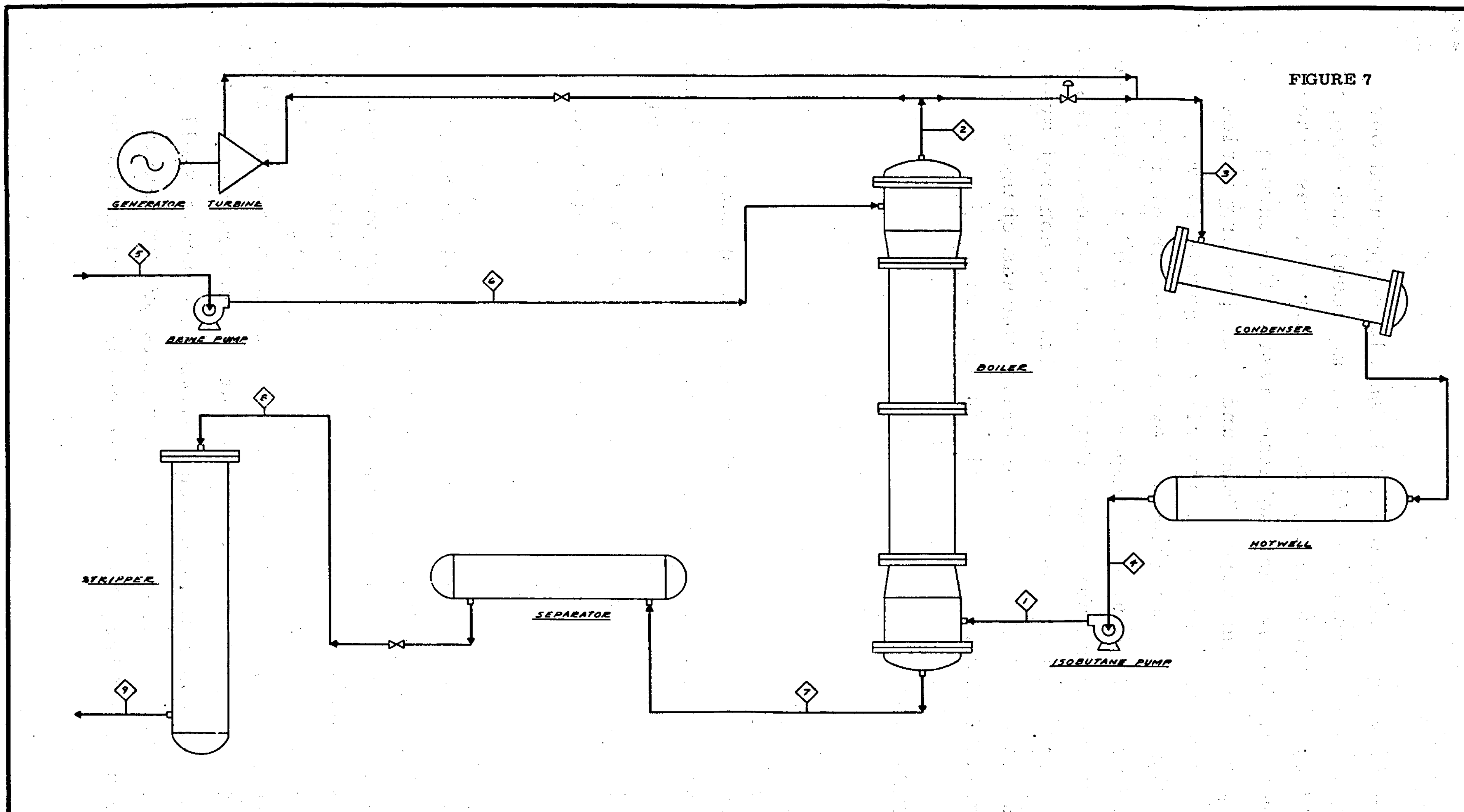
4.1 DIRECT CONTACT HEAT EXCHANGER TEST LOOP

4.1.1 Test Loop Components

The test loop consisted of the following major components: a direct contact heat exchanger, a combination preheater and boiler, an isobutane separator, a stripping column, a condenser, a hotwell, and two circulation pumps for isobutane and brine circulation. The general relationship of these components is illustrated in the process flow sheet and the P & I diagram.

4.1.2 Vessel Design

All test vessels were fabricated of carbon steel to Section VIII, Division I of the ASME Boiler and Pressure Vessel Code. Plate, standard weight pipe (sch. 40) and weld fittings were used for the shells. Nozzles were either schedule 80 pipe with 300# flanges or 3000# screwed couplings. Construction details are shown on the drawings in Appendix A. Overpressure protection was provided by safety valves installed behind stainless steel rupture discs. The condenser was a standard off-the-shelf item with brass shell and tubes, and a cast iron water box. Piping was predominantly carbon steel hydraulic tubing to facilitate bending and minimize



STREAM / PARAMETER		MATERIAL BALANCE — PROCESS STREAM NUMBER 1: ISOBUTANE FEED	2: ISOBUTANE VAPOR FROM BOILER	3: ISOBUTANE VAPOR TO CONDENSER	4: ISOBUTANE CONDENSATE FROM HOTWELL	5: GEOTHERMAL BRINE SUPPLY	6: BRINE FEED	7: BRINE FROM BOILER	8: BRINE FROM SEPARATOR	9: BRINE FROM STRIPPER
FLOW	LB/HR	2390	2390	2390	2390	2980	2980	2980	2980	2980
TEMPERATURE	°F	95	220	180	95	325	325	155	155	155
PRESSURE	PSIA	315	315	110	110	165	315	315	315	147
ENTHALPY	BTU/LB.	-787	[illegible]	-616	-787	295.63	295.63	122.96	122.96	122.96
DENSITY	LB/FT³	33.61	[illegible]	1.03	33.61	56.48	56.48	61.09	61.09	61.09

DATE	REV.	DESCRIPTION
REVISIONS		

DSS engineers, inc.

APPROVALS: P.S.

PROCESS FLOW DIAGRAM DCHX FOR TURBINE TEST AT EAST MESA

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SHEET 1 OF 1

Figure 7. Process flow diagram DCHX for turbine test at East Mesa.

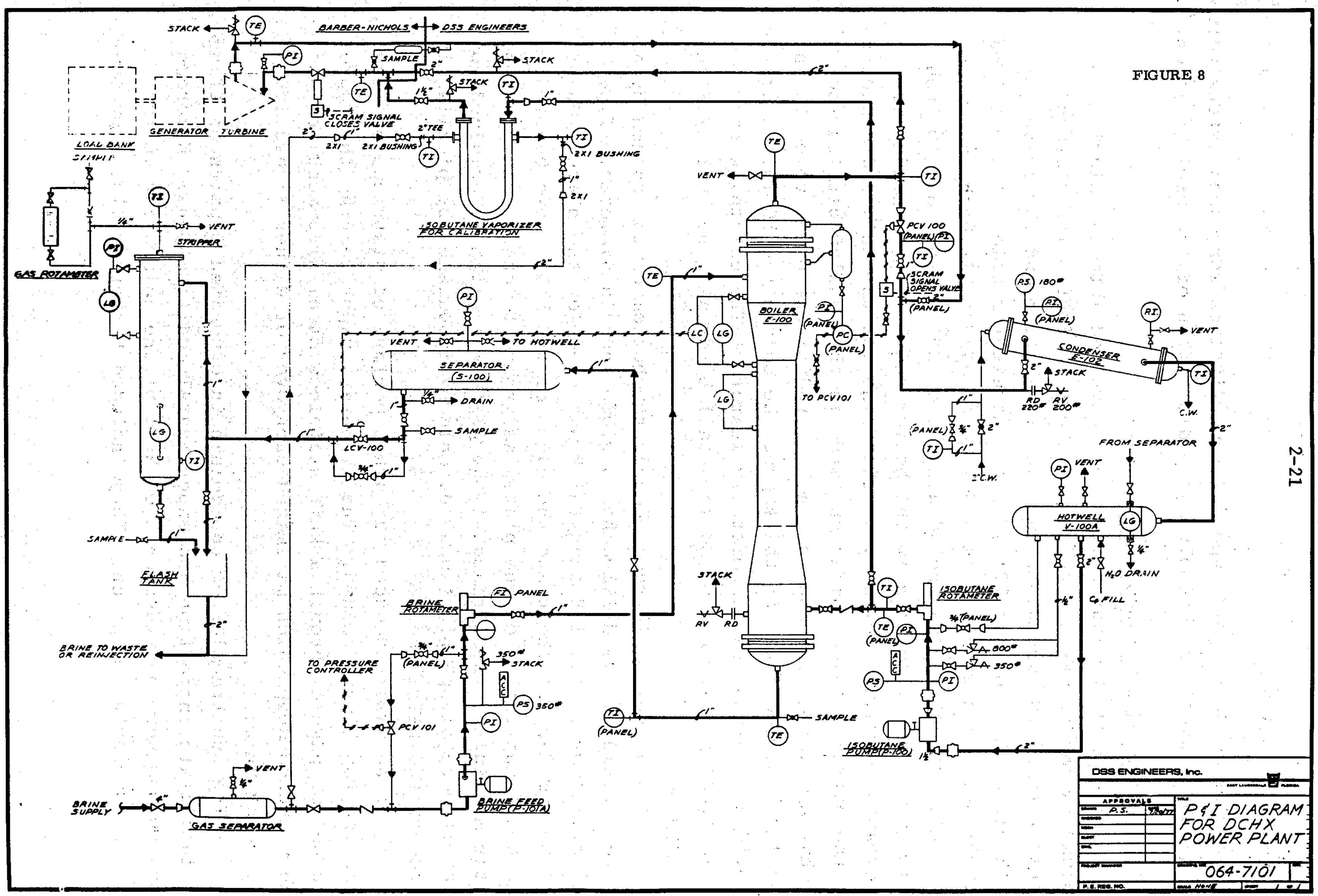

Figure 8. P & I diagram for DCHX power plant.

the number of joints. "Swagelok" type fittings were used extensively to allow frequent disassembly and modification of the piping system.

4.1.3 Direct Contact Heat Exchanger

The preheater and boiler were combined into one single column having an overall height of about 13 feet. The distance between the isobutane distributor at the bottom and the brine distributor at the top was 10' 4". The straight section of the column was 9' in length with an inside diameter of 6". At both ends a conical section with an included angle of 20^{o} flared to a 10 inch diameter. The transition at the bottom resulted in a gradually decreasing downward velocity of the brine and the enlarged diameter at the bottom allowed sufficient area in the annulus between the isobutane distributor and the wall to reduce the brine velocity in the straight section. At the top of the column the 10 inch diameter provided a large area for mist eliminators. The column is shown schematically in Figure 9 and in detail by drawing no. 051-2202 in the Appendix.

Isobutane was introduced near the bottom of the column by a plate containing 390 drilled holes 0.060 inch in diameter. Type 316 stainless steel, 1/8 inch thick was used for the plate. On the bottom face of the plate the edges of the holes were beveled while the upper face of the plate was ground to ensure sharp edges. After machining, the surface of the plate was slightly oxidized to make it more oleophobic (less wettable by isobutane) by pickling in concentrated nitric acid.

Brine was introduced through a perforated distribution ring near the top of the column. The isobutane vapor leaving the top of the column passed through a knit stainless steel wire mesh demister pad 6 inches deep which minimized carryover of either liquid isobutane or brine.

The construction of the column incorporated 5 pairs of flanges to facilitate inspection and experimental modifications. Numerous connections were provided on the shell of the column for vents, drains and instrumentation. They included 3 thermowells, 1 pressure

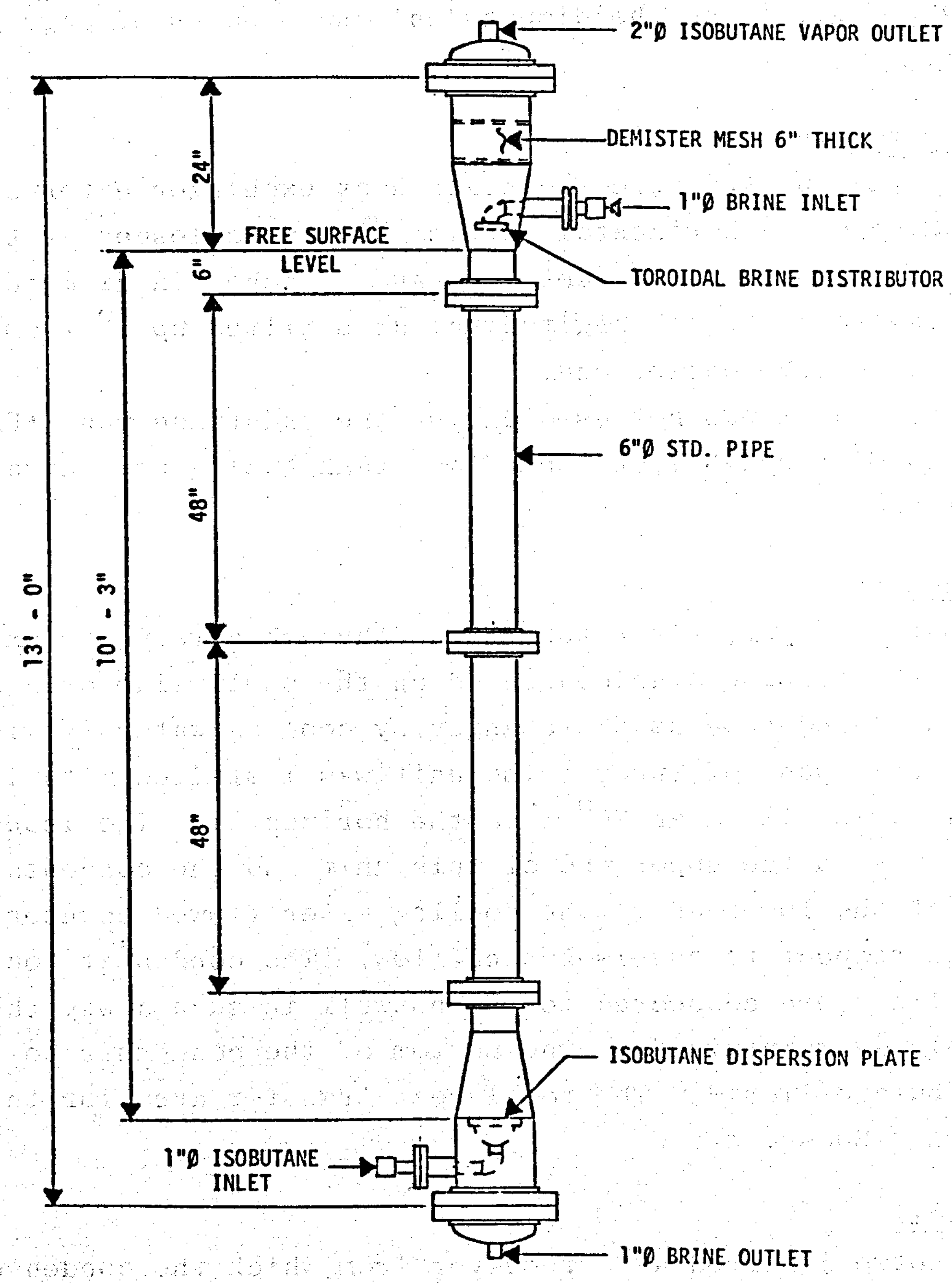

Figure 9. Direct contact heat exchanger

gage connection, 2 manometer taps, 4 high pressure windows (or "bulls-eyes") for viewing the dispersion, and 3 pairs of gage glass connections.

4.1.4 Separator

Brine leaving the direct contact heat exchanger entered the separator which was a horizontal settler with a coalescer at the inlet end. The vessel was 7 feet long and 8 inches in diameter, and was installed with its longitudinal axis tilted up 1^{o} from the horizontal toward the outlet end.

The separator was not used during the endurance run. Entering brine passed through it to the flash tank taking any entrained isobutane with it.

4.1.5 Condenser

Isobutane vapor, after throttling through a valve or expanding through the turbine, was condensed on the shell side of a conventional shell-and-tube heat exchanger by cooling water flowing through a single pass of tubes. The unit was installed with its longitudinal axis tilted up 20^{o} from the horizontal. The isobutane vapor inlet was at the upper end of this unit and the condensate outlet was at the lower end. The cooling water flowed counter current with respect to the isobutane flow. The condensate outlet and a vent line were connected to the hotwell in such a way that a liquid level was maintained in the bottom of the condenser to subcool isobutane liquid. The total heat transfer area for this condenser was 120 sq. ft.

4.1.6 Hotwell

The hotwell served as a receiver from which the condensed isobutane was pumped. As test unit operating conditions changed, the amounts of isobutane in the direct contact heat exchanger varied. The hotwell provided the inventory to cope with these operating changes as well as losses through leakage. Instrumentation on the hotwell included temperature and pressure indication and 1 gage glass. The vessel was 6 feet long and 10-5/8 inches in diameter.

It was installed with its longitudinal axis tilted up one inch from the horizontal toward the pump suction line. Therefore, the water condensate in the hotwell could be drained off at the lower end of the vessel. The hotwell is shown in detail in drawing 051-2208, Appendix A.

4.1.7 Stripping Column

The DSS loop was equipped with a stripping column to remove any additional isobutane from the brine leaving the separator. The stripper was isolated from the remainder of the DSS loop during the endurance run and, therefore, was not used.

4.1.8 Pumps

The isobutane was circulated by a 4 cylinder John Bean outside packed plunger pump (Model T-0410C) rated at 10 gpm capacity. The brine pump was an outside packed triplex pump (John Bean Model M-0910) with a rated capacity of 7 gpm. Both pumps were equipped with a nitrogen filled accumulator on the discharge side to reduce pressure pulsation. Flow rate was controlled by a valve on a bypass line from the discharge to suction side of the pump. A relief valve was provided on the discharge line to prevent overpressure damage to the pump or test unit.

Pumping requirements for the isobutane were especially demanding because of the combination of low available net positive suction head and high discharge head in a single pump. Isobutane liquid also had extremely low density and viscosity, placing further restrictions on the type of pump which could be used.

4.1.9 Modification to the Test Unit for Turbine Experiment

Several piping and control changes were made to the test unit to accommodate the turbine experiment. These changes were made so that the system could provide isobutane liquid at supercritical and subcritical pressures to the surface heat exchanger for calibration of the turbine.

Piping from the isobutane pump to the direct contact heat exchanger was provided with tees and block valves to direct isobutane to either the surface heat exchanger or to the direct contact heat exchanger. Since calibration with the surface heat exchanger was at both subcritical and supercritical conditions, two safety relief valves were provided on the pump discharge. Either safety valve could be selected: 650 psi for supercritical or 350 psi for subcritical.

Isobutane vapor piping was provided for the surface heat exchanger to the turbine inlet. The vapor line from the DCHX was also connected to the turbine inlet by means of appropriate isolating valves and tees. As a result of these changes, the mode of operation could be changed from calibration with the surface heat exchanger to operation with the DCHX without any piping changes.

A vapor line was provided between the turbine exhaust and the condenser. This line was tied into the original vapor line from the DCHX to the condenser. The original pressure control valve (PCV 100) between the DCHX and condenser remained operable to allow the turbine to shut down while the DCHX was still running. This provided a "short circuit" around the turbine directly to the condenser.

Separate brine supply and return pipes were installed to provide the surface heat exchanger with brine. The flow of brine to the heat exchanger was manually controlled.

All piping was hydrotested to 900 psi before testing began. Hot pipes were insulated with asbestos lagging.

An electrical relay was installed linking the turbine stop valve to the DCHX system. A signal closing the turbine stop valve also shut off the power to the DCHX pumps and opened the pressure control valve (PCV 100) between the boiler and condenser. This allowed the isobutane vapor in the DCHX to be vented to the condenser when the turbine inlet valve closed for any reason.

4.2 INSTRUMENTATION AND SAMPLING

4.2.1 Instrumentation - DSS Loop

The instrumentation on the DSS direct contact heat exchanger (DCHX) loop consisted of a multi-channel Leeds and Northrup strip chart recorder to record brine temperature into and out of the DCHX and isobutane temperature into and out of the DCHX. Other instrumentation consisted of pressure gages and thermometers located in the loop piping to monitor brine and butane temperatures and pressures at various locations, and two Brooks Instruments Division rotameters to measure brine and isobutane flow rates in gpm.

4.2.2 Instrumentation - Barber-Nichols Turbo-Generator

The turbo-generator instrumentation consisted of a Doric 412A-T-F Trendicator for displaying the turbine gearbox lube oil temperature, turbine bearing temperature, turbine inlet gas temperature, turbine exhaust temperature and generator shed temperature; a Newport Model 6110 electronic tachometer for displaying turbine output shaft speed in rpm; a Lebow Model 7525 digital indicator for displaying gearbox torque output; and turbine inlet pressure, exhaust pressure and oil pressure gages. Other instrumentation included a Dwyer Model 3000-00 shed pressure switch with a gage to indicate the amount of shed overpressure. The switch contained relays which would light a warning light and sound an alarm if the shed pressure dropped below 0.05 inches of water or rose above 0.15 inches of water (nominal was 0.10 inches of water).

The instruments were mounted on an aluminum plate and could be viewed through a plexi-glass window in the side of the generator shed. The window was hinged and the instrument panel edges were sealed to keep the shed pressurized when the window was open.

4.2.3 Sampling

The DSS loop was instrumented to obtain samples of isobutane flowing to the turbine inlet and samples of brine flowing from the DCHX to the separator. This instrumentation consisted of two 300 ml bombs. The isobutane vapor bomb was equipped with 350 psig gage. The brine bomb was connected to a tee on the brine outlet fitting of the DCHX. The bomb was mounted vertically and the line and bomb were purged for 5 minutes to equalize the temperature and bleed off excess gas before the sample was collected. The vapor bomb was connected in parallel with the turbine between the inlet and outlet pipe and was mounted horizontally. To collect a sample the bomb inlet valves were opened fully. The outlet valves were then opened until the bomb pressure dropped approximately 50 psi. The bomb was then purged for 5 minutes to equalize temperature and minimize condensation. The sample was then collected.

The samples were analyzed by the East Mesa chemistry lab to determine the amount of brine carryover to the turbine inlet and the amount of isobutane loss to the brine reinjection pond.

5.0 FIELD OPERATIONS

5.1 LOOP OPERATION

The test unit provided for the continuous circulation, contact, and separation of brine and isobutane. Heat from geothermal brine was transferred to the isobutane in a single direct contact heat exchanger.

Isobutane liquid was pumped from the hotwell to the direct contact heat exchanger where it was heated and vaporized by counter current contact with brine. The isobutane vapor and a small amount of water vapor was then throttled to the condensing pressure through the Barber-Nichols turbine. The vapor was condensed and the isobutane liquid returned to the hotwell to repeat the cycle. Water condensate in the hotwell was continuously drained off.

Hot brine from well Mesa 6-2 was pumped from the geothermal supply manifold to the direct contact heat exchanger where heat was extracted from it to heat and vaporize the isobutane. The high pressure brine was then reduced to atmospheric pressure and dumped into an open tank where dissolved isobutane in the brine was flashed out. No attempt was made to recover the isobutane at this point. Finally, the brine was sent to the reinjection system.

5.2 CALIBRATION TESTS

The primary goal of the calibration tests was to obtain turbine and cycle performance data with pure isobutane for use in evaluating results of the direct contact heat exchanger tests.

A calibration loop was assembled and instrumented to obtain turbo-gearbox and cycle efficiency over a range of cycle operating parameters. These operating parameters included turbo-gearbox output shaft speed, amount of superheat, turbine pressure ratio, and isobutane flow rate. The Calibration loop consisted of the DSS DCHX loop with the DCHX replaced by a conventional brine-to-isobutane heat exchanger.

The turbo-gearbox calibration tests were conducted at both subcritical and supercritical cycle conditions. Turbine inlet pressures of about 300 and 600 psia and temperatures of 220^{O}F and 300^{O}F were run. The upper limits of pressure and temperature correspond to the supercritical conditions.

The subcritical calibration tests were performed at isobutane flow rates of 8, 10 and 11.5 gpm, at turbine inlet temperatures of 220^{O}F through 290^{O}F, turbine inlet pressures of 220 psig through 350 psig, and at turbine output shaft speeds of 2400, 2600, 2800, 3000, and 3200 rpm. For each test, the isobutane flow rate was first set at a desired value by adjusting the isobutane pump bypass control. The turbine inlet temperature was then set by varying the brine flow through the hairpin heat exchanger. The turbine output shaft speed was then varied by adjusting the generator speed control. Gearbox output horsepower was calculated at each speed setting to assure that performance data was obtained at near peak output. Data were recorded at the five speeds mentioned above while holding the turbine inlet temperature and isobutane flow rate constant. Turbine inlet temperature was then adjusted to a new value and the test was repeated. When sufficient data had been collected at a number of different turbine inlet temperatures, the isobutane flow rate was changed and the test sequence was repeated.

The supercritical calibration tests were performed at isobutane flow rates of 8, 8.5, and 9 gpm, turbine inlet temperatures of 270^{O}F through 300^{O}F, turbine inlet pressures of 510 psig through 575 psig, and at turbine output shaft speeds of 2400 rpm through 3000 rpm. The single nozzle configuration was used in this test. The test sequence for the supercritical tests was the same as that used for the subcritical tests.

Upon completion of the subcritical and supercritical isobutane calibration tests, the hairpin heat exchanger was isolated and calibration tests were performed with the direct contact heat exchanger. Only subcritical isobutane/water tests were performed with the DCHX loop. These tests were performed at an isobutane flow rate of 10 gpm, a turbine inlet temperature of 225^{O}F, a turbine inlet pressure of 282 psig, and at turbine output shaft speeds of 2400, 2600, 2700, 2800, 3000, and 3600 rpm.

When the DCHX calibration tests were completed, the endurance run was started. The operating parameters for the endurance run were selected from results of the subcritical isobutate calibration with the hairpin heat exchanger and the DCHX subcritical isobutane and water tests.

5.3 ENDURANCE RUN

The goals of the endurance run were: 1) to obtain operating experience on a power producing loop utilizing an organic working fluid (isobutane) with some entrained geothermal brine vapor, 2) to evaluate cycle efficiency, 3) to accumulate sufficient run time to identify and/or evaluate potential problems, and (4) to identify loop or fluid degradation.

In discussing the endurance run and any equipment problems or malfunctions encountered, one point must be clarified. The intent of the endurance run was to demonstrate the performance of a direct contact heat exchanger in an electric power producing loop. The endurance run was not performed to test conventional heat exchangers, pumps, pressure switches, seals, or any other equipment or components. Although the Barber-Nichols turbo-generator was designed and assembled with the 500 hour endurance run in mind, the DSS direct contact heat exchanger loop was not. It was assembled on a "crash" basis nearly 2 years ago for a series of short heat transfer tests not lasting more than a few hours each.

The endurance run began at 0940 on July 26, 1977, and was completed at 1630 hours on August 24, 1977. During this time period the DCHX loop and the turbo-generator accumulated a total of 500 hours running time. Four personnel, two from Barber-Nichols Engineering Co. and two from DSS Engineers, manned the test loop on a 24 hour a day, 7 day a week basis. For the endurance run, the R. W. Holland heat exchanger (used for the calibration test) was isolated by closing appropriate valves and the DCHX was used.

Operating parameters for the endurance run were as follows:

Brine inlet temperature to the DCHX	$330 \pm 5^{\circ}F$
Butane outlet temperature from the DCHX	$245 \pm 5^{\circ}F$
DCHX operating pressure	300 ± 10 psig
Brine flow rate	5.4 ± 0.2 gpm
Butane flow rate	9.5 ± 0.3 gpm
Turbine speed	2750 ± rpm
Butane level in hotwell	greater than 1/2 full
Condenser pressure	95 ± 10 psig

These parameters resulted in:

Turbine inlet temperature	$232 \pm 3^{\circ}F$
Turbine inlet pressure	300 ± 10 psig
Turbine exhaust pressure	95 ± 10 psig

The turbine inlet temperature and pressure were very close to design conditions; however, the turbine exhaust pressure was approximately 25 psi higher than design. This occurred for a number of reasons: 1) some of the cooling water tubes in the condenser were plugged with debris, 2) with the high humidity at the test site, the cooling tower water was entering the condenser at $83^{\circ}F$ to $86^{\circ}F$, 3) water carryover and non-condensibles such as CO_2 increased the condenser pressure.

During the endurance run data were recorded once each hour. These data included gearbox lube oil temperature, gearbox bearing temperature, turbine inlet and exhaust temperature, shed temperature, turbine inlet and exhaust pressure, gearbox lube oil pressure, turbine output shaft speed and torque, hotwell temperature, brine and isobutane flow rate, and DCHX brine inlet and outlet temperature.

Since the gearbox was being purged with nitrogen to prevent isobutane buildup, its oil level and the nitrogen flow rate were checked once each hour.

During the endurance run, the DCHX loop and the turbogenerator were shut down a total of 26 times for various reasons. These reasons are summarized as follows:

Number of Times Shutdown	Cause
5	Isobutane flow problems
4	Brine pump pressure switch
3	Repack brine pump
2	Cooling tower
2	Gearbox oil change
2	Low isobutane level
1	Site power failure
1	High gearbox pressure
1	Blown DCHX burst disc
1	Ruptured brine pump discharge line
1	Switch to hairpin heat exchanger
1	Repack butane pump
1	Plugged reinjection line
1	Endurance run complete

The largest single shutdown problem was isobutane flow. This was normally caused by water or air in the butane pump or butane pump packing failure. The second largest shutdown failure occurred early in the endurance run and was traced to a faulty pressure switch. This was corrected by replacement of the pressure switch and addition of a cooling coil to the brine line to prevent overheating of the pressure switch diaphragm with hot brine. Brine pump packings were the third most common cause for shutting down the loop. The brine pump was designed for a maximum operating temperature of 160^{o}F. However, the brine was at 330^{o}F and 300 psig. The packings depended on water seepage for lubrication, and, since the brine flashed through the packings, they received no lubrication and failed every 50 to 100 hours.

Because the condenser was water cooled, the loop had to be shutdown twice to prevent condenser overpressure due to cooling tower problems.

The loop was shutdown twice to change gearbox oil, once because of water contamination caused by a leaking oil cooler at 311 hours and once due to isobutane contamination caused by a leaking gearbox turbine shaft face seal.

Although the loop was only shutdown once specifically to replace the isobutane pump packings, these packings were replaced a number of times when the loop was shut down for other reasons. The butane pump packings failed as often as the brine pump packings and for the same reason; no lubrication. Neither pump was designed for this application.

Since there were no automatic controls on the DSS loop, the brine flow rate, isobutane flow rate and water level in the hotwell had to be watched almost constantly. Normally if the water level in the hotwell was kept to a minimum and the isobutane level in the hotwell was kept above the half full mark, the isobutane flow rate remained constant.

The brine flow rarely changed, but it had to be readjusted periodically to compensate for changes in brine temperatures.

If the temperature of the isobutane leaving the DCHX was kept between 240^{o}F and 250^{o}F, the water carryover to the condenser remained fairly constant and the drain petcock on the hotwell sight glass could be set at one position and left alone for up to 5 hours.

Normally if the plots of isobutane inlet and exit temeprature and brine inlet and exit temperature remained constant on the strip chart recorder, the controls on the DSS loop could be left alone. This condition rarely persisted for more than 2 hours.

For the first 341 hours of the endurance run the Barber-Nichols turbo-generator ran trouble free. At 341 hours the turbine face seals started to leak isobutane into the gearbox. The high gearbox pressure alarm activated and the operator shutdown the test loop. The seal was replaced but never did completely stop leaking. This necessitated an oil change at 415 hours.

A total of 1315 pounds of isobutane were added to the DSS loop during the 500 hour endurance run. Figure 10 shows the cumulative isobutane added to the loop as a function of calendar days. As can be seen, the rate increased sharply at the 340 hour point and remained high until the 450 hour point. It then dropped back to the rate just prior to the seal failure. For the first 90 hours of operation the loop used 1 lb. of isobutane per hour. For the next 250 hours of operation it used 2.2 lbs. per hour. For the next 100 hours of operation it used 5.5 lbs. per hour and for the final 60 hours it used 2.1 lbs. per hour.

Not all the isobutane losses are attributable to operation with the direct contact heat exchanger. As discussed above, a substantial amount of isobutane was lost in numerous leaks in the test loop. Further discussion on this point is contained in Section 6.5.

5.4 TEARDOWN INSPECTION

After completion of the 500 hour endurance run a teardown inspection was conducted. This inspection included examination of the turbine, the DCHX, the condenser and the piping for corrosion or scaling.

The Barber-Nichols turbine nozzle block and turbine wheel were removed. They were photographed to show the areas of scale buildup. Samples of scale material were scraped from the nozzle block, the burbine wheel, and the exhaust housing. These samples were sent to LBL for analysis.

The DSS loop condenser heads were removed and the tubes were examined for corrosion, scale buildup and plugging caused by foreign debris. The heads and the tubes were also photographed to illustrate their condition.

The upper distributor tube and piping were removed from the DCHX and examined for corrosion and scale buildup. These items were photographed and samples of scale were sent to LBL for analysis.

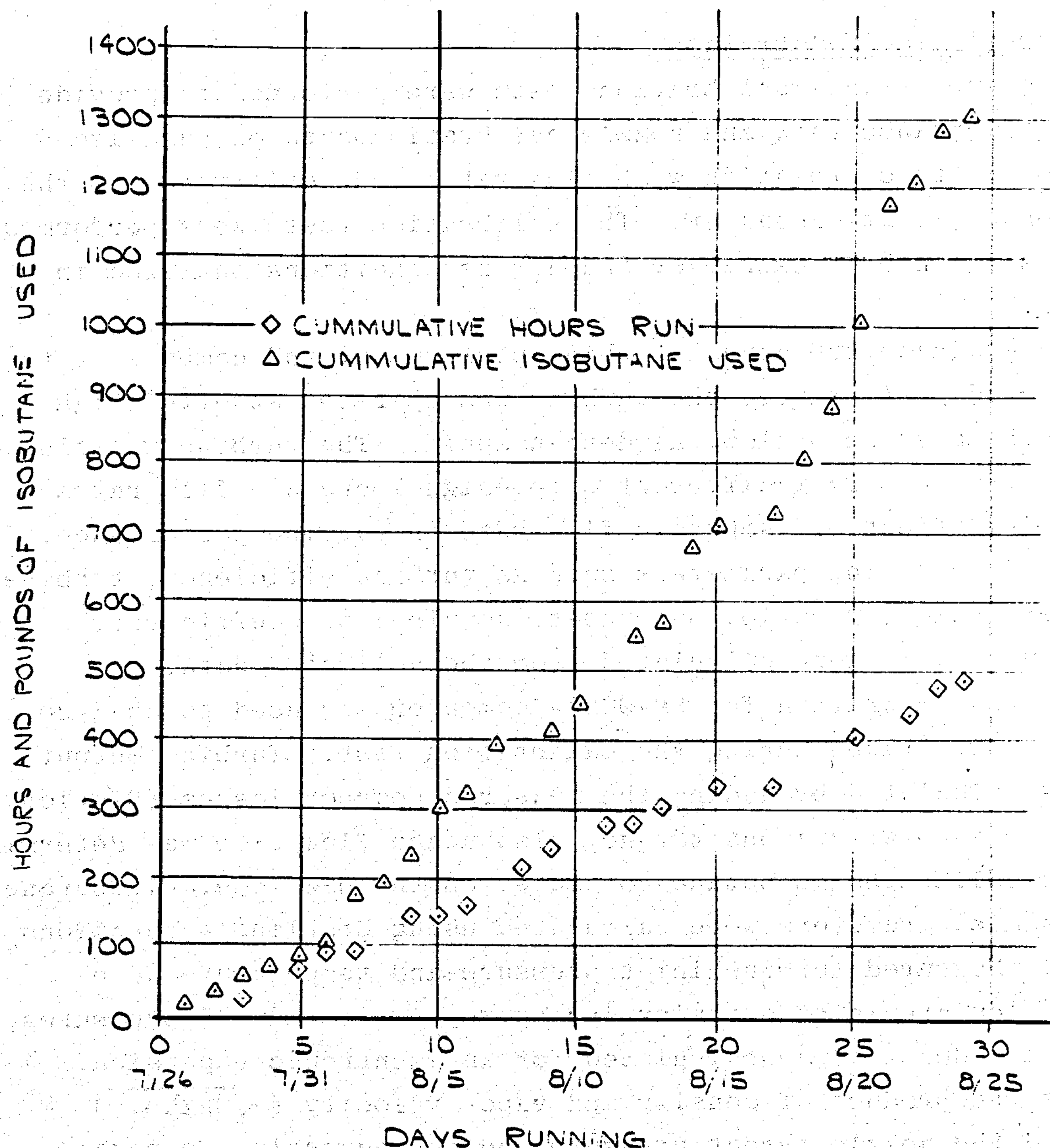

Figure 10. Cumulative run time and IC_4 added.

6.0 TEST RESULTS

6.1 TURBINE CALIBRATION - CONVENTIONAL HEAT EXCHANGER

6.1.1 Subcritical Calibration

The subcritical calibration tests were performed to provide turbine performance data and nozzle calibration data on pure isobutane for later correlation with flow rates and performance of the direct contact heat exchanger. The calibration tests were performed with the hairpin heat exchanger and at the conditions outlined in Section 5.1.

To evaluate the actual turbine performance and compare it to the predicted performance, the turbine was operated at off-design speeds both above and below the design speed. The turbine was also operated at a number of different off-design isobutane flow rates to determine effect of isobutane flow rate on turbine performance. A number of operating parameters such as turbine efficiency, turbine horsepower, velocity ratio, superheat, nozzle flow coefficient, and pressure ratio were calculated from the collected data.

A program written for HP-9825A computer was used to analyze and reduce data taken during the calibration test. Turbine output power was calculated by adding the measured gearbox losses (Section 3.4) to the generator input torque. Isobutane flow rate was determined using the calibrated isobutane rotameter on the DSS loop. Isobutane thermodynamic parameters were calculated using Starling's equations (Ref. 1). Measured turbine inlet pressure and temperature were used to calculate inlet enthalpy and entropy. At reduced pressures, enthalpy and density was calculated for an isentropic expansion. A maxima of the product of density and vapor velocity ($\sqrt{2g\Delta h_i'}$) determined the nozzle throat pressure and consequently the nozzle ideal flow rate. The nozzle flow coefficient (C_d) is defined:

$$C_d = \frac{\text{measured isobutane flow}}{\text{nozzle ideal flow}}$$

The above approach was necessary since the isobutane expansion occurred close to the saturated vapor dome, and significant compressibility effects were expected for the vapor expansion. Turbine available energy was calculated by continuing the isentropic expansion to the measured turbine exhaust pressure.

Turbine efficiency was defined:

$$\eta_t = \left[\frac{HP_o + HP_g}{\dot{w}\ \Delta H'}\right] \frac{550}{778}$$

where: HP_o = generator input shaft power

HP_g = measured gearbox power

$\dot{w}$ = measured isobutane flow, lb/sec

$\Delta H'$ = turbine available energy, B/lb.

Velocity ratio (U/C_o) was defined:

$$U/C_o = U/\sqrt{2g\Delta H'}$$

where: U = turbine pitch line velocity, ft/sec.

Both predicted and measured turbine efficiency were plotted vs. velocity ratio at various pressure ratios and are shown in Figure 11. In each case, the actual efficiency is compared to the predicted efficiency at actual pressure ratios of 3.4, 3.12, and 2.8. As may be noted, the peak turbine efficiency occurs at a velocity ratio close to the value predicted. This result indicates that indeed the available energy calculated from the thermodynamic analysis for isobutane is close to the value actually present in the expanding vapor. The variation in actual and predicted efficiency is due primarily to a higher than anticipated nozzle flow coefficient.

The nozzle flow coefficient calculated from the data in all cases was greater than unity. This was attributed to wetness in the isobutane. Figure 12 illustrates the effect of this wetness on the maximum turbine efficiency as a function of pressure ratio and isobutane flow rate.

Figure 13 shows the effect of nozzle inlet pressure and superheat on the nozzle flow coefficient. In general, the nozzle flow coefficient decreased toward its laboratory measured value of 0.962 as the nozzle inlet pressure and superheat increased to "dry out"

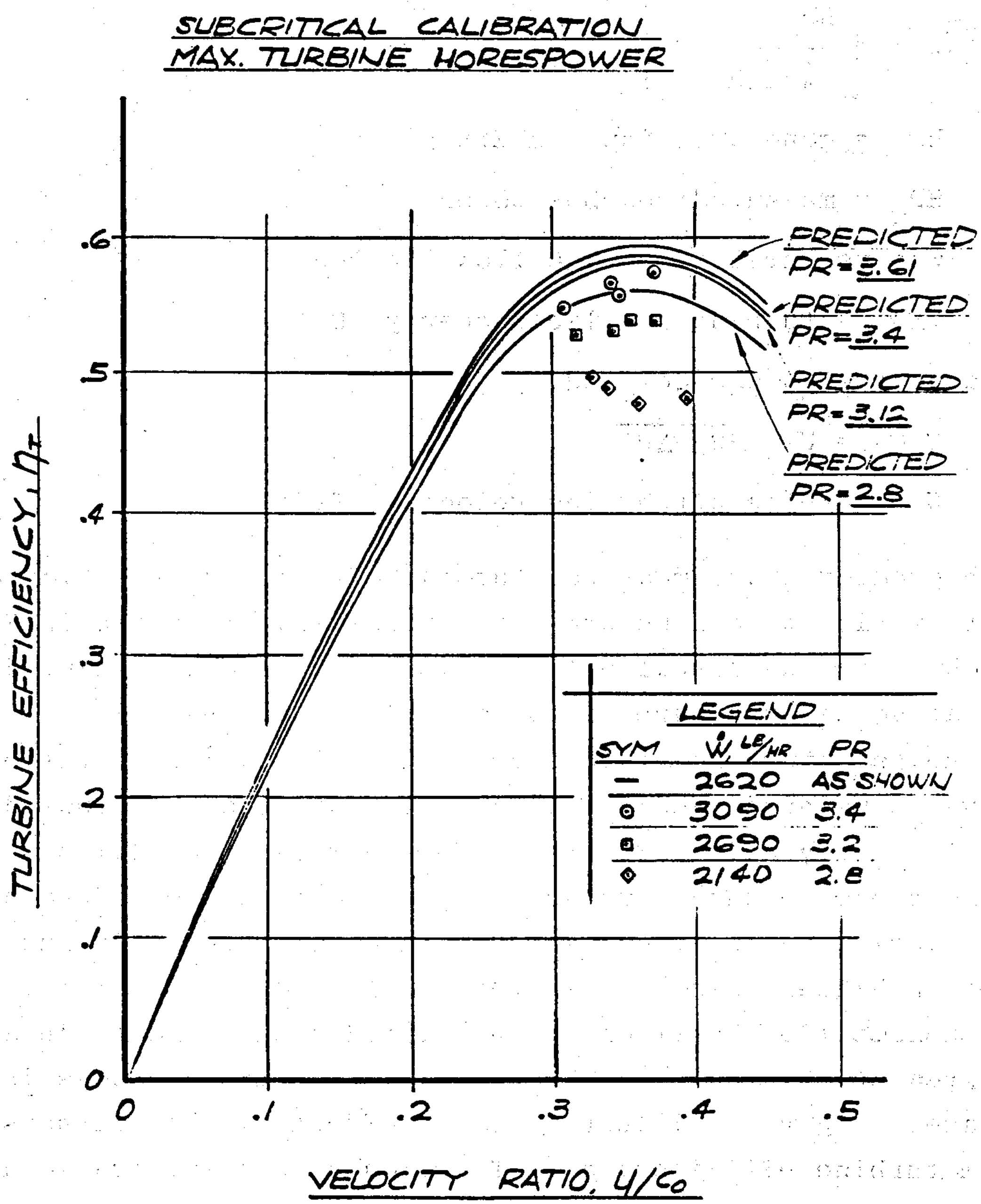

Figure 11. Variation of turbine efficiency with velocity ratio.

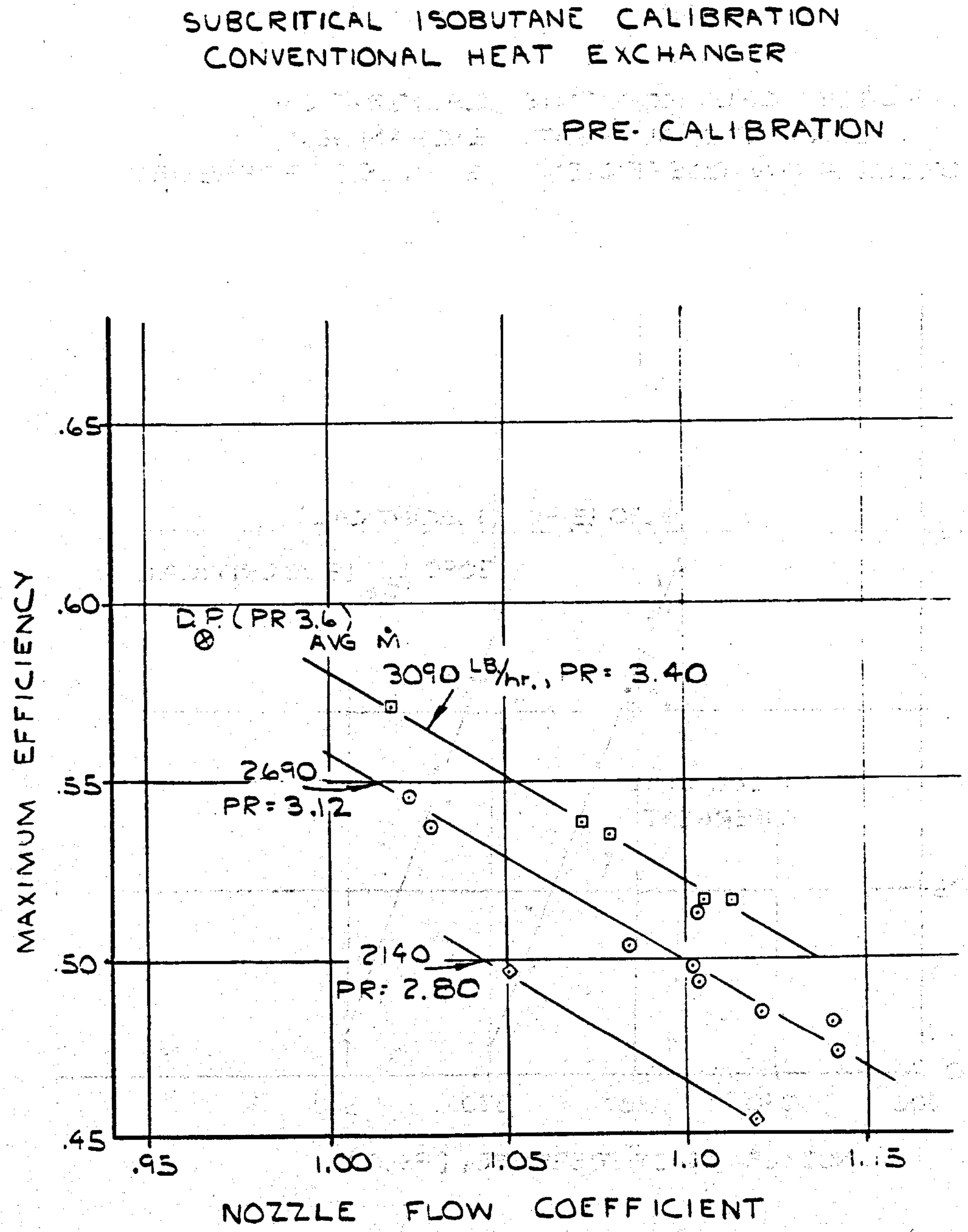

Figure 12. Effect of wetness on turbine efficiency.

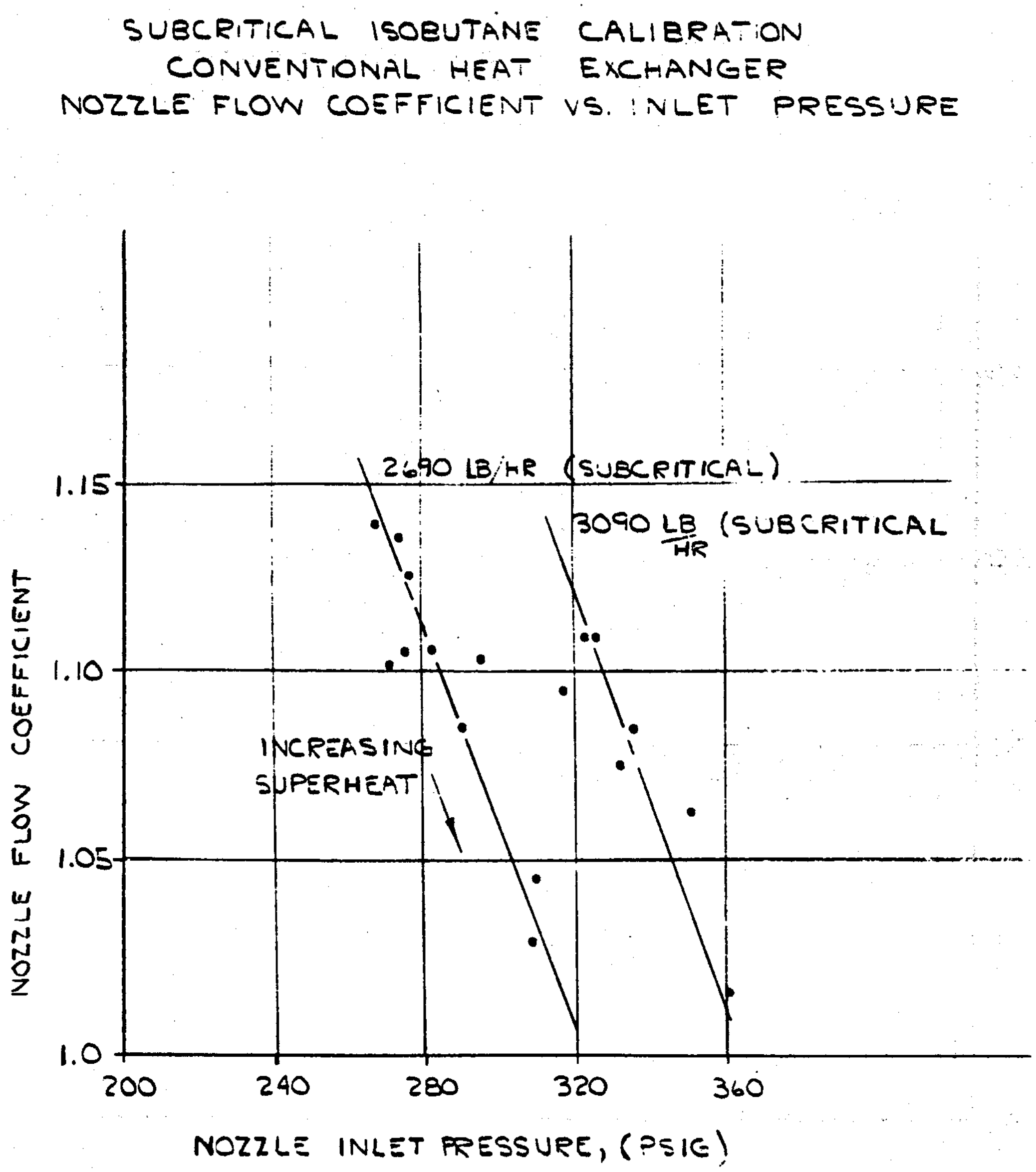

Figure 13. Nozzle flow coefficient vs. inlet pressure.

the isobutane. This figure can be used to explain the results shown in Figure 14.

Both predicted and actual maximum turbine output power are compared in this figure vs. superheat at three different values of isobutane flow rate and pressure ratio. At higher values of superheat, the actual output approaches the predicted output since the isobutane is drying out and the nozzle flow coefficient is approaching its measured value.

Turbine performance was not measured at values of superheat above 65^{O}F because of limitations imposed by the equipment.

6.1.2 <u>Supercritical Calibration</u>

The supercritical calibration tests were performed to obtain preliminary data in preparation for the Occidental tests: the supercritical nozzle was calibrated on pure isobutane using the hairpin heat exchanger and at the conditions outlined in Section 5.1.

The turbine was operated at a number of off-design speeds above and below the design speed to determine its peak efficiency and output horsepower. Since the isobutane pump flow rate was limited at the higher pressures, there was only a 1 gpm variation in flow rate (8 to 9 gpm) for the supercritical calibration.

Figure 15 shows the comparison between predicted and actual turbine output horsepower. Such lines of constant temperature and lines of constant pressure are almost parallel over the saturation dome, and degrees of superheat does not accurately depict the relationship between the fluid and the saturation dome, the turbine output was plotted vs. turbine inlet enthalpy. The position of each data point in relation to the dome is shown at the top of Figure 15.

As one moves to the left above the dome on the P-H diagram, the actual turbine output deviates further from the predicted and is greater than that predicted. At a point just above the right most extremity of the dome the actual and predicted output are in agreement, and as one moves to the right of this point, the actual turbine performance drops off again while the predicted performance continues to increase.

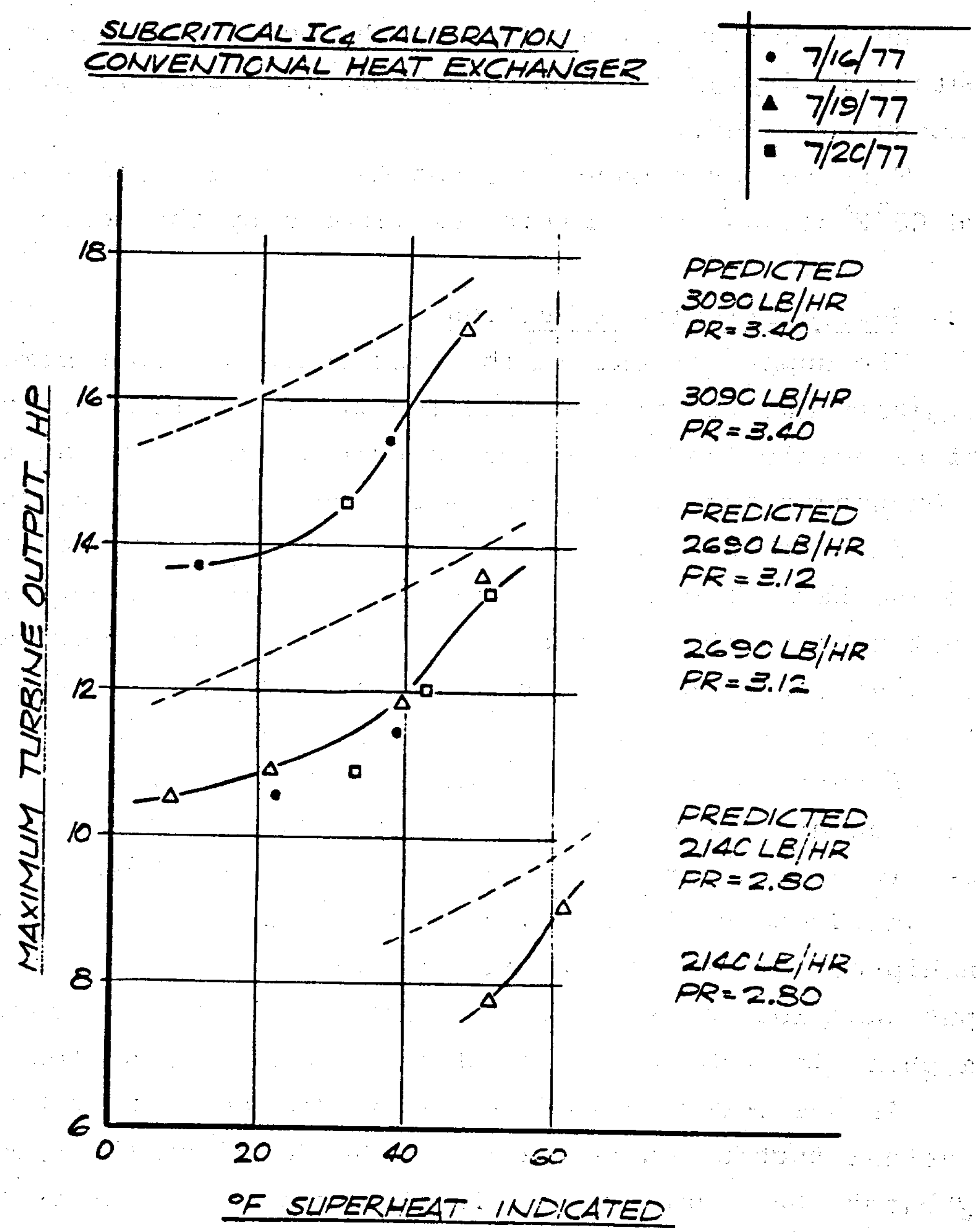

Figure 14. Maximum turbine output vs. superheat.

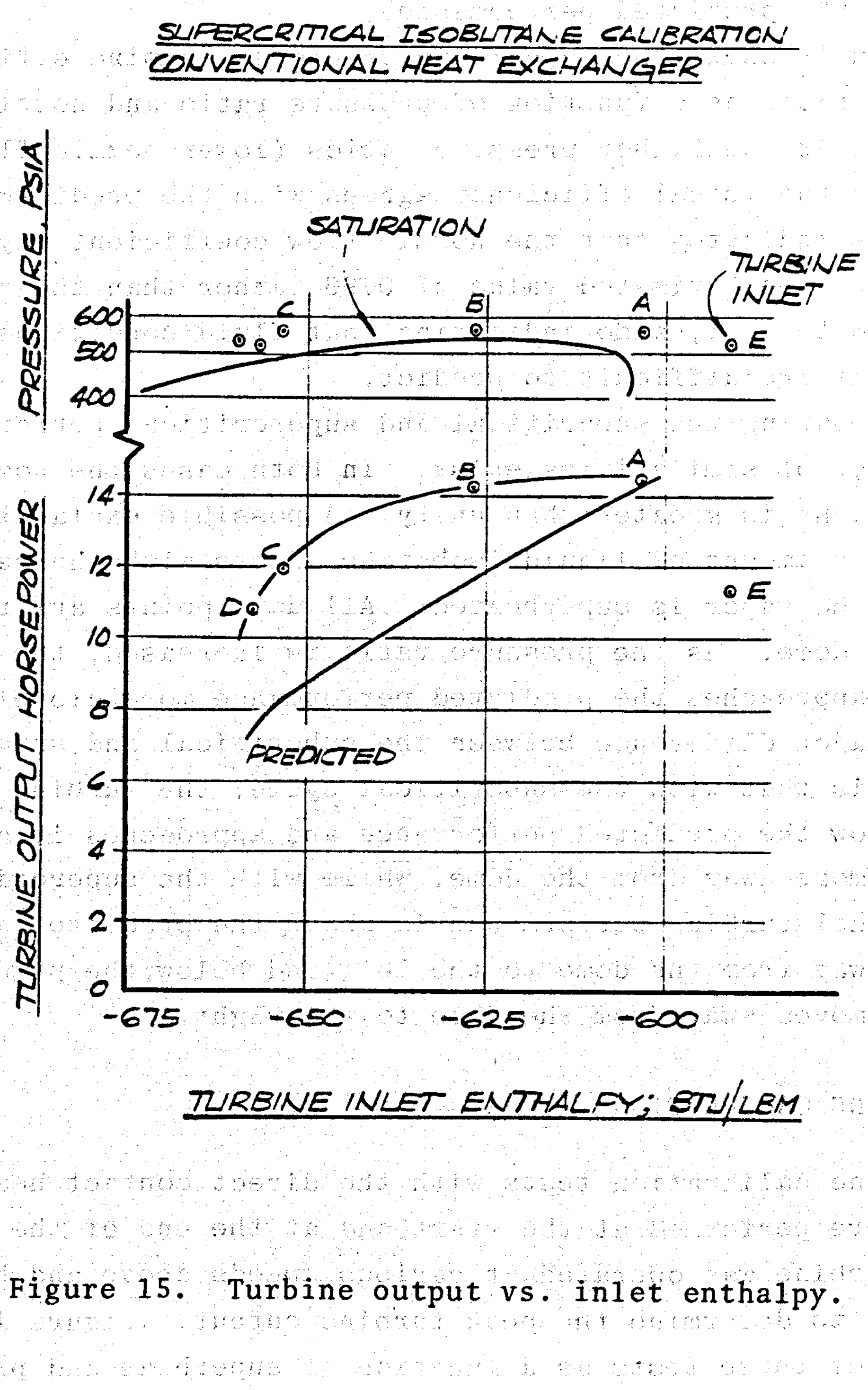

Figure 15. Turbine output vs. inlet enthalpy.

The reasoning behind this behavior is not clear. It is assumed that it results because of compressibility effects and the fluid properties just above the dome cannot be accurately predicted. Small changes in temperature or pressure certainly have a pronounced influence on the predicted performance.

Figure 16 shows the actual and predicted turbine efficiency vs. velocity ratio as a function of pressure ratio and nozzle flow coefficient. At the higher pressure ratios (lower nozzle flow coefficients) the actual efficiency agrees with the predicted efficiency. This indicates that the nozzle flow coefficient may be quite close to the estimated value of 0.96 rather than the calculated value of 1.08 to 1.11, also indicating that fluid conditions just above the dome are difficult to predict.

In comparing the subcritical and supercritical performance data, a number of similarities exist. In both cases the nozzle flow coefficient is greater than unity. A possible explanation is that a certain amount of liquid isobutane exists with the vapor even though the vapor is superheated. All data points are relatively close to the dome. As the pressure ratio is increased, the actual performance approaches the predicted performance more closely.

The major difference between the subcritical and supercritical calibration is that with the subcritical cycle, the turbine performance is below the predicted performance and approaches it as the data points move away from the dome, while with the supercritical data the actual turbine performance is above the predicted as the data moves away from the dome to the left and below the predicted as the data moves away from the dome to the right.

6.2 TURBINE CALIBRATION - DCHX

Turbine calibration tests with the direct contact heat exchanger were performed at the start and at the end of the endurance run. The turbine was operated at various speeds above and below the design speed to determine the peak turbine output. Figure 17 shows the results of these tests as a function of superheat and pressure ratio. As in the case of the subcritical and supercritical data,

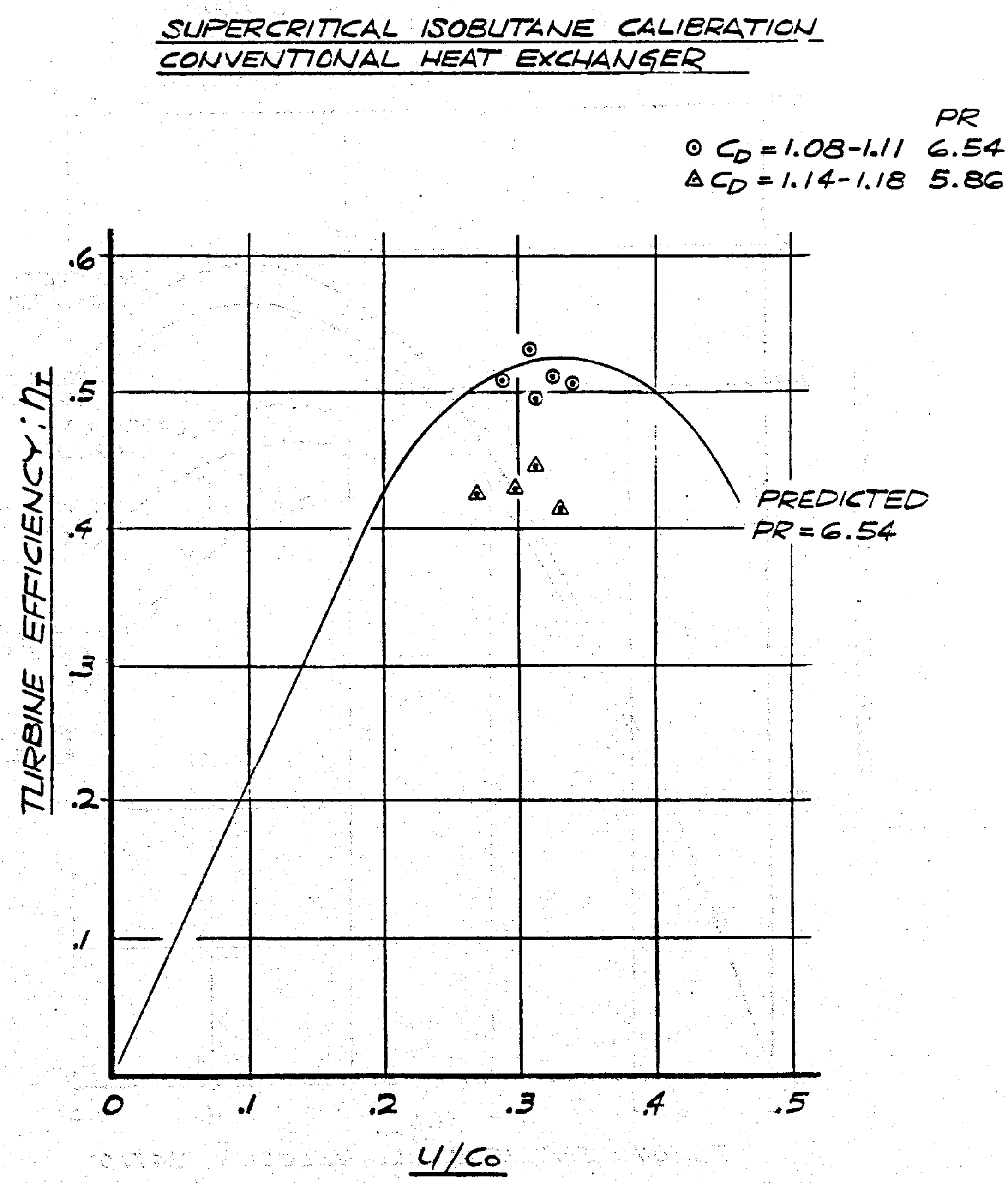

Figure 16. Turbine efficiency vs. velocity ratio.

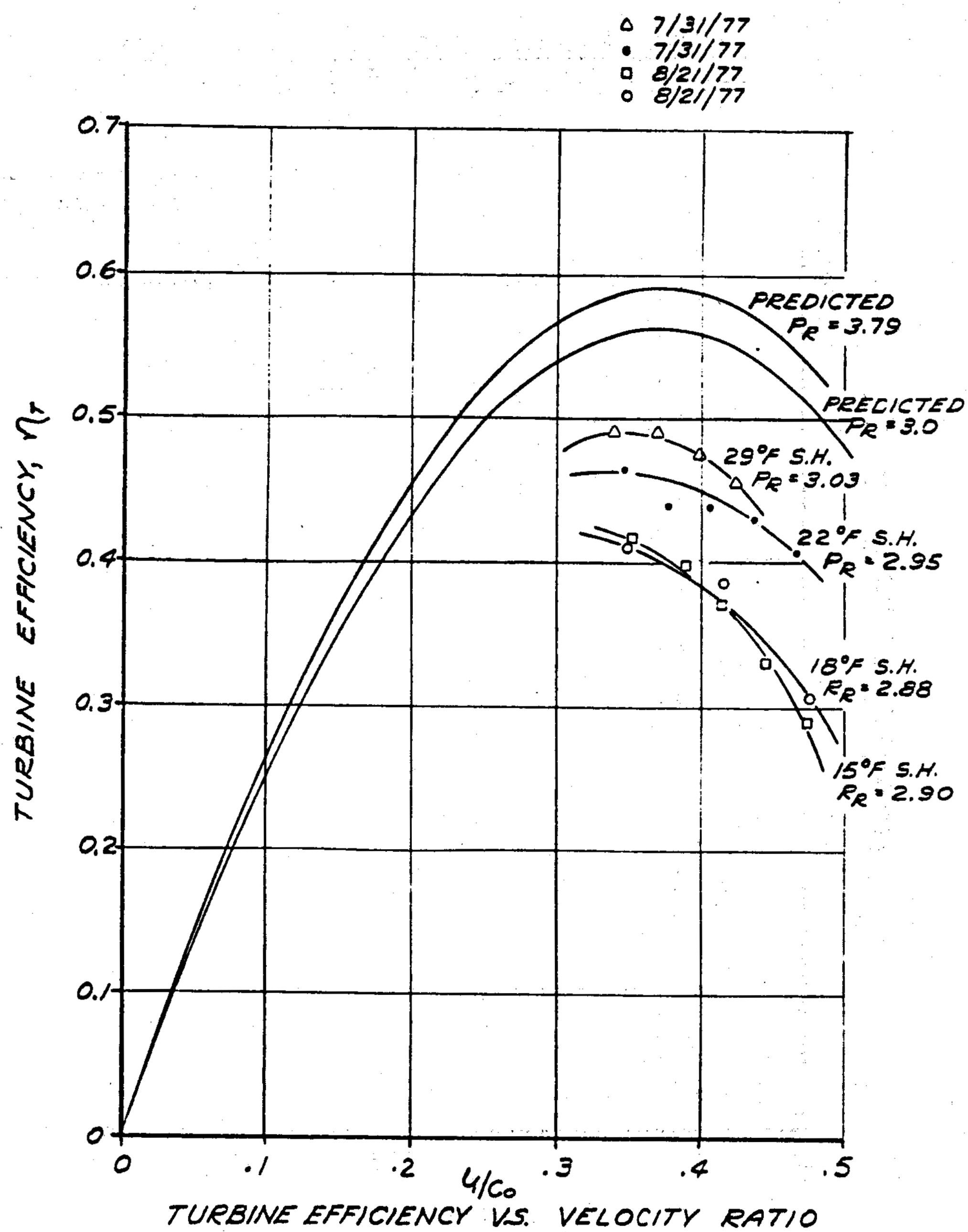

Figure 17. Subcritical IC_4/H_2O calibration, direct contact heat exchanger.

the actual performance is below the predicted performance and approaches the predicted as the amount of superheat is increased. This degradation in performance may also be attributed to wetness (nozzle flow coefficient of 1.11).

All calibration tests were performed at constant brine and isobutane flow rates and any change in the amount of superheat was due to the inlet brine temperature which could not be easily controlled. The brine and isobutane flow rates were set by the limits of the DSS direct contact heat exchanger loop capabilities and the temperature of the brine from the well.

6.3 ENDURANCE TEST

During the endurance test the direct contact heat exchanger maintained the same level of heat transfer performance throughout the 500 hours. The capacity to transfer heat was not effected by corrosion or scaling properties of the geothermal brine. Figure 18 shows a typical plot of temperature vs. percent of heat added to the isobutane during the endurance run. Approximately 94% of the heat from the brine was transferred to the isobutane. The remaining 6% was retained by the entrained water vapor. Any variations in the DCHX heat transfer performance were caused by variations in incoming brine temperature, brine flow rate and isobutane flow rate.

Scaling properties of the brine, however, did effect the performance of the turbine. Figures 19 and 20 illustrate the effect of scaling on turbine horsepower and efficiency as a function of time. The data from 0300 hours to 0600 hours each day was used because the maximum turbine output occurred between these hours more often than at any other time of day.

The turbine efficiency gradually changed from an average value of 49% at the beginning of the endurance run to 39% at the end of the endurance run. The turbine horsepower changed from an average value of 10.25 hp to 7 hp. The scatter in the data was caused by a number of factors. They included: (1) variations in inlet brine temperature which changed the amount of superheat and effected fluid wetness; (2) variations in atmospheric conditions which effected the

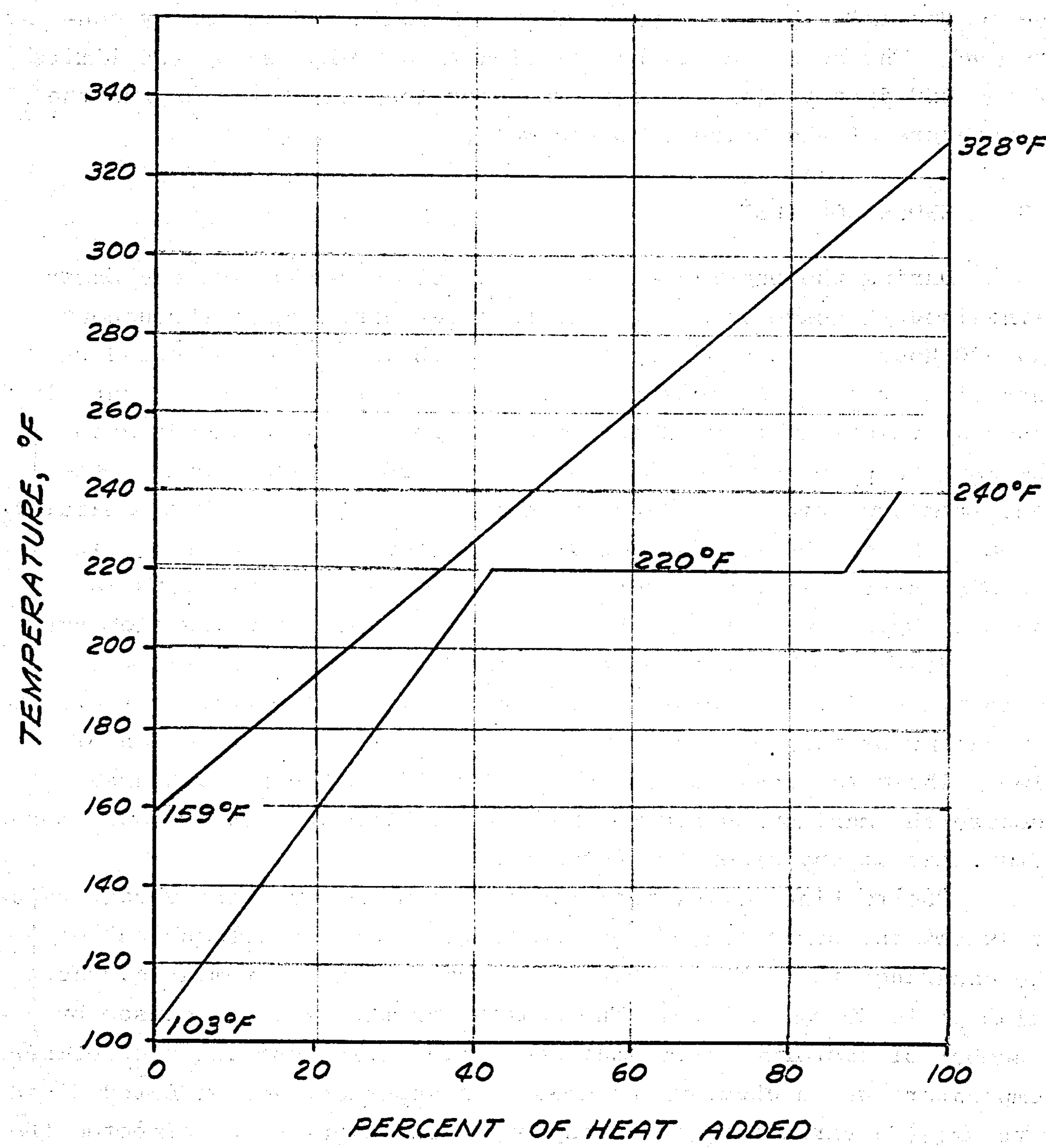

Figure 18. Typical plot of temperature vs. percent of heat added for the DCHX during the endurance run.

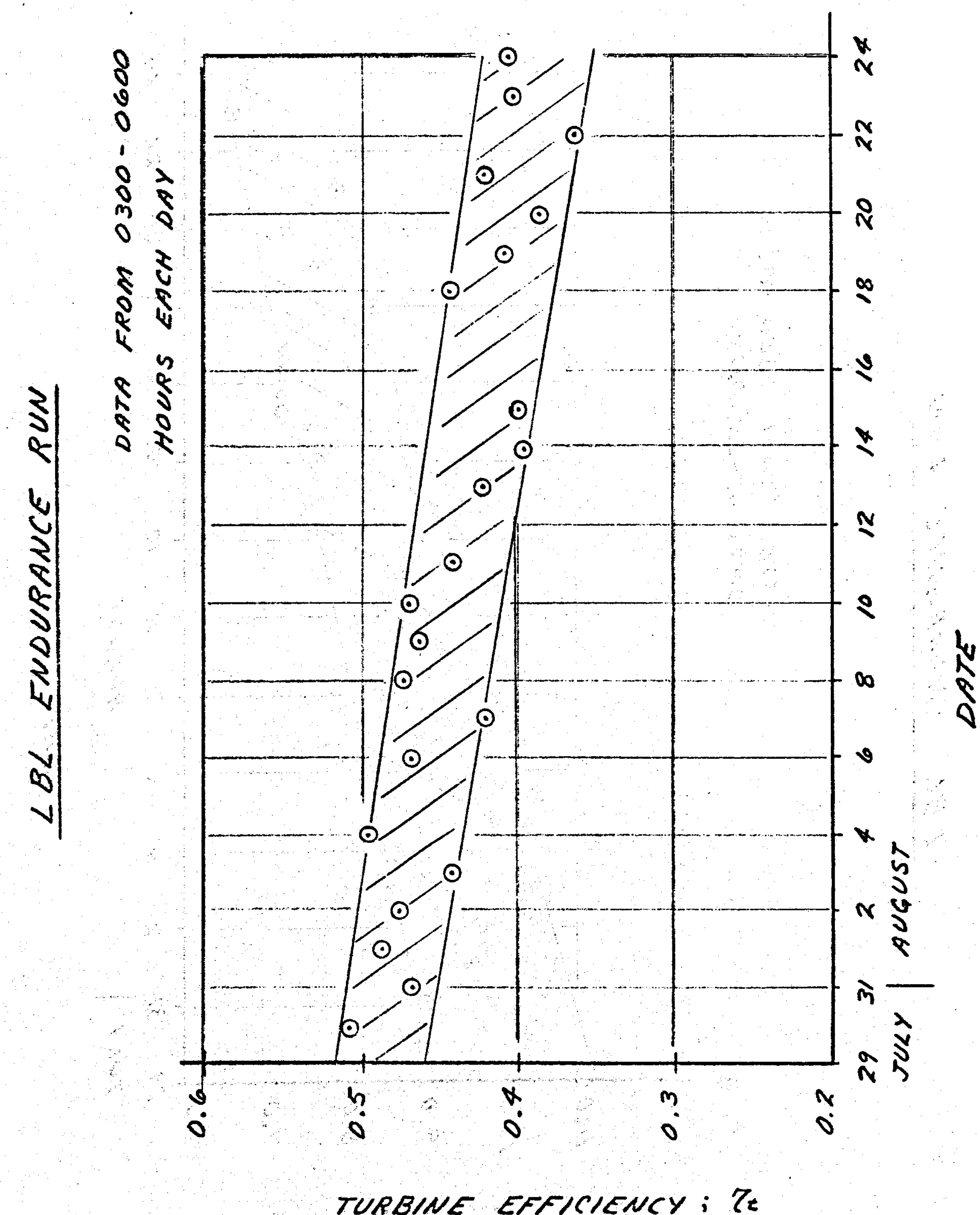

Figure 19. Turbine efficiency vs. time.

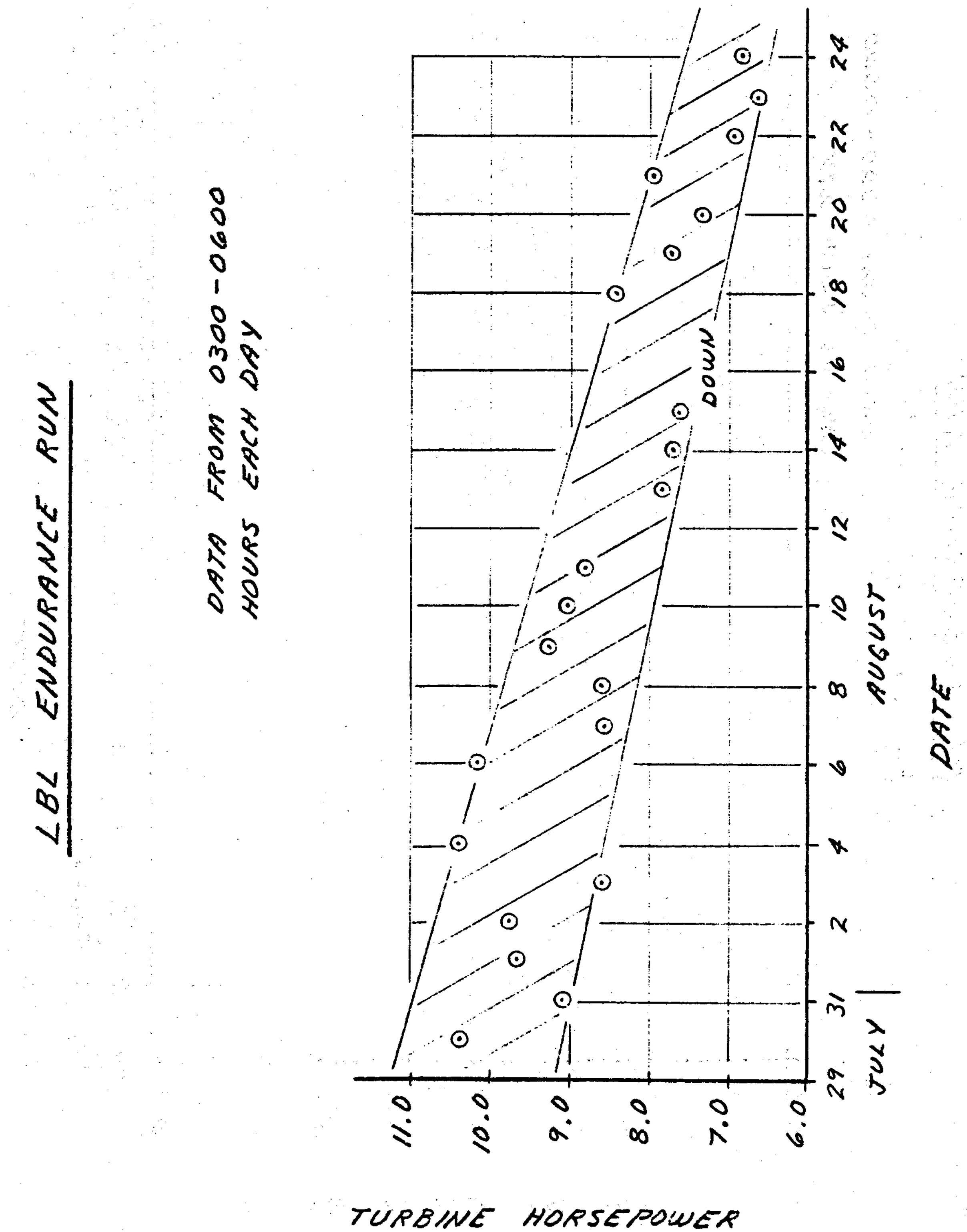

Figure 20. Turbine horsepower vs. time.

cooling tower water temperature and condenser pressure; (3) clogging of the condenser cooling water tubes which changed the turbine exhaust pressure; and (4) scale buildup on the turbine wheel, nozzles and exhaust housing.

6.4 POST CALIBRATION - CONVENTIONAL HEAT EXCHANGER

At the conclusion of the endurance test the turbine nozzles were again calibrated using the hairpin heat exchanger and the same isobutane flow rates as were used for the initial calibration. Figures 21 and 22 show the results of the post endurance calibration. The turbine performance at the end of the endurance run was much less than that originally measured during the initial calibration tests. Two factors appear to have caused the degradation in performance. One was the reduced pressure ratio (2.66 and 2.83 as opposed to 3.12 and 3.48 during initial calibration for the 2680 and 3070 lb/hr flow rates) and the scale buildup on the turbine components. The reduced pressure ratio and scale buildup resulted in a 16% reduction in turbine efficiency at the 3070 lb/hr flow rate and a 14% reduction at the 2680 lb/hr flow rate.

6.5 ISOBUTANE LOSS FRACTION

Brine samples were taken from the outlet of the direct contact heat exchanger in order to measure the concentration of isobutane in the brine and to estimate the isobutane loss. The brine samples were taken from a tee located directly below the DCHX outlet. The line was purged several times before connecting the sample cylinder in order to ensure that a clean sample would be taken and dirt or scale would not be swept into the cylinder. The sample bomb was oriented vertically to ensure a full liquid sample and avoid trapping vapor in the sample container. The brine samples were then cooled and analyzed by the chemistry laboratory at the East Mesa Test Facility.

The analysis of the brine samples for isobutane was made with a Perking-Elmer model 910 gas chromatograph. Prior to injecting the sample into the gas chromatograph, the brine samples were flashed

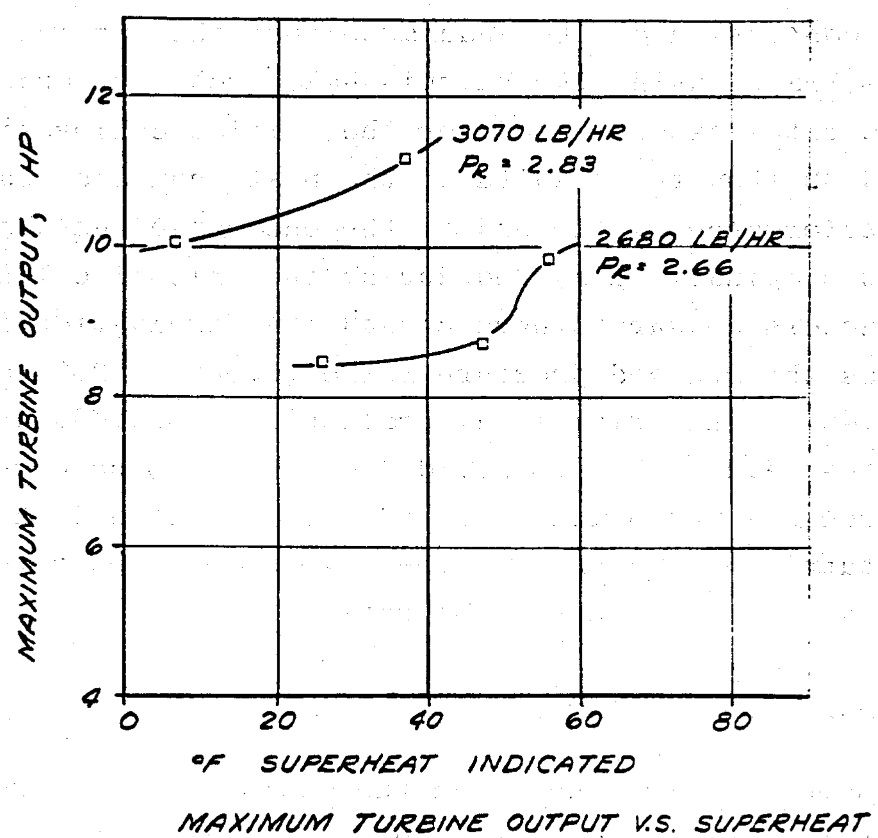

Figure 21. Subcritical isobutane calibration, conventional heat exchanger.

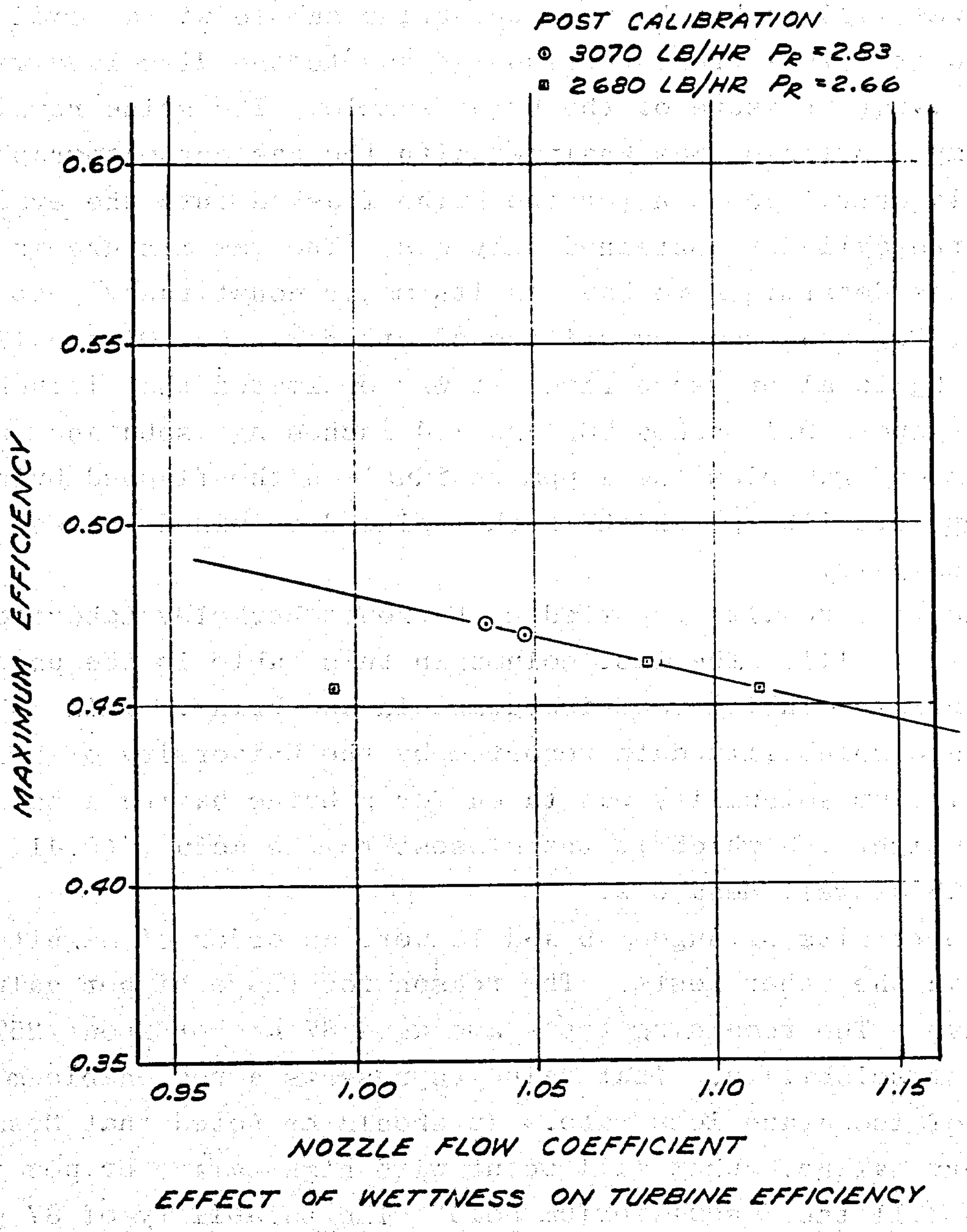

Figure 22. Subcritical IC_4 calibration, conventional heat exchanger.

into a second cylinder which had been evacuated to a vacuum of 26.5 inches hg. For the initial sample the pressure rise in the evacuated cylinder was 11.5 inches hg when the brine sample was allowed to flash into it. This pressure rise was due to the dissolved gases and water vapor pressure of the brine sample. The brine remaining in the sample cylinder was analyzed with the gas chromatograph and was usually only 1 ppm. After the brine flashed into the evacuated cylinder the cylinder contained only gas. The gas mixture in the cylinder was determined to have as its major constituents water vapor, isobutane, CO_2 and trace quantities of other gases. Of the 11.5 inches of hg total pressure rise, it was estimated that 1 inch hg was water vapor, 9.1 inches CO_2 and 1.4 inches hg isobutane vapor. This volume of gas plus the 1 ppm residual in the flashed brine translates into 110 ppm which is the value determined by gas chromatograph analysis.

The test results reported by Lawrence Berkeley Laboratory are shown in Table III. The last column in this table is the percent of equilibrium solubility of isobutane in the brine. This is based on isobutane solubility data reported by the University of Utah. The equilibrium solubility was taken for a brine having a salinity of 0% by weight TDS which is the closest to the actual (0.41% by weight) TDS of well Mesa 6-2.

The results of August 9 and 10 were an order of magnitude higher than the other tests. The reason for those higher values is speculative. The remaining tests averaged 87 ppm or about 25% of equilibrium solubility. That value represents a reasonable minimum estimate of isobutane loss rate. It should be noted that Mesa 6-2 is not very saline. What will occur with high saline brines is unknown. Will the % equilibrium hold? The solubility of 87 ppm is approximately the equilibrium solubility expected for a 250,000 ppm brine at 150°F and 315 psia. If the % equilibrium does hold for the higher saline brines, the anticipated dissolved isobutane will be quite low.

The 500 hour endurance run was accomplished in 29 days (703 hours). During this period the amount of isobutane added to

TABLE III

MEASUREMENT OF ISOBUTANE CONCENTRATIONS AT THE BRINE OUTLET OF DIRECT CONTACT HEAT EXCHANGERS

Date	Sampling Conditions Temp. °F	Pressure, PSIG	Observed Isobutane Concentration PPM	Percent Equilibrium Solubility*
July 25	150	293	110.0	29.3%
Aug. 9	157 F	297	1600	445. %
Aug. 10	154 F	305	940	256. %
Aug. 19	163	306	120.0	33.6%
Aug. 20	157	295	103.0	28.7%
Aug. 21	156	300	49.0	13.3%
Aug. 23	Not Recorded		50.0	
Aug. 24	164	300	80.5	23.2%
Aug. 24	167	315	89.2	25.0%

the system was measured. Figure 10 was prepared to show the cumulative amount of isobutane added to the loop as a function of time. For the entire 29-day period a total of 1300 lbs. of isobutane was added to the system which is much higher than was anticipated. For example, at an average solubility of 87 ppm and an average brine flow rate of 2900 lbs/hour, the total consumption of isobutane should have been $87 \times 10^{-6} \times 500 \times 2900 = 126$ lbs.

A substantial amount of the lost isobutane may be accounted for due to leaks in the test loop. The outside packed plunger pumps were probably the worst offender as they required repacking numerous times. The turbine seal began leaking considerable amounts of isobutane about 340 hours into the run and was replaced. Rupture discs were blown out and safety valves lifted on two occasions because of operator errors. Countless other undetected leaks were probably losing isobutane during the 29-day test. Exactly where and how the isobutane escaped would be pure speculation at this point. The isobutane being used is a technical grade (98% isobutane) and contains no ethyl mercaptan odorizers. Consequently the small leaks were not detected by the human nose which is the best isobutane detector when ethyl mercaptans have been added. For these reasons an improved estimate of isobutane losses cannot be made from the test data.

Additional tests with an active separator and stripping system is needed to allow economic evaluation of actual isobutane losses for the direct contact power loop.

6.6 INSPECTION

Following the endurance run the heat exchanger loop and the turbine were disassembled for inspection. This section presents the visual results and chemical analysis performed on scale or deposits found on the various components.

6.6.1 Direct Contact Loop

The primary test component of the heat exchanger loop was the direct contact boiler. No evidence of scaling or erosion due to the injected brine was observed in this component.

A reduced condenser heat transfer coefficient was evidenced during the endurance run by increased condenser pressure. The condenser had a thick mud coating on the cooling water side which is attributed to the quality of the cooling water available for use in the test. The mud was easily wiped from the inside of the condenser tubes, and no permanent damage was apparent. The shell or isobutane side appeared quite clean, and no evidence of scaling or corrosion was noted.

Throughout the loop the vapor carrying piping showed a thin coating of a soft black oxide, although the liquid lines were quite clean. Again, no distress was noted in any of the connecting pipes. In particular, the piping downstream of the pressure control valve also showed no evidence of scale build up.

6.6.2 Turbine and Gearbox

The turbine was disassembled on two occasions in this test sequence. The first was 350 hours into the endurance run and was required for the replacement of the turbine shaft seal. At this time a scale deposition was noted in the turbine nozzles. A portion of the nozzle throats were plugged and a significant build up of scale was apparent in the nozzle expansion cone. The scaling obviously reduced the nozzle throat area, thus increasing the inlet pressure level required to maintain the turbine flow rate. The deposition in the expansion cone degraded nozzle performance and overall turbine efficiency as discussed in Section 6.3. A similar deposition was in evidence in the exhaust housing surrounding the turbine rotor. In addition an accumulation of the black oxide material was found in the vicinity of the turbine shaft seal. This accumulation, however, is not felt to be responsible for the seal failure.

At the completion of the endurance run the turbine hardware was again disassembled. As before, evidence of scale deposition was apparent in the turbine nozzles, although the location and amount of scale had markedly changed. Figures 23 through 25 show photographs taken at this second disassembly.

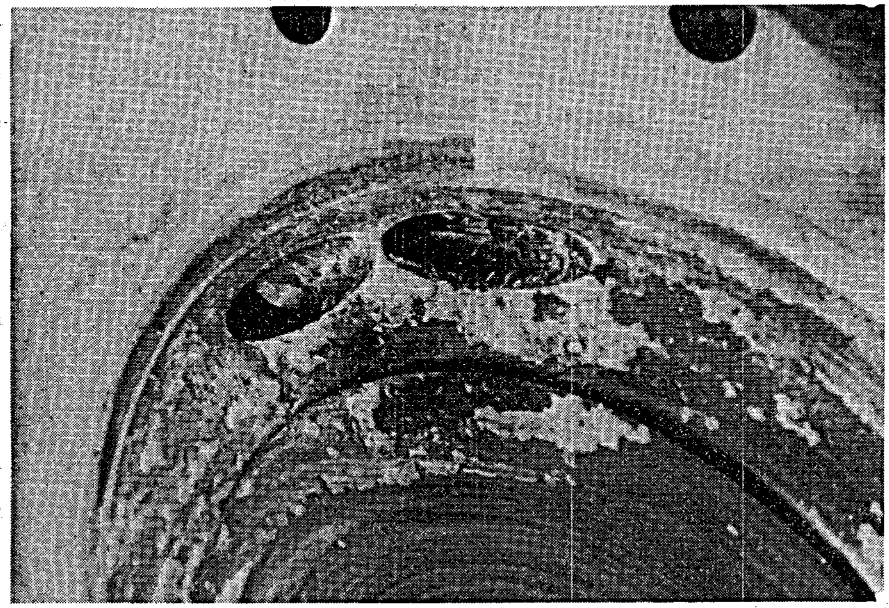

Figure 23. Subcritical turbine nozzle exit.

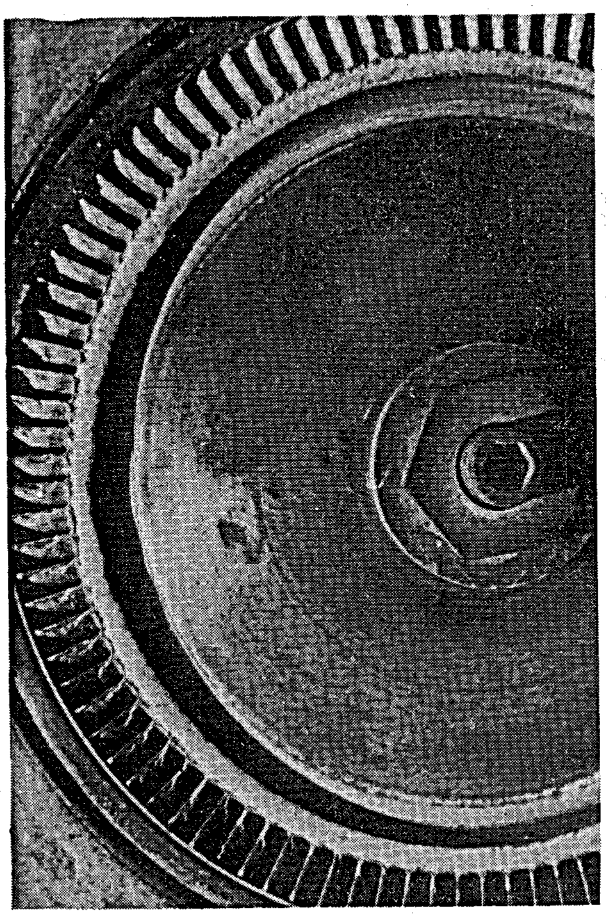

Figure 24. Turbine rotor inlet.

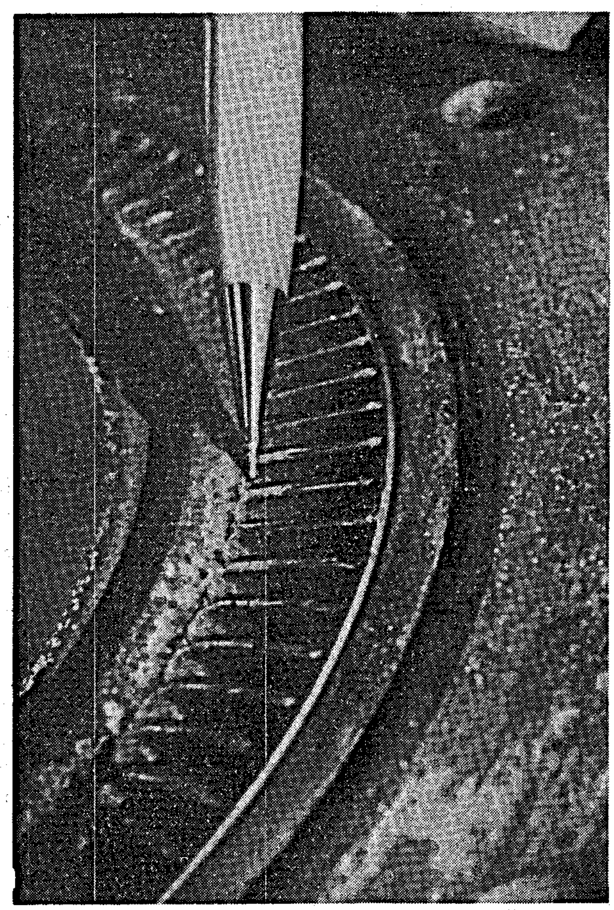

Figure 25. Rotor inlet blade leading edge.

Samples of the scale were collected at each disassembly and results of a chemical analysis of the material is presented in the following section.

6.6.3 Chemical Analysis

Samplings of the scale found in the turbine nozzles and scrapings from other components were taken during the disassembly and inspection of the power loop. Analysis of the turbine nozzle scale obtained at the 350 hour disassembly showed the primary constituent to be silicate. Spectrographic analysis of samples taken at the final inspection is shown in Table IV. Sample numbers in the table correspond to the following locations in the loop.

- \# 9 Material removed from brine sample at the discharge of the column.
- \#10 Deposit around brine inlet ring of direct contact column.
- \#12 Brine discharge of column.
- \#13 Turbine nozzle block scrapings.
- \#14 Top of direct contact column.

As may be noted, with the exception of #10, oxides of silica and iron compose the principal constituents of these samples. This result was expected from previous history at the East Mesa Facility. Sample #10 was identified as the remains of a polycarbonate ring previously installed in the direct contact heat exchanger. Consequently no unexpected results were obtained from the analysis.

7.0 CONCLUSIONS AND RECOMMENDATIONS

1) Stable long-term operation of the direct contact heat exchanger coupled with the turbine was achieved in the test program.

2) Tests indicated possible liquid carryover as evidenced by high nozzle discharge coefficients measured during the turbine calibration. At increased values of apparent vapor superheat (subcritical calibration) the nozzle discharge coefficient approached that expected prior to the test.

TABLE IV

SPECTROGRAPHIC ANALYSIS OF SCALE SAMPLES

FROM DIRECT CONTACT POWER LOOP

"P. C." indicates "Principal Constituent".
The following are reported as oxides of the elements indicated.

	#9	#10	#12	#13	#14
Si	P. C.	0.5 %	P. C.	P. C.	8.5 %
Ca	8.5 %	.05	7. %	6. %	1.5 %
B	2.5	-	1.5	1.5	-
Fe	8.5	1.	7.5	5.	P. C.
Na	1.5	<.3	1.75	2.	<.3
Al	.25	.08	.75	.35	.6
Cr	.35	.02	.05	.03	.015
Mn	.05	.005	.04	.025	.5
Mg	.1	.015	.2	.2	.3
Pb	.01	.15	.85	.1	.1
Ni	.2	.05	.15	.06	.025
Sn	.005	.04	.015	.01	.04
Cu	.1	.6	.6	.2	.07
Zn	.08	.07	.2	.15	.4
Sr	.5	.003	.5	.5	.03
Ba	.2	.003	.08	.07	.007
Sb	-	.05	<.03	-	<.03
Ti	-	.004	.005	.002	.015
K	-	-	-	<.5	-
Be	.001	-	.002	.002	<.001
Mo	.006	-	.006	-	<.003
Co	.003	-	-	-	.001
Bi	-	-	<.001	-	-
Ag	-	<.001	<.001	-	<.001

(Subsequent muffling indicates the following residues after ignition at 1100°F: #9------84%; #10------2.5%.)

3) Turbine output power measured during the supercritical calibration maintained a higher level than expected while expanding into the isobutane saturated vapor dome.

4) Laboratory measurement of isobutane carryover in the brine effluent from the direct contact heat exchanger averaged 87 ppm isobutane fraction in the latter stages of the endurance run. This quantity amounts to 25% of the equilibrium solubility expected for the brine from well Mesa 6-2. Isobutane losses from the test loop in pump packing and the turbine seal invalidated a measure of isobutane losses based on quantities of isobutane added to the loop.

5) No heat transfer problem in the direct contact heat exchanger was found due to scale or erosion from the injected brine. Scale deposition was noted, however, in the turbine nozzles, particularly in the area of the supersonic expansion cone.

Based on these results, it is recommended that additional tests be performed with a demister vessel located between the heat exchanger loop and the turbine. In this fashion the effect of the liquid mist carryover on nozzle discharge coefficient and scaling can be evaluated. Also, the isobutane loss fraction should be verified by continued testing with an operational separator and stripper added to the basic heat exchanger test loop.

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1) H. R. Jacobs, R. F. Boehm, A. C. Hansen, "Application of Direct Contact Heat Exchangers to Geothermal Power Production Cycles", Project Review Dec., 1974 - May, 1977, Mechanical Engineering Department, University of Utah, Salt Lake City.

2) W. B. Surratt, G. K. Hart; ORO 4893-1, "Study and Testing of Direct Contact Heat Exchangers for Geothermal Brines", DSS Engineers, Inc., 1977.

3) E. F. Wahl, F. Boncher, "Theory and Practice of Near Critical Pressure Direct Contact Heat Exchanger", SAN/1076-1, Occidental Research Corporation.

4) K. E. Starling, <u>Fluid Thermodynamic Properties for Light Petroleum Systems</u>, Gulf Publishing Company, 1973.

APPENDIX A

LBL STARTUP AND SHUTDOWN CONTROL CIRCUIT

Startup

Switch 1SS is placed on ON. Five seconds later, 1TR energizes. 1CR, 2CR, 3CR, and 4CR energize, 1LT, 5LT, and 7LT illuminate, and the fan starts. Switches 3SS and 4SS can now be placed to ON to turn on the building lights and start other electrical equipment. Relay 9CR energizes, locking out the alarm horn and alarm light.

Switch 2SS can be placed to ON at this time. When 5LT (low shed pressure light) goes out, the turbine start switch, 3PB, can be depressed. When the generator shaft speed reaches 2800 rpm switch 6SS can be placed to ON. 4TR, 5CR, 6CR, 7CR, and 8CR energize and 9 CR de-energizes, putting the alarm back in the circuit. However, 1CR, 2CR, and 8 CR are energized and 3CR is de-energized so the alarm will not sound. When lube oil pressure rises above 18 psig, 7LT goes out. After 30 seconds, relay 4TR de-energizes. Relay 6CR stays energized, holding relay 9CR de-energized to keep the alarm on the line.

Emergency Alarm and Shutdown

If shed pressure rises too high or drops too low (0.1 inch of water is normal pressure), switch 2PS will energize relay 3CR, turn on the high or low shed pressure light, and sound the alarm but the system will continue to run. If lube oil temperature rises above 200^{o}F, 1TS will open causing 2LT to illuminate and 1CR to de-energize, sounding the alarm. If gearbox pressure rises too high, switch 1PS will open to de-energize 2CR, light 3LT, and activate the alarm. If the turbine overspeeds or the lube oil pressure drops below 15 psig, switch 1OS or 3PS will open causing 4CR or 5CR to de-energize and 6LT or 7LT to illuminate. This will de-energize relay 8CR to sound the alarm, 1SV to shut down the turbine, and 7CR for the DSS equipment. After 0.1 second, 2TR or 3TR will energize to prevent a restart without depressing reset button 3PB. When 3PB is depressed, 2TR or 3TR will de-energize. Relay 6CR also de-energizes.

High Isobutane Level or Detector Failure

If the hydrocarbon detector is shut off or fails, relay 2CRE will de-energize, shutting down the system. If the hydrocarbon level outside the building rises too high (above 50% LEL), the alarm on the detector will sound but the system will continue to run. If the isobutane level inside the building rises above 20% LEL, the alarm on the detector will sound and relay contact 1CRE will open, shutting down the system.

Alarm

After relay 1TR energizes, the alarm horn and light can be tested by depressing alarm test switch 2PB. If the alarm should activate because of a fault, the audible alarm can be shut off by depressing switch 4PB.

If the system is to be operated with the door open, shed pressure will drop, energizing relay 3CR. This will cause the alarm to activate. However, the alarm horn can be de-activated by opening key switch 5SS. If any other fault should then cause an alarm, relay 10CR will energize and the alarm horn will sound. It can then be shut off by depressing switch 4PB.

Normal Shutdown

Shutdown is the reverse of startup. The alarms will activate when the gearbox oil pressure drops below 15 psig and switch 1PB must be depressed to reset the alarm circuit.

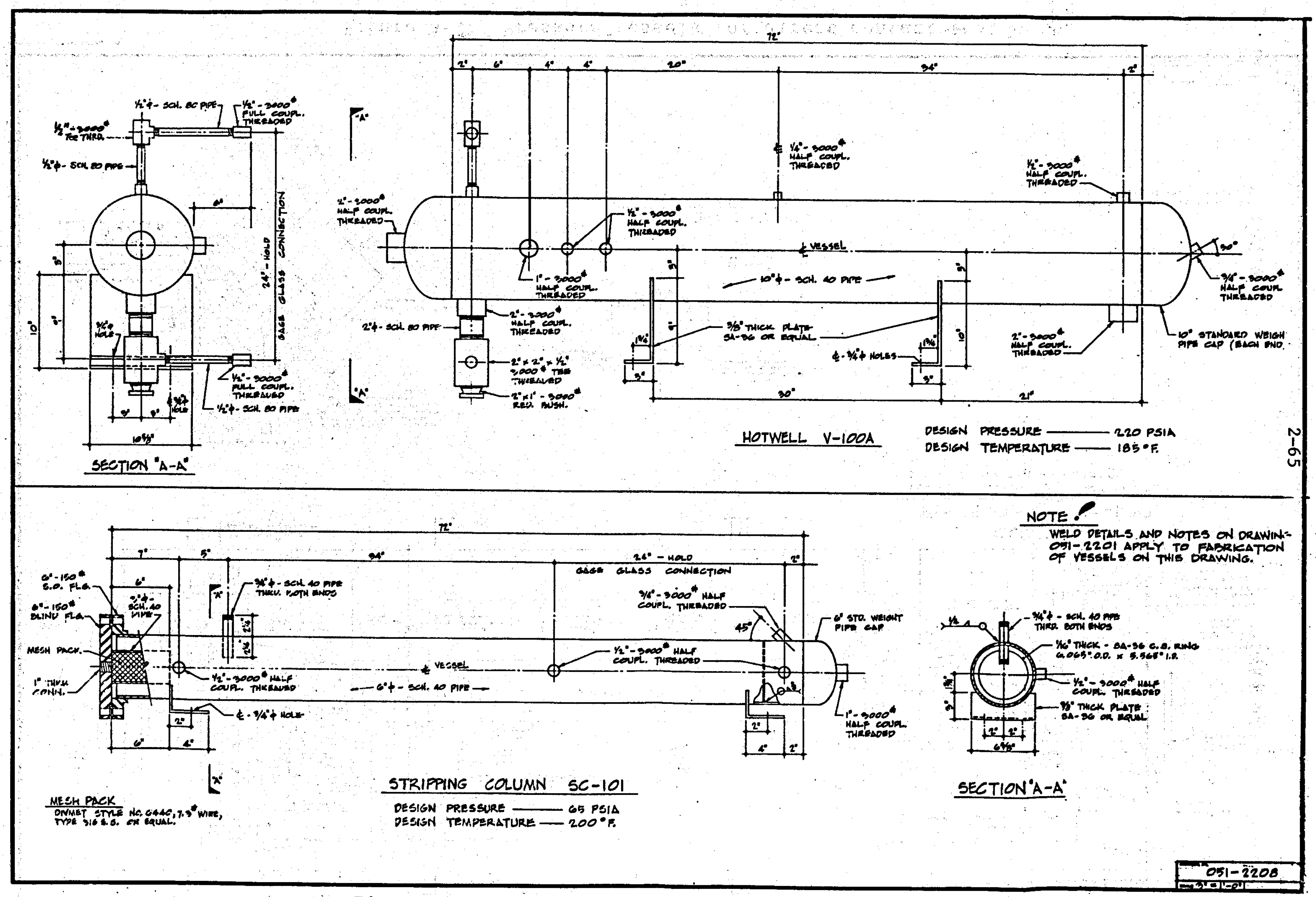

Figure A-1. Pressure vessel for direct contact test unit.

2-65

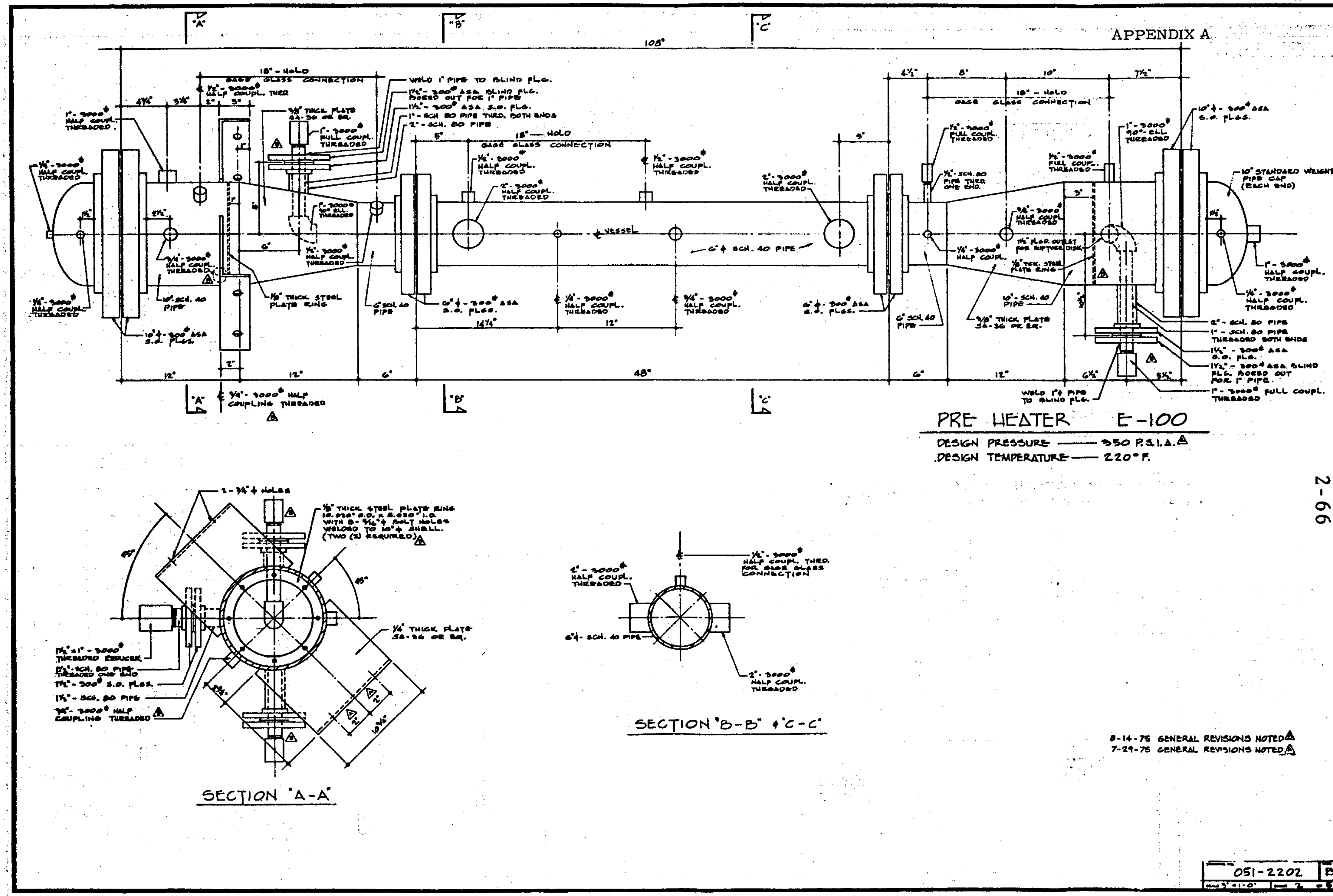

Figure A-2. Pressure vessels for direct contact test unit.

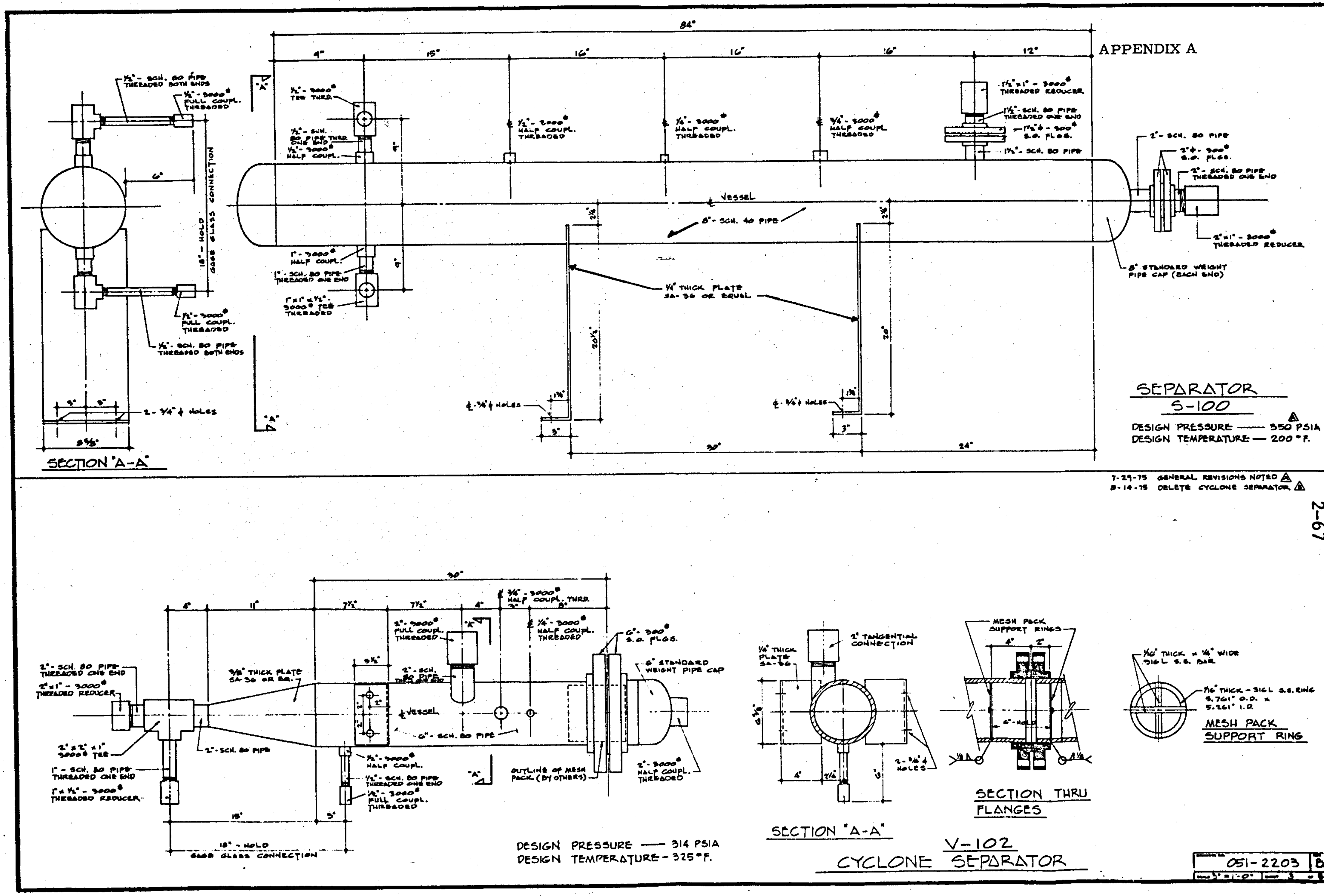

Figure A-3. Pressure vessels for direct contact test unit.

SECTION 3

DIRECT CONTACT HEAT EXCHANGER
10 kW POWER LOOP TEST SERIES No. 2

Prepared for:

Contract No. 4057702
Lawrence Berkeley Laboratory
University of California
Berkeley, California 94720

Prepared by:

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1.0 INTRODUCTION

In March and April, 1978, a second phase of testing was conducted at the East Mesa Component Test Facility near Holtville, California using a 10 kW Direct Contact Heat Exchanger Power Loop. The tests were designed to answer questions identified during earlier tests at the facility in July and August, 1977. (Phase I tests, Volume 1 of this report). Objectives of these Phase II tests may be summarized:

1) Improve turbine nozzle flow calibration.
2) Improve turbine performance calibration data at subcritical and supercritical conditions with the hairpin heat exchanger.
3) Evaluate turbine scaling problems associated with the DCHX with entrained moisture removed from the turbine vapor flow.

In addition, a separate loop was fabricated and information obtained as to the amount of CO_2 which could be removed from the incoming brine in a vent vessel. Concurrent data was also obtained as to the amount of isobutane which could be recovered in a separator and a stripper. This data was needed to aid in the design of the 500 kw pilot plant.

To achieve the above objectives and obtain information on isobutane loss fraction and CO_2 removal, a Phase II 200 hour endurance test was performed at East Mesa. For this endurance test the DCHX loop was modified in the following areas: 1) a demister vessel was located between the heat exchanger loop and the turbine, 2) a new subcritical nozzle was designed and fabricated for a lower pressure ratio, 3) an isobutane vapor flow orifice was added between the demister and the turbine inlet, 4) a brine flow orifice was added in the brine discharge line, 5) a calorimeter was added between the turbine inlet and exhaust piping, and 6) a vent vessel was added in a separate brine supply line.

Following the calibration tests performed with pure isobutane and a standard brine to isobutane heat exchanger, a 200 hour endurance run was completed using the existing hardware from the 500 hour endurance run. At the 42 hour, 120 hour, and 200 hour point the turbine nozzle was removed to check for scale buildup. At the end of the 200 hour run the nozzle and turbine blades were cleaned and an additional 40 hour endurance run was completed to determine what effect the scale buildup had on overall turbine efficiency and power output.

This report presents the results obtained during these tests and is submitted to fulfill the work defined in LBL Purchase Order No. 4057702.

2.0 SUMMARY

The geothermal power loop was modified for additional test effort addressing questions raised during Phase I testing (LBL-7036, Vol. 1). This report (Vol. 2 of LBL-7036) presents the results obtained during this testing. Additional instrumentation and a liquid knockout drum was added to the basic test loop. The power generation loop consisted of a brine pump supplying high pressure brine to a direct contact heat exchanger which transferred heat by intimate contact between the brine and the isobutane working fluid. The heated and vaporized isobutane was directed to a partial admission axial flow turbine which extracted energy from the working fluid to drive a 3600 rpm KATO generator. The working fluid then passed to a condenser and an isobutane feed pump and the cycle repeated. The heat exchanger and pump loop was designed and operated by DSS Engineers of Fort Lauderdale, Florida, and the turbo-generator package was supplied by Barber-Nichols Engineering Company. Details of the design geometries of this hardware are contained in Volume 1 of this report. The test program was conducted at the East Mesa Component Test Facility near Holtville, California.

Turbine performance was initially measured using a concentric tube "hairpin" heat exchanger in place of the direct contact heat exchanger. The purpose of these tests was to establish the turbine and system performance on a pure fluid not containing water vapor. Performance data was obtained on two subcritical nozzle configurations and a supercritical design. Data was taken at several flow rates and turbine inlet temperatures and at a range of turbine pressure ratios.

The two subcritical nozzle configurations differed primarily in the value of nozzle area ratio. The original nozzle was designed for an overall pressure ratio of 3.79. A nozzle expansion ratio of 1.567 was selected to match this pressure ratio. A peak turbine efficiency of 61% was measured at a pressure ratio of 4.25. This value of efficiency may be compared to an original estimate of 59% for this configuration. Data obtained at lower pressure ratios was found to duplicate efficiency measurements during Phase I tests. A second subcritical nozzle was designed with an expansion ratio of 1.306. The reduced nozzle area ratio compared to the original design was expected to reduce the optimum pressure ratio for the turbine. A maximum efficiency of 64% was achieved during these tests at a pressure ratio of 4.0. Slight improvement in nozzle efficiency due to surface finish or other factors could account for the difference in turbine efficiency noted between the subcritical nozzle configurations. Over the entire pressure ratio range examined, the new nozzle was found to outperform the original design, and was selected for use in the direct contact heat exchanger tests.

An objective of the subcritical turbine tests was to verify nozzle flow calibration particularly at different levels of superheat. An orifice type flow meter was installed in the turbine inlet line to provide a vapor flow measurement. Data obtained with this measurement provided much more reliable and consistent nozzle flow coefficients at levels of superheat from near zero to 60°F. Values of nozzle flow coefficient calculated from these data remained between 0.94 and 1.02 for the original nozzles and between 1.03 and 1.05 for the low area ratio nozzle.

Supercritical calibration tests were also completed using the hairpin heat exchanger during Phase II testing. Consistent data were obtained showing turbine efficiency and cycle performance as a function of turbine inlet enthalpy. Inlet enthalpy was varied to investigate the effect of vapor expansion close to the saturated vapor dome. A maximum turbine efficiency of 58% was measured consistently at high levels of inlet enthalpy. This efficiency may be compared to a design value of 52%. A reduction in efficiency to about 54% was finally noted when the turbine vapor expansion resulted in quality at the nozzle discharge. No reduction in efficiency was noted by expanding through a portion of the vapor dome as long as isentropic expansion to the turbine exhaust pressure resulted in at least saturated vapor conditions.

Cycle efficiency was a maximum at high inlet enthalpy. Performance declined consistently with inlet enthalpy even though the turbine inlet temperature remained fairly constant. Less available energy existed for the turbine expansion and the gross output power was reduced at lower values of inlet enthalpy.

The supercritical cycle can be used to improve utilization efficiency by eliminating the "pinch" temperature difference between the brine and working fluid and by minimizing the average temperature difference over the entire brine cooling curve. This advantage is offset, however, by the need to operate at high pressure levels corresponding to supercritical conditions. Consequently parasitic losses such as working fluid pumps and brine boost pumps are a large percentage of the output and high efficiency components are essential to show any net improvement over optimum subcritical cycles.

Turbine calibration tests were completed using the DCHX and the low area ratio subcritical nozzle at the beginning and end of the endurance runs. These tests showed good agreement with the hairpin data, with measured values of flow coefficient varying between 1.01 and 1.03. A maximum turbine efficiency of 59% was measured during the first calibration test at a pressure ratio of 3.54.

A 200 hour endurance run was started following the calibration tests. Both the DSS heat exchanger loop and the Barber-Nichols turbine-gearbox generator ran smoothly during the test. Only 10 days were needed to complete the 200 hour run. Inspections of the turbine component were made periodically during the endurance run. Very minor scaling was noted. noted up to 120 hours into the run. Between 120 hours and 200 hours significant scaling was formed, and a reduction in turbine efficiency from 54% to about 41% occurred. The nozzle flow coefficient dropped from 1.03 to a value of 0.92 during this time.

Following the 200 hour inspection the nozzle and rotor hardware was cleaned in a lye solution. A maximum turbine efficiency of 56% was measured after this cleaning, although the nozzle flow coefficient returned to a value of 1.02.

An additional 40 hour continuous run was then performed. Scaling was again noted, and apparently much more rapidly than during the previous test. At the end of the 40 hour run the turbine efficiency and flow coefficient dropped to values of 52% and 0.98, respectively.

The remainder of the piping and loop hardware was relatively clean, with no evidence of distress noted.

Due to the higher than anticipated condenser pressure observed during the DCHX test series, a simplified flash experiment was performed. Brine from well Mesa 6-2 was subjected to a controlled pressure drop, and the CO_2 and additional vapor generated was vented from a flash vessel. Flow pressure and temperature instrumentation was provided to set and monitor test conditions. Results of these tests showed that a significant amount of free and dissolved CO_2 can be removed from the brine stream with moderate flash temperature differences in the range of 2^oF to 7^oF. Also, significant carbonate scaling was observed at temperature drops of about 10^oF. At 5^oF extended operation of the flash vessel showed essentially no scaling problem.

These tests and the significance of the noncondensible accumulations problem has spurred additional detailed chemical analysis and field tests. Results of these investigations will be reported at a later date.

In conclusion, from a heat transfer and operability standpoint, the direct contact heat exchanger has demonstrated its viability as an alternative to conventional heat exchange equipment in an operating geothermal power generating loop. Three basic problems were identified, however, which require solution to improve the practical long term operation of the DCHX power loop. These may be summarized:

1) CO_2 or noncondensible contamination in the isobutane condenser.
2) Isobutane recovery following the direct contact heat exchanger.
3) Mineral carryover in working fluid vapor from the direct contact heat exchanger.

Reasonable solutions to these problems are proposed and are included in the design philosophy of the 500 kw pilot plant project.

3.0 POWER EQUIPMENT MODULE

The following is a partial list of equipment contained in the power equipment module:

1) Turbine, housings and connecting piping.
2) Gearbox.
3) Torque meter and readout.
4) Generator.
5) Speed control.
6) Pressure, temperature and speed instrumentation.
7) Safety controls and interlocks.
8) Air cooled resistance load bank.

Items 3 through 7 were located in a pressurized enclosure to circumvent any possibility of an explosive isobutane-air mixture in the vicinity of the generator. Item 8 was remotely located a safe distance from the isobutane source. The power equipment module was skid mounted for ease in transportation and was installed next to the direct contact heat exchanger test loop at the East Mesa facility. (See Figures 1 and 2.)

3.1 DESIGN PARAMETERS

3.1.1 Turbine

During the Phase I testing the turbine exhaust pressure was 95 to 105 psia rather than 83 psia as anticipated during the original design phase. To take this into account a new subcritical nozzle was designed with a reduced area ratio (1.3 instead of 1.57) corresponding to the increased turbine exhaust pressure (see Table I for design parameters).

TABLE I

NOZZLE DESIGN PARAMETERS

	Subcritical (Single Nozzle)
Number of nozzles	1
Throat diameter	0.306
Exit diameter	0.3521
Area ratio	1.306
Angle	16°
Edge thickness	N/A
Admission	27.34°
Exhaust	55°
Pressure ratio	3.32

The turbine was a partial admission axial flow design incorporating a single rotor and nozzle blocks with three nozzle geometries for subcritical and supercritical cycles. Design conditions and turbine geometry were presented in Volume 1 of this report.

3.1.2 Gearbox

The gearbox was a standard double reduction type (GR=6.05) manufactured by Barber-Nichols. Lubrication was provided by an internal chain driven oil pump, and the oil temperature was controlled with a water cooled oil cooler mounted on the gearbox housing.

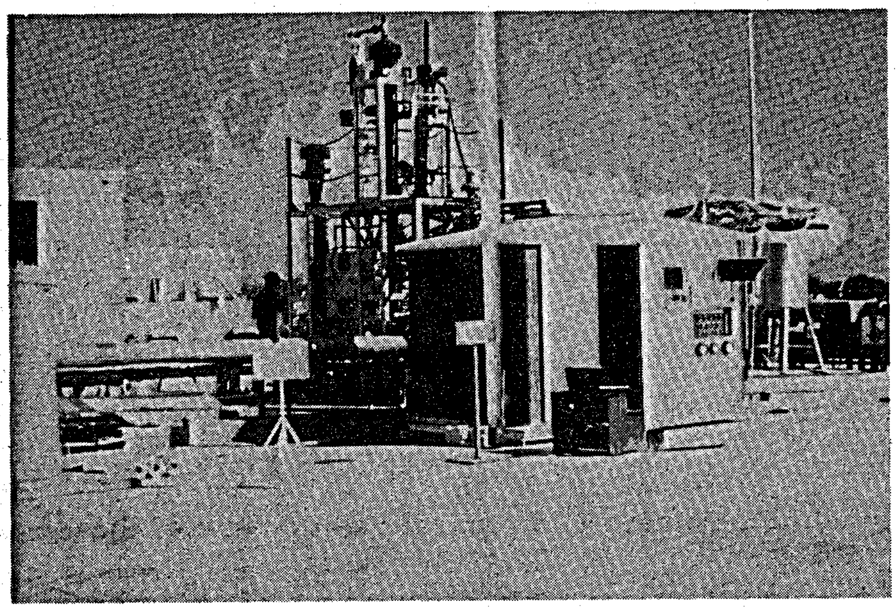

Figure 1. Generator shed and DSS loop.

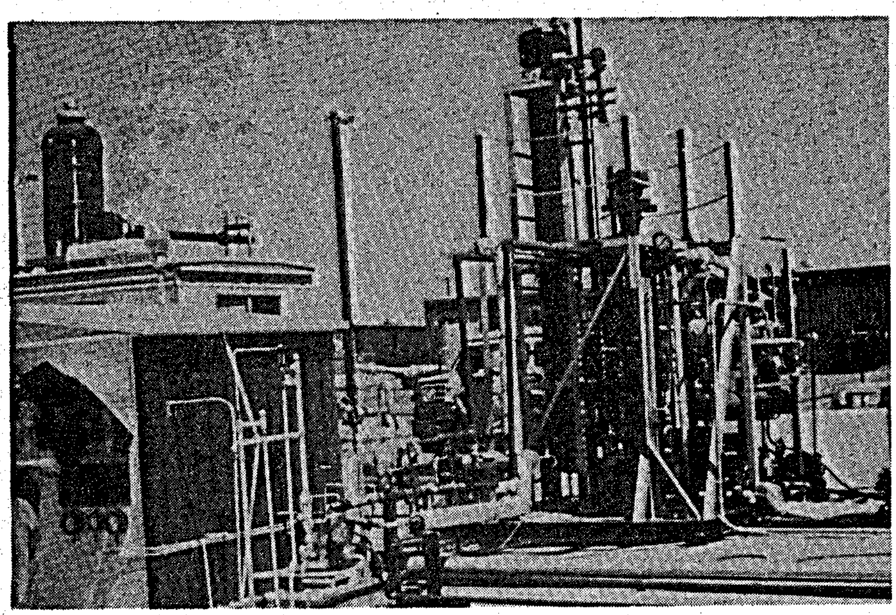

Figure 2. Generator shed and DSS loop.

4.0 LOOP CONFIGURATION

4.1 DCHX TEST LOOP

4.1.1 Test Loop Components

The test loop consisted of the following major components: a direct contact heat exchanger (Figure 3) which was a combination preheater and boiler to supply isobutane vapor to the turbine, an isobutane separator to separate any isobutane from the brine leaving the DCHX, a stripping column to remove the isobutane from the brine and return it to the hotwell, a condenser, a hotwell, two circulation pumps, one for isobutane circulation and one for brine feed to the DCHX, and a demister vessel to remove entrained water and isobutane liquid from the vapor entering the turbine. The general relationship of these components is illustrated in the P & I diagram, (Appendix A). .

4.1.2 DSS Loop Changes

The major changes to the DSS loop included an isobutane liquid bypass control valve to automatically control the isobutane liquid flow rate to the DCHX, a hotwell water level and water drain control, and an additional brine pump to be used in conjunction with the separator and stripping column.

4.1.3 Barber-Nichols Loop Changes

The major changes to the Barber-Nichols loop included a demister (see Figures 4 and 5) installed between the heat exchanger loop and the turbine to remove liquid from the IC_4/H_2O vapor entering the turbine, a calorimeter installed upstream of the turbine inlet to evaluate the enthalpy of vapor flow to the turbine, a flow orifice in the line between the demister and the turbine inlet to measure the vapor flow rate to the turbine, and a flow orifice in the brine discharge line to measure brine flow rate through the hairpin heat exchanger during calibration testing. A vent vessel was also added in a separate brine supply line to determine how much CO_2 could be removed by flashing the brine. (See Figure 6.)

4.2 INSTRUMENTATION AND SAMPLING

4.2.1 Flow Measurement

A "Daniel Industries" M-30RW orifice flow tube and a "Barton" model 200 (0-400" W.C.) ΔP gage were installed in the heat exchanger loop between

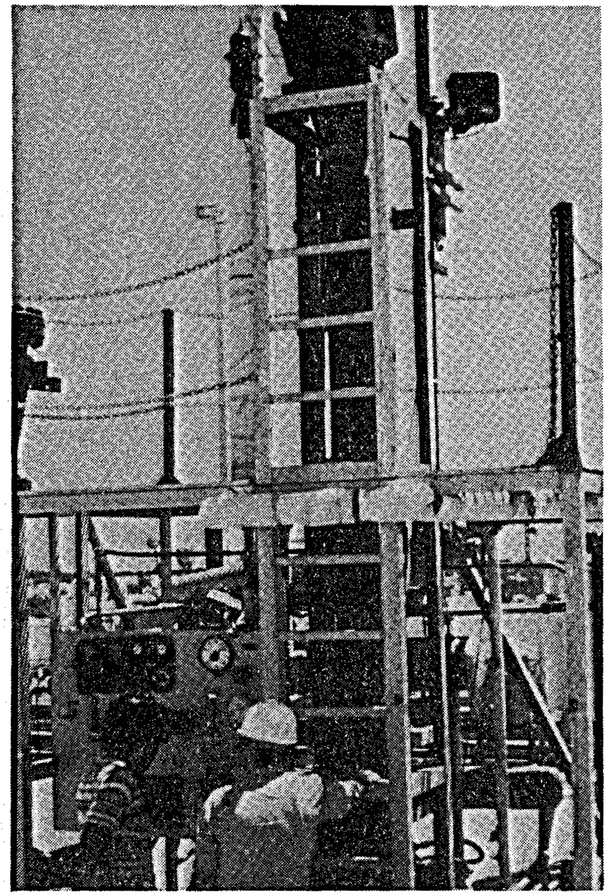

Figure 3. Direct contact heat exchanger.

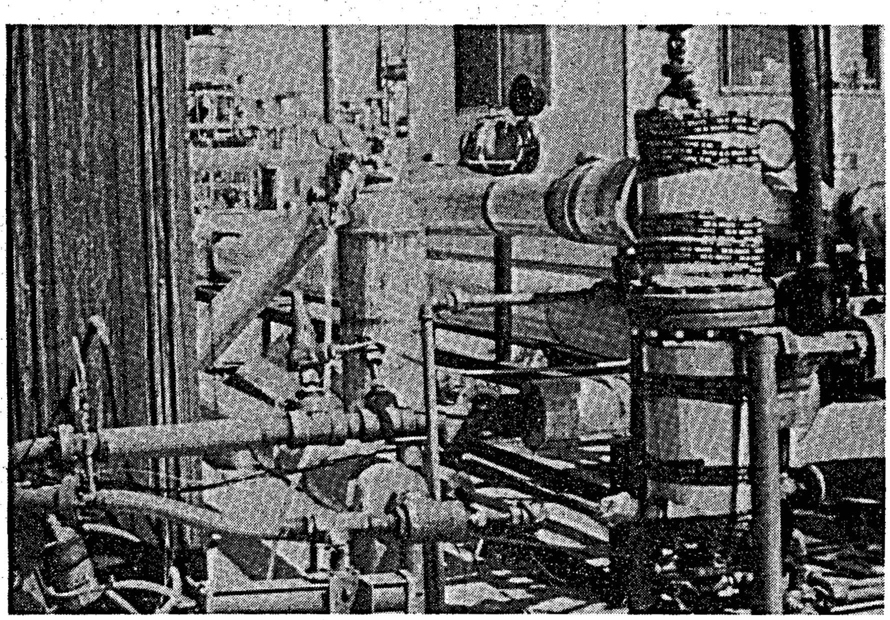

Figure 4. Demister vessel.

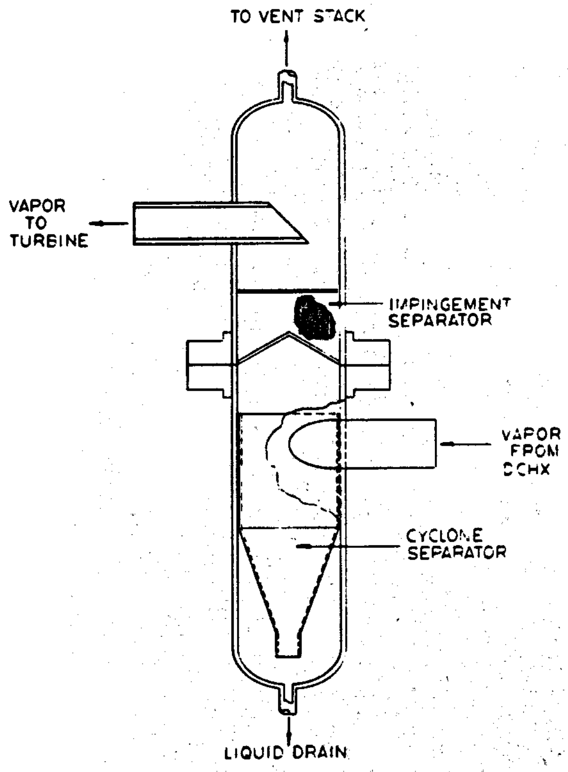

Figure 5. Demister vessel.

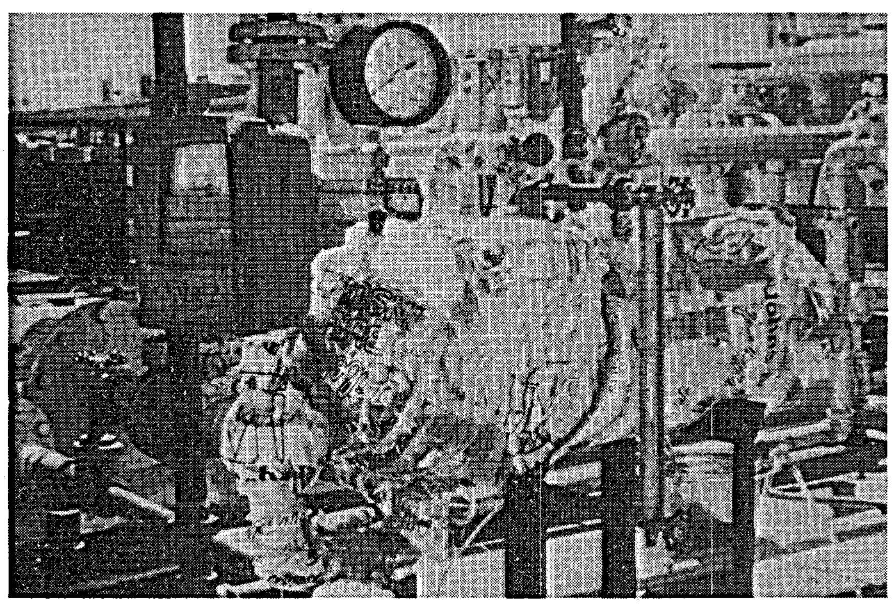

Figure 6. CO_2 flash tank.

the demister and the turbine inlet. The flow tube had a 0.70 inch orifice for subcritical isobutane testing and a 0.50 inch orifice for supercritical testing. The flow tube was used to verify turbine vapor flow rates for nozzle flow calibrations and to enable a heat balance around the hairpin heat exchanger.

A model M-30RW orifice flow tube and a "Barton" model 202A ΔP recorder (0-400" W. C.) were installed in the brine discharge line from the hairpin heat exchanger. The flow tube had a 0.40 inch orifice for subcritical testing and a 0.530 inch orifice for supercritical testing. The flow tube in the brine discharge line was used to enable a heat balance around the hairpin heat exchanger.

A calorimeter was fabricated from an 8 inch section of schedule 40 seamless pipe with schedule 40 pipe caps welded to each end. Instrumentation consisted of a model AA14P "Weksler" 0-200 psig gage and a model NSA-2-2-2-DH1-T-100T-2 1/2-304SS "Omega" thermocouple assembly. The calorimeter was used during calibration and endurance testing to evaluate the enthalpy of the vapor flow to the turbine nozzle(s).

4.2.2 CO_2 Handling

A small flash tank was installed adjacent to the DSS loop to determine how much CO_2 could be removed by preflashing the brine before it entered the brine feed pump. Brine was supplied to the flash tank from the well Mesa 6-2 supply line on the manifold. Brine samples were taken at the entrance and the exit of the brine flash tank and were analyzed by the East Mesa Chemistry Laboratory. To take a sample the flash tank brine conditions were first stabilized to obtain specified flow rate and temperature difference between the brine inlet and the fluid temperature in the tank with the tank half full. Brine flow rate and fluid level were controlled by manually adjusting the inlet and discharge throttle valves. The amount of brine flash and the pressure level in the vessel were controlled via the CO_2 and steam vent. When conditions had remained stable for at least 30 minutes samples were taken. To keep the brine from flashing in the sample containers, the sample liquid flowed through cooling coils immersed in ice water to lower their pressure and temperature below the flash point.

5.0 FIELD OPERATIONS

5.1 LOOP OPERATION

The test unit provided for the continuous circulation, contact, and separation of brine and isobutane. Heat from the geothermal brine was transferred to the isobutane in a single direct contact heat exchanger.

Isobutane liquid was pumped from the hotwell to the direct contact heat exchanger where it was heated and vaporized by counter current contact with the brine. The isobutane vapor and a small amount of water vapor then flowed to a demister vessel where any entrained liquid was removed. The "dry" vapor mixture was then expanded to condensing pressure through the Barber-Nichols turbine. The vapor was condensed and returned to the hotwell where the water condensate was continuously drained off. The liquid removed in the demister vessel was also returned to the hotwell where its water was drained off.

Hot brine from well Mesa 6-2 was pumped from the geothermal supply manifold to the direct contact heat exchanger where heat was extracted from it to heat and vaporize the isobutane working fluid. The high pressure brine was then reduced to atmospheric pressure and dumped into an open tank where entrained isobutane was flashed out. Except for a brief period of time when the separator and stripping column were being tested no attempt was made to recover the isobutane entrained in the brine leaving the direct contact heat exchanger. The brine, after leaving the open flash tank, was returned to a settling pond for reinjection.

5.2 CALIBRATION TESTS

The primary goal of the calibration tests was to operate the turbine and obtain data with both the original 1.57 area ratio subcritical nozzle and the new 1.3 area ratio subcritical nozzle as well as the original supercritical nozzle. The subcritical data was obtained at varying degrees of superheat, isobutane flow rates, and brine flow rates. Several turbine speeds were run to identify and measure peak power output. Turbine output power and efficiency, and nozzle flow coefficient were correlated with various cycle conditions run. The effect of turbine pressure ratio on overall efficiency was evaluated for each of the subcritical nozzles in these tests. With the demister in operation performance of the original nozzle was verified and the original turbine inlet state point was checked with the calorimeter and by performing a heat balance on the hairpin heat exchanger to determine how much heat was removed from the brine.

As with the subcritical nozzles, data was obtained for a range of speeds and isobutane flow rates with the single supercritical nozzle. Also, output power was determined with varying degrees of vapor expansion into the saturated vapor dome. The initial turbine inlet state point was checked by correlation with the heat removed from the brine.

Upon completion of the subcritical and supercritical isobutane calibration tests, the results of the subcritical calibrations were used to determine which of the subcritical nozzles to use for the DCHX endurance run and at what conditions of turbine inlet temperature, pressure and isobutane flow rate to operate during the endurance test. Since the efficiency of the turbine was approximately 4 points higher with the new 1.3 area ratio subcritical nozzle, it was used for the DCHX endurance run.

Direct contact heat exchanger calibration tests were run before and after the 200 hour endurance run and during the 40 hour endurance run. These calibration runs provided baseline data to be used in determining performance degradation due to scale buildup in the nozzle and on the rotor blades. The calibration tests were used to set operating parameters such as turbine inlet temperatures and pressures and turbine speeds (for maximum output horsepower) to be used during the 200 hour endurance run.

5.3 ENDURANCE RUNS

Two separate endurance tests were run, a 200 hour run during which the scale buildup on the nozzle and turbine blades was monitored by inspecting the turbine component three times during the run and a 40 hour endurance run which was performed after the nozzle block and rotor blades were descaled.

The primary goal of the 200 hour endurance run was to observe any difference in scaling in the turbine nozzle and rotor with the moisture removed in a demister vessel as compared to the original 500 hour endurance run when a mesh pack inside the DCHX was the only means to "dry out" the vapor.

The endurance runs began at 1342 hours on March 25, 1978, and were completed at 0700 hours on April 7, 1978. During this time period the DCHX loop and the turbo-generator accumulated a total of 240 hours running time. Six personnel, three from Barber-Nichols Engineering Company and three from DSS Engineers, manned the test loop on a 24 hour-a-day, 7 days-a-week basis. Unlike the 500 hour endurance run during which a Barber-Nichols employee and a DSS employee alternated 12 hour shifts, during the 200 hour and 40 hour endurance runs one Barber-Nichols and one DSS employee monitored the loop during each 8 hour shift.

TABLE II

Operating Parameters for the Endurance Runs

Brine inlet temperature to DCHX	322 ± 3°F
Isobutane outlet temperature from DCHX	230 ± 5°F
DCHX operating pressure	300 ± 10°F
Brine flow rate	5.4 ± 0.3 gpm
Isobutane flow rate	9.5 ± 0.4 gpm
Turbine speed	2750 ± 5 rpm
Isobutane level in hotwell	Greater than 1/2 full
Condenser pressure	65 ± 10 psig

These parameters resulted in:

Turbine inlet temperature	222 ± 3°F
Turbine inlet pressure	300 ± 20 psig
Turbine exhaust pressure	65 ± 10 psig

During the endurance run data were recorded once each hour. These data included gearbox lube oil temperature and pressure, gearbox bearing temperature, turbine inlet and exhaust temperature and pressure, isobutane orifice temperature, pressure and differential pressure, DCHX brine and isobutane inlet and outlet temperatures, demister temperature, hotwell temperature, calorimeter temperature and pressure, brine orifice differential pressure, turbine gearbox output shaft torque and rpm, and brine and isobutane liquid flow rate.

A substantial improvement in loop operation was noted during the 200 hour test compared to the phase I testing. Operation "up time" amounted to 92% of the total time available during the test.

During the endurance runs, the DCHX loop and the turbo-generator were shut down a total of 13 times for various reasons. These reasons are summarized as follows:

No. of Times Shutdown	
5	Gastech hydrocarbon detector drifting
2	Change wells
2	Normal shutdown at end of endurance run
1	Inspect turbine nozzle
1	Low wellhead temperature at well Mesa 8-1
1	Site power failure
1	Air compressor failure

The largest single shutdown problem was the "Gastech" hydrocarbon detector. The potentiometer circuit for the zero set point of the detector which monitored hydrocarbon level inside the pressurized shed kept drifting and shutdown the entire system if the drift was more than 5% downscale or 20% up scale on the detector meter. At no time was excess isobutane level found in the vicinity of the power generating system.

The loop was shut down twice to change wells. Well 6-2 was used for the first 42 hours of the endurance run. It was decided to use well 8-1 for the remainder of the endurance run since it is to be used to supply heat to the 500 kw pilot plant. The loop operated for approximately 5 hours on well 8-1. When its wellhead temperature dropped to 295^{O}F the loop could no longer be controlled. Therefore, the remainder of the endurance run was completed with well 6-2 which had a loop inlet temperature of 320^{O}F.

Unlike the 500 hour endurance run, there were no shutdowns due to pump packing failures. The brine pump was modified to supply additional cooling water to the packings. This increased their life to 200 hours. The isobutane pump packings were inspected at each shutdown and were replaced between the 200 hour and the 40 hour endurance runs.

A total of 596 pounds of isobutane was added to the DSS loop during the 200 hour endurance run. Figure 7 shows the cumulative isobutane added to the loop as a function of accumulated operating hours. As can be seen the rate

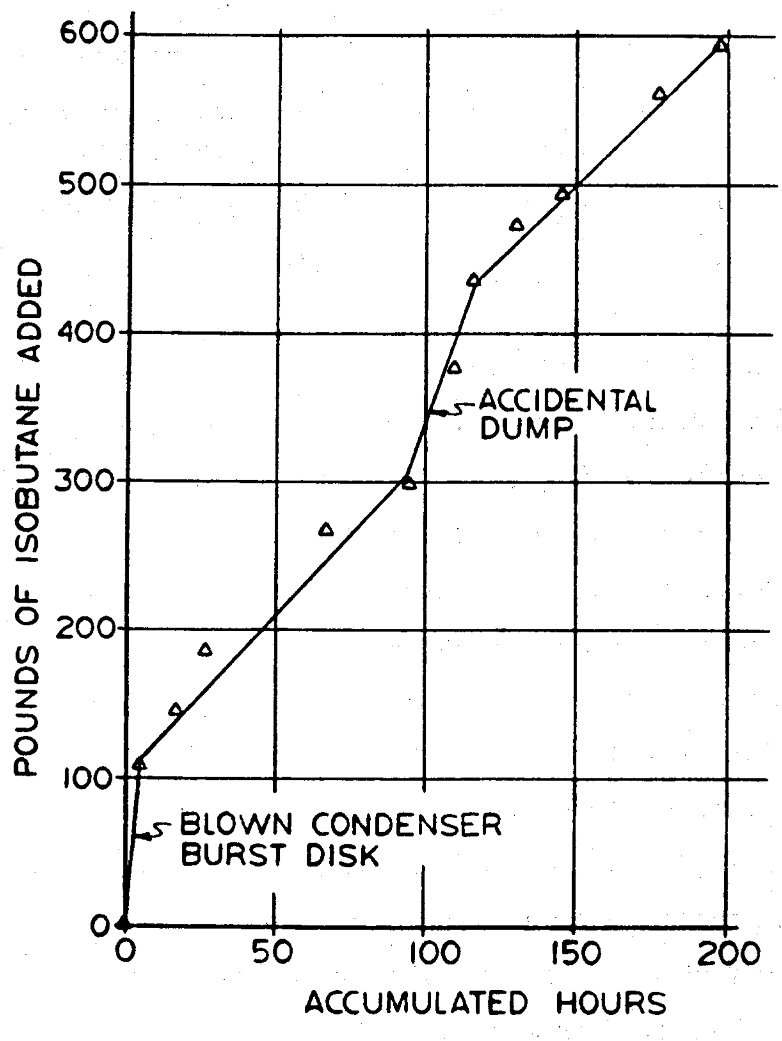

Figure 7. Isobutane utilization rate.

remained constant except at two points. During startup for the 200 hour endurance run the condenser burst disc ruptured. At the 99 hour point the DSS loop was accidentally shutdown while resetting the "Gastech" and a large quantity of isobutane was vented while restarting the loop. The make-up isobutane was added over a 15 hour period. The average isobutane usage throughout the endurance run was 2.2 pounds per hour.

5.4 ON LINE TIME

During the series 1 500 hour endurance run the system was started at 0940 hours on July 26, 1977 and the endurance run was completed at 1630 hours on August 24, 1977. This is a total of 703 hours. During this period the system accumulated a total of 500 hours of running time for an on line rate of 71%. During the series 2 200 hour endurance run the system was started at 1342 hours on March 25, 1978 and the endurance run was completed at 1509 hours on April 3, 1978. This is a total of 217 hours. During this period the system accumulated 200 hours running for an on line rate of 92%. If the time required to change from well 6-2 to well 8-1 and back to well 6-2 is deducted (a total of 8 hours) since this time was not due to a component failure, the on line rate increases to 95%. There were also a number of nuisance shutdowns due to the hydrocarbon detector circuits drifting. If these are deducted from the total down time, the on line rate is further increased to 97%. None of the shutdowns were due to Barber-Nichols or DSS equipment failures. Unlike the 500 hour endurance run, the pumps and turbine gearbox ran trouble free for the entire 240 hours of both endurance runs. During the 40 hour endurance run the system did not shutdown until the run was complete.

5.5 TEARDOWN INSPECTION

Teardown inspections of the turbine nozzle and rotor were conducted at the 42 hour, 120 hour, 200 hour and 240 hour points in the two endurance runs.

The nozzle block and turbine rotor were photographed after the 200 hour endurance run to show the area of scale buildup (see Figures 8, 9, and 10). Samples of scale material were removed from the nozzle block and turbine wheel. These samples were sent to LBL for analysis.

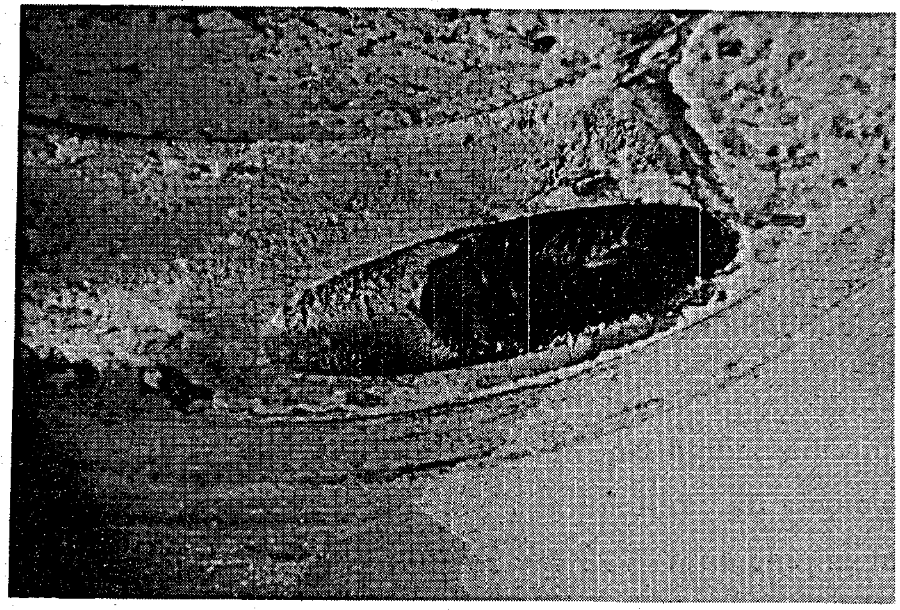

Figure 8. Subcritical turbine nozzle exit.

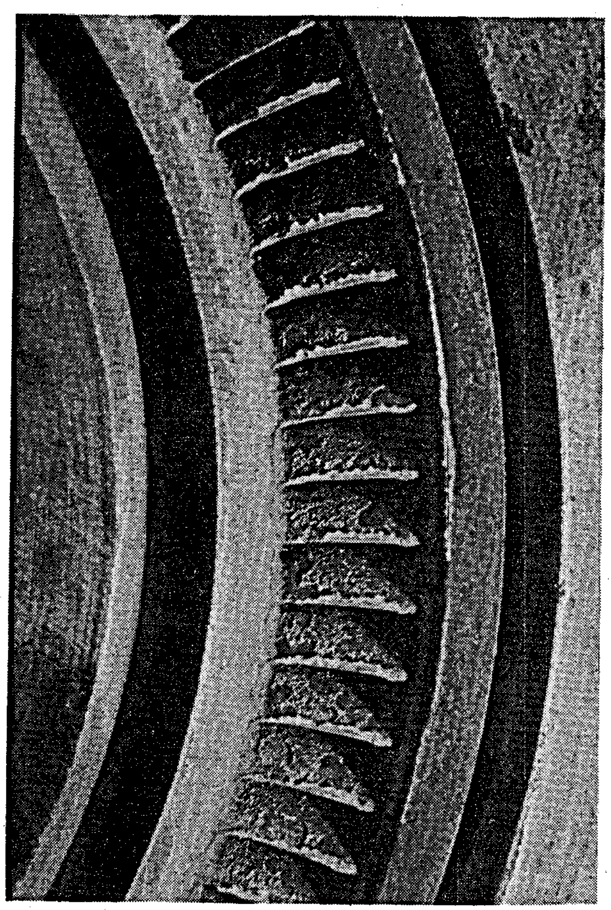

Fig. 9. Rotor inlet blade leading edge.

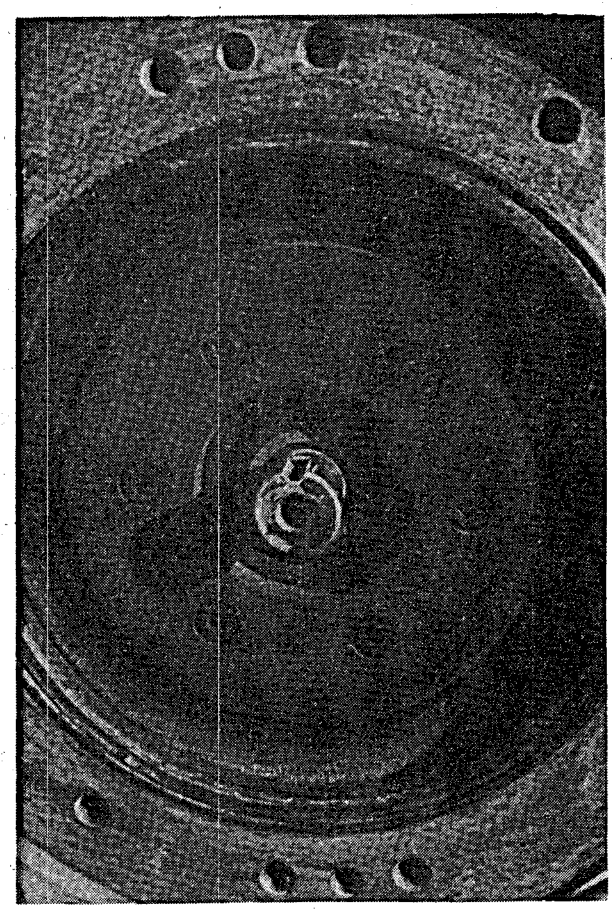

Fig. 10. Turbine exhaust housing.

6.0 TEST RESULTS

6.1 TURBINE CALIBRATION - CONVENTIONAL HEAT EXCHANGER

6.1.1 Subcritical Calibration

Calibration tests were performed with two subcritical nozzle configurations utilizing the hairpin heat exchanger. The tests were designed to verify the turbine efficiency data obtained during the previous test series (Ref. 1) with the original nozzles (area ratio = 1.57) and to obtain more consistent values of nozzle flow coefficient. The subcritical data was obtained at varying degrees of superheat from 10-60°F, isobutane flow rates of 8, 10, and 11.5 gpm, and brine flow rates of 5.5 and 8 gpm.

The calibration tests with the second subcritical nozzle (area ratio = 1.3) were also used to evaluate the effect of area ratio on turbine performance and to select the subcritical nozzle to be used during the DCHX tests. Since greater pressure ratios were achieved during these calibration tests because of reduced cooling water temperatures and condenser pressures as compared to the previous tests, it was not obvious which nozzle would perform the best with the DCHX.

As with the previous tests, turbine efficiency, turbine output horsepower, velocity ratio, superheat, nozzle flow coefficient, and pressure ratio were calculated at each data point using a program written for a Hewlett-Packard 9825A calculator. Isobutane thermodynamic parameters were calculated using equations in the program which were obtained from Starling (Ref. 2).

The turbine output power was obtained by calculating gearbox output horsepower from the measured values of shaft rpm and output torque and adding the gearbox horsepower loss obtained during laboratory calibration of the gearbox. The isobutane flow rate was measured with an orifice flow tube and a differential pressure gage. The nozzle flow rate (Wn) in pounds per hour was obtained using the equation:

$$Wn = 359CFd^2FaY\sqrt{hw\,\gamma}$$

where: C = the coefficient of discharge

F = velocity of approach factor $(1/\sqrt{1-\beta^4})$

(β = orifice diameter/pipe inside diameter) d = orifice diameter

Fa = thermal expansion coefficient for the orifice

Y = an expansion factor for square edged orifices, ratio of the discharge coefficient for a gas to that for a liquid at the same Reynolds number.

hw = differential pressure gage reading in inches of water

γ = fluid density

Measured turbine inlet temperature and pressure were used to calculate inlet enthalpy and entropy. At reduced pressures, enthalpy and density were calculated for an isentropic expansion. A maxima of the product of density and vapor spouting velocity ($\sqrt{2gJ\Delta h'}$) determined the nozzle throat pressure and consequently the nozzle ideal flow rate. The nozzle flow coefficient (Cd) was then calculated as:

$$Cd = \frac{\text{measured isobutane flow rate}}{\text{nozzle ideal flow rate}}$$

The above approach was necessary since the isobutane expansion occurred close to the saturated vapor dome, and significant compressibility effects were expected for the vapor expansion. Turbine available energy was calculated by continuing the isentropic expansion to the measured turbine exhaust pressure.

The turbine efficiency was calculated from:

$$\eta t = [(HPo + HPg)/\dot{w}\,\Delta h'] \quad 550/778$$

where: HPo = generator input shaft power

HPg = measured gearbox losses

$\dot{w}$ = measured isobutane flow rate, lb/sec

$\Delta h'$ = turbine available energy, Btu/lbm

Velocity ratio was calculated from:

$$U/C_o = U/\sqrt{2gJ\Delta h'}$$

where: U = turbine pitch line velocity, ft/sec.

Measured values of turbine efficiency are shown plotted against velocity ratio for the two subcritical nozzles in Figures 12 and 13. A comparison of the results obtained with both nozzle configurations is shown in Figure 11. A peak efficiency of 61% was measured with the original nozzle configuration at a turbine pressure ratio of 4.5. Data at lower pressure ratios, 2.8 to 3.5, showed the same efficiencies as that obtained during the first test series. Efficiency data with the 1.3 area ratio nozzle is improved over the original nozzles by about 4 points. The original nozzle block consisted of a two nozzle set with an area ratio of 1.57 and a design pressure ratio of 3.79. The 1.3 area ratio nozzle block consisted of only one nozzle with a design pressure ratio of 3.32. The new lower pressure ratio nozzle was designed and fabricated because during the 500 hour endurance run the actual turbine exhaust pressure was 95 to 105 psia instead of 85 psia as anticipated. As expected, the maximum efficiency occurs at a lower pressure ratio corresponding to the reduced expansion ratio. The best efficiency point for this configuration may be seen to occur at a pressure ratio of 4.0.

The reduced expansion ratio was expected to improve turbine efficiency at low pressure ratios, but the overall improvement observed in the test was higher than expected. Slight improvement in nozzle efficiency due to better surface finish or alignment could account for the difference. Because the primary goals of the test were centered around operation of the turbine with the direct contact heat exchanger no attempt was made to identify the reason for the improved efficiency.

Values of nozzle flow coefficient calculated from these data were more consistent compared to the previous tests. Over a wide range of superheat and pressure ratios the calculated nozzle flow coefficient remained between 0.99 and 1.02 for the original nozzles and between 1.03 and 1.05 for the low area ratio nozzle. Consistent values of flow coefficient greater than 1.0 are felt to be the result of inaccuracies in measuring actual nozzle throat areas or uncertainties in isobutane vapor thermodynamic properties. Turbine efficiency data were also much more consistent during these tests. Primary differences between the test

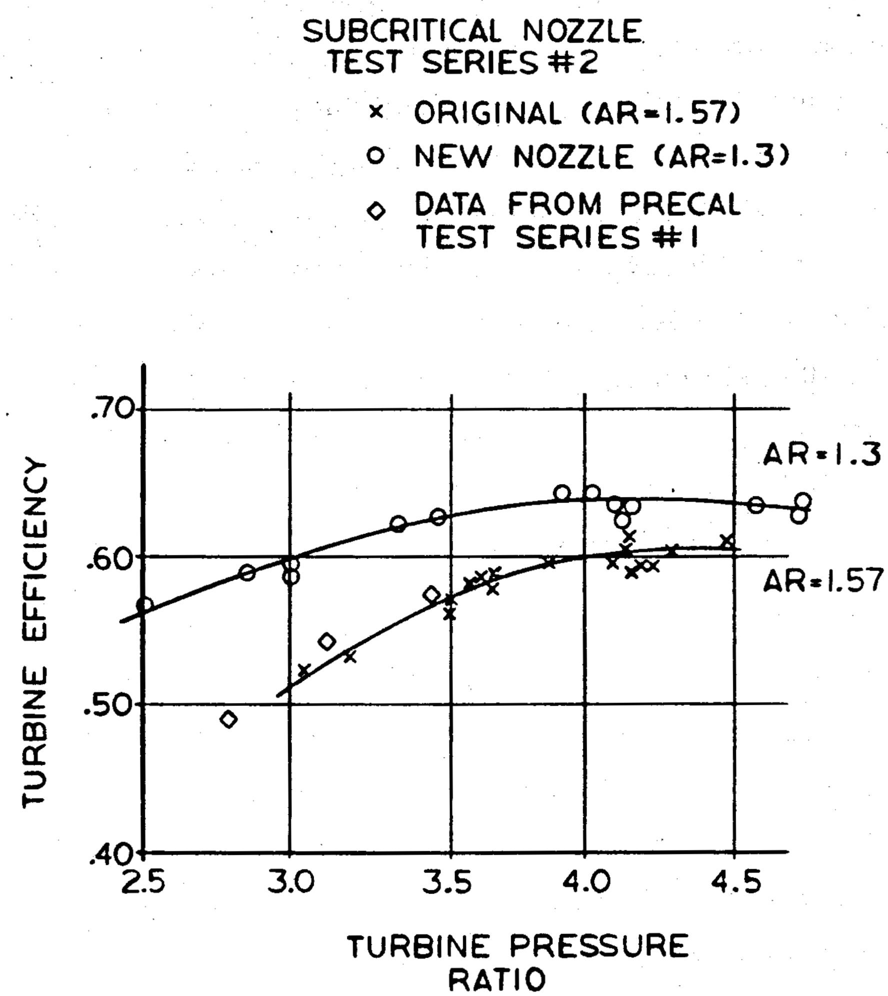

Figure 11. Pure IC_4 calibration test (using hairpin heat exchanger).

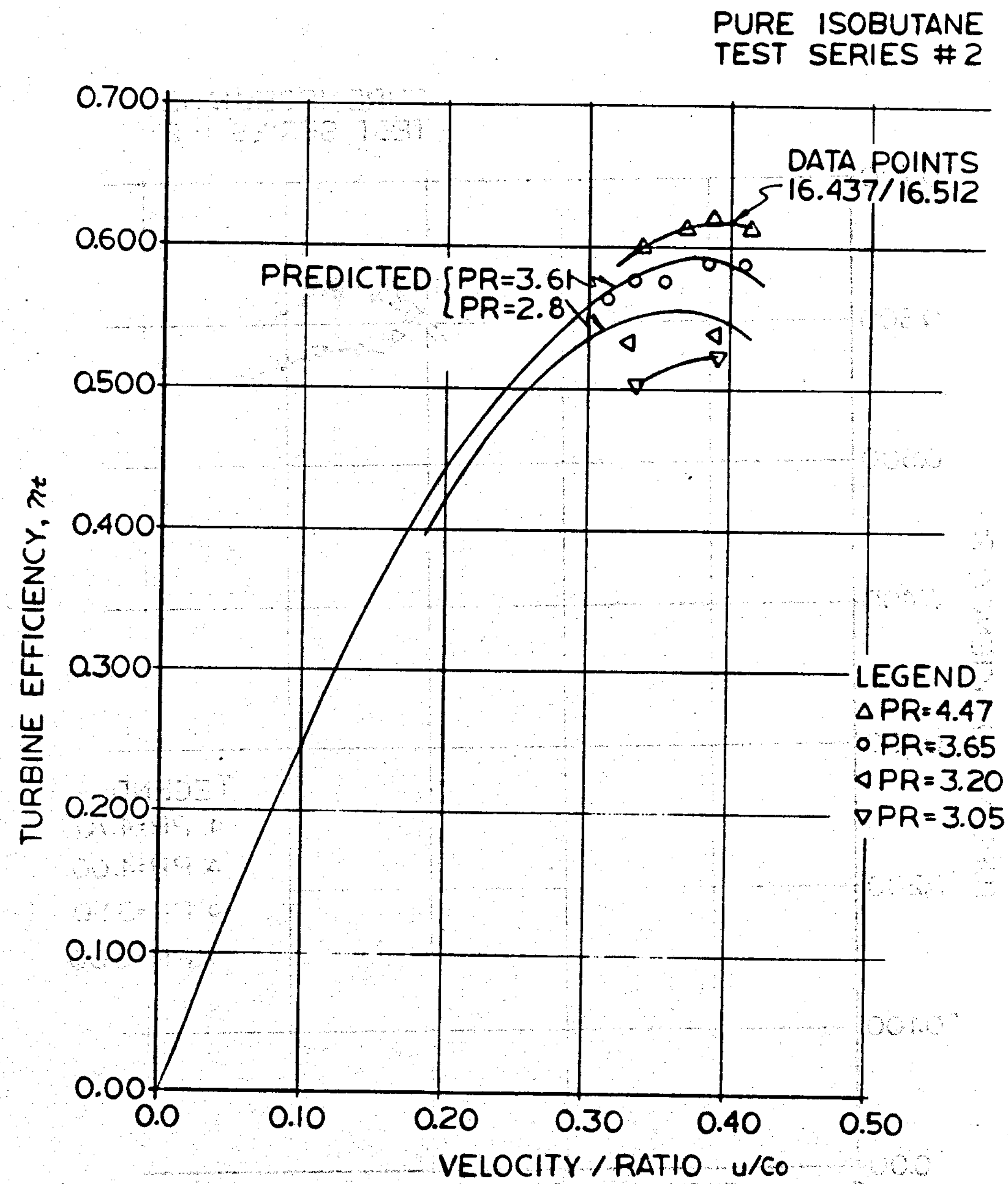

Figure 12. Subcritical calibration tests; efficiency vs. u/Co nozzle AR = 1.57 (two nozzles).

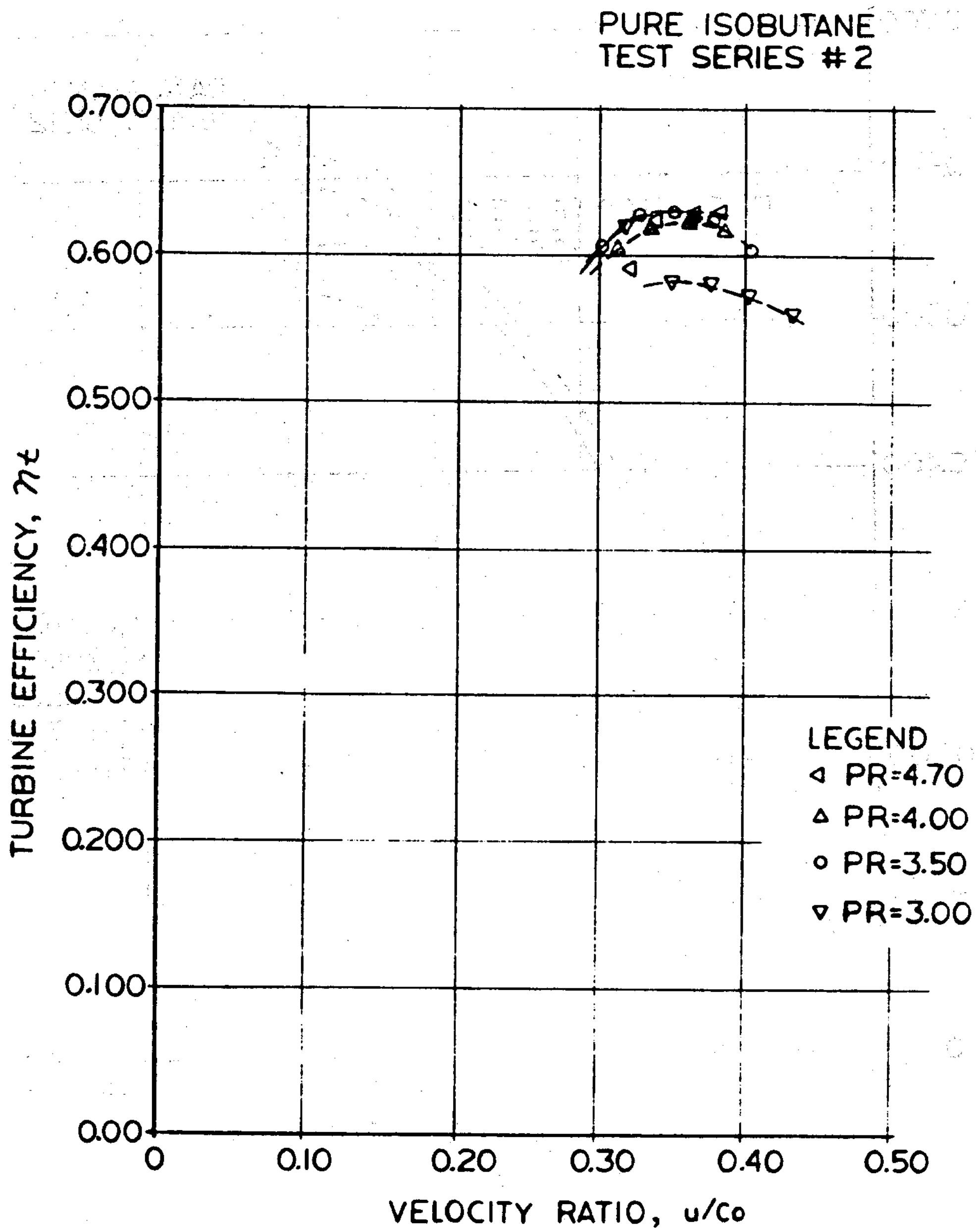

Figure 13. Subcritical calibration tests; efficiency vs. u/Co nozzle AR = 1.30 (single nozzle).

series was a liquid knockout vessel installed between the heat exchanger and turbine and also the isobutane vapor flow rate was determined using a standard orifice installed in the vapor line leading to the turbine. During the previous tests a calibrated rotameter was used to measure liquid isobutane flow rates upstream of the heat exchanger.

It may be hypothesized that not all the liquid supplied to the heat exchanger was vaporized and the liquid fraction either was trapped in the knockout drum or was passed through the drum in the form of a fine mist and went undetected through the orifice and turbine. In either case, the orifice provided a good determination of the vapor flow to the turbine and no evidence of reduced turbine performance due to entrained liquid was observed during these tests.

6.1.2 Supercritical Calibration

Calibration tests were performed with the original supercritical nozzle utilizing the hairpin heat exchanger. The tests were designed to obtain performance data expanding through the saturated vapor dome and to use the calorimeter data to verify inlet enthalpy and turbine efficiency data obtained during the previous test series (Ref. 1).

Calibration data were collected at pressure ratios ranging from 7.7 to 8.7. The data collected and the method of data reduction was identical to that used with the subcritical data.

Measured values of turbine efficiency are shown plotted against velocity ratio for the supercritical nozzle in Figure 14. The predicted curve presented in the Phase I report is also shown for comparison. During the Phase II testing the condenser pressure was 68 psia as compared to an original design pressure of 83.2 psia. A turbine pressure ratio of 6.54 was measured during the Phase I testing. The increased pressure ratio for this test series resulted in an increase of 2.5 to 5.5 percent in turbine efficiency.

Values of nozzle flow coefficient calculated from these data varied from 1.08 to 1.14 over a wide range of pressure ratios using the calorimeter data to determine inlet enthalpy. The calorimeter was able to improve the consistancy of the data used to calculate turbine flow rates as compared to those obtained using temperature and pressure measurements. Values of flow coefficient greater than 1.0 are felt to be the result of inaccuracies in measuring actual nozzle throat areas or uncertainties in isobutane vapor thermodynamic properties used

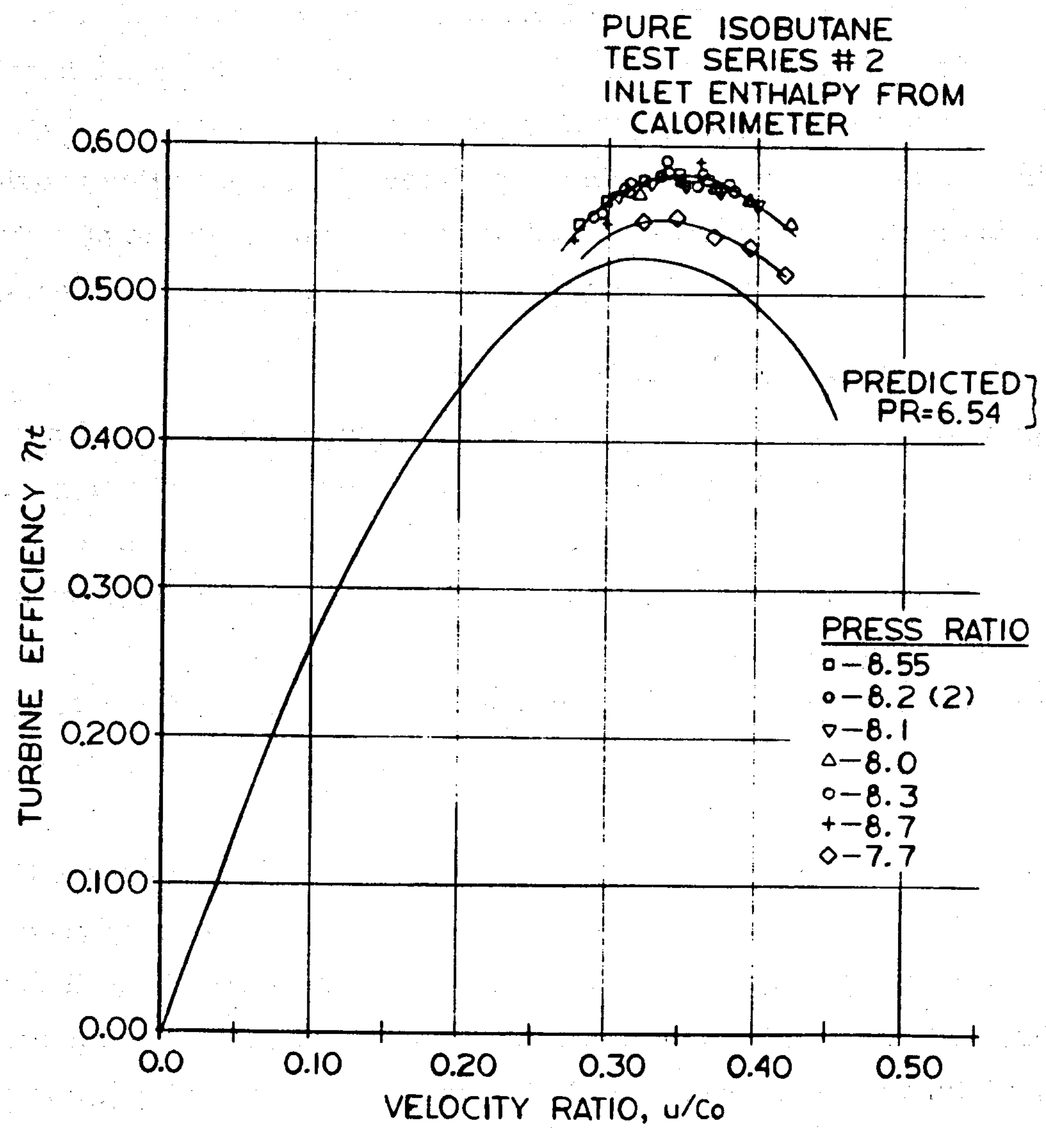

Figure 14. Supercritical calibration tests: efficiency vs. u/Co nozzle AR = 2.87 (single nozzle).

to calculate the ideal flow rates. Turbine efficiency data were also found to be very consistent during these tests. As with the subcritical tests, a knock-out drum was used between the heat exchanger and the turbine and isobutane and brine flow rates were measured with orifice flow meters. Turbine output horsepower is shown versus inlet enthalpy in Figure 15. As may be seen the variation in output horsepower with inlet enthalpy reasonably follows the predicted variation over the range of enthalpy examined. Figure 16 shows the variation in maximum turbine efficiency and maximum cycle efficiency with turbine inlet enthalpy. As can be seen, the calorimeter measurements resulted in a consistent set of data. The turbine efficiency varied from 55% to 59%. The predicted efficiency over the range of inlet enthalpies presented was 52.4%. The reduction in efficiency at low values of inlet enthalpy is felt to be the result of expanding into the saturated vapor dome.

6.2 TURBINE CALIBRATION - DCHX

Turbine calibration tests with the DCHX were performed at the start of the 200 hour endurance run and at the start and finish of the 40 hour endurance run. At the beginning of the 200 hour endurance run the subcritical calibration data of the hairpin tests were examined to determine which subcritical nozzle to use. Since the low pressure ratio (PR=1.3) nozzle gave a 4 point higher turbine efficiency it was selected for use during the endurance runs.

All calibration tests were performed at constant brine and isobutane flow rates. Since the turbine inlet enthalpy obtained from the turbine inlet temperature and pressure agreed with the enthalpy obtained from the calorimeter data, the turbine inlet temperature and pressure were used in conjunction with the measured values of gearbox output shaft torque and rpm to determine turbine efficiency as a function of velocity ratio.

Figures 17 and 18 present the turbine calibration data collected before and after each endurance run. The calibration tests prior to the 200 hour endurance run are presented in Figure 17. They were performed at pressure ratios of 3.4, 3.54, and 3.70. The nozzle flow coefficient varied from 1.01 to 1.03. The turbine efficiency at the beginning of endurance run #1 (200 hour) agrees very well with the predicted efficiency shown for a pressure ratio of 3.79. This improvement in agreement between the predicted and actual performance is due to the improved flow measurement with the isobutane flow orifice.

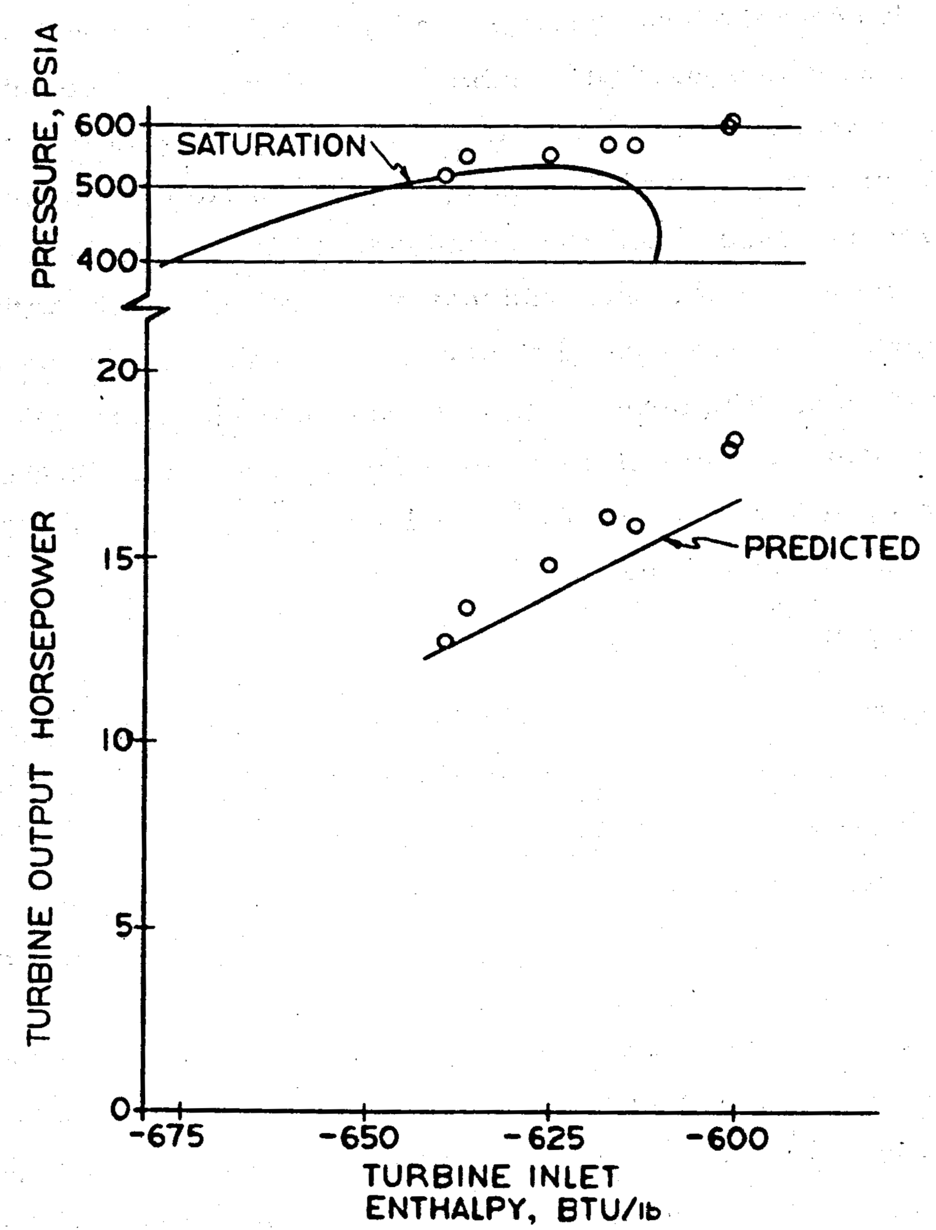

Figure 15. Turbine output vs. inlet enthalpy. Supercritical isobutane calibration inlet enthalpy based on calorimeter.

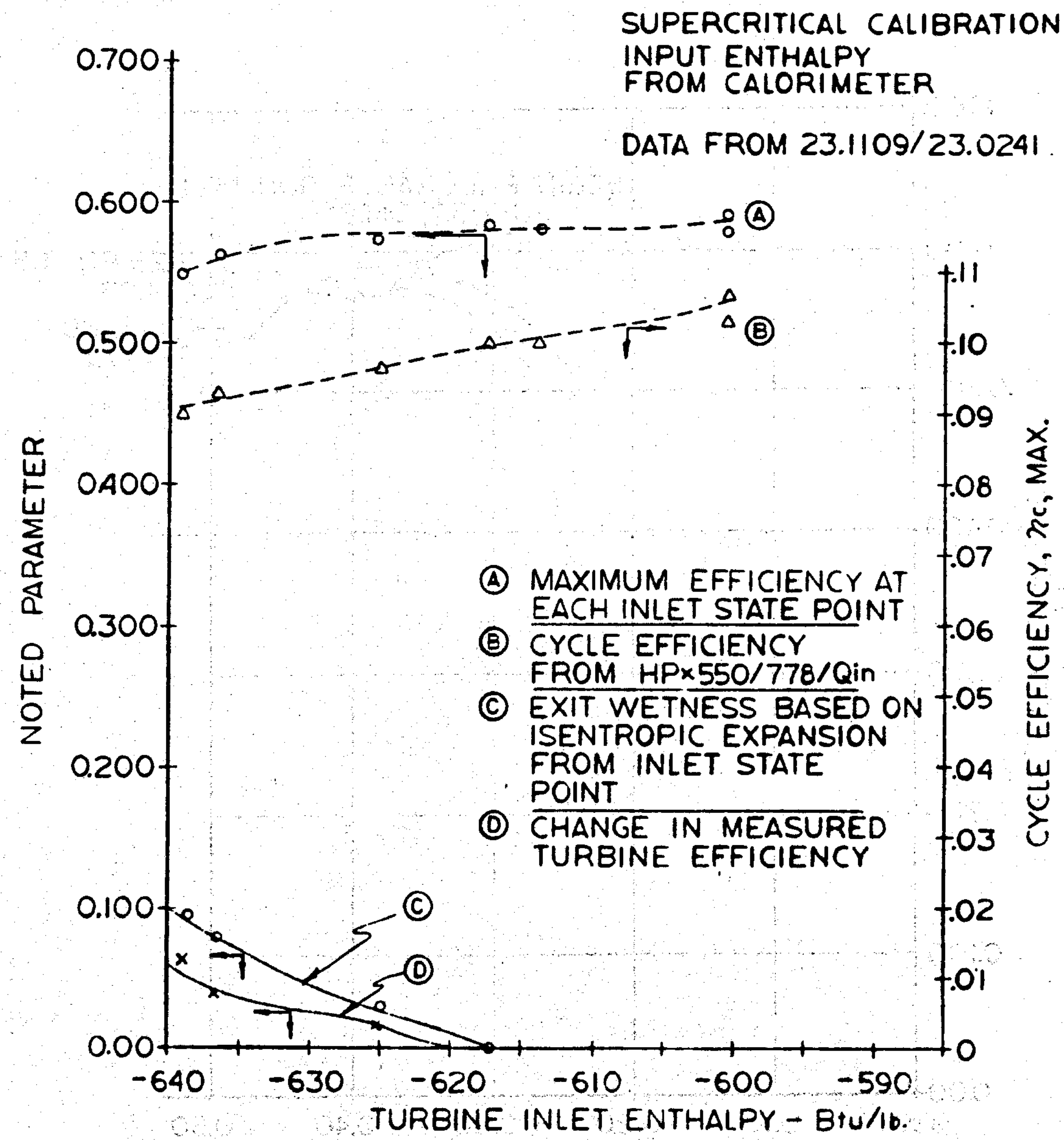

Figure 16. Variation of turbine performance with inlet enthalpy.

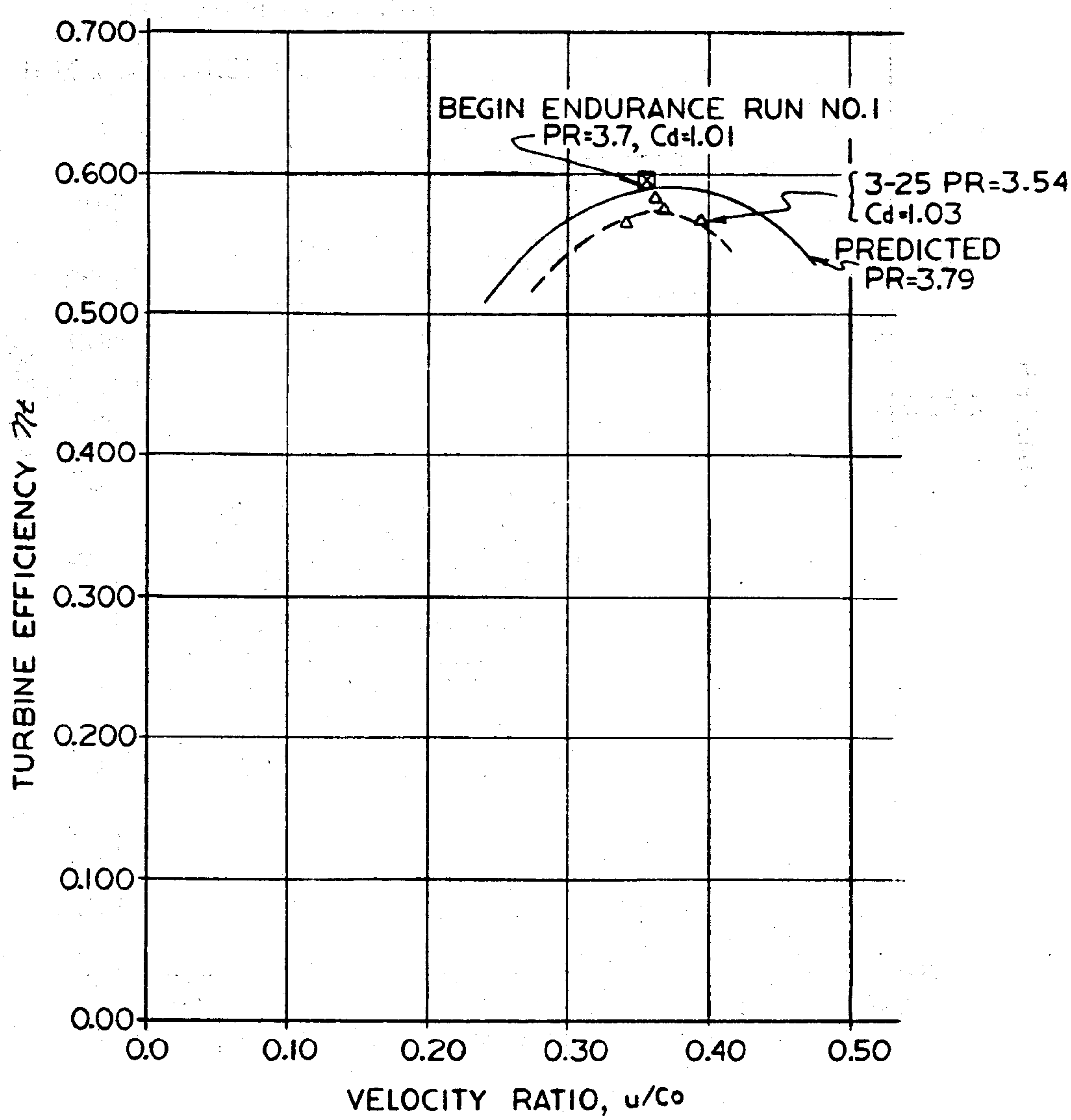

Figure 17. Turbine calibration with DCHX prior to endurance run, nozzle AR = 1.3.

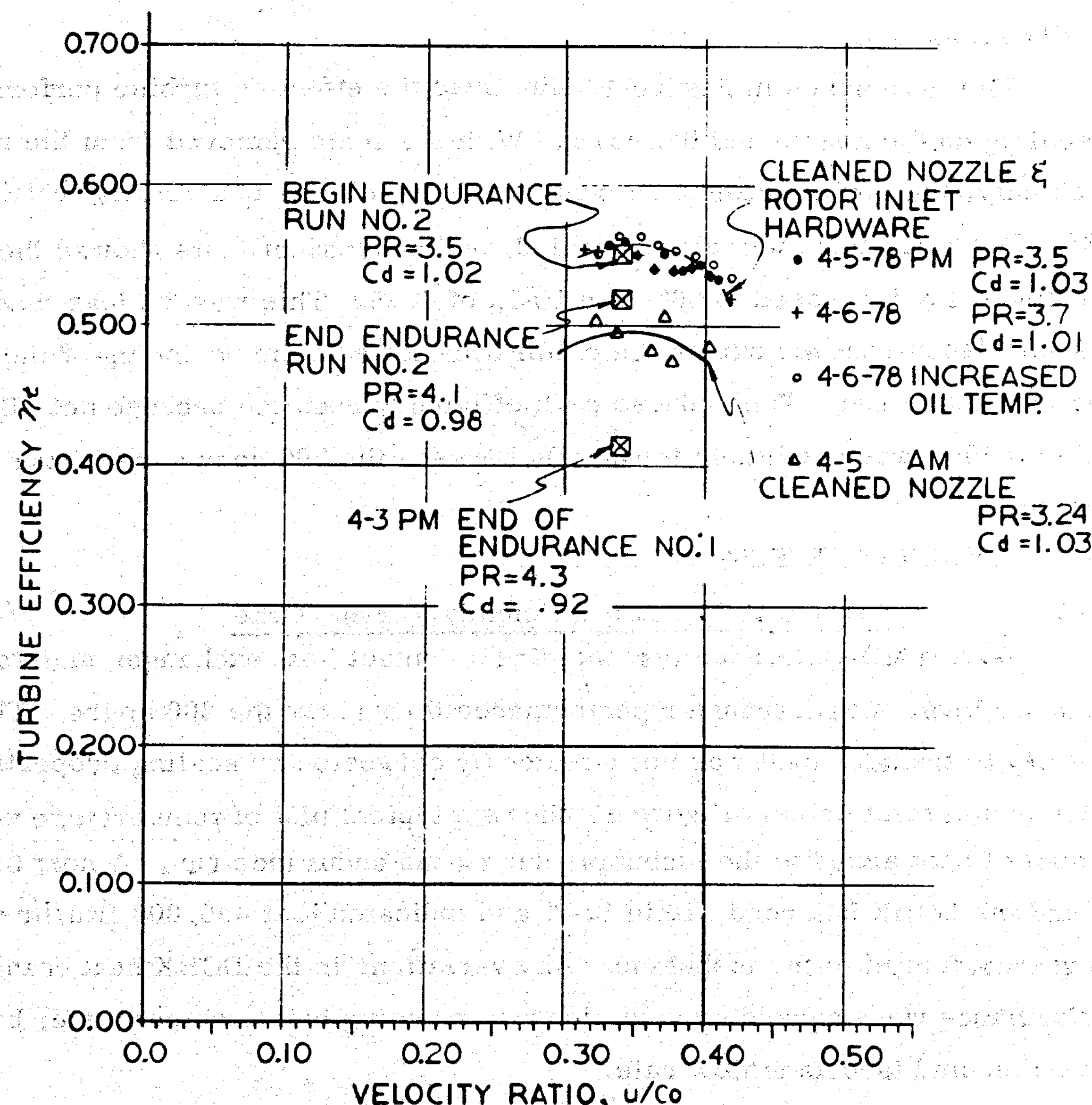

Figure 18. Turbine calibration with DCHX after endurance run, nozzle AR = 1.3.

Figure 18 shows the efficiencies measured at the end of the 200 hour run, at the beginning and end of the 40 hour endurance run, and the efficiency versus velocity ratio measured during the calibration tests at the beginning of the 40 hour endurance run. The nozzle flow coefficient at the end of the 200 hour endurance run was 0.92 down from 1.03 indicating that the nozzle was heavily scaled.

The two curves in Figure 18 illustrate the effect on turbine performance of scaling on the nozzle and the rotor. With the scale removed from the nozzle block only, the turbine reached a peak efficiency of 50% at a velocity ratio of 0.35. The rotor inlet was also descaled, and subsequent tests showed the turbine efficiency increased to 56% at a U/C_o of 0.35. This was 2% less than the peak efficiency reached during the calibration tests at the beginning of the 200 hour endurance run. The reduced peak efficiency resulted because not all of the scale could be removed from the nozzle between the 200 hour and 40 hour endurance runs.

6.3 ENDURANCE TEST

6.3.1 Turbine Output and Efficiency Versus Time

During the endurance test the direct contact heat exchanger maintained the same level of heat transfer performance throughout the 200 hours. The capacity to transfer heat was not affected by corrosion or scaling properties of the geothermal brine. Figure 19 shows a typical plot of temperature versus amount of heat added to the isobutane during the endurance run. A heat balance around the DCHX balanced within 1.4% and indicated that 490,000 Btu/hr was being transferred to the isobutane. Any variations in the DCHX heat transfer performance were caused by variations in incoming brine temperature, brine flow rate, and isobutane flow rate.

Scaling properties of the brine did affect the performance of the turbine. Figures 20 and 21 illustrate the effect of scaling on gearbox output horsepower and turbine efficiency as a function of time. Turbine gearbox horsepower and turbine efficiency were calculated for each data point recorded during the endurance run.

The turbine efficiency gradually changed from an average value of 58% at the beginning of the 200 hour endurance run to 41% at the end of the run. The

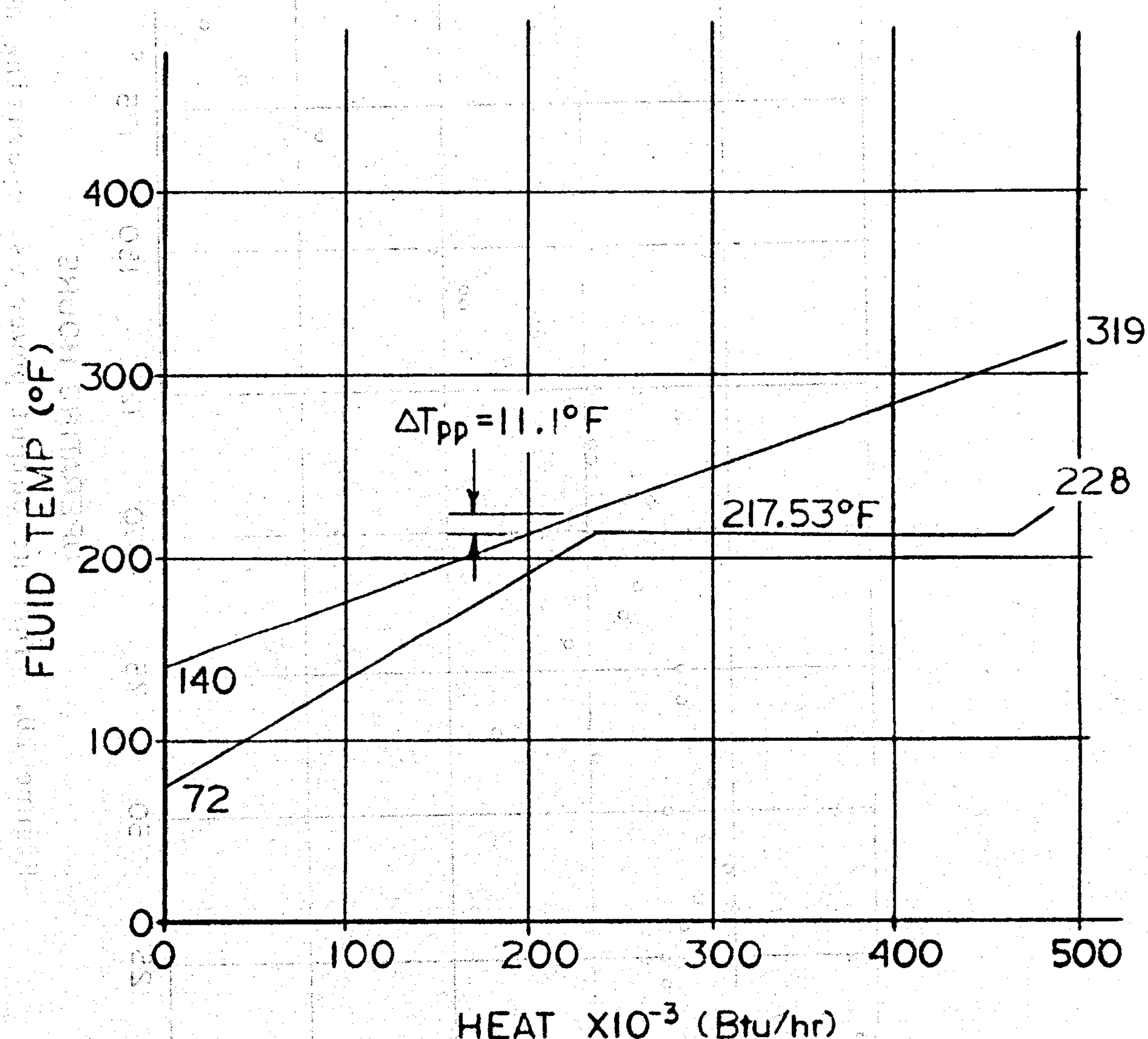

Figure 19. Boiler pinch-point calculation.

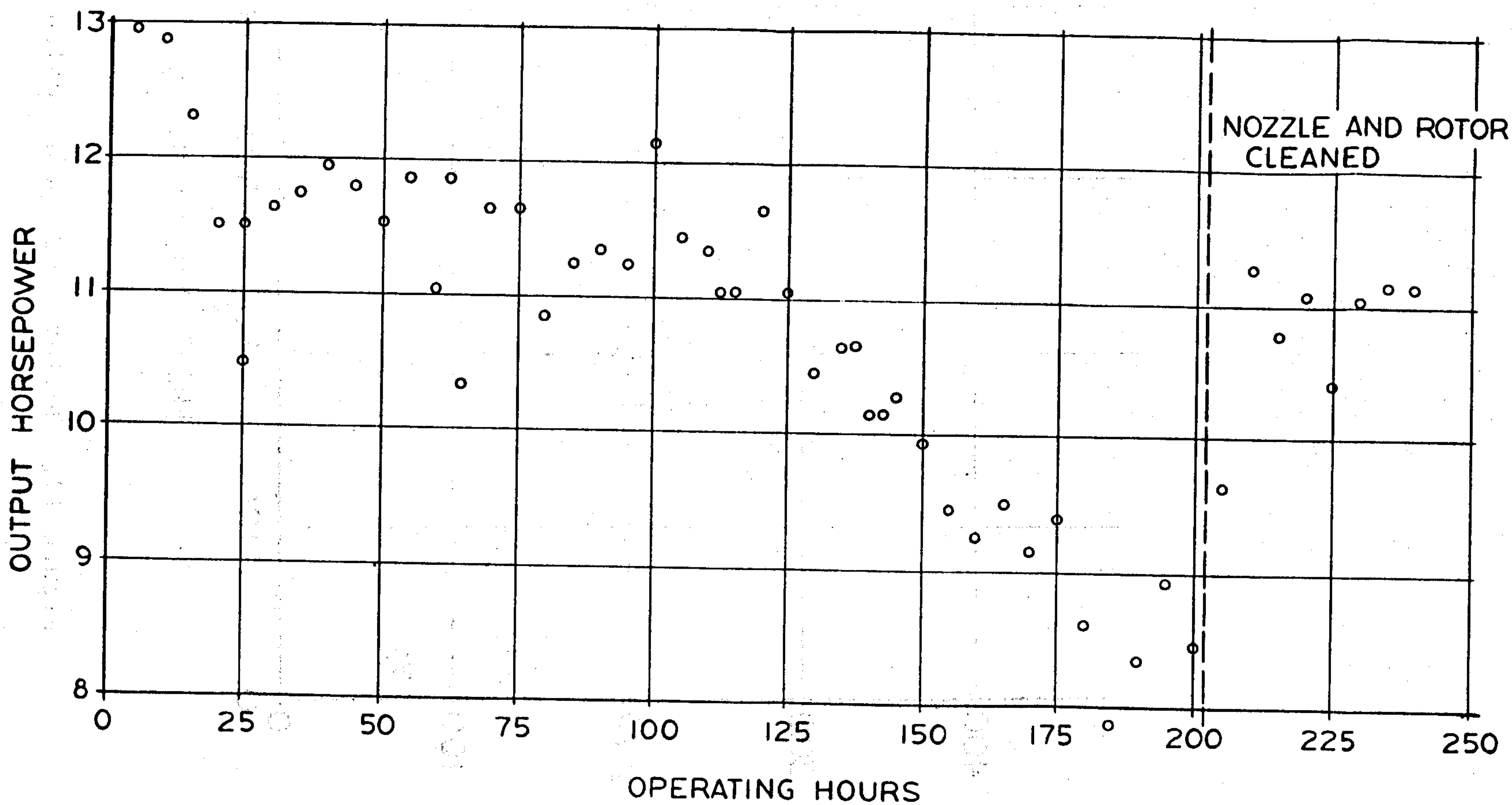

Figure 20. Turbine output power vs. operating hours.

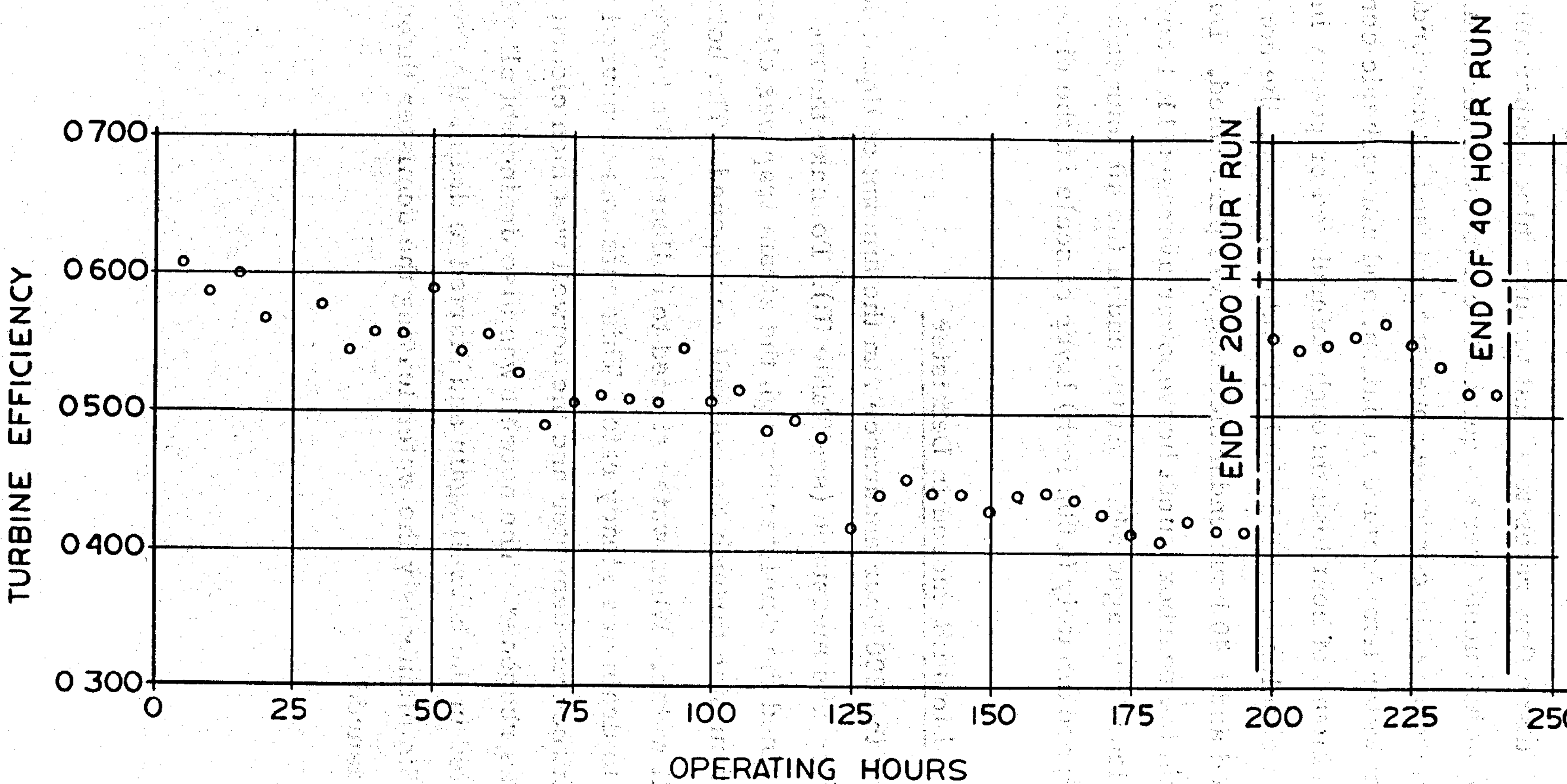

Figure 21. Turbine efficiency vs. operating hours.

turbine gearbox output horsepower decreased from 12.5 horsepower to 8.6 horsepower. The hourly variation in the data was a result of changing the brine and isobutane flow rates to adjust turbine inlet temperature and pressure. These adjustments were required to compensate for changes in brine temperature, changes in condenser pressure caused by noncondensibles, cooling tower water temperature variation, and atmospheric conditions which changed the amount of heat loss through uninsulated or poorly insulated piping.

At the conclusion of the 200 hour endurance run, the nozzle and rotor were descaled and a 40 hour endurance test was performed. During the 40 hour endurance run the gearbox output horsepower averaged 11.1 horsepower and the turbine efficiency averaged 54%. At the end of the 40 hour endurance run there was a thin (approximately 0.002 inch) layer of scale on the divergent section of the nozzle.

6.3.2 Liquid Removed in Demister

During the 200 hour endurance run the amount of liquid being removed by the demister was measured (see Table III). To make the measurement the valve that drained the demister through the steam trap was closed and the valve to drain the demister through the hot well was opened. The hotwell was then drained of all water. When water started to collect in the hotwell again, it was allowed to drain at a steady rate. This rate was measured and recorded. The valve between the demister and the hotwell was then closed and the steam trap valve was reopened. The hotwell was also drained of all water. When water collected in the hotwell again and started to drain, its flow rate was measured. Simultaneously the water leaving the demister through the steam trap was measured.

TABLE III

MEASURED WATER CARRYOVER FLOW RATES

Demister Valve to Hotwell Opened	Closed	Flow Rate, gpm Hotwell	Demister	Total
x		0.294	*	0.294
x		0.455	*	0.455
x		0.435	*	0.435
x		0.400	*	0.400
	x	0.071	0.238	0.309
	x	0.167	0.158	0.325
x		0.253	*	0.253
x		0.111		0.111
x		0.136		0.136
x		0.227		0.227
	x	0.089	0.198	0.287
x		0.278		0.278
x		0.278		0.278

As shown above, the total flow rate of water being carried over with the isobutane vapor varied from 0.111 gpm to 0.455 gpm with an average flow rate of 0.291 gpm. Comparing the columns for the water removed from the hotwell with those for the water removed from the demister when both flow rates were measured separately, the demister removed between 50% and 75% of the entrained geothermal brine from the vapor stream flowing to the turbine. On the basis of 9.5 gpm of isobutane at a nominal temperature of 230°F and a total pressure of 315 psia, the equilibrium water vapor fraction is 0.011 by weight. This weight fraction corresponds to a calculated flow rate of water of 0.08 gpm which is reasonably close to the hotwell flow rate with the demister valve closed.

6.3.3 Measurement of Flow Rate with IC_4/H_2O Orifice

The working fluid flow rate through the turbine nozzle was measured with a 0.701 inch orifice having a β of 0.339. During the 200 hour endurance run the flow rate through the nozzle started out at 2351 lb/hr and dropped to 2150 lb/hr as scale collected on the nozzle divergent section and in the throat area. The nozzle flow coefficient decreased from 1.01 to 0.92 during the 200 hour endurance run.

At the end of the 200 hour endurance run the nozzle was cleaned. The flow rate returned to the 2351 lb/hr value and the flow coefficient was 1.02. A 40 hour endurance run was completed. At the end of this run the flow rate was 2271 lb/hr and the flow coefficient was 0.99, indicating the nozzle was scaling up again.

TABLE IV

DCHX HEAT BALANCE
Data at 1330 on 4/2/78

Brine In	319°F	
Brine Out	140°F	Boiler pressure = 325 psia
IC_4 In	72°F	
IC_4 Out	228°F	
Q IC_4	9.1 in = 8.913 gpm = 2485.75 lb/hr	
Q Brine	5.3 in = 5.96 gpm = 2710.175 lb/hr	

Brine

$$Q = \dot{m}\Delta H = 2710.175\ (289.64 - 107.89) = 492{,}574.31 \text{ Btu/hr}$$
$$C_p = 1.015$$

Partial pressure of 228°F water = 20.016 psia

Partial pressure of saturated IC_4 = 325 - 20 = 305 psia
$T_{sat}\ IC_4 = 217.53°F$

Superheat = 228 - 217.53 = 10.47°F

ΔH boil = 88.88 Btu/lbm
Q boil = 2485.75 (88.88) = 220,921.03 Btu/hr

ΔH ph = (-703.63) - (-801.74) = 98.120 Btu/lbm
Q ph = 2485.75 (98.120) = 243,889.36 Btu/hr

ΔH sh = -606.39 - (-614.75) = 8.360 Btu/lbm
Q sh = 2485.75 (8.360) = 20,780.87 Btu/hr

Q total = 485,591.26 Btu/hr

Balance within 1.42%

6.4 INSPECTION

6.4.1 Turbine Gearbox

At the 42, 100, 200 and 240 hour points in the two endurance runs, the nozzle block was removed from the turbine and inspected for scale buildup. At the 42 hour point there was virtually no scale on the nozzle block. This was probably due to the fact that the nozzle block was new and still contained a coating of oil from the machining process. At the 100 hour and 200 hour points a scale deposition was noted on the expansion cone of the nozzle and the nozzle throat was partially plugged (approximately 10%) with scale. The scaling reduced the nozzle throat area and the mass flow rate as indicated by the isobutane orifice. The deposition in the expansion cone degraded nozzle performance and overall turbine efficiency as discussed in section 6.3.

At the completion of the 200 hour endurance run the nozzle block and turbine rotor were descaled and a 40 hour endurance run was performed to check the effect of cleaning the nozzle and rotor on overall performance and to check scale buildup in the first 40 hours of run time. Samples of scale were collected at the completion of the 200 hour and 40 hour endurance runs and results of a chemical analysis of the scale material is presented in the following section.

6.4.2 Chemical Analysis

Sampling of the scale found in the turbine nozzles and scrapings from the rotor blades and turbine housing were taken during the disassembly and inspection of the turbine. Analysis of the scale obtained showed the primary ingredient, 90% or more, to be silicon dioxide indicating that the source of all the scale was brine entrainment from the DCHX. The rest of the constituents present were typical also of the brine composition, see Table V. Sample numbers in the table correspond to the locations noted in Table V.

Evaluation of the scale in sample #2 also revealed the presence of excessive copper and zinc indicating possible corrosion of a brass fitting somewhere in the system. All of the samples contained iron which indicates some corrosion of the steel piping, though the principle source would be the brine.

6.5 RESULTS OF CO_2 FLASH EXPERIMENT

Condensing pressure for the entire direct contact test series was significantly higher than one would expect based on the cooling water temperature available and the IC_4 temperature measured in the hotwell vessel. Typically, values of IC_4 subcooling in the range of 20^oF to 30^oF were measured. This overpressure condition was attributed in part to an accumulation of noncondensible vapors in the condenser. A primary source of noncondensible vapor was

TABLE V

SPECTROGRAPHIC ANALYSIS OF SCALE SAMPLES FROM DCHX TURBINE

	#1	#1A	#2	#3	#4
Si	Principle constituent in each sample.				
B	1.5%	2.0%	2.5%	2.0%	1.5%
Fe	1.75	1.25	2.5	1.25	1.75
Na	.6	2.5	.6	1.5	.6
Ca	1.0	2.0	1.5	1.0	1.0
K	-	2.5	-	2.0	-
Al	.05	.2	.03	.3	.05
Cu	.08	.06	.4	.04	.08
Mg	.04	.1	.08	.08	.04
Sr	.15	.4	.3	.15	.15
Cd	.015	.02	.05	.04	.015
Zn	.05	.05	.3	.05	.05
Mn	.025	.02	.04	.01	.025
Pb	.01	.015	.006	.04	.01
Ba	.015	.05	.05	.02	.015
Cr	.008	.01	.008	.05	.008
Ni	.025	.02	.01	.03	.025
Ge	.005	.01	.015	.003	.005
Sn	.002	.003	-	.002	.002
Ga	.001	.001	-	.002	.001
Be	.001	.001	.002	.002	.001
Ti	-	.002	-	.002	-

#1	Found inside nozzle	(4/4/78)
#1A	Found inside nozzle	(4/4/78)
#2	Turbine nozzle	(4/7/78)
#3	Turbine housing	(4/4/78)
#4	Turbine rotor	(4/4/78)

CO_2 from the brine stream which when entering the direct contact column gave up a portion of the dissolved CO_2 to the IC_4 stream. The CO_2 accumulated in the condenser raising its total pressure.

To investigate the amount of CO_2 which can be removed from the brine stream a small flash vessel was fabricated and installed at the test pad. Brine from well Mesa 6-2 was directed to the flash vessel and subjected to a controlled pressure drop. Flow, pressure, and temperature instrumentation were provided around the flash vessel to set and monitor test conditions. Brine sample ports were provided upstream and in the vessel to evaluate the CO_2 removed in the flash process. The brine samples were analyzed by the chemistry lab at the East Mesa Component Test Facility. Although these tests were not intended to provide a complete answer to the complex CO_2-brine chemistry interactions, two basic results were obtained:

1) Free and dissolved CO_2 contaminates can be substantially reduced by moderate flash temperature differences in the brine stream. Flash temperature differences as low as 2°F to 3°F showed substantial reductions in CO_2 content.

2) Flash temperature drops of up to 10°F resulted in significant carbonate scaling problems in the flow meter and vessel downstream of the flash tank. Flash temperature drops of about 5°F were run for extended periods (2 to 3 days) and no evidence of scaling was observed.

As a result of these tests, detailed chemical analyses and additional field tests were initiated. Results of these investigations will be reported at a later date.

7.0 CONCLUSIONS AND RECOMMENDATIONS

1) Turbine efficiency and nozzle flow coefficients measured during this phase of testing verified performance prediction techniques and design methods for partial admission axial flow hydrocarbon turbines. The influence of entrained water in the working fluid vapor flow on turbine performance during direct contact heat exchanger testing was shown to be reasonably predictable.

2) Improved performance data obtained during this phase of testing as compared to that obtained during previous tests was attributed to the use of a knockout vessel located between the brine-to-isobutane heat exchangers and the turbine. The knockout drum removed liquid droplets from the vapor stream to the turbine. Also, the vapor flow rate to the turbine was determined using a standard orifice. This provided improved flow data as compared to that obtained by measuring the liquid isobutane feed to the heat exchangers with a rotameter.

3) A turbine efficiency improvement of 4 points was obtained with the single low area ratio nozzle (AR=1.3) as compared to that obtained with the original nozzle design (AR=1.57).

4) Supercritical test results showed that turbine and cycle efficiency was reduced slightly by expanding inside the isobutane saturated vapor dome as compared to data obtained with expansion resulting in superheat at the turbine exhaust. Brine utilization efficiency was improved, however, because of the additional ΔT removed from the brine. Overall performance of the supercritical cycle was not as good as the subcritical cycle because of the additional required pump work.

5) Nozzle scaling was again found to occur during the endurance operation with the direct contact heat exchanger. Although the knockout drum removed large liquid droplets, some entrained mineral content was carried to the turbine resulting in a silica scale formation. As in previous tests no significant scaling was observed in the direct contact heat exchanger or in the connecting pipes.

6) A practical means of substantially reducing the free and dissolved CO_2 content of the East Mesa brines was demonstrated by allowing the brine to flash 2 to 3°F in a flash tank. Very low amounts of carbonate scaling were observed at this condition while laboratory analysis indicated only residual amounts of CO_2 remaining in the brine samples taken downstream of the flash tank. The majority of CO_2 remained in a bicarbonate form (HCO_3) in these samples.

To achieve practical long term operation with the direct contact power loop, improvements are needed in the following three areas:

1) CO_2 or noncondensible contamination in the isobutane condenser.
2) Isobutane recovery following the direct contact heat exchanger.
3) Mineral carryover in working fluid vapor from the direct contact heat exchanger.

Items 1 and 2 are not unrelated since noncondensibles contained in the brine influence the amount of isobutane which can be stripped from the brine in a flash recovery vessel. Noncondensible content also determines the difficulty involved in condensing and recovering the isobutane collected.

Suggested means of dealing with the above problems are:

1) Install a flash tank to pre-flash the brine upstream of the direct contact heat exchanger. Noncondensibles removed at this point reduce the severity of noncondensible accumulation in the condenser. Analysis indicates that if the CO_2 content in the brine can be reduced to about 50 ppm, less than 2 psi partial pressure of CO_2 would accumulate in the condenser.
2) The present scheme for isobutane recovery involves flashing the brine to a subatmospheric pressure level and collecting the freed gas. Operating this system in the absence of noncondensibles in the brine may reduce the effectiveness of the flash process in removing isobutane, although the recovery condenser would be less of a problem.

3) Improvements in demister design and operation can reduce or eliminate mineral carryover in the working fluid vapor. It is suggested that the design be modified to incorporate a hot water wash of the demister mesh to improve removal of the water carryover. Chevron baffles following the mesh (preferably in a horizontal configuration) would also improve the efficiency of the demister element.

It is recommended that these unknown effects be evaluated in test.

8.0 REFERENCES

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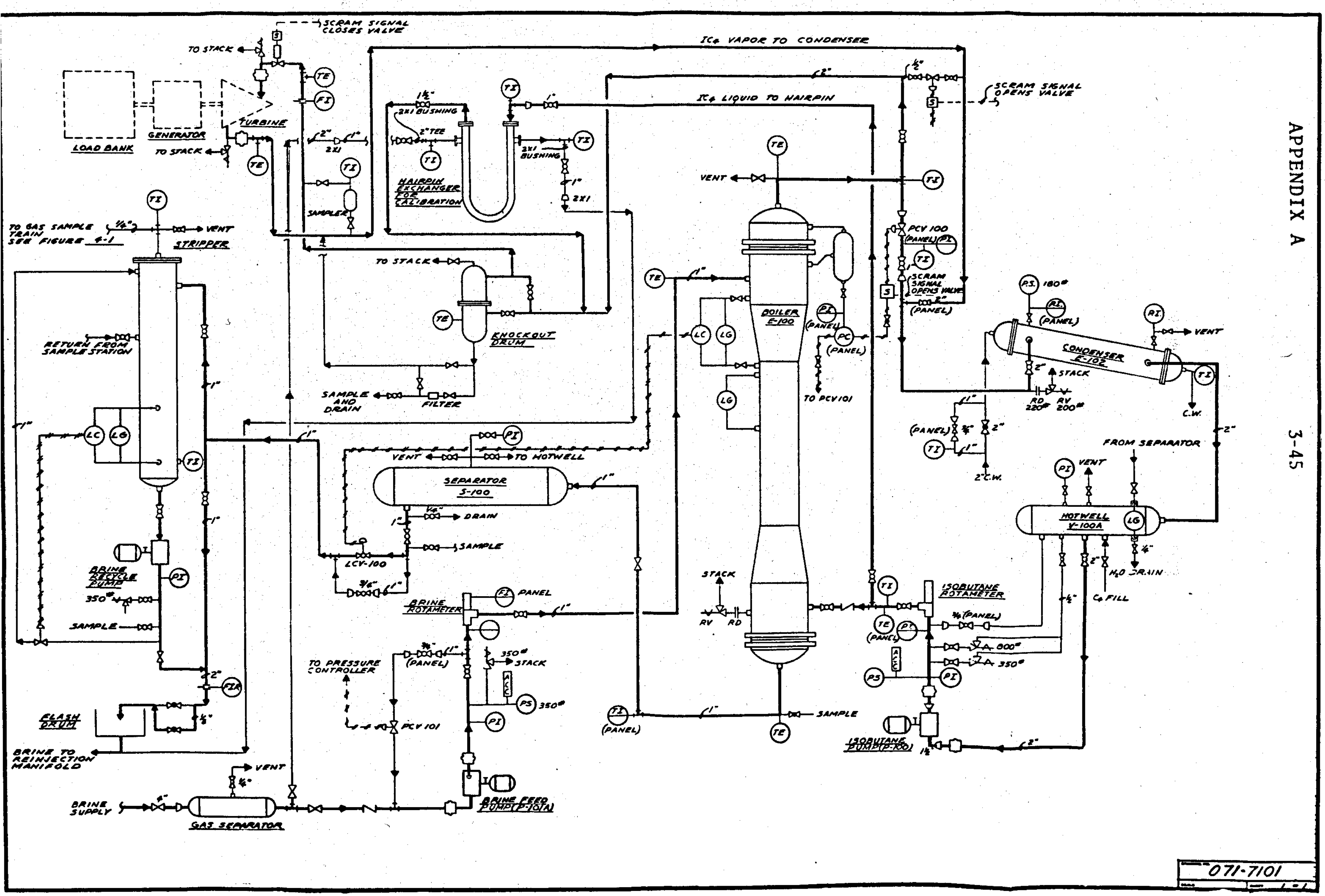

Fig. A-1. P & I diagram for DCHX power plant.

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