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COMPONENT DEVELOPMENT

Claude Harvey, Warren McBee and Hugh B. Matthews

Sperry Research Center, Sudbury, MA 01776

SPERRY LOW TEMPERATURE GEOTHERMAL
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FOREWORD

This final report covers the work performed under the "Sperry Low Temperature Geothermal Conversion System" project, both Phase I under Contract No. DE-AC03-78-ET27125 (formerly ET-78-C-02-4633) and Phase II under Contract No. DE-AC03-79-ET27131.

Vols. I and II, submitted in 1982 covered a portion of the work accomplished under Phase I. Vols. III, IV, V, and VI submitted at this time cover the balance of work performed under Phase I and work performed under Phase II.

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5.0 INTRODUCTION

The fundamental inventions which motivate this program are system concepts centered on a novel heat engine cycle and the use of down-well heat exchange. However, the implementation of these system concepts required development of new components and subsystems, and research in materials and fluid dynamics. A major research topic was organic working fluid properties, reported in Vol. I of this Final Report. Another major research topic was compatibility between the three fluids involved in the system (R-114, geothermal brine, lubricating oil) and between these fluids and the materials of construction, especially elastomers, used in seals. This topic is reported in Vol. II. The remaining component and material development efforts are reported in this volume. As would be expected, the primary emphasis is on down-well hardware. The only surface equipment included is the surface portion of the instrumentation and control systems covered in sections 5.1.3, 5.1.4 and 5.1.5. Down-well instrumentation is reported in section 5.1.2.

Down-well conduits and techniques for installing, connecting and sealing them are covered in Section 5.2.

The downwell turbine-pump unit (TPU) is a critical component since it is relatively inaccessible and operates in a hostile environment. Its development is reported in Section 5.3. The TPU for the gravity-head system requires a different type of turbine because of the large flow-rate through it and the small pressure difference across it. The design study for a Francis turbine to meet these requirements is reported in Section 5.4.

A feature of these systems is use of a down-well heat exchanger. There were extensive studies, described in Section 5.5, of tube-bundle configuration, tube-sheet seals, structural integrity, and flow and heat transfer, as well as the research on welded connections and sliding elastomeric seals described in Section 5.2.

Another innovative component in these systems is the enthalpy recovery unit (ERU). This direct-contact heat exchanger compensates for under-cooling in the condenser and superheat in the main turbine exhaust. It is described in Section 5.6.

5.1 INSTRUMENTATION AND CONTROL

5.1.1 Introduction

The instrumentation and control systems for the pump system were completed and installed at the E. Mesa test site and were ready for operation in June, 1981, when the well was lost. These I&C systems consisted of three subsystems: 1) a down-well data acquisition and transmission system, 2) an operation surface system and 3) a scientific data surface system. These subsystems are described in sections 5.1.2, 5.1.3 and 5.1.4 below.

The instrumentation and control system for the gravity head system was not fabricated per se, but it would have been derived from the pump I&C system for the most part. The principal addition would have been associated with the turbine-generator controls and output electric power switchgear. The plan for these additions is described briefly in section 5.1.5 below.

5.1.2 Down-well Data Acquisition Subsystem

The functional schematic of this subsystem is shown in Figure 1. Its function is to sense and transmit the following data:

- o Pump Inlet & Outlet Pressures
- o Turbine Inlet & Outlet Pressures
- o Lubrication Oil Pressure
- o Pump Outlet Temperature
- o Turbine Inlet & Outlet Temperatures
- o Heat Exchanger Inlet & Outlet Temperatures
- o (Both Brine & Working Fluid)
- o Transducer Manifold Temperatures
- o Instrument Cannister Temperatures
- o Turbine-Pump RPM (2 Channels)
- o Zero & Full-Scale Calibration Signals

The block flow diagram of the down-well package is shown in Figure 2. This portion of the system is located in a sealed canister inside the turbine exhaust pipe just above the turbine. Electrical connection to the surface is by means of a coaxial logging cable which was threaded through the lubrication oil tube. The instrumentation cable and lube oil tube are continuous and are stabbed into the turbine exhaust pipe by means of a special fitting above the oil cooler. This is done after the TPU and its R114 conduits have been installed in the well. Due to loss of the well this installation operation was never tried. However, many tests were made in the lab of the stabbing connection and it went through several design modifications as a result of these tests. The hydraulically operated lowering and hoisting drum for the combined lube oil tube and instrumentation cable is shown in Figure 3.

The electronic components and sensors for the down-well instrument package were tested separately for satisfactory performance under the severe down-well conditions, particularly the soak at maximum brine temperature when the TPU is not operating and there is no cooling by the turbine exhaust. The complete down-well instrument package was operated successfully in the turbine exhaust line during laboratory testing of the TPU.

5.1.3 Surface I&C System

The complete piping and instrumentation diagram for the pump system is shown in Figure 4. The items in a double circle are associated with the scientific data system. All other I&C items in this figure are part of the independent operating subsystem. The primary control system is very simple. The primary controlled parameters are brine flow and organic working fluid flow via feed pump control. There is also a system for automatically transferring from external power supply to an emergency generator with a battery-powered uninterruptable instrument power supply to bridge the switchover time gap. Figure 5. is a control schematic. The system control panel is a mimic board. A layout drawing is shown in Figure 6 and a photograph of the installed panel in Figure 7.

All surface instrumentation and control components were installed, wired and checked out. A complete set of instrumentation loop diagrams was prepared and is in the drawing file.

5.1.4 Scientific Data System

This system was designed to acquire, display and record all data required to define performance and compare with theoretical predictions. The overall system configuration is shown in Figure 8. The processor is a Sperry UNIVAC V70-600. The multiplexed surface data acquisition system block diagram is shown in Figure 10. The multiplexer was procured from Honeywell. A block diagram of the data storage and control portion of the system is shown in Figure 11. A complete operating software package was written and debugged. The system was operational on the E. Mesa site in its own van at the time the well was lost. The operating instructions for the system are contained in Appendix A.

5.1.5 Gravity Head I&C System

The instrumentation and control system for the pump system was to be used for the gravity head system also. For example, all I&C associated with the brine-handling system would be identical except for adjustments in sensor ranges to match different parameter values. The pump surface data system was designed to accommodate the additional sensors needed for the gravity-head system because of the additional evaporative condensers and the surface turbine.

The principal change would be in the control system, which would include the main turbine-generator controls and output electric power switchgear. The gravity-head system control schematic is shown in Figure 11. The basic control function is to sense generator load conditions and condenser status and to adjust the brine flow, turbine throttle, turbine

by-pass, and R-114 flow, and/or valves to accommodate changes in load and wet-bulb temperature, or an emergency shut-down. The same system would execute a normal start-up or shut-down procedure.

The operating panel mimic board layout is shown in Figure 12. It was planned to be even simpler than the one for the pump system. The main single line diagram is shown in Figure 13. The system was to have been capable of exercising fully automatic control under the supervision of the scientific data system processor to demonstrate the feasibility of unattended or remote operation.

5.2 DOWNWELL CONDUITS AND SEALS

5.2.1 Introduction

A schematic drawing of the gravity-head system is shown in Figure 14. Unique features of the system are the down-well heat exchanger, turbine-pump unit (TPU) and insulated riser. Also required is a small conduit to supply lubrication oil and a coaxial cable for instrumentation from the surface to the TPU. The upper portion of the down-well heat exchanger consists of tube bundle sections as shown in Figure 15. in order to provide the required heat exchange area per foot of length. The design installation, connection and sealing of these conduits pose engineering problems not encountered previously in down-well equipment design and operation. An extensive development effort was carried out to address and solve these problems.

The reporting of this effort is divided into four sections: section 5.2.2 treats the development of welded connections between heat exchanger sections, section 5.2.3. covers the elastomeric sliding seals, section 5.2.4 discusses the lubrication and instrument lines and section 5.5 covers the tube sheet seal development as well as other aspects of heat exchanger development.

5.2.2 Conduit Welded Connections

The design of the connections between sections of the down-well conduits is driven by three primary requirements:

- (1) structural integrity
- (2) negligible leakage
- (3) rapid, low-cost installation and removal with several workover cycles.

The proposed automatic welding and de-welding (cutting) procedures were developed at Sperry Research Center to offset disadvantages experienced when joining large down-hole pipe or casing with threaded couplings. In spite of high cost, "leakproof" designs, and use of a variety of sealants, large threaded joints typically leaked and were difficult to unscrew when coming out of the well because of thread corrosion by brine. Sealing welds have been applied to threaded couplings by machine with an H and M Pipe Beveling Machine Company pipe lathe which has been split and hinged to permit clamping around the pipe rather than slipping over the end. The lathe holds, positions, and transports the torch during welding and has been proven reliable in line pipe operations which are in many ways as difficult as rig operations. The pipe lathe is also used for cutting if removal of a section is required. The lathe production model was not precise or sturdy enough for this application and was enhanced by several changes, including:

1. Heavy base plates were attached to the carriage rings.

2. Inter-ring struts were enlarged in cross section.
3. Hinges for frame and ring gear were made heavier and more precise.
4. Clamps and alignment means for the carriage and ring gear separation were improved.
5. Carriage-to-pipe clamping means were enlarged and improved.
6. Means were developed to locate the frame on the pipe so that the ring gear rotates in a plane parallel to the weld face when welding and yet perpendicular to the pipe axis when Arcairing. This is to prevent cumulative increase in error of weld plane-to-pipe axis after repeated weld/de-weld operations.
7. A larger and more rugged drive motor with integral gear box rotates the ring gear directly through a fiber gear. It delivers a 7-fold increase in torque in the operating range to give a constant speed even if the ring binds because of dirt or misalignment. In addition, a remotely located power supply with IR compensation was added which yields a 99% speed regulation over the torque range.

A photograph of the modified welding lathe set-up for 8" tests is shown in Figure 16.

Welding is done by the MIG (metal inert gas) process which is also referred to as GMAW (gas-metal-arc welding), while de-welding or cutting is done by arc gouging using an Arcair torch. Currently both procedures have been used successfully on hundreds of test runs on 8-, 13-, and 16-inch diameter pipe with wall thicknesses of 1/2-inch in both bench and field tests. For example this process and equipment were used to weld the threaded couplings on one end of all the down well conduit sections fabricated for the pump system test. Welding rates of about 10 inch/min and cutting rates of 20 inch/min have been demonstrated. Weld quality has been reported consistently high as determined by:

- o Visual inspection
- o Metallographic sections
- o Pressurization with water to 5000 psi

The welding procedure is easily learned by an experienced welder and satisfactory welds with rusted and greasy, as well as clean, pipe have been achieved.

Visual inspection of finished welds for a smooth uniform surface has proved adequate to identify satisfactory welds. The metallographic section of welded casing shown in Figures 17 and 18 illustrates that the welded joints are of high quality with no porosity or slag in the weld

zone. Rewelded sections previously cut by arc gouging also exhibit high quality.

An important aspect of the experience to date with all conduit welds is that they are seal welds on threaded joints. Full structural in combination with sealing aspects of the welds have not been tested. However, a properly made weld should withstand expected stresses and seal properly. The highest stresses expected will be generated by the pulling capability of the rig and will be exerted along the axis of the conduit. *If a 1/2-inch fillet weld is made with E60 weld metal in a 1/2-inch wall thickness conduit, the American Welding Society code allows 6.37×10^3 lbf/inch of weld. For a 24-inch casing, the total load which could be supported without failure is

$$W = 6.37 \times 10^3 () (24) = 480 \times 10^3 \text{ lbf.}$$

This represents 10,000 lb more than the yielding load of the pipe, based on a yield strength of 25 ksi (1000 lbf/inch squared of cross section), and should be safe. On larger and thicker casing, such as the 30-inch OD casing with a 3/4-inch wall thickness, the allowed load is 9.55×10^3 lbf/inch for a 3/4 inch fillet. The weld can support

$$W = 9.55 \times 10^3 () (30) = 0.9 \times 10^3 \text{ lbf ,}$$

which is 160,000 lbs more than the yield load of the pipe, again based on a yield strength of 25 ksi. Properly made welds are essentially as strong as the conduit with a nominal yield strength of 25 ksi.

The scale-up required to handle up to 30-inch diameter with 3/4-inch wall pipe or casing should be feasible based on equipment availability and current knowledge. Larger pipe lathes are commercially available and could be modified as described earlier. The preferred approach to minimize scale-up difficulty would be the use of threaded couplings with a weld for sealing. The threaded couplings need not be of the sealing type but would provide alignment and support prior to welding as well as structural integrity in the event of poor weld quality. As welding expertise and confidence improves, consideration could be given to all welded joint construction (i.e., no threads).

Corrosion rates in weld regions should not be significantly different than for the base metal because severe compositional gradients are not generated in the weld area. The E60 weld rod employed does not vary appreciably from the nominal composition of the base metal.

5.2.3 Packers and Sliding Seals

5.2.3.1 Introduction

As can be seen by referring back to Figure 14. there are three large down-well fluid flows: 1) upward flow of geothermal brine between the well casing and the heat exchanger, 2) downward flow of R-114 in the heat exchanger, and 3) upward flow of R-115 inside the riser. These flows are contained by three large conduits which are at different temperatures at different times, therefore there must be at least two sliding seals to accommodate differential thermal expansion. Also, there must be a seal

between the pump and the well casing, across which the pump added pressure appears. This packer can be fixed, in which case there must be a sliding seal somewhere else in the down-comer (heat exchanger) conduit, or the packer can perform this function also by making a sliding seal to the well casing. Both approaches were studied and are discussed in sections 5.2.3.2 and 5.2.3.3 below. The other sliding seal is in the R-114 riser. This is discussed in section 5.2.3.4 below.

5.2.3.2 Sliding Packer

Since there must be an elastomeric packer seal to the well at the pump casing, it is an attractive approach to make this packer a sliding seal as well to accommodate differential thermal expansion between the down-well heat exchanger and the well casing. In this approach, the down-well hardware is suspended from the well-head. Such a sliding packer is not an off-the-shelf component. There are fixed packers available with a sliding-seal bore. In fact, such an assembly was used for the earlier Sperry pump test at Heber and proved unsatisfactory because of jamming of the close-tolerance sliding-seal assembly by sand and debris falling from above. If a fixed packer is used, it is believed better to locate the sliding-seal at well-head where it is not susceptible to such sources of jamming. This approach was studied and is reported below in section 5.2.3.3. Thus, development of a packer which can provide a sliding-seal against the well casing was a task objective. The approach taken was to modify oil-well swab cup designs and materials to meet the larger-size, higher-temperature, higher-pressure, corroded-casing requirements of the geothermal application.

The configuration selected for development consisted of three redundant packer cups to be attached to the pump inlet tube to form a sliding seal against the 13-3/8 inch casing string. Each cup was to have the approximate shape of an inverted bell. The bell lip would form an initial seal with the pump off. After pump start-up the differential pressure would expand the cup and increase the sealing contact area. Maximum design differential pressure would be 625 psid. The rated operating brine temperature was 350°F.

Regal International was contracted to assist in the sliding packer design and to fabricate the experimental packer cups. A test facility was constructed at Barber-Nichols Company to test the cups both statically and dynamically in brine at 350°F. Six generations of experimental cups were designed and fabricated before a successful combination of design, materials, and manufacturing technique was established.

The early designs failed to establish an initial seal properly. Later versions sealed initially, but failed structurally at higher differential pressures. When the cups were strengthened with metal reinforcing fingers, to overcome the strength of material problems, material bonding problems developed which again resulted in structural failures. Ultimately, the proper combination of bonding agents and

manufacturing techniques was established which resulted in a successful test of three sets of cuts in the Barber-Nichols test jig at full temperature and pressure.

5.2.3.2.1 Packer Formulation Development

This section concerns the development of the elastomer compound to be used in the cup type packing unit. Existing elastomer formulations were known to be incapable of lasting in the geothermal environment. The development of a new elastomer was begun on 1-9-79. The parameters for development of the new formulation are shown in Table I.

The first task was to research available literature and consult with suppliers' technical personnel in an effort to identify polymers with the greatest potential for development. A great deal of research was performed and several possibilities were defined for elastomer candidates as well as some promising methods for formulation development. One thing we quickly found was that very little information exists relating directly to geothermal work. However, related work on formulations to be used in steam was found to be helpful. EPDM or Fluorocarbon elastomers were found to be promising and were chosen for development work.

In order to evaluate the suitability of these formulations for the end use, a test media similar to the actual working media was needed. This media, which had been used in other geothermal development work, was formulated as an aqueous solution of H_2S (300 ppm), CO_2 (1,000 ppm) and NaCl (25,000 ppm). All tests were run at 204°C for varying durations. Initial tests were conducted for 22 hours and established three of eighteen formulations as candidates for further development (Table II). The 22 hour tests proved to be a disaster for 2 fluoroelastomer formulations which had been compounded for steam resistance. Due to time limitations and the fact that the EPDM formulations worked so well, further development of fluoroelastomer formulations was suspended.

As the program progressed, the duration of the tests was expanded from the initial 22 hours to 48 hours, 70 hours, 120 hours and finally 576 hours. The most hydrothermal chemically stable formulations from each group were selected for further work with the goal being to achieve a percentage change of unaged to aged physical properties of 15% or less in any category. A total of 39 EPDM and 2 fluoroelastomer formulations were designed and tested before the desired results were obtained. These results are shown in Table III.

Field experience with elastomer packers indicated a problem of casing cure, a situation in which an elastomer seal under heat and pressure will adhere to the casing wall. To determine if this might be a problem with the formulation developed, we clamped a sample between two pieces of steel plate. One plate was placed in a pan of synthetic brine solution at 70°C for 24 hours and allowed to rust. Half of this plate was then bead blasted and cleaned and a sample of the compound was clamped over both the rusty and clean areas. The clamped arrangement was aged in the brine solution for 24 days at 204°C and showed no adhesion of rubber to

metal in any area. Consequently, we do not feel that casing cure will be a severe problem, if anything, considering the desired one-year lifetime.

During our development work, we found that this same formulation also works very well in a standard compressive seal ring configuration holding water at 1000 psi and 175°C for 24 hours. A seal of this type is needed in the same geothermal conversion system although it will be used in a slightly different environment. The environmental requirements for this compressive seal are good aging resistance to the brine solution plus resistance to R-114, 1,2 dichlorotetrafluoroethane.

TABLE I

Temperature Range: 1600 - 1800°C
 Fluid Composition: Brine solution consisting of chlorides (Na, Ca), H₂S, CO₂, CO₃, = very little dissolved O₂.
 Pressure Differential: 1000 psi
 Life Expectancy: 1 year

TABLE II

CHANGE*		<u>%</u>		
	<u>UNITS</u>	<u>#3</u>	<u>#10</u>	
<u>#17</u>				
Tensile	psi	+2	-8.6	-
19.7				
Ultimate Elongation	%	+18.4	+22.0	
+13.8				
Modulus, 100%	psi	-1.2	-19.3	-
5.3				
200%	psi	-6.4	-23.0	-
16.4				
Tear, die C	pli	+6	-1.1	
+6.0				
Hardness	Shore A	0	-3	0
*Aged 24 Hours at 204°C, Synthetic Brine.				

TABLE III

	UNITS	% CHANGE*
Tensile	psi	-6.3
Ultimate Elongation	%	+15.2
Modulus, 100%	psi	-7.1
200%	psi	-7.7
Tear, die C	pli	+5.0
Hardness	Shore A	+1
*Aged 576 hours at 204°C, Synthetic Brine.		

REFERENCES

- (1) Baldwin, F.P. and Ver Strate, G., Rubber Chem. Technol. 45, 709 (1972).
- (2) Dunn, J. R., Rubber Chem. Technol. 41,304 (1968).
- (3) Hirasuna, A. R., et. al., December 1977 "Geothermal Elastomeric Materials - Twelve-Months Progress Report, October 1, 1976-September 30, 1977), L'Garde, Inc., Newport Beach, Calif. 92663, SAN/1308-2, U.S. Department of Energy, Geothermal Energy.
- (4) Murray, K., Rubber Chem. Technol., 44,1441 (1971).
- (5) Ranney, M.W. and Pagano, C.A., Rubber Chem. Technol.44, 1080 (1971).
- (6) Sperry Research Center, February 21, 1978, "Low Temperature Geothermal Conversion Systems", SCRC-317-16C, Sudbury, Mass 01776.

5.2.3.2.2 Packer Cut Test Procedure

A test facility was designed and constructed by Barber-Nichols Engineering Co. to test packer cups under realistic conditions. A photo is shown in Figure 19.

The objective was to operate the packer cups under simulated well conditions to verify performance in terms of: 1) leakage before and after experiencing cyclical operating conditions, 2) the forces involved in moving the cups, and 3) the ability to seal and reseal.

The approach was to: 1) install packer cups in a section of water-filled, unconditioned well casing, 2) maintain the water temperature, water differential pressure, and the presence of well sand that simulates well conditions, and 3) put the cups through the cyclical motion typical of system startup, operation, and restart. During the tests, which were accelerated greatly relative to actual field-test time variations, the temperature, pressure, leakage, and forces were monitored. The differential pressure was maintained constant during cup motion, which was imposed by an independent hydraulic system. Two cups were mounted on a shaft, opposing each other, so that the differential pressure across the cups would not transmit a force to the hydraulic system.

The test fixture was oriented vertically. The water within the fixture was heated by electrical band heaters around the outside of the fixture and wrapped with blanket insulation to retain the heat. The temperature was maintained at 350°F for all tests.

The tests were done in two parts, one with sand, and one without sand. The test procedures were as follows.

Tests Without Sand

Set the pressure in the cavity between the two cups at 1050 psig. Set the pressure on the opposite sides of the cups at 425 psig (for a nominal design pressure across the cups of 625 psi). Go through the following steps.

- 1) Leave packer in initial position for five minutes, compensating for any leakage to the low pressure sides by manually bleeding off water

to maintain the set pressure of 425 psig. To determine leakage rates, collect bleed water and record the times involved.

2) Move packer approximately one inch to a new position by using the hydraulic system to slowly increase the force on the packer assembly until it moves. Record the peak pressure at which movement occurs. Repeat step 1.

At each step, record water temperature, all pressures, times, packer position, peak hydraulic pressure at which packer moves, and volumes of bleed water. Note any irregularities that occur.

3) Repeat step 2 three times.

4) Repeat step 2 three times, but using 2-inch instead of 1-inch displacements.

5) Repeat step 2 three times using 3-inch displacements.

6) Repeat step 2 one time using 10-inch displacement.

7) Repeat steps 3 through 6 ten times.

8) Repeat steps 1 through 7 for a low-pressure setting of 750 psig, corresponding to a pressure across the cups of 300 psi.

9) Move packer approximately 2 inches, reduce all pressures to zero psig, wait for five minutes, then repressurize to the original 625 psid conditions of step 1. Check leakage rates for five minutes.

10) If leakage rates not excessive in step 9, leave in position for one week. Maintain pressures and monitor regularly. After one week, repeat steps 3 through 6 one time. Disassemble and inspect.

Tests With Sand

Cover bottom cup with special well sand to a depth of approximately 2 inches and reassemble. Repeat steps 1 through 10.

5.2.3.2.3 Packer Cup Design

The initial packer cup design is shown in Figure 20. This design failed in Laboratory Testing because of inelastic deformation of the spring-wire fingers and separation of elastomer from the fingers. Attempts to use tube-reinforced wire fingers also failed and ultimately the design of the fingers was changed to the much stronger T-bars shown in Figure 21. These were placed in the cup as shown in Figure 22.

5.2.3.2.4 Cup Test Results

Approximately 12 packer cups were tested. These represented three different structural design approaches for the support fingers: 1) the initial spring-wire fingers which failed at 300 psid 2) tube-reinforced spring-wire fingers which failed at 500 psid and 3) T-Bar fingers which passed the test procedure in at least 4 cups. There was also an early problem in establishing an initial seal, which was solved by changed lip design and the fit of the cup in the casing. There was also an interim problem with elastomer bonding to the base-plate and the fingers, which was solved by a process change instituted by Regal.

Three different cups passed the total test procedure and were assembled into a three-cup assembly, which was at well-site for use in the initial pump test at the time the well liner collapsed.

5.2.3.3 Fixed Packer Approach

Because the sliding packer cups were still failing tests in the fall of 1980, it was decided to pursue a back-up approach to achieving the seal to the well-casing at the pump and the sliding seal to accommodate differential expansion between the well casing and the heat exchanger.

The design of the fixed Baker packer is shown in Figure 23. This packer rests on the tapered transition between the 20" and 13-3/8" well casings. It extends into the 13-3/8" casing and makes a fixed seal to the I.D. of that casing. The seal is established by a limited compression of an elastomer ring produced by the weight of the down-well equipment. The production tube hangs from the packer assembly. A fixed packer was fabricated and available for the pump test if needed.

Because the packer is fixed, there must be a sliding seal somewhere in the heat exchanger conduit. The preferred location is at wellhead. There are several reasons for this choice. The full weight of the down-well equipment then rests on the fixed packer assembly to set it. The sliding seal can be replaced easily without removing any other down-well hardware. The sliding seal is located where sand and debris will not settle into it causing jamming and leaks. A sketch of this well-head sliding seal assembly is shown in Figure 24. The 18" O.D., 0.938" wall tube is welded to the top heat exchanger section and has a polished bore to receive the seal assembly. The seal assembly is shown in more detail in Figure 25. This is captured between the casing flange and the next wellhead spool lower flange. EPDM seal rings are bonded into the six grooves. The vent between the upper and lower groups of three seal rings provides leak detection and is a positive protection against brine leakage into the R-114 system. A seal was fabricated and pressure-tested and was available for the pump test if needed.

5.2.3.4 Riser Sliding Seal

The other large sliding seal is in the riser pipe. For practicality of installation there is one such seal for each heat exchanger section. Since these seals separate R-114 at low pressure from R-114 at high

pressure, leakage is not a source of fluid contamination and slow leakage can be tolerated. High leakage would degrade system performance however, so a positive seal is required. A drawing of the downwell heat exchanger coupling is shown in Figure 26. The sliding seal need accommodate only the differential expansion of one section. Sliding seals of this type were fabricated and tested successfully by Regal Tool and Rubber Co. Only one was needed for the pump test since it used a one-section heat exchanger. It was located just above the turbine pump unit. The EPDM elastomer was used here also.

5.2.3.5 Pump System Down-Comer Seal

For the pump system only, there was a requirement for a stabbled sliding seal at the bottom end of the R-114 4" down-comer. The receptacle for this is above the one-section heat exchanger. A seal assembly was designed, fabricated and tested by Regal and was available for the pump system test. A sketch is shown in Figure 27.

5.2.4 Lube and Instrument Lines

Both systems require make-up lubrication oil conduit and a one-conductor electrical lead for instrumentation from the surface to the down-well TPU. The design approach taken was to thread the electrical cable (a small standard 1-conductor, teflon-insulated, armored logging cable) through a 1/2" OD X 0.035" Wall Stainless Steel tube of sufficient length to extend from wellhead to TPU. A combined electrical and lubricating oil stabbing fixture and weight are affixed to the lower end and a feed-through is on center-line at the top of the wellhead. Oil line and cable is run-in as a unit after all other down-well hardware is in place.

One question with this approach was the practicality of threading the electrical cable through the tubing. By laying the tubing out straight on the ground it was found to be easy to blow a pig with a light line through a 2300' long tube and to pull the logging cable through.

Another question was the potential failure of the tube after several installations with repeated spooling and unspooling from a storage and installation drum. Tests to check this were made by Mustang and are reported in Appendix B.

Another design problem was the downwell stabbing fixture. The version of this fixture for the pump system is shown in Figure 28.. Pressure and temperature tests were conducted by Mustang Engineering on the elastomer seal/electrical connection portion of this assembly. Their report is attached as Appendix C.

5.3 TURBINE-PUMP-UNIT DEVELOPMENT

5.3.1 Introduction

A key component of the gravity-head and pump system is the down-well turbine-pump unit (TPU). The TPU performance requirements must be tailored to the system parameters. Also, the cost of retrieval for maintenance dictates extreme reliability.

The development of the TPU was subcontracted to Barbher-Nichols Engineering Co., who had developed the similar, steam-driven TPU used successfully in the earlier down-well pump system test at Heber, CA.

Two technical reports submitted under this contract by Barber-Nichols are attached as Appendices D. and E. Appendix D. covers the design and predicted performance of the TPU. Appendix E. covers the test programs conducted on both components of the TPU and a complete TPU. Figure 29 is a layout drawing of the TPU which was fabricated and tested in the laboratory and was available at field test site for the pump system field layout.

5.4 FRANCIS TURBINE DESIGN STUDY

The down-well turbine-pump unit (TPU) for the gravity-head system has many operating parameters and constraints in common with the TPU for the pump system. For example, the pressure-added, flow-rate and, therefore, the power required of the pump are nearly the same and the overall diameter and required reliability are the same. Therefore, many design features of the pump system TPU can be carried over to the gravity-head system, such as pump design, bearings, shaft seals, static seals, lubrication system, materials selection, etc. However, there is an essential difference in the turbine design parameters. Compared with the pump system the gravity-head system requires a much larger working fluid flow-rate through the TPU turbine and a much smaller difference across the TPU turbine in working fluid pressure and enthalpy. These differences dictate an entirely different design approach to the gravity-head system TPU turbine. Questions of compatibility with the pump speed, the rotating group design and the diameter constraint must be addressed.

Fortunately, choice of a Francis-type turbine resulted in favorable answers to all these questions. In fact, it leads to an even better match between the desired operating speeds of turbine and pump than was possible for the pump-system TPU.

A design study for a Francis turbine for the gravity-head TPU was completed by Barber-Nichols Engineering Co. Their study report is attached as Appendix F. Figure 30. is a Machine Assembly drawing of the Francis turbine and Figure 31. is a drawing of a single blade.

5.5 HEAT EXCHANGER DEVELOPMENT

5.5.1 INTRODUCTION

The downwell heat exchanger is perhaps the most characteristic novel feature of the gravity head system. There were several study topics in the program aimed at generating design information for the downwell heat exchanger. Some of these have been covered earlier in this report. These include R-114 properties in Vol. I, the stability of R-114 and its compatibility with other materials in Vol. II, calculation of heat exchange requirements in Vol. III and welded connections and sliding seals in Section 5.2 of this volume. In attempting to calculate heat transfer in the supercritical region it became apparent that empirical data were required and an experimental task was planned to obtain data on heat transfer from brine to R-114 and transport properties of R-114 for the complete range of pressure, temperature and density to be encountered in the heat exchanger. The experimental apparatus designed and built for this purpose was called a "ten-pound loop" because its charge of R-114 weighed about ten pounds and it involved closed-loop flow of R-114 with both heating and cooling. This task is reported in Section 5.5.2 below.

The configuration of the downwell heat exchanger has two distinct portions. There is a lower portion consisting simply of two concentric tube strings. Other than the welded and sliding seal connections reported on in Section 5.2 above, this portion has no unusual design problems. The upper portion however consists of an annular tube bundle in the space between well casing and R-114 riser as shown in Fig. 15. This poses two interesting problems. One, how can roughly uniform brine flow be achieved throughout the tube bundle, and, two, how can one prevent any leakage at the joints between tubes and tube sheets? These tasks are reported in Sections 5.5.3 and 5.5.4 below.

5.5.2 TEN-POUND LOOP

An experimental apparatus called a "ten-pound loop" was designed and built to obtain fluid transport property and heat transfer data on R-114. A schematic diagram is shown in Fig. 32. A photograph of the completed apparatus is shown in Fig. 33. The long structure in the photograph contains the concentric tube heat exchanger portion of the apparatus. It is about 25 feet long and can be positioned horizontally or vertically to vary the gravity-head of the fluid over the length of the exchanger from zero to a maximum value. This feature permits separation of pressure differences due to flow loss and gravity-head (density). Another feature of the apparatus is use of flow meters for both R-114 and water which measure mass flow rate directly to avoid the assumption of viscosity values necessary in an aperture-type flow meter. A description of the flow meter used is given in Appendix G.

The design parameters for the ten-pound loop were as follows:

- o Inner HX Tube: 0.5" O.D. X 0.049" Wall, SS
- o Outer HX Tube: 0.75" O.D. X 0.050" Wall, SS
- o Length of HX Between Sensors: 20 Ft.
- o Nominal HX DT: 20°F.
- o Nominal Heat Exchanger Rate: 210 BTU/Min.
- o End Loss Heat Rate: 0.003 BTU/Min.
- o p = R-114: Nominal 3.48 psi @ 5'/sec.
- Water: Nominal 0.78 psi @ 3.5'/sec.

Two approaches to obtaining thermal conductivity measurements were considered. The first involves directly calculating k from the overall heat duty of the annular test section. These calculations, which are explained in detail later, are the primary data reduction means since a single experiment provides all of the necessary input data. As a check on these measurements, the Wilson plot technique is proposed as a secondary means of data reduction. The technique involves performing a series of experiments at a constant heat flux, operating pressure, and temperature in the organic loop; but at a series of mass flow rates in the "brine" loop. Only one thermal conductivity calculation is performed on this data set. The technique is useful in that data taken for the primary calculation scheme can be used as input for the Wilson plot technique after it has been categorized appropriately. A Wilson plot also provides an estimate of the degree of scatter in a given data set.

DIRECT THERMAL CONDUCTIVITY CALCULATIONS:

Measured Parameters: $T_{114 \text{ in}}, T_{114 \text{ out}}, T_B \text{ in}, T_B \text{ out}, \dot{m}_{114}, \dot{m}_B$

System Constants: $A_C, C, A_B, A_{114}, A_W, R_W$

Parameters determined from the above information:

$\text{LMTD}, \Delta T_{114}, C_{p114}, H_B, H_{114}$

The heat exchanger log-mean temperature difference and the heat transfer coefficient on the R-114 side of the loop are determined by:

$$\text{LMTD} = \frac{(T_{H_2O \text{ in}} - T_{114 \text{ in}}) - (T_{H_2O \text{ out}} - T_{114 \text{ out}})}{\ln \frac{(T_{H_2O \text{ in}} - T_{114 \text{ in}})}{(T_{H_2O \text{ out}} - T_{114 \text{ out}})}}$$

$$H_{114} = \left[\frac{(LMTD) A_{114}}{C (\dot{m} C_p \Delta T)_{114}} - \frac{1}{H_B (A_B / A_{114})} - \frac{R_w}{A_w / A_{114}} \right]^{-1}$$

For water on the organic side of the loop, k is determined by:

$$K_{H_2O} = (\alpha) (H_{H_2O})^{1.5} (\dot{m}_{H_2O})^{-1.2} (\nu_{H_2O})^{0.70} (C_{p_{H_2O}})^{-0.5}$$

where thermodynamic and transport properties are determined at the mean fluid temperature,

$$T_m = 1/2(T_{in} + T_{out})$$

and where $\alpha = 1.69 \times 10^{-4}$ (α is determined by system geometry constants)

For R-114 in organic side of loop, k is determined by:

$$K_{114} = (\alpha) (H_{114})^{1.5} (\dot{m}_{114})^{-1.2} (\nu_{114})^{0.70} (C_{p_{114}})^{-0.5}$$

where $\alpha = 1.69 \times 10^{-4}$.

THERMAL CONDUCTIVITY CALCULATION USING WILSON PLOT

The Wilson plot is a graphical interpretation of the overall heat transfer coefficient determining equation:

$$\frac{1}{U} = \frac{1}{H_0} + R_w + \frac{1}{H_i}$$

There are several tacit assumptions involved in using this technique which must be understood in order to accurately interpret test data. When these assumptions are recognized, however, the technique is a useful check on other means of data reduction. The assumptions involved in determining H_i from a data set are:

- Constant heat flux, Q/A
- Constant mass flow, $\dot{m}$ (or constant velocity, v_i)
- As wide a variation in mass flow, $\dot{m}_0$ (or v_0), as possible with the experimental apparatus.
- Known tube-wall materials so that R_w can be determined
- Heat transfer coefficient, H_0 , is a function of $(v_0)^{0.8}$.

The Wilson plot technique derives from the fact that the ideal way to measure the tube-side heat transfer coefficient, H_i , in a shell-and-tube heat exchanger would be to set the shell-side heat transfer coefficient, H_0 , infinitely high. In this limit, the overall heat transfer coefficient would be simply

$$\frac{1}{U} = \frac{1}{H_i} + R_w$$

In order to approach this limit, the Wilson plot technique suggests repeating overall heat transfer measurements at a constant overall heat flux and a constant tube-side velocity, while varying the shell-side velocity over a wide range. Inherent in this technique is the assumption that the individual heat transfer coefficients are primarily a function of Reynolds number, $(Re)^{0.8}$, or velocity, $v^{0.8}$; an assumption which is made in most empirical heat transfer coefficient correlations. For each constant heat flux data set, $(1/v_0^{0.8})$ is plotted on the abscissa vs. $(1/U)$ on the ordinate. The resulting plot is a straight line with a positive y - intercept, which by inspection is the value of $(1/U)$ for infinite $v^{0.8}$ (or infinite heat transfer coefficient, H_0). Therefore, the y - intercept equals $(1/H_1 + R_w)$. Since R_w is known, H_1 has been determined.

An advantage of the Wilson plot is that it provides an estimation of the degree of scatter in the test data and determines H_1 as the "best-fit" of all data taken. A disadvantage of the technique is that a large data set is required for accurate data interpretation. This fact makes the technique more useful as a check rather than as a primary means of data reduction.

NOMENCLATURE

A_B	Brine-side heat transfer area in annular heat exchanger (ft^2)
A_{114}	R-114 - side heat transfer area in annular heat exchanger (ft^2)
A_w	Tube-wall heat transfer area of inner tube in annular heat exchanger (ft^2)
C	Conversion factor (min/hr)
C_p	Specific heat at constant pressure (Btu/lb $^{\circ}\text{F}$)
H	Heat transfer coefficient (Btu/hr - ft^2 - $^{\circ}\text{F}$)
k	Thermal conductivity (Btu/hr - ft - $^{\circ}\text{F}$)
LMTD	Heat exchanger log-mean temperature difference ($^{\circ}\text{F}$)
$\dot{m}$	Mass flow rate (lb_m/min)
Q	Heat duty (Btu/hr)
R_e	Reynolds number (dimensionless)
R_w	Tube-wall heat transfer resistance of inner tube in annular heat exchanger ((Btu/hr - ft^2 - $^{\circ}\text{F}$) $^{-1}$)
T	Temperature ($^{\circ}\text{F}$)
ΔT	Temperature difference ($^{\circ}\text{F}$)
U	Overall heat transfer coefficient (Btu/hr - ft^2 - $^{\circ}\text{F}$)
v	Fluid velocity (ft/sec)
α	Conversion factor ($\text{ft}^{2.7}(\text{hr}/\text{min})^{1.2}$)
μ	Velocity (lb_m/Hr - ft)

Subscripts:

B	Brine or brine side of test loop
114	Organic or organic side of test loop
H_2O	Water
i	Quantity referenced to the inside of the HX tube
in	Annular heat exchanger inlet
m	Mean
o	Quantity referenced to the outside of the HX tube
out	Annular heat exchanger outlet

The ten-pound loop apparatus was constructed and placed in initial operation. However, contract funds were curtailed before good data were obtained. The apparatus has been transferred with DOE permission to the Mechanical Engineering Dept. of Brown University, Providence, R.I.. They have set it up in their laboratory and intend to use it to obtain data on R-114 and other working fluids of interest.

5.5.3 Tube Bundle Geometry

In the pump system, there is only one heat exchanger section, a tube bundle unit located just above the downwell turbine - pump unit. Since the ratio of R-114 flow rate to brine flow rate is < 1 and the average temperature difference between R-114 and brine is large, the heat exchanger design is non-critical and easy. However for the gravity-head system, the ratio of R-114 flow-rate to brine flow-rate is > 1 , the temperature difference is small and the heat transfer rate per unit of length must be tailored to a desired function. Therefore, the tube bundle design is critical and difficult. Since the area for R-114 flow must be $>$ the area for brine flow, the tubes must be tightly packed and there must be small clearance with the well casing. Also the space for brine flow must be evenly distributed between the tubes to ensure uniform brine flow velocity and heat transfer. Computer programs were written to generate tube bundle designs under constraints, such as tube diameter, ratio of R-114 to brine flow area, maximum and minimum gap dimensions between tubes, inscribed and circumscribed circle diameters, R-114 flow area, etc. These programs would generate tube bundle layouts and derived parameters, such as number of tubes, brine flow area, etc. typical computer-generated tube layout

is shown in Figure #34. Note that it is not possible to achieve uniform spacing for brine flow. This is not a problem for the pump system, but would be for the gravity-head system. An alternative design approach for the tube bundles for the gravity-head heat exchanger is to spiral the tubes through $1/2$ turn over their length, with adjacent layers spiraled in opposite sense. This configuration would be expected to result in turbulence and thorough mixing of the brine flow. Heat exchanger manufacturers were consulted concerning the feasibility and cost of fabricating such a configuration. They agreed that it could be done and that it was a more practical and less expensive approach than trying to add baffles in such close-packed bundles. However, no such design was fabricated prior to termination of funding.

5.5.4 Tube Sheet Seals

One of the major concerns of the downhole equipment design was to minimize the possibility of leaks because much of the equipment would be inaccessible for repairs. A tubesheet leak, for example, might require pulling the entire downhole string in order to gain access to the defective part. Since tubesheets are notorious for becoming the "weak link" in the integrity of tubular heat exchangers, particular attention was paid to their design in the Gravity Head downwell system.

A double tubesheet was designed to accept double-rolled, welded tubes and, as a backup, to accept and contain an injected sealing compound between the two sheets. Test sections of the design were fabricated and experiments were run using various rolling, welding, and backup sealant combinations. The experiments, which are described in Appendix H, "Double Tubesheet Leak Test Report" from Mustang Engineering Company, resulted in a successful design which was utilized in the fabrication of the downwell heat exchanger which was to be used with the Stand Alone Organic Turbine Pump Unit (TPU). The same design approach was planned to be used in the gravity head system also.

5.6 ENTHALPY RECOVERY UNIT

The "enthalpy recovery unit" or ERU is not essential for the operation of a binary-type geothermal plant. However, from a thermodynamic viewpoint, the inclusion of an ERU produces certain advantages, the main one being an improvement in the cycle efficiency. The design of the downhole heat exchanger is predicted on the "enthalpy recovery" to be provided by the ERU.

The ERU is shown in Fig. 35. The basic purpose of the ERU is to capture some of the energy (heat) remaining in the R-114 exhaust gas leaving the turbine. Since this gas is still relatively hot (approx. 106°F), it can be used to preheat the R-114 liquid prior to its being piped to the main downhole heat exchanger. The temperature of the R-114 liquid as it leaves the BAC units is about 85°F. In many applications a unit such as the ERU is called an exhaust-gas heat recuperator.

The design calls for a direct-contact type unit in which the exhaust gas is allowed to bubble up through a falling liquid stream. Excellent heat transfer can be expected as long as the tray remains free of contaminants that might clog the holes through which the fluids must pass.

The barometric leg is supplementary to the ERU. By allowing for an elevation difference between the BAC outlet and the liquid level in the ERU, this component can restore nearly all the pressure that was lost in the coils of the BAC due to friction.

The use of the ERU reduces the heating duty of the geothermal brine. Assuming a constant brine flow rate, this means that the exit temperature of the brine as it emerges from the downhole heat exchanger will be somewhat higher than if the ERU were not used. This may have certain chemical advantages with regard to scale formation.

Mechanically, the ERU is a straightforward system similar apparatus is commonly found in chemical process plants. With proper maintenance it should perform reliably.

Fig. 1 Downwell Data Acquisition

Fig. 3 Hoisting Drum for Lube/Signal Line

LEGEND

AR ALARM RECORDER
 ATR AIR TEMPERATURE RECORDER
 ATT AIR TEMP. TRANSMITTER
 COU COUPON (CORROSION MEASUREMENT)
 CPL CORROSION PROBE (MULTIPOINT RECORDER)
 CPR CORROSION PROBE (PORTABLE INSTRUMENT)
 D DRYER
 CZ CHECK VALVE

E- HEAT EXCHANGER

F- FILTERS & DRIERS
 FAL FLOW ALARM LOW
 FCV FLOW CONTROL VALVE
 FE FLOW ELEMENT
 FI FLOW INDICATOR
 FIC FLOW INDICATING CONTROLLER
 FR FLOW RECORDER
 FRC FLOW RECORDER CONTROL
 FSH FLOW SWITCH HIGH
 FSL FLOW SWITCH LOW
 FT FLOW TRANSMITTER

GV GAUSS VALVE
 H HEATER
 HV HAND VALVE
 LC LEVEL CONTROLLER
 LAH LEVEL ALARM HIGH
 LAL LEVEL ALARM LOW
 LIS LEVEL INDICATOR SWITCH
 LG LEVEL GAUGE
 LR LEVEL RECORDER
 LSH LEVEL SWITCH HIGH
 LT LEVEL TRANSMITTER
 LCV LEVEL CONTROL VALVE
 LIS LEVEL INDICATING SWITCH
 MS MASTER SWITCH
 PAH PRESSURE ALARM HIGH
 N.C. NORMALLY CLOSED
 N.O. NORMALLY OPEN
 PAR PRESSURE ALARM RECORDER
 P- PUMPS (COMPRESSORS)
 PAL PRESSURE ALARM LOW
 PCV PRESSURE CONTROL VALVE
 PDI PRESSURE DIFFERENTIAL INDICATOR
 PDR PRESSURE DIFFERENTIAL RECORDER
 PDT PRESSURE DIFFERENTIAL TRANSMITTER
 PI PRESSURE INDICATOR
 PR PRESSURE RECORDER
 PRV PRESSURE RELIEF VALVE
 PSH PRESSURE SWITCH HIGH
 PSL PRESSURE SWITCH LOW
 PSV PRESSURE SAFETY VALVE
 PT PRESSURE TRANSMITTER
 PIC PRESSURE INDICATING CONTROLLER
 RD ROTURE DISC
 S- STRAINER
 SG SIGHT GLASS
 ST STATUS SIGNAL
 TI TEMPERATURE INDICATOR
 TR TEMPERATURE RECORDER
 TT TEMPERATURE TRANSMITTER

V- VESSELS & TANKS

WBR WET BULB RECORDER
 WBT WET BULB TRANSMITTER
 WR WATT RECORDER
 WT WATT TRANSMITTER

PAH PRESSURE ALARM HIGH
 X SPECIAL EQUIPMENT
 PS PRESSURE SWITCH
 LSL LEVEL SWITCH LOW
 RPMR RPM RECORDER

✕ HAND VALVE

✕ GLOBE VALVE

⌞ CHECK VALVE

⊙ BALL VALVE

⌞ BUTTERFLY VALVE

△ ANGLE VALVE

⌞ PSV-PRV-PCV

⌞ HAND-THROTTLED VALVE

⌞ CONTROL VALVE

⌞ BACKPRESSURE REGULATED VALVE SELF-CONTAINED

⌞ PRESSURE REDUCING REGULATED VALVE SELF-CONTAINED

⌞ BACKPRESSURE REGULATED VALVE WITH EXTERNAL TAP

⌞

→ PIPING LINE

→ INSTRUMENT AIR

--- ELECTRICAL SIGNAL

○ LOCAL INSTRUMENT

○ MULTIFUNCTION INSTRUMENT

○ PANEL MOUNTED INST.

○ SCIENTIFIC INST.

○ ROTAMETER-TYPE FLOW INDICATOR

○ ROPTURE DISC

○ QUICK-CHANGE FLOW ELEMENT

Y- Y-STRAINER

⌞ FLEXIBLE HOSE

⌞ HOSE COUPLING

○ SCIENTIFIC INSTRUMENT ALSO ON PANEL

○ NORMAL STOP RELAY 286

○ HIGH PRESSURE SHUT DOWN RELAY 286

○ R-114 EVACUATION & SHUT DOWN RELAY 386

2/27/74	DCX	△	ADDED CPL, CPR NUMBERS
12/1/73	DWB	△	MISC. REVISIONS
9/25/73	HHY	△	MISC. REVISIONS
8/12/73	HHY	△	MISC. REVISIONS
7/30/73	HHY	△	MISC. REVISIONS
5/21/73	HHY	△	MISC. REVISIONS
5/14/73	HHY	△	REMOVED H.O. MAKE-UP ON E1, E2
4/7/73	SEW	△	REVISED V-11, V-15, INSTRUMENTS
4/2/73	D.M.R.	△	MISC. REVISIONS
4/1/73	CF	△	ADDED P-16 & 17
4/1/73	CF	△	REVISED B-10TH, B-11TH ADDED (P) 1
3/1/73	DWB	△	MISC. REVISIONS
2/7/73	CF	△	REVISED VACUUM SYSTEM
2/1/73	DWB	△	MISC. REVISIONS
1/2/73	CF	△	MISC. REVISIONS
1/1/73	DWB	△	MISC. REVISIONS
1/1/73	CF	△	MISC. REVISIONS
1/1/73	CF	△	ADD V-77 RELATED PIPING - B-10TH, B-11TH
1/1/73	CF	△	REDRAW

SAC DWG# G-80-79 REV. 2

REFERENCES

REVISIONS

FIG. 4

PUMP TEST SYSTEM			
EAST MEA, CALIF.			
SPERRY RESEARCH CENTER			
SUDBURY, MASS.			
Mustang ENGINEERING COMPANY			
13220 CHAMBERLAIN STREET			
DANTA RE, SPRINGFIELD, CALIFORNIA 95830			
DESIGNED BY	C. F. FINE	DATE	11/1/73
CHECKED BY	W. E. ELLIS	SCALE	
APPROVED BY			
7200-R-004A

FIG. 5 Organic Pump System Control Schematic.

FIG. 6 Mimic Board Layout.

Fig. 7 Pump System Control Panel

Fig. 8 Geothermal Pump Instrumentation System

Fig. 9 Surface Data Acquisition

Fig. 10 Data Storage and Control

FIG. 11 Gravity Head System Control Schematic.

BRINE _____
 R-114 _____
 ELECT _____

Fig. 12 Mimic Board Layout

SUDBURY, MASS.

TITLE

MIMIC BOARD LAYOUT

SCALE

NONE

DATE

11-13-79

DWG. NO.

G-72E-79

REV. NO.

0

FIG. 13 Main Single Line Diagram.

FIG. 14 Gravity Head System Schematic.

Fig. 15 Typical Tubular Heat Exchanger Section

Fig. 16 Automated Welding Lathe

Casing

Fusion Zone

Weld Filler Metal

75X

Figure 17: Photomicrograph of welded section of casing.

The weld is sound, free of porosity, and relatively free of slag. The vertical black streaks readily visible in the fusion zone are inclusions (probably silicate) normally present in the pipe. They are elongated during forming of the pipe.

Casing

Fusion Zone

Weld Filler Metal

150X

Figure 18. A higher magnification photo of the region shown in Figure 17. Inclusions present in the casing are more visible but the weld is of a high quality.

Fig. 19 Packer Cup Test Facility

Attn: Hugh Matthews

FIG. 21 T-Bar Fingers for Packer Cup.

ATTN
HUGH MATTHEWS

FIG. 22 Modified Cup Design Cross-Section.

NAME **BRINE WATER COMPRESSION
PACKER - PROD. 420-99-**

DWG.
NO.

**243
995**

DRAWN **T. GARDUNO**

APPROVED *J. K.*

DATE **11-10-80**

REVISION DATES:

DWG.
NO.

**243
995**

ENG. REPORT NO.

Fig. 23 Fixed Baker Packer

PREPARED BY B. JOCKES / REZAL TER	SPERRY RESEARCH CENTER ENGINEERING CALCULATION FORM	PAGE 1 OF 1
CHECKED BY		PROGRAM AND/OR TASK NO.
NO SCALE	SUBJECT SPERRY PUMP SYSTEM WELLHEAD SLIDING SEAL.	DATE 10.6.80

PRELIMINARY

18" O.D. PIPE

I.D. 16.115"

ROUGH .02830"

FINISHED ID 16.155"

RADIAL GAP .078"

STABBING PIPE + SEAL 16.25" O.D.

18" O.D., 2480
10.938" WALL

16" O.D. x 14 3/4" I.D.

16" O.D. x 14" I.D.

BLEED PORT

44"

GROUND LEVEL

11/28/79

DATE OF ISSUE **DEC 17 1980**

- ☒ PRELIMINARY
- ☐ CHECK PRINT
- ☐ ENGINEERS APPROVAL
- ☐ CUSTOMER
- ☐ CONSTRUCTION
- ☐ CERTIFIED DRAWING

BY _____

☐ AS BUILT

SECTION A-A

Fig. 25 Wellhead Sliding Seal Assembly

**16" CASING HANGER, REGAL SLIDING SEAL
WITH 3/4" SPLIT TUBE**

ORDER $\frac{1}{2}$ " = 1'-0"	APPROVED BY:	DRAWN BY DEL
DATE:		REVISED

**SPERRY RESEARCH CENTER
SUDBURY, MASS.**

**MUSTANG ENGINEERING CO.
13230 CAMBRIDGE ST.
SANTA FE SPRINGS, CAL. 90670**

DRAWING NUMBER
7975-B-912

Fig. 26 Downwell Heat Exchanger Coupling

01.03.101

SPERRY RESEARCH CENTER			
SUDBURY, MASS.			
TITLE			
SCALE	DATE	DWG. NO.	REV. NO.

FIG. 27 R-114 Downcomer Seal.

FIG. 29 Pump System TPU Layout.

Y AXES	X AXES	R ₁	φ	R ₂	θ	NOSE DIA.
-0.91	-1.72	1.72 ± .14	-80.8°	0 ± .14	94.6°	.076
-1.27	-1.22	1.22 ± .06	-79.7°	0 ± .06	92°	
-1.66	-1.90	1.90	-77.8°	.04	87.8°	
-1.86	-1.90	1.90	-77.9°	.12	87.8°	
-2.17	-2.06	2.06	-76.1°	.20	85°	.076
-2.46	-2.14	2.14	-72.8°	.28	82.6°	.062
-2.61	-2.23	2.23	-68.7°	.37	80°	.050
-2.81	-2.23	2.23	-65.9°	.43	78.4°	
-2.92	-2.27	2.27	-47.1°	.51	76°	
-2.90	-2.27	2.27	-46°	.55	75°	
-2.64	-2.42	2.42	-45°	.56	70°	
-2.70	-2.54	2.54	-44°	.61	67°	
-2.90	-2.51	2.55	-45°	.66	64°	.050

NOTE: SEE BLADE TABULATION FOR SECTION CONFIGURATIONS.

Fig. 31

UNLESS OTHERWISE SPECIFIED		ITEM NO.	QTY	PART NO.	DESCRIPTION	SPECIFICATION
INTERPRET PER ANSI Y14.5-1973 ANGULAR DIMENSIONS = 9° SURFACE ROUGHNESS 32-INT/IN TOLERANCES ON DECIMALS 1 PLACE 2 PLACE 3 PLACE .005 .002 .001 REMOVE BURRS SPEAK EDGES 1/32" MIN 1/16" MAX		THIRD ANGLE PROJECTION MATERIAL NOTED - SH-1 HEAT TREATMENT PROCESS		APPROVED DATE 2-11-81 2-15-81		BARBER-BROOKS ENGINEERING
TITLE PROTOTYPE, TURBINE BLADE GRAVITY HEAD TPU						SIZE CODE IDENT NO. DRAWING NO. E 57377 60P-1700
SCALE 1/1 FWT ACT CALC SHEET 2 OF 3						60P-1700

NOTE: Δ INDICATES TRANSITION ZONE @ RADIUS R_0 .

NOTE: Δ INDICATES TRANSITION ZONE @ RADIUS R_0 .

[illegible]

NOTE: SEE BLADE NOMENCLATURE FOR ENGINEERING DATA.

Figure 32 Fluid Test Apparatus

Figure 33. Ten Pound Loop

MECHANICAL DESIGN

08/27/79 14:05:01 PRODUCTION SECTION .750 278 23 17.7500 .9375 20.7500 .9375

Figure Fig. 34 Typical Tube Bundle Layout

81-435

FIG. 35 Emthalpy Recovery Unit.

APPENDIX A

OPERATING INSTRUCTIONS
FOR
SCIENTIFIC INSTRUMENTATION
SYSTEM

OPERATING INSTRUCTIONS
FOR
SCIENTIFIC INSTRUMENTATION
SYSTEM

GEO THERMAL PUMP PLANT

MAY 29, 1981

D. Peterson
P. Schottler

INTRODUCTION

This manual contains operating instructions for the scientific instrumentation system for the geothermal pump plant. The scientific instrumentation system is a time-division multiplexed data acquisition and control system which consists of four main parts: 1. A Univac V77/600 minicomputer with 128K words of 16 bit memory, a 10M byte disk, and one magnetic tape drive, 2. A Cutler-Hammer multiplex system consisting of two mux stations (one in the control box and one near the ERU) and a communications station (in the instrumentation van), 3. Various discrete and analog sensors as specified on the pump plant P&ID, and 4. The downwell instrument package. The computer and Cutler-Hammer communications station are located in the Sperry Instrumentation Van, and the scientific instrumentation system is run from the computer terminals located in the van.

This manual is organized into seven sections. The content of the various sections is as follows:

- Section 1: Contains instructions for starting-up the scientific instrumentation system.
- Section 2: Contains instructions for running the scientific instrumentation system from the computer consoles. The various options available to an operator are illustrated as well as the prompting available at the consoles from the computer software.
- Section 3: Contains instructions for shutting-down the instrumentation system.
- Section 4: The various signals that are recorded by the scientific instrumentation system from the surface plant are related to the corresponding transmitters, control box signals, and local field indicators as shown on the pump plant P&ID.
- Section 5: The Cutler-Hammer mux system can be used to display data from the surface-plant sensors independently of the computer-driven system. Section 5 contains instructions for displaying data on the Cutler-Hammer monitor panel (located in the Sperry van).

Section 6: The computer software makes use of three disk files which contain reference information on the signals and sensors which tie into the scientific instrumentation system. These three files are (i) Cutler-Hammer file (CHFILE), which contains information on all of the signals which tie into the scientific instrumentation system via the Cutler-Hammer mux, (ii) Downwell file (DWFILE), which contains information on the signals from the downwell instrument package, and (iii) Engineering file (ENGFIL), which specifies the ranges and units of the analog (4-20 ma.) signals in CHFILE and all of the signals in DWFILE. Listings of these three files are included in Section 6 for reference purposes.

Section 7: Contains an octal-decimal integer conversion table.

SECTION 1

DATA ACQUISITION START - U P

The purpose of this document is to summarize the start up procedure to enable the computer to take data. There are 3 sub-procedures: power, computer operating system, and the data collection programs.

A) Internal Van Power

- 1) Turn on the power switches on the back right hand wall of the van.
 - Red - Computer power
 - Orange - Peripheral power (tape, Cutter-Hammer, terminals, (There is an additional circuit breaker in the blue rack behind the Cutter-Hammer monitor panel. All the peripherals are on it. It is normally on. If the Cutler-Hammer monitor panel (blue rack) shows "MEMORY PARITY ADDRESS 7777", switch peripheral power off and then on again. System normal (green) LED should be lit on the monitor panel.)
- 2) Computer
 - Turn the four position key located on the computer panel to ON.
 - *** WARNING DO NOT RUN THE COMPUTER WITH OUT AIR CONDITIONING
- 3) Disk
 - Turn disk power to ON. Position thumb wheel switch to 1.
 - Press the RUN/STOP button. The computer system will be ready when the READY light on the disk is ON (3-5 min.).
- 4) Tape
 - Press the top RED power switch on the mag tape unit.
- 5) Terminals & Printers
 - Most terminals and printers have power switches on the back, some on the front, turn them all on.
- 6) Cutler-Hammer Mux stations
 - Mux-1: Turn on the red circuit breaker on mux-1 inside the control box. (normally on)
 - Mux-2: Turn on the AC power and the fan.

B) Computer Operating System - VORTEX

- 1) First VORTEX must be loaded from the disk into the computer memory (booting). This is done using the buttons on the computer's front panel.

PANEL MODEL

D x xxx xxx xxx xxx x1x
 x C x L
xxx xxxxxPrxxxxxx R Sx

C - Clear register select
D - Display clear
L - Load
P - P-register
R - Reset
r - Register select
S - Step/run
1 - Boot device address
x - Not used

VORTEX will not boot if the disk is not READY

VORTEX BOOT

- Press R (Reset)
- Press P (P-register)
- Press D (Display clear)
- Press r (register select)
- Press C (Clear register select)
- Press D (Display clear)
- Press 1 (boot device address)
- Press R (Reset)
- Press S (Step/run)
- Press L (Load)

Boot summary : R P D r C D 1 R S L

VORTEX will now boot. There will be some disk activity
and some messages on the OPCOM terminal. ##

The booting procedure is completed on the OPCOM terminal.

* to erase an incorrect line type \ (back slash) then retype*

Enter date: MM/DD/YY ex. 04/28/81 (return)

Enter time: HHMM ex. 1304 (return)

Type ^I (control I ... while pressing down the CTRL key type I),

Enter /OPEN,LO,CRT4,244 (return)

This completes the booting of VORTEX. We'll now log on
to terminal CRT1 and start the data collection programs.

C) Data Collection Programs

We'll take you through a sample start up and exercise
one of the program's control options. In the example user
responses will be followed with " (return) ", and comments will be

bracketed by *. Log on to CRT1.

(return)

USERID:

GEO (return) * User ids are secret and DONOT print on the terminal *

PASSWORD: * if typos are made hit return until asked for USERID:*

(return) * to delete a line type \, a character type backspace *

* log on completed *

* Assign data collection program files type: *

AS,#1,CHFILE,DT (return)

AS,#2,DWFILE,DT (return)

AS,#3,ONELIN,DT (return)

AS,#4,ENGFIL,DT (return)

* Mount a tape for the acquisition program *

* Place a tape with a write ring on the lower drive. *

* Thread the tape using the picture in the upper left *

* of the mag tape unit. *

* Press the LOAD button twice to bring the tape to *

* load point. The tape should stop at load point in a *

* few moments. If the load point was missed press REWIND. *

* Press ON LINE. *

*** TNSLCT can be run even if the mag tape is broken. You could ***

*** use the programs for display only. To do this on OPCOM type ***

*** /ASSIGN,18,DUM (return) ***

* Run the collection program TNSLCT. (sample start up) *

R,TNSLCT,L1 (return)

SELECT JOB TYPE:

0 - STOP

1 - SURFACE AND DOWN WELL WITH DATA COLLECTION

2 - DOWN WELL ONLY WITH DATA COLLECTION

3 - SURFACE ONLY WITH DATA COLLECTION

4 - CALIBRATION

3 (return) * default type = 1 *

SELECT GLOBAL TASK PARTITION ASSIGNMENT OPTION:

1 - PROCEED WITH DEFAULT TASK PARTITIONS

2 - EXAMINE/MODIFY TASK PARTITIONS

(return) * default = 1 *

* TNSCLT is now active and collecting data *

HELP (return) * Type HELP to view and select program options. *

Select change code:

0 - Stop

1 - Change TCHACQ sample rate

2 - Change Cutler-Hammer outputs

3 - Change delays between formatted outputs on devices

- 4 - Change Cutler-Hammer read status word flag
- 5 - Write END OF FILE on output tape
- 6 - Indicate new tape loaded
- 7 - Change delay for unsolicited inputs
- 8 - Change DWACQ sample rate
- 9 - Print Cutler-Hammer error counters
- 10 - Suspend/resume tape writing
- 11 - Compose a line of sensor names for single line printout
- 12 - Force output of the single line print
- 13 - Retrieve pre-stored single line printout sensor names
- 14 - Flip flop outputs between octal and engineering units

SECTION 2

DATA ACQUISITION PROGRAMS (TNSLCT)

The data acquisition programs are written to collect surface and down well data from the geothermal test site. A good deal of human engineering went into the design, to make a friendly, informative system. We can monitor, collect and display any of the surface and down well sensors. There is also a control "output" capability built in. We can change any of the analog and discrete outputs on the Cutler-Hammer system. Presently the control is manual, future designs will include automated control.

The program is "gorilla" proof. By that I mean users should not be afraid to type on the keyboard. Incorrect entries are ignored and the user is requested to retry with informative prompts. Program control options may always be viewed by typing HELP.

We'll take you through a sample start up and exercise each of the program's control options. In the example user responses will be followed with " (return) ", and comments will be bracketed by *. Start the computer as described in the start up procedure, Section 1. Then log on to CRT1:

(return)

USERID:

GEO (return)

PASSWORD:

(return)

* log on completed *

* Assign data collection program files,type: *

AS,#1,CHFILE,DT (return)

AS,#2,DWFILE,DT (return)

AS,#3,ONELIN,DT (return)

AS,#4,ENGFIL,DT (return)

* Mount a tape for the acquisition program *

• Place a tape with a write ring on the lower drive. *

• Thread the tape using the picture in the upper left *

• of the mag tape unit. *

• Press the LOAD button twice to bring the tape to *

• load point. The tape should stop at load point in a *

• few moments. If the load point was missed press REWIND. *

• Press ON LINE. *

* Run the collection program TNSLCT. (sample start up) *

R,TNSLCT,L1 (return)

SELECT JOB TYPE:

0 - STOP

- 1 - SURFACE AND DOWN WELL WITH DATA COLLECTION
- 2 - DOWN WELL ONLY WITH DATA COLLECTION
- 3 - SURFACE ONLY WITH DATA COLLECTION
- 4 - CALIBRATION

3 (return) * default type = 1 *

SELECT GLOBAL TASK PARTITION ASSIGNMENT OPTION:

- 1 - PROCEED WITH DEFAULT TASK PARTITIONS
- 2 - EXAMINE/MODIFY TASK PARTITIONS

(return) * default = 1 *

* TNSCLT is now active and collecting data *

HELP (return) * Type HELP to view and select program options. *

Select change code:

- 0 - Stop
- 1 - Change TCHACQ sample rate
- 2 - Change Cutler-Hammer outputs
- 3 - Change delays between formatted outputs on devices
- 4 - Change Cutler-Hammer read status word flag
- 5 - Write END OF FILE on output tape
- 6 - Indicate new tape loaded
- 7 - Change delay for unsolicited inputs
- 8 - Change DWACQ sample rate
- 9 - Print Cutler-Hammer error counters
- 10 - Suspend/resume tape writing
- 11 - Compose a line of sensor names for single line printout
- 12 - Force output of the single line print
- 13 - Retrieve pre-stored single line printout sensor names
- 14 - Flip flop outputs between octal and engineering units

Option 0: Stop

Halts the data collection program (TNSLCT).

To restart TNSLCT, go back to R,TNSLCT,L1.

Option 1: Change TCHACQ sample rate.

Modify or examine the computer's sampling rate of the Cutler-Hammer sensors. The rate is entered in minutes and 5 milli-sec ticks.

EXAMPLE:

1 (return) * Change rate to 5 min *

0 (MINUTES) 2000 (5 MSEC INTERVALS) = CURRENT TCHACQ SAMPLE RATE

0 = MINUTES ACCEPT - TYPE RETURN MODIFY TYPE - DIGITS

* Cutler-Hammer data is being sample every 0 min

2000 (5msec intervals)/200 (intervals/sec) = 10 sec *

5 (return) * change minute portion of sample rate to 5 minutes *

2000 = 5 MSEC INTERVALS ACCEPT-TYPE RETURN MODIFY TYPE DIGITS

0 (return) * cahnge 5 msec portion of sample rate to 0 *

Option 3: Change delays between formatted outputs on devices

Each of our 3 outputs; surface, down-well, or one-liners can have its own print rate to the terminal screen or the printer.

Option 3 will first ask for the device to change the print

rate for, then the print type for that device, followed by the print rate in secs.

EXAMPLE: *Print one liners every 10 sec on the CRT screen *

3 (return) * Choose option 3 *

SELECT OUTPUT DEVICE RATE EXAMINE/MODIFY OPTION:

0 - USE ESTABLISHED DELAYS BETWEEN PRINTOUTS

1 - EXAMINE/MODIFY SCREEN TIMING

2 - EXAMINE/MODIFY PRINTER TIMING

1 (return) * Modify screen printout rate *

SELECT OUTPUT FORMAT ON THIS DEVICE:

1 - SURFACE DATA SET

2 - DOWN WELL DATA SET

3 - SINGLE LINE OF SELECTED VARIABLES

3 (return) * Choose single line printouts *

60 = SEC. BETWEEN OUTPUTS ACCEPT - TYPE RETURN MODIFY - TYPE DIGITS

10 (return) * Choose 10 sec. *

SELECT OUTPUT DEVICE RATE EXAMINE/MODIFY OPTION:

0 - USE ESTABLISHED DELAYS BETWEEN PRINTOUTS

1 - EXAMINE/MODIFY SCREEN TIMING

2 - EXAMINE/MODIFY PRINTER TIMING

0 (return) * Quit option 3, we'll use the 10 sec we established *

Option 4: Change Cutler-Hammer read status word flag

Enable or disable the reading of the Cutler-Hammer status word

If there is a problem acknowledging faults from the monitor

panel then reading the status word flag should be disabled.

EXAMPLE: * Disable reading the status word flag *

4 (return) * Choose option 4 *

RSWFLG= 1 Enter new value (1->TRUE, 0->FALSE):

0 (return) * RSWFLG = false i.e. do not read the status word *

Option 5: Write END OF FILE on output tape

The current tape will be closed and rewound allowing a new tape to be mounted.

EXAMPLE:

5 (return) * Choose option 5: the tape will rewind *

Option 6: Indicate new tape loaded

When the program detects a new tape should be mounted because

the last one was full, or the operator did option 5, or

the last tape had a write error, it will inform the operator

that its waiting. Option 6 will indicate that a new tape is

loaded and ready for writing on.

EXAMPLE:

6 (return) * Choose option 6 *

Operator requested tape rewind.

TAPE LOG CH data from 05/05/81 14:15:17

to 05/05/81 18:30:00

TAPE LOG no DW data on tape

PRESS RETURN WHEN READY

* Label the tape with the TAPE LOG data, then press return *

(return)

Option 7: Change delay for unsolicited inputs

You might modify the time the program gives you to type in an option. Because the program first waits for you to enter an option and then prints out waiting information, decreasing the wait time could speed up print out.

EXAMPLE:

7 (return) * Choose option 7 cahnge time to 10 sec. *

CURRENT DELAY IS: 15 SEC ... NEW DELAY ?

10 (return) * Enter 10 sec *

Option 8: Change TDWACQ sample rate.

Modify or examine the computer's sampling rate of the down well sensors. The rate is entered in minutes and 5 milli-sec ticks.

Option 9: Print Cutler-Hammer error counters

Print all the accumulated communication errors between the Cutler-Hammer and the computer, include total scan count.

Option 10: Suspend/resume tape writing

Tape writing can be enabled or disabled with this switch.

You might want to display data only, or wait before saving data.

Option 11: Compose a line of sensor names for single line printout

This option incorporates a one line editor to create or change sensor names for single line print out. The editor works thus:

/string1/string2/ means search the line for "string1" and replace it with "string2".

ex. The sensor names are:

TIME(CH) AR13 PAL6 RPMR1

/6/6 STR5/ *to add STR5 after PAL6*

TIME(CH) AR13 PAL6 STR5 RPMR1

In order to insert sensor names use //new names/.

This causes "new names" to be added to the left of the current line. Old names shifted off to the right are lost.

In our above example to add PDR2 and WR1 type:

//PDR2 WR1 / * would yield *

PRD2 WR1 TIME(CH) AR13 PAL6 STR5 RPMR1

You may choose any combination of sensor names including TIME(CH) and TIME(DW) for Cutler-Hammer and down well sample times.

/// terminates option 11.

Option 12: Force output of the single line print

You might have had the sensor names scroll off the top of the screen. Option 12 will reprint them.

Option 13: Retrieve pre-stored single line printout sensor names

File ONELIN,DT contains some frequently used single line sensor names. This option will allow you to retrieve them instead of needing option 11 to make new ones each time.

Option 14: Flip-flop outputs between octal and engineering units; switching to engineering units.

SENSOR NAMES

Cutler-Hammer:

Name	Function	Type
AR1:	R114 LEVEL ALARM LOW IN ERU	, DI
AR2:	TPU LUBE OIL PRESSURE ALARM LOW	, DI
AR3:	WELLHEAD BRINE PRESSURE ALARM HIGH	, DI
AR4:	BRINE PUMP P13 SUCTION PRESS ALARM LOW	, DI
AR6:	BRINE PUMP P14 DSCHRG PRESS ALARM HIGH	, DI
AR7:	TPU LUBE OIL LEVEL ALARM LOW	, DI
AR8:	R114 TO WELL FLOW ALARM LOW	, DI
AR9:	R114 TO WELL PRESSURE ALARM LOW	, DI
AR11:	MAKE-UP WATER SUPPLY PRESS ALARM LOW	, DI
AR12:	INSTRUMENT AIR PRESSURE ALARM LOW	, DI
AR13:	BAROMETRIC LEG WATER LEVEL ALARM HIGH	, DI
PAL6:	R114 AUX PUMP P1 SUCTION PRESS ALRM LOW	, DI
PAR1:	R114 FROM WELL PRESSURE ALARM HIGH	, DI
AR186:	MASTER SHUTDOWN RELAY 186 ALARM	, DI
AR286:	MASTER SHUTDOWN RELAY 286 ALARM	, DI
AR386:	MASTER SHUTDOWN RELAY 386 ALARM	, DI
ST1R:	BRINE PRESS CNTRL VLVE PCV4 OPEN	, DI
ST1G:	BRINE PRESS CNTRL VLVE PCV4 CLOSED	, DI
ST2R:	BRINE PUMP P14 ON	, DI
ST3R:	BRINE PUMP P13 ON	, DI
ST4R:	R114 TO WELL FLOW CNTRL VALVE FCV1 OPEN	, DI
ST11R:	BRINE PRESS CONTROL VALVE PCV3 OPEN	, DI
ST5R:	R114 AUX PUMP P1 ON	, DI
ST6R:	R114 FEED PUMP P2 ON	, DI
ST7R:	BAC E2 ON	, DI
ST8R:	BAC E1 ON	, DI
ST9R:	R114 OUT PRESS CNTRL VALVE PCV5 OPEN	, DI
ST9G:	R114 OUT PRESS CNTRL VALVE PCV5 CLOSED	, DI
ST10R:	R114 TO WELL HAND VALVE HV16 OPEN	, DI
ST10G:	R114 TO WELL HAND VALVE HV16 CLOSED	, DI
FIC1:	R114 TO WELL FLOW CNTRLER FIC1 SETPOINT	, AO
PIC1:	BRINE BACK PRESS CNTRLER PIC1 SETPOINT	, AO
PR6:	R114 TO WELL PRESSURE	, AI
TR10:	R114 TO WELL TEMPERATURE	, AI
FR2:	R114 TO WELL FLOW	, AI
TR7:	R114 FROM WELL TEMPERATURE	, AI
PR3:	R114 FROM WELL PRESSURE	, AI
PR5:	BRINE WELLHEAD PRESSURE	, AI
FR1:	BRINE WELLHEAD FLOW	, AI
TR5:	BRINE WELLHEAD TEMPERATURE	, AI
RPMR1:	TPU RPM	, AI
LR1:	TPU LUBE OIL SUPPLY LEVEL	, AI
TR2:	R114 GAS TEMP ABOVE ERU BUBBLE TRAY	, AI
TR3:	R114 GAS TEMP BELOW ERU BUBBLE TRAY	, AI
TR4:	R114 LIQUID TEMP BELOW ERU BUBBLE TRAY	, AI
TR6:	R114 LIQUID TEMP ABOVE ERU BUBBLE TRAY	, AI
PDR1:	R114 DIFF GAS PRESS ACROSS ERU BUB TRAY	, AI

PR4:	R114 GAS PRESS ABOVE ERU BUBBLE TRAY	, AI
PR11:	R114 FEED PUMP P2 SUCTION PRESSURE	, AI
PR9:	TPU LUBE OIL SUPPLY PRESSURE	, AI
PR2:	R114 GAS PRESS AT BAC E1 INLET	, AI
TR8:	R114 LIQUID TEMP AT BAC E1 OUTLET	, AI
PDR2:	R114 DIFF PRESS ACROSS BAC E1	, AI
FR4:	R114 LIQUID RETURN FLOW BAC'S TO ERU	, AI
WR1:	BAC E1 SPRAY PUMP POWER CONSUMPTION	, AI
WR2:	BAC E1 COOLING FANS POWER CONSUMPTION	, AI
FR3:	MAKE-UP WATER FLOW TO BAC'S	, AI
TR9:	COOLING WATER RESERVOIR TEMP BAC E1	, AI
FR8:	BLEED WATER FLOW FROM BAC'S	, AI
PR10:	BRINE PRESSURE AT PCV4 DISCHARGE	, AI
WBR1:	WET BULB TEMPERATURE	, AI
ATR1:	AIR TEMPERATURE	, AI
VR1:	120 VAC FROM 480 VAC BUS	, AI
CR1:	CORR RATE AT BRINE VALVE PCV4 DSCHRG	, AI
CR2:	CORR RATE AT BRINE INLET TO P13&P14	, AI
CR3:	CORR RATE AT WELLHEAD BRINE DISCHARGE	, AI
CR4:	CORR RATE AT BAC-E1 COOLING SPRAY	, AI
TR15:	MUX 2 STATION TEMPERATURE	, AI

Down well:

Name	Function
----	-----
CAL1:	ZERO CALIBRATION
CAL2:	FULL SCALE CALIBRATION
PIP:	PUMP INLET PRESSURE
POP:	PUMP OUTLET PRESSURE
TIP:	TURBINE INLET PRESSURE
TOP:	TURBINE OUTLET PRESSURE
LUBP:	LUBRICATION OIL PRESSURE
POT:	PUMP OUTLET TEMPERATURE
TIT:	TURBINE INLET TEMPERATURE
TOT:	TURBINE OUTLET TEMPERATURE
MANT:	TRANSDUCER MANIFOLD TEMPERATURE
CANT:	INSTRUMENT CANNISTER TEMPERATURE
HXBI:	HX BRINE INLET TEMPERATURE
HXBO:	HX BRINE OUTLET TEMPERATURE
HXOI:	HX ORGANIC INLET TEMPERATURE
HXOO:	HX ORGANIC OUTLET TEMPERATURE
RPM1:	TPU RPM
RPM2:	TPU RPM

SECTION 3

DATA ACQUISITION SHUT-DOWN

The purpose of this document is to summarize the shut down procedure to halt the data collection programs and shut the computer off. There are 3 sub-procedures; halting the data collection programs, halting the computer operating system, and power down.

A) Data Acquisition Programs

If TNSLCT is running type 0 (return) on CRT1.
Wait until TNSLCT has completed and the tape is rewound.
Dismount, label, and save the data tape.

Dismount tape by:

- press ON LINE (light should be off)
- press REWIND (LOAD light will come on)
- Press REWIND

Type CP HALT (return) #optional#

Type LOG (return) #optional#

The data collection programs are now halted

B) Computer Operating System - VORTEX

The computer operating system is halted at the computer's front panel.

- 1) At the computer front panel press RESET.

PANEL MODEL

```

X  X XXX XXX XXX XXX XXX
      X           X X X
      xxx xxxxxxxxxxxx R xx
```

R - Reset

x - not used

The computer operating system VORTEX is halted

C) Power down

1) Computer

Turn the 4 position key to HOLD.

(Memory initialization on the V77 computer is complex, therefore; the computer should NEVER have its power turned completely OFF.)

2) Disk

Turn the thumb wheel switch to 0.

Press the RUN/STOP button.

Turn the thumb wheel switch to 1.
Wait for the SAFE light to come on then turn power off.

3) Tape

Dismount tape by: (if still mounted)
- press ON LINE (light should be off)
- press REWIND (LOAD light will come on)
- Press REWIND
Press RED power switch.

4) Terminals & Printers

Most terminals and printers have power switches on the back, some on the front, turn them all off.

5) Cutler-Hammer Mux Stations

If the instrumentation system is going to be off for an extended period of time, then power down Mux 1 and Mux 2.

Mux 1: Turn the red circuit breaker off (inside the control box).

Mux 2: Turn off the AC power and the fan.

6) Internal Van Power

The van power switches are on the back right hand wall.

Red - Computer power ... NEVER turn off the computer power unless you plan to re-initialize memory.

Orange - Shut down all peripheral power.

SECTION 4

CORRESPONDENCE AMONG P&ID INSTRUMENTATION SIGNAL NAMES

VAN NAME ----	CONTROL NAME -----	FIELD NAME -----	XMTR NAME ----	SIGNAL -----
AR1	: LAL2	:	: LS1	: R114 LEVEL ALARM LOW IN ERU
AR2	: PAL1	:	: PSL6	: TPU LUBE OIL PRESSURE ALARM LOW
AR3	: PAH1	:	: PSH1	: WELLHEAD BRINE PRESSURE ALARM HIGH
AR4	: PAL2	:	: PSL8	: BRINE PUMP P13 SUCTION PRESS ALARM LOW
AR6	: PAH2	:	: PSH4	: BRINE PUMP P14 DSCHRG PRESS ALARM HIGH
AR7	: LAL1	:	: LSL1	: TPU LUBE OIL LEVEL ALARM LOW
AR8	: FAL1	:	: FSL1	: R114 TO WELL FLOW ALARM LOW
AR9	: PAL5	:	: PSL7	: R114 TO WELL PRESSURE ALARM LOW
AR11	: PAL3	:	: PSL3	: MAKE-UP WATER SUPPLY PRESS ALARM LOW
AR12	: PAL4	:	: PSL4	: INSTRUMENT AIR PRESSURE ALARM LOW
AR13	: LAH2	:	: LSH1	: BAROMETRIC LEG WATER LEVEL ALARM HIGH
PAL6	: PAL6	:	: PSL2	: R114 AUX PUMP P1 SUCTION PRESS ALARM LOW
PAR1	: PAH3	:	: PSH6	: R114 FROM WELL PRESS ALARM HIGH
AR186	:	:	:	: MASTER SHUTDOWN RELAY 186 ALARM
AR286	:	:	:	: MASTER SHUTDOWN RELAY 286 ALARM
AR386	:	:	:	: MASTER SHUTDOWN RELAY 386 ALARM
ST1R	:	:	:	: BRINE PRESS CNTRL VLVE PCV4 OPEN
ST1G	:	:	:	: BRINE PRESS CNTRL VLVE PCV4 CLOSED
ST2R	:	:	:	: BRINE PUMP P14 ON
ST3R	:	:	:	: BRINE PUMP P13 ON
ST4R	:	:	:	: R114 TO WELL FLOW CNTRL VLVE FCV1 OPEN
ST11R	:	:	:	: BRINE PRESS CNTRL VLVE PCV3 OPEN
ST5R	:	:	:	: R114 AUX PUMP P1 ON
ST6R	:	:	:	: R114 FEED PUMP P2 ON
ST7R	:	:	:	: BAC E2 ON
ST8R	:	:	:	: BAC E1 ON
ST9R	:	:	:	: R114 OUT PRESS CNTRL VLVE PCV5 OPEN
ST9G	:	:	:	: R114 OUT PRESS CNTRL VLVE PCV5 CLOSED
ST10R	:	:	:	: R114 TO WELL HAND VALVE HV16 OPEN
ST10G	:	:	:	: R114 TO WELL HAND VALVE HV16 CLOSED

FIC1 : FIC1 : : FIC1 : R114 TO WELL FLOW CONTROLLER
 FIC1 SETPOINT
 PIC1 : PIC1 : : PIC1 : BRINE BACK PRESSURE CONTROLLER
 PIC1 SETPOINT

PR6 : PR7 : PI22 : PT4 : R114 TO WELL PRESSURE
 TR10 : TR12 : TI10 : TT9 : R114 TO WELL TEMPERATURE
 FR2 : FR6 : PDI6 : FT1 : R114 TO WELL FLOW
 TR7 : TR11 : TI4 : TT2 : R114 FROM WELL TEMPERATURE
 PR3 : PR3 : PI2 : PT3 : R114 FROM WELL PRESSURE
 PR5 : PR8 : PI16 : PT2 : BRINE WELLHEAD PRESSURE
 FR1 : FR7 : FR5 : FT2 : BRINE WELLHEAD FLOW
 TR5 : TR1 : TI2 : TT1 : BRINE WELLHEAD TEMPERATURE
 RPMR1 : RPMR1 : : RPMT : TPU RPM
 LR1 : MV DSPLY : LI1 : LT1A & LT1B : TPU LUBE OIL SUPPLY
 LEVEL

TR2 : : TI7 : TT6 : R114 GAS TEMP ABOVE ERU
 BUBBLE TRAY
 TR3 : : TI5 : TT4 : R114 GAS TEMP BELOW ERU
 BUBBLE TRAY
 TR4 : : TI8 : TT3 : R114 LIQUID TEMP BELOW ERU
 BUBBLE TRAY
 TR6 : : TI6 : TT5 : R114 LIQUID TEMP ABOVE ERU
 BUBBLE TRAY
 PDR1 : : : PDT1 : R114 DIFFERENTIAL GAS PRESS
 ACROSS ERU BUBBLE TRAY
 PR4 : : PI12 : PT5 : R114 GAS PRESS ABOVE ERU
 BUBBLE TRAY
 PR11 : : PI5 : PT8 : R114 FEED PUMP P2 SUCTION PRESS
 PR9 : : PI8 : PT1 : TPU LUBE OIL SUPPLY PRESSURE
 PR2 : : : PT6 : R114 GAS PRESS AT BAC E1 INLET
 TR8 : : TI11 : TT7 : R114 LIQUID TEMP AT BAC E1 OUTLET
 PDR2 : : : PDT2 : R114 DIFFERENTIAL PRESS ACROSS
 BAC E1
 FR4 : : : FT4 : R114 LIQUID RETURN FLOW BAC'S
 TO ERU
 WR1 : : : WT1 : BAC E1 COOLING WATER SPRAY PUMP
 POWER CONSUMPTION
 WR2 : : : WT2 : BAC E1 COOLING FANS POWER
 CONSUMPTION
 FR3 : : : FT5 : MAKE-UP WATER FLOW TO BAC'S
 TR9 : : : TT8 : COOLING WATER RESERVOIR TEMP
 BAC E1
 FR8 : : : FT6 : BLEED WATER FLOW FROM BAC'S
 PR10 : : : PT7 : BRINE PRESSURE AT PCV4 DSCHRG
 WBR1 : : : WBT1 : WET BULB TEMPERATURE

ATR1	:	:	:	ATT1	:	AIR TEMPERATURE
VR1	:	:	:	VT1	:	120 VAC FROM 480 VAC BUS
CR1	:	:	CME1	:	CPL1	CORROSION RATE AT BRINE VLVE PCV4 DISCHARGE
CR2	:	:	CME2	:	CPL2	CORROSION RATE AT BRINE INLET TO P13 AND P14
CR3	:	:	CME3	:	CPL3	CORROSION RATE AT WELLHEAD BRINE DISCHARGE
CR4	:	:	CME4	:	CPL4	CORROSION RATE AT BAC E1 COOLING SPRAY
TR15	:	:	:	:	TT15	MUX 2 STATION TEMPERATURE

SECTION 5

INSTRUCTIONS FOR DISPLAYING DISCRETES ON CUTLER-HAMMER MONITOR PANEL

0. EXECUTE STEPS A1(PERIPHERAL POWER ONLY) AND A6 OF SECTION 1.
1. HIT FUNCTION KEY(BLACK 1 KEY) REPEATEDLY UNTIL DISPLAY APPEARS IN THE LEFT HALF OF THE MESSAGE WINDOW.
2. HIT FORM KEY (BLACK 2 KEY) REPEATEDLY UNTIL DISCRETE APPEARS IN THE RIGHT HALF OF THE MESSAGE WINDOW.
3. HIT ADDRESS KEY (BLACK 3 KEY) ONCE.
4. ENTER MODULE ADDRESS USING WHITE KEYS. ADDRESS WILL APPEAR IN ADDRESS WINDOW. HIT ADDRESS KEY AGAIN. DATA WILL APPEAR IN DATA WINDOW.
5. TO DISPLAY A DISCRETE AT A DIFFERENT MODULE ADDRESS, REPEAT STEP 4.

DISCRETE	MODULE ADDRESS	BIT #	CONDITION
AR1	000	0	1: LEVEL OK 0: LOW LEVEL
AR2	000	1	1: PRESSURE OK 0: PRESSURE LOW
AR3	000	2	1: PRESSURE OK 0: PRESSURE HIGH
AR4	000	3	1: PRESSURE OK 0: PRESSURE LOW
AR6	000	4	1: PRESSURE OK 0: PRESSURE HIGH
AR7	000	5	1: LEVEL OK 0: LEVEL LOW
AR8	000	6	1: FLOW OK 0: FLOW LOW
AR9	000	7	1: PRESSURE OK 0: PRESSURE LOW
AR11	001	0	1: PRESSURE OK 0: PRESSURE LOW
AR12	001	1	1: PRESSURE OK 0: PRESSURE LOW

AR13	001	2	1: LEVEL OK 0: LEVEL HIGH
PAL6	001	3	1: PRESSURE OK 0: PRESSURE LOW
PAR1	001	4	1: PRESSURE OK 0: PRESSURE HIGH
AR186	002	0	0: NORMAL 1: SHUTDOWN
AR286	002	1	0: NORMAL 1: SHUTDOWN
AR386	002	2	0: NORMAL 1: SHUTDOWN
ST1R	002	4	1: PCV4 NOT FULLY CLOSED 0: PCV4 FULLY CLOSED
ST1G	002	5	1: PCV4 NOT FULLY OPEN 0: PCV4 FULLY OPEN
ST2R	002	6	1: P14 ON 0: P14 OFF
ST3R	003	0	1: P13 ON 0: P13 OFF
ST4R	003	2	1: FCV1 NOT FULLY CLOSED 0: FCV1 FULLY CLOSED
ST11R	003	3	1: PCV3 NOT FULLY CLOSED 0: PCV3 FULLY CLOSED
ST5R	003	4	1: P1 ON 0: P1 OFF
ST6R	003	6	1: P2 ON 0: P2 OFF
ST7R	004	0	1: E2 ON 0: E2 OFF
ST8R	004	2	1: E1 ON 0: E1 OFF
ST9R	004	4	1: PCV5 NOT FULLY CLOSED 0: PCV5 FULLY CLOSED
ST9G	004	5	1: PCV5 NOT FULLY OPEN 0: PCV5 FULLY OPEN
ST10R	004	6	1: HV16 NOT FULLY CLOSED 0: HV16 FULLY CLOSED
ST10G	004	7	1: HV16 NOT FULLY OPEN 0: HV16 FULLY OPEN

INSTRUCTIONS FOR DISPLAYING ANALOGS ON
CUTLER-HAMMER MONITOR PANEL

0. EXECUTE STEPS A1(PERIPHERAL POWER ONLY) AND A6 OF SECTION 1.
1. HIT FUNCTION KEY (BLACK 1 KEY) REPEATEDLY UNTIL DISPLAY APPEARS IN THE LEFT HALF OF THE MESSAGE WINDOW.
2. HIT FORM KEY (BLACK 2 KEY) REPEATEDLY UNTIL ANALOG APPEARS IN THE RIGHT HALF OF THE MESSAGE WINDOW.
3. HIT ADDRESS KEY (BLACK 3 KEY) ONCE.
4. ENTER CHANNEL ADDRESS USING WHITE KEYS. ADDRESS WILL APPEAR IN ADDRESS WINDOW. HIT ADDRESS KEY AGAIN. DATA WILL APPEAR IN DATA WINDOW.
5. TO DISPLAY AN ANALOG AT A DIFFERENT CHANNEL ADDRESS, REPEAT STEP 4.
6. AN ANALOG VALUE WILL APPEAR AS A THREE-DIGIT NUMBER IN THE RANGE 0.-100. WHICH REPRESENTS A PERCENTAGE OF THE RANGE COVERED BY THE CORRESPONDING SIGNAL. SUPPOSE THAT THE SIGNAL VALUE VARIES FROM VLOW TO VHIGH AS THE SIGNAL CURRENT VARIES FROM 4 TO 20 MA. IF THE ANALOG DISPLAY SHOWS A VALUE R, THEN THE CORRESPONDING SIGNAL HAS A VALUE

$$V = (R/100)*(VHIGH-VLOW) + VLOW$$

7. TO DISPLAY AN ANALOG AS AN OCTAL NUMBER:
 - 7.1. PERFORM STEP 1 ABOVE.
 - 7.2. HIT FORM KEY (BLACK 2 KEY) REPEATEDLY UNTIL BUFFER APPEARS IN THE RIGHT HALF OF THE MESSAGE WINDOW.
 - 7.3 HIT ADDRESS KEY (BLACK 3 KEY) ONCE.
 - 7.4 ENTER BUFFER ADDRESS USING WHITE KEYS. ADDRESS WILL APPEAR IN ADDRESS WINDOW. HIT ADDRESS KEY AGAIN. DATA WILL APPEAR IN DATA WINDOW.
 - 7.5. TO DISPLAY AN ANALOG AT A DIFFERENT BUFFER ADDRESS, REPEAT STEP 7.4.
 - 7.6. AN ANALOG VALUE WILL APPEAR AS A SIX-DIGIT OCTAL NUMBER, 01 02 03 04 05 06. THE CORRESPONDING 12-BIT BINARY NUMBER, B0-B11, IS OBTAINED AS FOLLOWS:

01: - - B0 (B0 = MSD)
02: B1 B2 B3
03: B4 B5 B6
04: B7 B8 B9
05: B10 B11 - (B11 = LSD)
06: - - -

ANALOG -----	CHANNEL ADDRESS -----	BUFFER ADDRESS -----	SIGNAL RANGE (VLOW TO VHIGH) -----
MUX 1 -----			
FIC1	0130	4000	
PIC1	0131	4001	
PR6	0060	4004	0-500 PSIA
TR10	0061	4005	50-150 DEGF
FR2	0062	4006	0-600 "H2O
TR7	0063	4007	0-400 DEGF
PR3	0070	4010	0-300 PSIA
PR5	0071	4011	0-500 PSIA
FR1	0072	4012	0-86 SQ RT "H2O
TR5	0073	4013	0-400 DEGF
RPMR1	0100	4014	0-15000 RPM
LR1	0101	4015	0-750 LBS
TST1	0102	4016	

MUX 2

TR2	0200	4020	50-150 DEGF
TR3	0201	4021	50-150 DEGF
TR4	0202	4022	50-150 DEGF
TR6	0203	4023	50-150 DEGF
PDR1	0210	4024	0-30 "H2O
PR4	0211	4025	0-100 PSIA
PR11	0212	4026	0-200 PSIA
PR9	0213	4027	0-200 PSIA
PR2	0220	4030	0-50 PSIA
TR8	0221	4031	50-150 DEGF
PDR2	0222	4032	0-100 "H2O
FR4	0223	4033	0-15 "H2O
WR1	0230	4034	0-3.75 KW
WR2	0231	4035	0-7.5 KW
FR3	0232	4036	9-100 GPM
TR9	0233	4037	50-150 DEGF
FR8	0240	4040	9-100 GPM
PR10	0241	4041	0-700 PSIA
WBR1	0242	4042	0-100 DEGF
ATR1	0243	4043	0-150 DEGF
VR1	0250	4044	108-130 VAC
CR1	0251	4045	0-100 M"/Y
CR2	0252	4046	0-100 M"/Y
CR3	0253	4047	0-100 M"/Y
CR4	0260	4050	0-100 M"/Y

TR15
TST2

0261
0262

4051
4052

-25 - +75 DEGC

SECTION 6

CHFILE

5/22/81

AR1:	R114 LEVEL ALARM LOW IN ERU	,	0,	0,00,DI	,	
AR2:	TPU LUBE OIL PRESSURE ALARM LOW	,	0,	0,01,DI	,	
AR3:	WELLHEAD BRINE PRESSURE ALARM HIGH	,	0,	0,02,DI	,	
AR4:	BRINE PUMP P13 SUCTION PRESS ALARM LOW	,	0,	0,03,DI	,	
AR6:	BRINE PUMP P14 DSCHRG PRESS ALARM HIGH	,	0,	0,04,DI	,	
AR7:	TPU LUBE OIL LEVEL ALARM LOW	,	0,	0,05,DI	,	
AR8:	R114 TO WELL FLOW ALARM LOW	,	0,	0,06,DI	,	
AR9:	R114 TO WELL PRESSURE ALARM LOW	,	0,	0,07,DI	,	
AR11:	MAKE-UP WATER SUPPLY PRESS ALARM LOW	,	01,	01,00,DI	,	
AR12:	INSTRUMENT AIR PRESSURE ALARM LOW	,	01,	01,01,DI	,	
AR13:	BAROMETRIC LEG WATER LEVEL ALARM HIGH	,	01,	01,02,DI	,	
PAL6:	R114 AUX PUMP P1 SUCTION PRESS ALRM LOW	,	01,	01,03,DI	,	
PAR1:	R114 FROM WELL PRESSURE ALARM HIGH	,	01,	01,04,DI	,	
AR186:	MASTER SHUTDOWN RELAY 186 ALARM	,	02,	02,00,DI	,	
AR286:	MASTER SHUTDOWN RELAY 286 ALARM	,	02,	02,01,DI	,	
AR386:	MASTER SHUTDOWN RELAY 386 ALARM	,	02,	02,02,DI	,	
ST1R:	BRINE PRESS CNTRL VLVE PCV4 OPEN	,	02,	02,04,DI	,	
ST1G:	BRINE PRESS CNTRL VLVE PCV4 CLOSED	,	02,	02,05,DI	,	
ST2R:	BRINE PUMP P14 ON	,	02,	02,06,DI	,	
ST3R:	BRINE PUMP P13 ON	,	03,	03,00,DI	,	
ST4R:	R114 TO WELL FLOW CNTRL VALVE FCV1 OPEN	,	03,	03,02,DI	,	
ST11R:	BRINE PRESS CONTROL VALVE PCV3 OPEN	,	03,	03,03,DI	,	
ST5R:	R114 AUX PUMP P1 ON	,	03,	03,04,DI	,	
ST6R:	R114 FEED PUMP P2 ON	,	03,	03,06,DI	,	
ST7R:	BAC E2 ON	,	04,	04,00,DI	,	
ST8R:	BAC E1 ON	,	04,	04,02,DI	,	
ST9R:	R114 OUT PRESS CNTRL VALVE PCV5 OPEN	,	04,	04,04,DI	,	
ST9G:	R114 OUT PRESS CNTRL VALVE PCV5 CLOSED	,	04,	04,05,DI	,	
ST10R:	R114 TO WELL HAND VALVE HV16 OPEN	,	04,	04,06,DI	,	
ST10G:	R114 TO WELL HAND VALVE HV16 CLOSED	,	04,	04,07,DI	,	
FIC1:	R114 TO WELL FLOW CNTRLER FIC1 SETPOINT	,	04000,	013,00,AO,		0, 0
PIC1:	BRINE BACK PRESS CNTRLER PIC1 SETPOINT	,	04001,	013,01,AO,		0, 0
PR6:	R114 TO WELL PRESSURE	,	04004,	06,00,AI	,	
TR10:	R114 TO WELL TEMPERATURE	,	04005,	06,01,AI	,	
FR2:	R114 TO WELL FLOW	,	04006,	06,02,AI	,	
TR7:	R114 FROM WELL TEMPERATURE	,	04007,	06,03,AI	,	
PR3:	R114 FROM WELL PRESSURE	,	04010,	07,00,AI	,	
PR5:	BRINE WELLHEAD PRESSURE	,	04011,	07,01,AI	,	
FR1:	BRINE WELLHEAD FLOW	,	04012,	07,02,AI	,	
TR5:	BRINE WELLHEAD TEMPERATURE	,	04013,	07,03,AI	,	
RPMR1:	TPU RPM	,	04014,	010,00,AI	,	
LR1:	TPU LUBE OIL SUPPLY LEVEL	,	04015,	010,01,AI	,	
TST1:	TEST 1	,	04016,	010,02,AI	,	
TR2:	R114 GAS TEMP ABOVE ERU BUBBLE TRAY	,	04020,	020,00,AI	,	

TR3:	R114 GAS TEMP BELOW ERU BUBBLE TRAY	,04021,	020,01,AI	,	,
TR4:	R114 LIQUID TEMP BELOW ERU BUBBLE TRAY	,04022,	020,02,AI	,	,
TR6:	R114 LIQUID TEMP ABOVE ERU BUBBLE TRAY	,04023,	020,03,AI	,	,
PDR1:	R114 DIFF GAS PRESS ACROSS ERU BUB TRAY	,04024,	021,00,AI	,	,
PR4:	R114 GAS PRESS ABOVE ERU BUBBLE TRAY	,04025,	021,01,AI	,	,
PR11:	R114 FEED PUMP P2 SUCTION PRESSURE	,04026,	021,02,AI	,	,
PR9:	TPU LUBE OIL SUPPLY PRESSURE	,04027,	021,03,AI	,	,
PR2:	R114 GAS PRESS AT BAC E1 INLET	,04030,	022,00,AI	,	,
TR8:	R114 LIQUID TEMP AT BAC E1 OUTLET	,04031,	022,01,AI	,	,
PDR2:	R114 DIFF PRESS ACROSS BAC E1	,04032,	022,02,AI	,	,
FR4:	R114 LIQUID RETURN FLOW BAC'S TO ERU	,04033,	022,03,AI	,	,
WR1:	BAC E1 SPRAY PUMP POWER CONSUMPTION	,04034,	023,00,AI	,	,
WR2:	BAC E1 COOLING FANS POWER CONSUMPTION	,04035,	023,01,AI	,	,
FR3:	MAKE-UP WATER FLOW TO BAC'S	,04036,	023,02,AI	,	,
TR9:	COOLING WATER RESERVOIR TEMP BAC E1	,04037,	023,03,AI	,	,
FR8:	BLEED WATER FLOW FROM BAC'S	,04040,	024,00,AI	,	,
PR10:	BRINE PRESSURE AT PCV4 DISCHARGE	,04041,	024,01,AI	,	,
WBR1:	WET BULB TEMPERATURE	,04042,	024,02,AI	,	,
ATR1:	AIR TEMPERATURE	,04043,	024,03,AI	,	,
VR1:	120 VAC FROM 480 VAC BUS	,04044,	025,00,AI	,	,
CR1:	CORR RATE AT BRINE VALVE PCV4 DSCHRG	,04045,	025,01,AI	,	,
CR2:	CORR RATE AT BRINE INLET TO P13&P14	,04046,	025,02,AI	,	,
CR3:	CORR RATE AT WELLHEAD BRINE DISCHARGE	,04047,	025,03,AI	,	,
CR4:	CORR RATE AT BAC-E1 COOLING SPRAY	,04050,	026,00,AI	,	,
TR15:	MUX 2 STATION TEMPERATURE	,04051,	026,01,AI	,	,
TST2:	TEST 2	,04052,	026,02,AI	,	,

CAL1:	ZERO CALIBRATION	,	0
CAL2:	FULL SCALE CALIBRATION	,	01
PIP:	PUMP INLET PRESSURE	,	02
POP:	PUMP OUTLET PRESSURE	,	03
TIP:	TURBINE INLET PRESSURE	,	04
TOP:	TURBINE OUTLET PRESSURE	,	05
LUBP:	LUBRICATION OIL PRESSURE	,	06
POT:	PUMP OUTLET TEMPERATURE	,	07
TIT:	TURBINE INLET TEMPERATURE	,	010
TOT:	TURBINE OUTLET TEMPERATURE	,	011
MANT:	TRANSDUCER MANIFOLD TEMPERATURE	,	012
CANT:	INSTRUMENT CANNISTER TEMPERATURE	,	013
HXB1:	HX BRINE INLET TEMPERATURE	,	014
HXB0:	HX BRINE OUTLET TEMPERATURE	,	015
HX01:	HX ORGANIC INLET TEMPERATURE	,	016
HX00:	HX ORGANIC OUTLET TEMPERATURE	,	017
RPM1:	TPU RPM	,	020
RPM2:	TPU RPM	,	021

FIC1: R114 TO WELL FLOW CNTRLER FIC1 SETPOINT,	,AO,0,4095,0.,4095.,0.
PIC1: BRINE BACK PRESS CNTRLER PIC1 SETPOINT	,AO,0,4095,0.,4095.,0.
PR6: R114 TO WELL PRESSURE	,PSIA,AI,0,4095,0.,500.,5.
TR10: R114 TO WELL TEMPERATURE	,DEGF,AI,0,4095,50.,150.,1.5
FR2: R114 TO WELL FLOW	, "H2O,AI,0,4095,0.,600.,6.0
TR7: R114 FROM WELL TEMPERATURE	,DEGF,AI,0,4095,0.,400.,4.
PR3: R114 FROM WELL PRESSURE	,PSIA,AI,0,4095,0.,300.,3.
PR5: BRINE WELLHEAD PRESSURE	,PSIA,AI,0,4095,0.,500.,5.
FR1: BRINE WELLHEAD FLOW	, "H2O** .5,AI,0,4095,0.,86.,.86
TR5: BRINE WELLHEAD TEMPERATURE	,DEGF,AI,0,4095,0.,400.,4.
RPMR1: TPU RPM	,RPM ,AI,0,4095,0.,15000.,150.
LR1: TPU LUBE OIL SUPPLY LEVEL	,LBS ,AI,0,4095,0.,750.,7.5
TR2: R114 GAS TEMP ABOVE ERU BUBBLE TRAY	,DEGF,AI,0,4095,50.,150.,1.5
TR3: R114 GAS TEMP BELOW ERU BUBBLE TRAY	,DEGF,AI,0,4095,50.,150.,1.5
TR4: R114 LIQUID TEMP BELOW ERU BUBBLE TRAY	,DEGF,AI,0,4095,50.,150.,1.5
TR6: R114 LIQUID TEMP ABOVE ERU BUBBLE TRAY	,DEGF,AI,0,4095,50.,150.,1.5
PDR1: R114 DIFF GAS PRESS ACROSS ERU BUB TRAY	, "H2O,AI,0,4095,0.,30.,.3
PR4: R114 GAS PRESS ABOVE ERU BUBBLE TRAY	,PSIA,AI,0,4095,0.,100.,1.
PR11: R114 FEED PUMP P2 SUCTION PRESSURE	,PSIA,AI,0,4095,0.,200.,2.
PR9: TPU LUBE OIL SUPPLY PRESSURE	,PSIA,AI,0,4095,0.,200.,2.
PR2: R114 GAS PRESS AT BAC E1 INLET	,PSIA,AI,0,4095,0.,50.,.5
TR8: R114 LIQUID TEMP AT BAC E1 OUTLET	,DEGF,AI,0,4095,50.,150.,1.5
PDR2: R114 DIFF PRESS ACROSS BAC E1	, "H2O,AI,0,4095,0.,100.,1.
FR4: R114 LIQUID RETURN FLOW BAC'S TO ERU	, "H2O,AI,0,4095,0.,15.,.15
WR1: BAC E1 SPRAY PUMP POWER CONSUMPTION	,KW ,AI,0,4095,0.,3.75,.0375
WR2: BAC E1 COOLING FANS POWER CONSUMPTION	,KW ,AI,0,4095,0.,7.5,.075
FR3: MAKEUP WATER FLOW TO BAC'S	,GPM ,AI,0,4095,9.,100.,1.
TR9: COOLING WATER RESERVOIR TEMP BAC E1	,DEGF,AI,0,4095,50.,150.,1.5
FR8: BLEED WATER FLOW FROM BAC'S	,GPM ,AI,0,4095,9.,100.,1.
PR10: BRINE PRESSURE AT PCV4 DISCHARGE	,PSIA,AI,0,4095,0.,700.,7.
WBR1: WET BULB TEMPERATURE	,DEGF,AI,0,4095,0.,100.,1.
ATR1: AIR TEMPERATURE	,DEGF,AI,0,4095,0.,150.,1.5
VR1: 120 VAC FROM 480 VAC BUS	,VOLT,AI,0,4095,108.,130.,1.3
CR1: CORR RATE AT BRINE VALVE PCV4 DSCHRG	,M"/Y,AI,0,4095,0.,100.,1.
CR2: CORR RATE AT BRINE INLET TO P13&P14	,M"/Y,AI,0,4095,0.,100.,1.
CR3: CORR RATE AT WELLHEAD BRINE DISCHARGE	,M"/Y,AI,0,4095,0.,100.,1.
CR4: CORR RATE AT BAC-E1 COOLING SPRAY	,M"/Y,AI,0,4095,0.,100.,1.
TR15: MUX 2 STATION TEMPERATURE	,DEGC,AI,0,4095,-25.,75.,.5
CAL1: ZERO CALIBRATION	,CNTS,,,0.,0.,1.
CAL2: FULL SCALE CALIBRATION	,CNTS,,,3968.,3968.,1.
PIP: PUMP INLET PRESSURE	,PSIA,,,0.,750.,1.
POP: PUMP OUTLET PRESSURE	,PSIA,,,0.,1500.,1.
TIP: TURBINE INLET PRESSURE	,PSIA,,,0.,750.,1.
TOP: TURBINE OUTLET PRESSURE	,PSIA,,,0.,400.,.1
LUBP: LUBRICATION OIL PRESSURE	,PSIA,,,0.,1500.,1.
POT: PUMP OUTLET TEMPERATURE	,DEGC,,,0.,200.,.1
TIT: TURBINE INLET TEMPERATURE	,DEGC,,,0.,200.,.1
TOT: TURBINE OUTLET TEMPERATURE	,DEGC,,,0.,200.,.1
MANT: TRANSDUCER MANIFOLD TEMPERATURE	,DEGC,,,0.,200.,.1

CANT:	INSTRUMENT CANNISTER TEMPERATURE	,DEGC,,,0.,200.,.1
HXBI:	HX BRINE INLET TEMPERATURE	,DEGC,,,0.,200.,.1
HXBO:	HX BRINE OUTLET TEMPERATURE	,DEGC,,,0.,200.,.1
HXOI:	HX ORGANIC INLET TEMPERATURE	,DEGC,,,0.,200.,.1
HXOO:	HX ORGANIC OUTLET TEMPERATURE	,DEGC,,,0.,200.,.1
RPM1:	TPU RPM/10	,RPM/10,,,0.,1500.,.1.
RPM2:	TPU RPM/10	,RPM/10,,,0.,1500.,.1.

SECTION 7

OCTAL-DECIMAL INTEGER CONVERSION TABLE

0000 to 8777
(Octal) (Decimal)

Octal Decimal
10000 - 4096
20000 - 8192
30000 - 12288
40000 - 16384
50000 - 20480
60000 - 24576
70000 - 28672

	0	1	2	3	4	5	6	7
0000	0000	0001	0002	0003	0004	0005	0006	0007
0010	0008	0009	0010	0011	0012	0013	0014	0015
0020	0016	0017	0018	0019	0020	0021	0022	0023
0030	0024	0025	0026	0027	0028	0029	0030	0031
0040	0032	0033	0034	0035	0036	0037	0038	0039
0050	0040	0041	0042	0043	0044	0045	0046	0047
0060	0048	0049	0050	0051	0052	0053	0054	0055
0070	0056	0057	0058	0059	0060	0061	0062	0063
0100	0064	0065	0066	0067	0068	0069	0070	0071
0110	0072	0073	0074	0075	0076	0077	0078	0079
0120	0080	0081	0082	0083	0084	0085	0086	0087
0130	0088	0089	0090	0091	0092	0093	0094	0095
0140	0096	0097	0098	0099	0100	0101	0102	0103
0150	0104	0105	0106	0107	0108	0109	0110	0111
0160	0112	0113	0114	0115	0116	0117	0118	0119
0170	0120	0121	0122	0123	0124	0125	0126	0127
0200	0128	0129	0130	0131	0132	0133	0134	0135
0210	0136	0137	0138	0139	0140	0141	0142	0143
0220	0144	0145	0146	0147	0148	0149	0150	0151
0230	0152	0153	0154	0155	0156	0157	0158	0159
0240	0160	0161	0162	0163	0164	0165	0166	0167
0250	0168	0169	0170	0171	0172	0173	0174	0175
0260	0176	0177	0178	0179	0180	0181	0182	0183
0270	0184	0185	0186	0187	0188	0189	0190	0191
0300	0192	0193	0194	0195	0196	0197	0198	0199
0310	0200	0201	0202	0203	0204	0205	0206	0207
0320	0208	0209	0210	0211	0212	0213	0214	0215
0330	0216	0217	0218	0219	0220	0221	0222	0223
0340	0224	0225	0226	0227	0228	0229	0230	0231
0350	0232	0233	0234	0235	0236	0237	0238	0239
0360	0240	0241	0242	0243	0244	0245	0246	0247
0370	0248	0249	0250	0251	0252	0253	0254	0255

	0	1	2	3	4	5	6	7
0400	0256	0257	0258	0259	0260	0261	0262	0263
0410	0264	0265	0266	0267	0268	0269	0270	0271
0420	0272	0273	0274	0275	0276	0277	0278	0279
0430	0280	0281	0282	0283	0284	0285	0286	0287
0440	0288	0289	0290	0291	0292	0293	0294	0295
0450	0296	0297	0298	0299	0300	0301	0302	0303
0460	0304	0305	0306	0307	0308	0309	0310	0311
0470	0312	0313	0314	0315	0316	0317	0318	0319
0500	0320	0321	0322	0323	0324	0325	0326	0327
0510	0328	0329	0330	0331	0332	0333	0334	0335
0520	0336	0337	0338	0339	0340	0341	0342	0343
0530	0344	0345	0346	0347	0348	0349	0350	0351
0540	0352	0353	0354	0355	0356	0357	0358	0359
0550	0360	0361	0362	0363	0364	0365	0366	0367
0560	0368	0369	0370	0371	0372	0373	0374	0375
0570	0376	0377	0378	0379	0380	0381	0382	0383
0600	0384	0385	0386	0387	0388	0389	0390	0391
0610	0392	0393	0394	0395	0396	0397	0398	0399
0620	0400	0401	0402	0403	0404	0405	0406	0407
0630	0408	0409	0410	0411	0412	0413	0414	0415
0640	0416	0417	0418	0419	0420	0421	0422	0423
0650	0424	0425	0426	0427	0428	0429	0430	0431
0660	0432	0433	0434	0435	0436	0437	0438	0439
0670	0440	0441	0442	0443	0444	0445	0446	0447
0700	0448	0449	0450	0451	0452	0453	0454	0455
0710	0456	0457	0458	0459	0460	0461	0462	0463
0720	0464	0465	0466	0467	0468	0469	0470	0471
0730	0472	0473	0474	0475	0476	0477	0478	0479
0740	0480	0481	0482	0483	0484	0485	0486	0487
0750	0488	0489	0490	0491	0492	0493	0494	0495
0760	0496	0497	0498	0499	0500	0501	0502	0503
0770	0504	0505	0506	0507	0508	0509	0510	0511

1600 to 1777
(Octal) (Decimal)

	0	1	2	3	4	5	6	7
1000	0512	0513	0514	0515	0516	0517	0518	0519
1010	0520	0521	0522	0523	0524	0525	0526	0527
1020	0528	0529	0530	0531	0532	0533	0534	0535
1030	0536	0537	0538	0539	0540	0541	0542	0543
1040	0544	0545	0546	0547	0548	0549	0550	0551
1050	0552	0553	0554	0555	0556	0557	0558	0559
1060	0560	0561	0562	0563	0564	0565	0566	0567
1070	0568	0569	0570	0571	0572	0573	0574	0575
1100	0576	0577	0578	0579	0580	0581	0582	0583
1110	0584	0585	0586	0587	0588	0589	0590	0591
1120	0592	0593	0594	0595	0596	0597	0598	0599
1130	0600	0601	0602	0603	0604	0605	0606	0607
1140	0608	0609	0610	0611	0612	0613	0614	0615
1150	0616	0617	0618	0619	0620	0621	0622	0623
1160	0624	0625	0626	0627	0628	0629	0630	0631
1170	0632	0633	0634	0635	0636	0637	0638	0639
1200	0640	0641	0642	0643	0644	0645	0646	0647
1210	0648	0649	0650	0651	0652	0653	0654	0655
1220	0656	0657	0658	0659	0660	0661	0662	0663
1230	0664	0665	0666	0667	0668	0669	0670	0671
1240	0672	0673	0674	0675	0676	0677	0678	0679
1250	0680	0681	0682	0683	0684	0685	0686	0687
1260	0688	0689	0690	0691	0692	0693	0694	0695
1270	0696	0697	0698	0699	0700	0701	0702	0703
1300	0704	0705	0706	0707	0708	0709	0710	0711
1310	0712	0713	0714	0715	0716	0717	0718	0719
1320	0720	0721	0722	0723	0724	0725	0726	0727
1330	0728	0729	0730	0731	0732	0733	0734	0735
1340	0736	0737	0738	0739	0740	0741	0742	0743
1350	0744	0745	0746	0747	0748	0749	0750	0751
1360	0752	0753	0754	0755	0756	0757	0758	0759
1370	0760	0761	0762	0763	0764	0765	0766	0767

	0	1	2	3	4	5	6	7
1400	0768	0769	0770	0771	0772	0773	0774	0775
1410	0776	0777	0778	0779	0780	0781	0782	0783
1420	0784	0785	0786	0787	0788	0789	0790	0791
1430	0792	0793	0794	0795	0796	0797	0798	0799
1440	0800	0801	0802	0803	0804	0805	0806	0807
1450	0808	0809	0810	0811	0812	0813	0814	0815
1460	0816	0817	0818	0819	0820	0821	0822	0823
1470	0824	0825	0826	0827	0828	0829	0830	0831
1500	0832	0833	0834	0835	0836	0837	0838	0839
1510	0840	0841	0842	0843	0844	0845	0846	0847
1520	0848	0849	0850	0851	0852	0853	0854	0855
1530	0856	0857	0858	0859	0860	0861	0862	0863
1540	0864	0865	0866	0867	0868	0869	0870	0871
1550	0872	0873	0874	0875	0876	0877	0878	0879
1560	0880	0881	0882	0883	0884	0885	0886	0887
1570	0888	0889	0890	0891	0892	0893	0894	0895
1600	0896	0897	0898	0899	0900	0901	0902	0903
1610	0904	0905	0906	0907	0908	0909	0910	0911
1620	0912	0913	0914	0915	0916	0917	0918	0919
1630	0920	0921	0922	0923	0924	0925	0926	0927
1640	0928	0929	0930	0931	0932	0933	0934	0935
1650	0936	0937	0938	0939	0940	0941	0942	0943
1660	0944	0945	0946	0947	0948	0949	0950	0951
1670	0952	0953	0954	0955	0956	0957	0958	0959
1700	0960	0961	0962	0963	0964	0965	0966	0967
1710	0968	0969	0970	0971	0972	0973	0974	0975
1720	0976	0977	0978	0979	0980	0981	0982	0983
1730	0984	0985	0986	0987	0988	0989	0990	0991
1740	0992	0993	0994	0995	0996	0997	0998	0999
1750	1000	1001	1002	1003	1004	1005	1006	1007
1760	1008	1009	1010	1011	1012	1013	1014	1015
1770	1016	1017	1018	1019	1020	1021	1022	1023

Octal-Decimal Integer Conversion Table

	0	1	2	3	4	5	6	7
2000	1024	1025	1026	1027	1028	1029	1030	1031
2010	1032	1033	1034	1035	1036	1037	1038	1039
2020	1040	1041	1042	1043	1044	1045	1046	1047
2030	1048	1049	1050	1051	1052	1053	1054	1055
2040	1056	1057	1058	1059	1060	1061	1062	1063
2050	1064	1065	1066	1067	1068	1069	1070	1071
2060	1072	1073	1074	1075	1076	1077	1078	1079
2070	1080	1081	1082	1083	1084	1085	1086	1087
2100	1088	1089	1090	1091	1092	1093	1094	1095
2110	1096	1097	1098	1099	1100	1101	1102	1103
2120	1104	1105	1106	1107	1108	1109	1110	1111
2130	1112	1113	1114	1115	1116	1117	1118	1119
2140	1120	1121	1122	1123	1124	1125	1126	1127
2150	1128	1129	1130	1131	1132	1133	1134	1135
2160	1136	1137	1138	1139	1140	1141	1142	1143
2170	1144	1145	1146	1147	1148	1149	1150	1151
2200	1152	1153	1154	1155	1156	1157	1158	1159
2210	1160	1161	1162	1163	1164	1165	1166	1167
2220	1168	1169	1170	1171	1172	1173	1174	1175
2230	1176	1177	1178	1179	1180	1181	1182	1183
2240	1184	1185	1186	1187	1188	1189	1190	1191
2250	1192	1193	1194	1195	1196	1197	1198	1199
2260	1200	1201	1202	1203	1204	1205	1206	1207
2270	1208	1209	1210	1211	1212	1213	1214	1215
2300	1216	1217	1218	1219	1220	1221	1222	1223
2310	1224	1225	1226	1227	1228	1229	1230	1231
2320	1232	1233	1234	1235	1236	1237	1238	1239
2330	1240	1241	1242	1243	1244	1245	1246	1247
2340	1248	1249	1250	1251	1252	1253	1254	1255
2350	1256	1257	1258	1259	1260	1261	1262	1263
2360	1264	1265	1266	1267	1268	1269	1270	1271
2370	1272	1273	1274	1275	1276	1277	1278	1279

	0	1	2	3	4	5	6	7
3000	1536	1537	1538	1539	1540	1541	1542	1543
3010	1544	1545	1546	1547	1548	1549	1550	1551
3020	1552	1553	1554	1555	1556	1557	1558	1559
3030	1560	1561	1562	1563	1564	1565	1566	1567
3040	1568	1569	1570	1571	1572	1573	1574	1575
3050	1576	1577	1578	1579	1580	1581	1582	1583
3060	1584	1585	1586	1587	1588	1589	1590	1591
3070	1592	1593	1594	1595	1596	1597	1598	1599
3100	1600	1601	1602	1603	1604	1605	1606	1607
3110	1608	1609	1610	1611	1612	1613	1614	1615
3120	1616	1617	1618	1619	1620	1621	1622	1623
3130	1624	1625	1626	1627	1628	1629	1630	1631
3140	1632	1633	1634	1635	1636	1637	1638	1639
3150	1640	1641	1642	1643	1644	1645	1646	1647
3160	1648	1649	1650	1651	1652	1653	1654	1655
3170	1656	1657	1658	1659	1660	1661	1662	1663
3200	1664	1665	1666	1667	1668	1669	1670	1671
3210	1672	1673	1674	1675	1676	1677	1678	1679
3220	1680	1681	1682	1683	1684	1685	1686	1687
3230	1688	1689	1690	1691	1692	1693	1694	1695
3240	1696	1697	1698	1699	1700	1701	1702	1703
3250	1704	1705	1706	1707	1708	1709	1710	1711
3260	1712	1713	1714	1715	1716	1717	1718	1719
3270	1720	1721	1722	1723	1724	1725	1726	1727
3300	1728	1729	1730	1731	1732	1733	1734	1735
3310	1736	1737	1738	1739	1740	1741	1742	1743
3320	1744	1745	1746	1747	1748	1749	1750	1751
3330	1752	1753	1754	1755	1756	1757	1758	1759
3340	1760	1761	1762	1763	1764	1765	1766	1767
3350	1768	1769	1770	1771	1772	1773	1774	1775
3360	1776	1777	1778	1779	1780	1781	1782	1783
3370	1784	1785	1786	1787	1788	1789	1790	1791

	0	1	2	3	4	5	6	7
2400	1280	1281	1282	1283	1284	1285	1286	1287
2410	1288	1289	1290	1291	1292	1293	1294	1295
2420	1296	1297	1298	1299	1300	1301	1302	1303
2430	1304	1305	1306	1307	1308	1309	1310	1311
2440	1312	1313	1314	1315	1316	1317	1318	1319
2450	1320	1321	1322	1323	1324	1325	1326	1327
2460	1328	1329	1330	1331	1332	1333	1334	1335
2470	1336	1337	1338	1339	1340	1341	1342	1343
2500	1344	1345	1346	1347	1348	1349	1350	1351
2510	1352	1353	1354	1355	1356	1357	1358	1359
2520	1360	1361	1362	1363	1364	1365	1366	1367
2530	1368	1369	1370	1371	1372	1373	1374	1375
2540	1376	1377	1378	1379	1380	1381	1382	1383
2550	1384	1385	1386	1387	1388	1389	1390	1391
2560	1392	1393	1394	1395	1396	1397	1398	1399
2570	1400	1401	1402	1403	1404	1405	1406	1407
2600	1408	1409	1410	1411	1412	1413	1414	1415
2610	1416	1417	1418	1419	1420	1421	1422	1423
2620	1424	1425	1426	1427	1428	1429	1430	1431
2630	1432	1433	1434	1435	1436	1437	1438	1439
2640	1440	1441	1442	1443	1444	1445	1446	1447
2650	1448	1449	1450	1451	1452	1453	1454	1455
2660	1456	1457	1458	1459	1460	1461	1462	1463
2670	1464	1465	1466	1467	1468	1469	1470	1471
2700	1472	1473	1474	1475	1476	1477	1478	1479
2710	1480	1481	1482	1483	1484	1485	1486	1487
2720	1488	1489	1490	1491	1492	1493	1494	1495
2730	1496	1497	1498	1499	1500	1501	1502	1503
2740	1504	1505	1506	1507	1508	1509	1510	1511
2750	1512	1513	1514	1515	1516	1517	1518	1519
2760	1520	1521	1522	1523	1524	1525	1526	1527
2770	1528	1529	1530	1531	1532	1533	1534	1535

	0	1	2	3	4	5	6	7
3400	1792	1793	1794	1795	1796	1797	1798	1799
3410	1800	1801	1802	1803	1804	1805	1806	1807
3420	1808	1809	1810	1811	1812	1813	1814	1815
3430	1816	1817	1818	1819	1820	1821	1822	1823
3440	1824	1825	1826	1827	1828	1829	1830	1831
3450	1832	1833	1834	1835	1836	1837	1838	1839
3460	1840	1841	1842	1843	1844	1845	1846	1847
3470	1848	1849	1850	1851	1852	1853	1854	1855
3500	1856	1857	1858	1859	1860	1861	1862	1863
3510	1864	1865	1866	1867	1868	1869	1870	1871
3520	1872	1873	1874	1875	1876	1877	1878	1879
3530	1880	1881	1882	1883	1884	1885	1886	1887
3540	1888	1889	1890	1891	1892	1893	1894	1895
3550	1896	1897	1898	1899	1900	1901	1902	1903
3560	1904	1905	1906	1907	1908	1909	1910	1911
3570	1912	1913	1914	1915	1916	1917	1918	1919
3600	1920	1921	1922	1923	1924	1925	1926	1927
3610	1928	1929	1930	1931	1932	1933	1934	1935
3620	1936	1937	1938	1939	1940	1941	1942	1943
3630	1944	1945	1946	1947	1948	1949	1950	1951
3640	1952	1953	1954	1955	1956	1957	1958	1959
3650	1960	1961	1962	1963	1964	1965	1966	1967
3660	1968	1969	1970	1971	1972	1973	1974	1975
3670	1976	1977	1978	1979	1980	1981	1982	1983
3700	1984	1985	1986	1987	1988	1989	1990	1991
3710	1992	1993	1994	1995	1996	1997	1998	1999
3720	2000	2001	2002	2003	2004	2005	2006	2007
3730	2008	2009	2010	2011	2012	2013	2014	2015
3740	2016	2017	2018	2019	2020	2021	2022	2023
3750	2024	2025	2026	2027	2028	2029	2030	2031
3760	2032	2033	2034	2035	2036	2037	2038	2039
3770	2040	2041	2042	2043	2044	2045	2046	2047

2000 1024
to to
2777 1535
(Octal) (Decimal)

Octal Decimal
10000 - 4096
20000 - 8192
30000 - 12288
40000 - 16384
50000 - 20480
60000 - 24576
70000 - 28672

3000 1536
to to
3777 2047
(Octal) (Decimal)

Octal-Decimal Integer Conversion Table

4000 2048
to to
4777 2559
(Octal) (Decimal)

Octal Decimal
10000 - 4096
20000 - 8192
30000 - 12288
40000 - 16384
50000 - 20480
60000 - 24576
70000 - 28672

	0	1	2	3	4	5	6	7
4000	2048	2049	2050	2051	2052	2053	2054	2055
4010	2056	2057	2058	2059	2060	2061	2062	2063
4020	2064	2065	2066	2067	2068	2069	2070	2071
4030	2072	2073	2074	2075	2076	2077	2078	2079
4040	2080	2081	2082	2083	2084	2085	2086	2087
4050	2088	2089	2090	2091	2092	2093	2094	2095
4060	2096	2097	2098	2099	2100	2101	2102	2103
4070	2104	2105	2106	2107	2108	2109	2110	2111
4100	2112	2113	2114	2115	2116	2117	2118	2119
4110	2120	2121	2122	2123	2124	2125	2126	2127
4120	2128	2129	2130	2131	2132	2133	2134	2135
4130	2136	2137	2138	2139	2140	2141	2142	2143
4140	2144	2145	2146	2147	2148	2149	2150	2151
4150	2152	2153	2154	2155	2156	2157	2158	2159
4160	2160	2161	2162	2163	2164	2165	2166	2167
4170	2168	2169	2170	2171	2172	2173	2174	2175
4200	2176	2177	2178	2179	2180	2181	2182	2183
4210	2184	2185	2186	2187	2188	2189	2190	2191
4220	2192	2193	2194	2195	2196	2197	2198	2199
4230	2200	2201	2202	2203	2204	2205	2206	2207
4240	2208	2209	2210	2211	2212	2213	2214	2215
4250	2216	2217	2218	2219	2220	2221	2222	2223
4260	2224	2225	2226	2227	2228	2229	2230	2231
4270	2232	2233	2234	2235	2236	2237	2238	2239
4300	2240	2241	2242	2243	2244	2245	2246	2247
4310	2248	2249	2250	2251	2252	2253	2254	2255
4320	2256	2257	2258	2259	2260	2261	2262	2263
4330	2264	2265	2266	2267	2268	2269	2270	2271
4340	2272	2273	2274	2275	2276	2277	2278	2279
4350	2280	2281	2282	2283	2284	2285	2286	2287
4360	2288	2289	2290	2291	2292	2293	2294	2295
4370	2296	2297	2298	2299	2300	2301	2302	2303

	0	1	2	3	4	5	6	7
4400	2304	2305	2306	2307	2308	2309	2310	2311
4410	2312	2313	2314	2315	2316	2317	2318	2319
4420	2320	2321	2322	2323	2324	2325	2326	2327
4430	2328	2329	2330	2331	2332	2333	2334	2335
4440	2336	2337	2338	2339	2340	2341	2342	2343
4450	2344	2345	2346	2347	2348	2349	2350	2351
4460	2352	2353	2354	2355	2356	2357	2358	2359
4470	2360	2361	2362	2363	2364	2365	2366	2367
4500	2368	2369	2370	2371	2372	2373	2374	2375
4510	2376	2377	2378	2379	2380	2381	2382	2383
4520	2384	2385	2386	2387	2388	2389	2390	2391
4530	2392	2393	2394	2395	2396	2397	2398	2399
4540	2400	2401	2402	2403	2404	2405	2406	2407
4550	2408	2409	2410	2411	2412	2413	2414	2415
4560	2416	2417	2418	2419	2420	2421	2422	2423
4570	2424	2425	2426	2427	2428	2429	2430	2431
4600	2432	2433	2434	2435	2436	2437	2438	2439
4610	2440	2441	2442	2443	2444	2445	2446	2447
4620	2448	2449	2450	2451	2452	2453	2454	2455
4630	2456	2457	2458	2459	2460	2461	2462	2463
4640	2464	2465	2466	2467	2468	2469	2470	2471
4650	2472	2473	2474	2475	2476	2477	2478	2479
4660	2480	2481	2482	2483	2484	2485	2486	2487
4670	2488	2489	2490	2491	2492	2493	2494	2495
4700	2496	2497	2498	2499	2500	2501	2502	2503
4710	2504	2505	2506	2507	2508	2509	2510	2511
4720	2512	2513	2514	2515	2516	2517	2518	2519
4730	2520	2521	2522	2523	2524	2525	2526	2527
4740	2528	2529	2530	2531	2532	2533	2534	2535
4750	2536	2537	2538	2539	2540	2541	2542	2543
4760	2544	2545	2546	2547	2548	2549	2550	2551
4770	2552	2553	2554	2555	2556	2557	2558	2559

5000 2560
to to
5777 3071
(Octal) (Decimal)

	0	1	2	3	4	5	6	7
5000	2560	2561	2562	2563	2564	2565	2566	2567
5010	2568	2569	2570	2571	2572	2573	2574	2575
5020	2576	2577	2578	2579	2580	2581	2582	2583
5030	2584	2585	2586	2587	2588	2589	2590	2591
5040	2592	2593	2594	2595	2596	2597	2598	2599
5050	2600	2601	2602	2603	2604	2605	2606	2607
5060	2608	2609	2610	2611	2612	2613	2614	2615
5070	2616	2617	2618	2619	2620	2621	2622	2623
5100	2624	2625	2626	2627	2628	2629	2630	2631
5110	2632	2633	2634	2635	2636	2637	2638	2639
5120	2640	2641	2642	2643	2644	2645	2646	2647
5130	2648	2649	2650	2651	2652	2653	2654	2655
5140	2656	2657	2658	2659	2660	2661	2662	2663
5150	2664	2665	2666	2667	2668	2669	2670	2671
5160	2672	2673	2674	2675	2676	2677	2678	2679
5170	2680	2681	2682	2683	2684	2685	2686	2687
5200	2688	2689	2690	2691	2692	2693	2694	2695
5210	2696	2697	2698	2699	2700	2701	2702	2703
5220	2704	2705	2706	2707	2708	2709	2710	2711
5230	2712	2713	2714	2715	2716	2717	2718	2719
5240	2720	2721	2722	2723	2724	2725	2726	2727
5250	2728	2729	2730	2731	2732	2733	2734	2735
5260	2736	2737	2738	2739	2740	2741	2742	2743
5270	2744	2745	2746	2747	2748	2749	2750	2751
5300	2752	2753	2754	2755	2756	2757	2758	2759
5310	2760	2761	2762	2763	2764	2765	2766	2767
5320	2768	2769	2770	2771	2772	2773	2774	2775
5330	2776	2777	2778	2779	2780	2781	2782	2783
5340	2784	2785	2786	2787	2788	2789	2790	2791
5350	2792	2793	2794	2795	2796	2797	2798	2799
5360	2800	2801	2802	2803	2804	2805	2806	2807
5370	2808	2809	2810	2811	2812	2813	2814	2815

	0	1	2	3	4	5	6	7
5400	2816	2817	2818	2819	2820	2821	2822	2823
5410	2824	2825	2826	2827	2828	2829	2830	2831
5420	2832	2833	2834	2835	2836	2837	2838	2839
5430	2840	2841	2842	2843	2844	2845	2846	2847
5440	2848	2849	2850	2851	2852	2853	2854	2855
5450	2856	2857	2858	2859	2860	2861	2862	2863
5460	2864	2865	2866	2867	2868	2869	2870	2871
5470	2872	2873	2874	2875	2876	2877	2878	2879
5500	2880	2881	2882	2883	2884	2885	2886	2887
5510	2888	2889	2890	2891	2892	2893	2894	2895
5520	2896	2897	2898	2899	2900	2901	2902	2903
5530	2904	2905	2906	2907	2908	2909	2910	2911
5540	2912	2913	2914	2915	2916	2917	2918	2919
5550	2920	2921	2922	2923	2924	2925	2926	2927
5560	2928	2929	2930	2931	2932	2933	2934	2935
5570	2936	2937	2938	2939	2940	2941	2942	2943
5600	2944	2945	2946	2947	2948	2949	2950	2951
5610	2952	2953	2954	2955	2956	2957	2958	2959
5620	2960	2961	2962	2963	2964	2965	2966	2967
5630	2968	2969	2970	2971	2972	2973	2974	2975
5640	2976	2977	2978	2979	2980	2981	2982	2983
5650	2984	2985	2986	2987	2988	2989	2990	2991
5660	2992	2993	2994	2995	2996	2997	2998	2999
5670	3000	3001	3002	3003	3004	3005	3006	3007
5700	3008	3009	3010	3011	3012	3013	3014	3015
5710	3016	3017	3018	3019	3020	3021	3022	3023
5720	3024	3025	3026	3027	3028	3029	3030	3031
5730	3032	3033	3034	3035	3036	3037	3038	3039
5740	3040	3041	3042	3043	3044	3045	3046	3047
5750	3048	3049	3050	3051	3052	3053	3054	3055
5760	3056	3057	3058	3059	3060	3061	3062	3063
5770	3064	3065	3066	3067	3068	3069	3070	3071

Octal-Decimal Integer Conversion Table

	0	1	2	3	4	5	6	7
6000	3072	3073	3074	3075	3076	3077	3078	3079
6010	3080	3081	3082	3083	3084	3085	3086	3087
6020	3088	3089	3090	3091	3092	3093	3094	3095
6030	3096	3097	3098	3099	3100	3101	3102	3103
6040	3104	3105	3106	3107	3108	3109	3110	3111
6050	3112	3113	3114	3115	3116	3117	3118	3119
6060	3120	3121	3122	3123	3124	3125	3126	3127
6070	3128	3129	3130	3131	3132	3133	3134	3135
6100	3136	3137	3138	3139	3140	3141	3142	3143
6110	3144	3145	3146	3147	3148	3149	3150	3151
6120	3152	3153	3154	3155	3156	3157	3158	3159
6130	3160	3161	3162	3163	3164	3165	3166	3167
6140	3168	3169	3170	3171	3172	3173	3174	3175
6150	3176	3177	3178	3179	3180	3181	3182	3183
6160	3184	3185	3186	3187	3188	3189	3190	3191
6170	3192	3193	3194	3195	3196	3197	3198	3199
6200	3200	3201	3202	3203	3204	3205	3206	3207
6210	3208	3209	3210	3211	3212	3213	3214	3215
6220	3216	3217	3218	3219	3220	3221	3222	3223
6230	3224	3225	3226	3227	3228	3229	3230	3231
6240	3232	3233	3234	3235	3236	3237	3238	3239
6250	3240	3241	3242	3243	3244	3245	3246	3247
6260	3248	3249	3250	3251	3252	3253	3254	3255
6270	3256	3257	3258	3259	3260	3261	3262	3263
6300	3264	3265	3266	3267	3268	3269	3270	3271
6310	3272	3273	3274	3275	3276	3277	3278	3279
6320	3280	3281	3282	3283	3284	3285	3286	3287
6330	3288	3289	3290	3291	3292	3293	3294	3295
6340	3296	3297	3298	3299	3300	3301	3302	3303
6350	3304	3305	3306	3307	3308	3309	3310	3311
6360	3312	3313	3314	3315	3316	3317	3318	3319
6370	3320	3321	3322	3323	3324	3325	3326	3327

	0	1	2	3	4	5	6	7
6400	3328	3329	3330	3331	3332	3333	3334	3335
6410	3336	3337	3338	3339	3340	3341	3342	3343
6420	3344	3345	3346	3347	3348	3349	3350	3351
6430	3352	3353	3354	3355	3356	3357	3358	3359
6440	3360	3361	3362	3363	3364	3365	3366	3367
6450	3368	3369	3370	3371	3372	3373	3374	3375
6460	3376	3377	3378	3379	3380	3381	3382	3383
6470	3384	3385	3386	3387	3388	3389	3390	3391
6500	3392	3393	3394	3395	3396	3397	3398	3399
6510	3400	3401	3402	3403	3404	3405	3406	3407
6520	3408	3409	3410	3411	3412	3413	3414	3415
6530	3416	3417	3418	3419	3420	3421	3422	3423
6540	3424	3425	3426	3427	3428	3429	3430	3431
6550	3432	3433	3434	3435	3436	3437	3438	3439
6560	3440	3441	3442	3443	3444	3445	3446	3447
6570	3448	3449	3450	3451	3452	3453	3454	3455
6600	3456	3457	3458	3459	3460	3461	3462	3463
6610	3464	3465	3466	3467	3468	3469	3470	3471
6620	3472	3473	3474	3475	3476	3477	3478	3479
6630	3480	3481	3482	3483	3484	3485	3486	3487
6640	3488	3489	3490	3491	3492	3493	3494	3495
6650	3496	3497	3498	3499	3500	3501	3502	3503
6660	3504	3505	3506	3507	3508	3509	3510	3511
6670	3512	3513	3514	3515	3516	3517	3518	3519
6700	3520	3521	3522	3523	3524	3525	3526	3527
6710	3528	3529	3530	3531	3532	3533	3534	3535
6720	3536	3537	3538	3539	3540	3541	3542	3543
6730	3544	3545	3546	3547	3548	3549	3550	3551
6740	3552	3553	3554	3555	3556	3557	3558	3559
6750	3560	3561	3562	3563	3564	3565	3566	3567
6760	3568	3569	3570	3571	3572	3573	3574	3575
6770	3576	3577	3578	3579	3580	3581	3582	3583

6000 3072
to to
6777 3583
(Octal) (Decimal)

Octal Decimal
10000 - 4096
20000 - 8192
30000 - 12288
40000 - 16384
50000 - 20480
60000 - 24576
70000 - 28672

	0	1	2	3	4	5	6	7
7000	3584	3585	3586	3587	3588	3589	3590	3591
7010	3592	3593	3594	3595	3596	3597	3598	3599
7020	3600	3601	3602	3603	3604	3605	3606	3607
7030	3608	3609	3610	3611	3612	3613	3614	3615
7040	3616	3617	3618	3619	3620	3621	3622	3623
7050	3624	3625	3626	3627	3628	3629	3630	3631
7060	3632	3633	3634	3635	3636	3637	3638	3639
7070	3640	3641	3642	3643	3644	3645	3646	3647
7100	3648	3649	3650	3651	3652	3653	3654	3655
7110	3656	3657	3658	3659	3660	3661	3662	3663
7120	3664	3665	3666	3667	3668	3669	3670	3671
7130	3672	3673	3674	3675	3676	3677	3678	3679
7140	3680	3681	3682	3683	3684	3685	3686	3687
7150	3688	3689	3690	3691	3692	3693	3694	3695
7160	3696	3697	3698	3699	3700	3701	3702	3703
7170	3704	3705	3706	3707	3708	3709	3710	3711
7200	3712	3713	3714	3715	3716	3717	3718	3719
7210	3720	3721	3722	3723	3724	3725	3726	3727
7220	3728	3729	3730	3731	3732	3733	3734	3735
7230	3736	3737	3738	3739	3740	3741	3742	3743
7240	3744	3745	3746	3747	3748	3749	3750	3751
7250	3752	3753	3754	3755	3756	3757	3758	3759
7260	3760	3761	3762	3763	3764	3765	3766	3767
7270	3768	3769	3770	3771	3772	3773	3774	3775
7300	3776	3777	3778	3779	3780	3781	3782	3783
7310	3784	3785	3786	3787	3788	3789	3790	3791
7320	3792	3793	3794	3795	3796	3797	3798	3799
7330	3800	3801	3802	3803	3804	3805	3806	3807
7340	3808	3809	3810	3811	3812	3813	3814	3815
7350	3816	3817	3818	3819	3820	3821	3822	3823
7360	3824	3825	3826	3827	3828	3829	3830	3831
7370	3832	3833	3834	3835	3836	3837	3838	3839

	0	1	2	3	4	5	6	7
7400	3840	3841	3842	3843	3844	3845	3846	3847
7410	3848	3849	3850	3851	3852	3853	3854	3855
7420	3856	3857	3858	3859	3860	3861	3862	3863
7430	3864	3865	3866	3867	3868	3869	3870	3871
7440	3872	3873	3874	3875	3876	3877	3878	3879
7450	3880	3881	3882	3883	3884	3885	3886	3887
7460	3888	3889	3890	3891	3892	3893	3894	3895
7470	3896	3897	3898	3899	3900	3901	3902	3903
7500	3904	3905	3906	3907	3908	3909	3910	3911
7510	3912	3913	3914	3915	3916	3917	3918	3919
7520	3920	3921	3922	3923	3924	3925	3926	3927
7530	3928	3929	3930	3931	3932	3933	3934	3935
7540	3936	3937	3938	3939	3940	3941	3942	3943
7550	3944	3945	3946	3947	3948	3949	3950	3951
7560	3952	3953	3954	3955	3956	3957	3958	3959
7570	3960	3961	3962	3963	3964	3965	3966	3967
7600	3968	3969	3970	3971	3972	3973	3974	3975
7610	3976	3977	3978	3979	3980	3981	3982	3983
7620	3984	3985	3986	3987	3988	3989	3990	3991
7630	3992	3993	3994	3995	3996	3997	3998	3999
7640	4000	4001	4002	4003	4004	4005	4006	4007
7650	4008	4009	4010	4011	4012	4013	4014	4015
7660	4016	4017	4018	4019	4020	4021	4022	4023
7670	4024	4025	4026	4027	4028	4029	4030	4031
7700	4032	4033	4034	4035	4036	4037	4038	4039
7710	4040	4041	4042	4043	4044	4045	4046	4047
7720	4048	4049	4050	4051	4052	4053	4054	4055
7730	4056	4057	4058	4059	4060	4061	4062	4063
7740	4064	4065	4066	4067	4068	4069	4070	4071
7750	4072	4073	4074	4075	4076	4077	4078	4079
7760	4080	4081	4082	4083	4084	4085	4086	4087
7770	4088	4089	4090	4091	4092	4093	4094	4095

7000 3584
to to
7777 4095
(Octal) (Decimal)

APPENDIX B

REPORT ON TEST FOR $\frac{1}{2}$ " TUBE
(LUBE LINE)

REPORT ON TEST FOR 1/2" TUBE

(LUBE LINE)

"PURPOSE":

To determine the strength and endurance of stainless steel tube 1/2" outside diameter and 0.035 wall thickness through tension and bending.

"APPARATUS":

Flat bar 1/2" x 2 1/2" x 4'-9" long bent to 3'-0" radius. Frame to attach bent flat bar. 1/2" sch 80 pipe used as tube sleeve at each end of the tube. 1/2" stainless steel tube; five steel discs each weighting 219.84 lbs, (total 1099.2 lbs.).

DESCRIPTION:

A piece of 12'-0" long 1/2" stainless steel tube was hung to the top of the bent flat bar by means of steel plate and top sleeve. The bottom sleeve was then welded and a hook type assembly was also welded to the bottom sleeve. The weight of the five steel discs was then hung from the hook attached to the bottom sleeve. In the first phase of the experiment only one bent flat bar was used. Swinging of the hung weight was made until the tube made a 15° angle with the vertical and an elapse time of 12 minutes passed, then the weight was stopped and measures of the tube were made, this was done twice for the first phase of the tube. (See Fig. 1.)

For the second phase of the experiment a second flat bar was added opposite to the first one (Fig. II). The weight was swung so the tube could make the same 15° angle with the vertical. Weight was stopped at five minutes intervals and measures were taken up to an elapsed time of 30 minutes. Three marks were made in the tube: A, B, C, (See Fig. III for location).

FIG I

Mustang Engineering Co.

13230 CAMBRIDGE STREET
SANTA FE SPRINGS, CA. 90670

DR'N	REVISIONS			
CHK.	NO.	DATE	BY	
SCALE				
DATE				
APP'D.				

1/2" ϕ TUBE TESTING
ASSEMBLY

JOB NO.	DWG. NO.
7975	A-313

FIG II

Mustang Engineering Co.

13230 CAMBRIDGE STREET
SANTA FE SPRINGS, CA. 90670

DR'N. *DLI*

REVISIONS

CHK.

NO. DATE BY

SCALE

NONE

DATE *9-4-80*

APP'D.

1/2" TUBE TESTING
ASSEMBLY

JOB NO.

DWG. NO.

7975

A-312

FIG III

Mustang Engineering Co.

13230 CAMBRIDGE STREET
SANTA FE SPRINGS, CA. 90670

DR'N. *DRX*

REVISIONS

CHK.

NO.

DATE

BY

SCALE *NONE*

DATE *9-4-80*

APP'D.

1/2" ϕ TUBE TESTING
ASSEMBLY

JOB NO.

DWG. NO.

7975

A-314

1/2" TUBE TEST

TEST DATA

<u>DATE</u>	<u>BEGINNING DIA./IN.</u>	<u>TIME ELAPSED IN MIN.</u>	<u>CYCLES PASSED</u>	<u>TUBE LG. IN IN.</u>	<u>DIA. @ A IN IN.</u>	<u>DIA. @ B IN IN.</u>	<u>DIA. @ C IN IN.</u>	<u>REMARKS</u>
8/25/80	0.507	0	0	43.25	0.57A	0.57	0.57	Only one bent bar (See Fig. 1)
8/25/80	0.507	12	325	43.50	0.501A	0.5015	0.5020	
8/25/80	0.507	10	255	43.937	0.501A	0.5015	0.5020	Test stopped
8/26/80	0.507	5	145	44.687	0.499A	0.501	0.5015	
8/26/80	0.507	5	145	44.875	0.496A	0.501	0.5015	See Fig. II Dble bent bar used
8/26/80	0.507	5	145	45.0	0.425A' 0.496A	0.501	0.5015	Dim @ A is 90° to other A dia. measured
8/26/80	0.507	5	145	45.0625	0.415A' 0.490A	0.501	0.5015	
8/26/80	0.507	5	145	45.125	0.411A' 0.501A	0.501	0.5015	
TOTAL		52	1450	1.875	0.096	0.006	0.005	

SAMPLE CALCULATION

$$\text{Stress in tube} = \frac{P}{A} + \frac{M}{Z}$$

$$\text{But } \frac{P}{A} = \sigma_T \text{ and } \sigma_T = \frac{L \times E}{L} = \frac{P}{A}$$

Test of 8/25/80:

$$= \frac{(0.25") \times 29 \times 106 \text{ lbs.}}{138} = 52,536 \text{ psi}$$

For $\frac{M}{Z}$ See conclusions and results

1/2" TUBE TEST

TABULAR SUMMARY

<u>DATE</u>	<u>TIME ELAPSED IN MIN.</u>	<u>BEGINNING LG/INCHES</u>	<u>ENDING LG IN INCHES</u>	<u>ELONGATION IN INCHES</u>	<u>MODULUS OF ELASTICITY</u>	<u>STRESS IN TUBE /PSI</u>	<u>REMARKS</u>
8/25/80	0	138.250	138.250	0	29×10^6	21,515	Using one bent bar only
8/25/80	12	138.250	138.500	0.250	29×10^6	52,536	See conclusions & results
8/25/80	10	138.500	138.937	0.437	29×10^6	91,833	
8/26/80	5	138.937	139.687	0.750	29×10^6	157,609	Dble bent bar
8/26/80	5	139.687	139.875	0.188	29×10^6	39,507	See conclusions & results
8/26/80	5	139.875	140.000	0.125	29×10^6	26,268	
8/26/80	5	140.000	140.0625	0.0625	29×10^6	13,134	
8/26/80	5	140.0625	140.0625	0.0625	29×10^6	-	
8/26/80	5	140.0625	140.125	0.0625	29×10^6	13,134	

CONCLUSIONS & RESULTS

The test was not conducted under the most strict method. Therefore there is a large margin for error.

The stress calculated in the tabular summary is the tensile stress of tube after some bending was applied to the tube. But the exact bending stress can not be calculated since the force applied to the weight hung was not a constant force and the angle the tube made with the vertical was not always fifteen degrees. The acceleration of the weight was not a constant one since impulse took a big factor, because the weight was pushed with an uneven force.

If the purpose of the test has been to calculate the bending moment, it would had been a complete failure, even though a more sophisticated mechanism could be used if this were the case.

The purpose of the test was to find the stress and endurance of the 1/2" tube, and these were accomplished with certain accuracy. Since the stress shown in the tabular data shows the effect of bending after it has been applied.

Flattening of the tube occurred in the direction of motion of the weight. Bending in this test was occurring at the same location in the tube, therefore diminishing the strength of the tube. It can be seen that the endurance limit of the tube was never reached and after 52 minutes and approximately 1450 cycles the tube was still holding and no cracks had occurred in the tube.

In actuality, bending will occur all throughout the length and rotated directions of the tube and not at a specific spot as happened during the test.

The test indicated the tube selection is adequate if the injection and retrieval apparatus subjects the tube to no more sever forces.

APPENDIX C

REPORT ON PRESSURE/TEMPERATURE TEST OF
STABBING FIXTURE SEALS

REPORT ON
PRESSURE/TEMPERATURE TEST OF
STABBING FIXTURE SEALS

The seal test was divided in two parts:

a) Pressurizing with water at ambient temperatures

The first tests showed a failure of the backup rings. This appeared to be due to the insufficient weight on the stabbing fixture. Pressures over 1400 psi pushed the stabbing fixture up until liquid was escaping which decreased the pressure. This resulted in dropping of the stabbing fixture which sheared the backup rings. After securing the stabbing fixture with a U-bolt test pressures up to 2200 psi were reached, without failure or detectable leak.

b) Pressurizing with oil at elevated temperatures

For this test the stabbing fixture was filled with light motor oil. By means of a Chromalox bandheater and temperature controller the temperature at the lower part of the stabbing fixture was raised up to 410°F. During heating up the pressure was kept between 1200 and 1500 psi. Because of heat loss through the test stand this resulted in a seal temperatures of 290 - 310°F. When the temperature stabilized, the pressure was raised to 1600-1800 psi and occasionally up to 2000 psi.

This test was done three times in three consecutive days. At the end of the test a small leak was detected at the 2 1/2" O-rings. Because of their close proximity to the bandheater the Viton O-rings were damaged. The local temperature must have been well over 410°F. No visible leak was detected at the 1/2" stabbing fixture seals.

When the stabbing fixture was taken apart it appeared that the two lower backup seals were damaged. Out of experience from the previous tests our conclusion is that this damage was caused by partially lifting the stabbing fixture at pressures over 1800 psi or shearing the backup rings when inserting the stabber.

CONCLUSIONS AND RECOMMENDATIONS

The stabbing fixture seals are capable of holding pressures up to 1800 psi at temperature to at least 310°F. The usual operating temperature will be around 170°.

It seems advisable to increase the 2 1/2" OD, 1" ID weight tube to prevent problems as encountered in the test stand. We increased the weight to 550# (40') which should prevent lifting the weight up to 1900 psi. This will help prevent any seal damage, if any pressure surges occur.

It appears also to be very important that after stretching the backup rings over the stabber on the o-ring they are forced back to their original dimensions in order to avoid shearing the backup rings when inserting the stabber. A final check on this before lowering the 1/2" tube with the stabbing fixture in the well is recommended.

AUG 6 1980

SPERRY
RESEARCH CENTER

SPOSBURY, MASS.

TITLE ASSEMBLY
ELECTRICAL STABBING FIXT.

SCALE	DATE	DWG. NO.	REV. NO.
FULL	8-6-80	G-170-80	0

APPENDIX D

DESIGN AND PREDICTED PERFORMANCE OF
SPERRY STAND-ALONE TURBOPUMP UNIT

BARBER-NICHOLS

ENGINEERING

**DESIGN AND PREDICTED PERFORMANCE OF
SPERRY STAND-ALONE TURBO PUMP UNIT**

November, 1979

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1.0 INTRODUCTION

This document summarizes the Barber-Nichols Engineering design of the turbopump unit (TPU) for the Sperry stand-alone pump system. The subjects discussed include system design criteria, pump and turbine performance analysis, bearing selection, and the major aspects of mechanical design.

2.0 DESIGN CRITERIA

The basic TPU design criteria are given in Table I. The nominal values of well productivity index (PI) and brine temperature are those estimated for the test well; however, the TPU was designed to pump at least 1000 gpm over as wide a range of well conditions as possible.

The TPU frame size of 14 1/2 inches is compatible with a standard 16-inch well casing. Although this constraint has been met, it is not firm because future wells will probably be at least 24 inches in diameter.

Figure 1 shows the estimated brine pressure profile in the well for design flow conditions, and for maximum pressure conditions when the pump is momentarily dead-headed at design power. The relationships defining the pressures are also given.

The reference wellhead pressure P_{wh0} for a hot, non-flowing, unpumped well is assumed to be 130 psig; the desired flowing wellhead pressure P_{wh} is 175 psig. For a setting depth h of 2180 ft, an average brine density ρ of 56.1 lb/ft³, and a friction drop ΔP_f of 25 psi, the sum of the wellhead pressure and ΔP_f (see equations on Figure 1) determines the pump outlet pressure of 1050 psig for the design flow condition. The pump design pressure rise ΔP_d is the sum of the drawdown for 1050 gpm and a PI of 1.89, the additional desired wellhead pressure of 45 psi, and ΔP_f . Thus P_d is 625 psi, or 1600 ft of head. The pump inlet pressure is consequently 425 psig (the NPSH is 300 psia).

The pump inlet pressure P_3 for the deadhead condition is the sum of the non-flowing wellhead pressure and the gravity head. The deadhead pressure rise ΔP_{dh} is 27% higher than the design pressure rise, resulting in 1775 psig at the pump outlet and 925 psig at the wellhead.

The above considerations formed the basis for proceeding with the pump and turbine design.

3.0 DESIGN AND PERFORMANCE ANALYSIS

3.1 PUMP

TABLE I
TPU DESIGN CRITERIA

Nominal Brine Flow Rate	1000 gpm
Brine Temperature	340-350°F
Working Fluid	R-114
Maximum Diameter	14 1/2 in.
Setting Depth	2180 ft.
Well Productivity Index	1.9 gpm/psi
Minimum Wellhead Pressure	175 psig
Nominal Ambient Wet Bulb Temperature	75°F
Maximize Overall Efficiency:	
● Pump Goal	75%
● Turbine Goal	70%

*Actual Efficiency = 75%
Overall was 61% including
bearing loss.*

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PROJECT
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11-7-79

SUBJECT
PRESSURE PROFILE
IN WELL

DEADHEAD CONDITION

DESIGN FLOW CONDITION

$$\Delta P_1 = \frac{Ph}{144} + \Delta P_F = \frac{56.1(2180)}{144} + 25 = 875 \text{ PSI}$$

$$\Delta P_2 = \frac{Ph}{144} = 850 \text{ PSI}$$

$$\Delta P_D = \frac{Q}{F_I} + \Delta P_{WH} + \Delta P_F = \frac{1050}{1.69} + (175 - 130) + 25 = 625 \text{ PSI}$$

$$\Delta P_{DH} = 1.27 \Delta P_D = 795 \text{ PSI}$$

$$P_3 = P_{WH} + \Delta P_2 = 130 + 850 = 980 \text{ PSI}$$

FIGURE 1

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SUBJECT

PUMP PERFORMANCE

FIGURE 2

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DATE
11-1-79

SUBJECT
PUMP HEAD
COEFFICIENTS
VERSUS FLOW

FIGURE 3

TABLE II
PUMP CHARACTERISTICS

Design Specifications

Brine Flow Rate	1050 gpm
Well Production Index	1.89 gpm/psi
Head	1600 ft.
Efficiency Goal	75%
Maximum Speed	12,000 rpm

Design Parameters

Type	Centrifugal
Impeller Diameter	7.00 in.
Hub Diameter	1.25 in.
Eye Diameter	3.50 in.
Number of Pump Vanes	6
Type of Pump Vanes	25°, Backward Curved
Number of Diffuser Vanes	11
Number of Thrust - Balance Vanes	11
Speed	10,500 rpm
Specific Speed (CFS Units)	63.5
Suction Specific Speed (GPM Units)	2334
NPSH	767 ft.
Input Power	506 hp
Flow Rate	1050 gpm
Pressure Rise (330°F Brine)	625 psi
Head Coefficient, ψ	0.502
Flow Factor, ϕ	0.078
Input Pump Power	485 hp
Thrust Balance Vane Windage Loss	21 hp
Efficiency	76%

TABLE III
TURBINE CHARACTERISTICS

Design Specifications

Working Fluid	R-114
Inlet Temperature	330 ^o F
Ambient Wet Bulb Temperature	75 ^o F
Speed	10,500 rpm
Efficiency Goal	70%
Output Power	525 hp

Design Parameters

Wheel Mean Diameter	7.884 in.
Blade Height	0.829 in.
Blade Chord	0.460 in.
Number of Blades	86
Blade Tip Radial Clearance	0.010 in.
Specific Speed (CFS Units)	51.3
Um/Co	0.454
Nozzle Type	Rectangular, Straight
Nozzle Throat Area	1.21 in. ²
Number of Nozzles	23
Nozzle Height	0.650 in.
Nozzle Exit Angle	15 ^o
Output Power	525 hp
Efficiency (To static exit condition)	76%
Condensing Temperature	90 ^o F
Ambient Wet Bulb Temperature	75 ^o F

Design Flow Conditions

Exhaust Pressure	64.2 psia
Exhaust Temperature	171 ^o F
Isentropic Head	12.6 BTU/lb
Reaction Fraction	0.15
Blade Tip Leakage	1.5%
R-114 Flow Rate	38 lb/sec.
Inlet Pressure	570 psia

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GDP

BY

DATE
10-23-79

SUBJECT
TURBINE
PERFORMANCE

FIGURE 4

DESIGN PARAMETERS

- 10,500 RPM
- 525 HP
- 8.712 INCH TIP DIA.
- 830°F INLET TEMP.
- 90°F CONDENSING TEMP.
- 76% EFFICIENCY
- 570 PSIA INLET PRESSURE

PROJECT

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7-12-79

SUBJECT

TURBINE EFFICIENCY
VERSUS
VELOCITY
RATIO

FIGURE 5

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BY

DATE
10-25-79

SUBJECT
R-114 FLOW RATE VS
TURBINE INLET PRESS.

FIGURE 6

Pump pressure rise versus flow rate is shown for several horsepower levels in Figure 2. Pump pressure rise required to achieve 175 psig wellhead pressure for various PI values is also indicated. The range of anticipated PI's is from 1.5 to 3 gpm/psi. Required flow rates center about 1000 gpm, with some bias toward the high side. The range of pump power considered was limited to 500 hp. Therefore, the design point was chosen on the best-efficiency-pump (BEP) curve at the maximum horsepower. This biased the design toward the flow rates and makes allowance for the uncertainty in the well PI.

Figure 3 gives the predicted performance of the pump. These data will be verified by laboratory tests on the pump impeller and the diffuser and flow-straightening vanes.

Table II summarizes the pump characteristics at the design point indicated in Figure 2. The flow factor ϕ refers to the ratio of the radial velocity of the brine at the impeller tip to the impeller tip speed.

3.2 TURBINE

The turbine was designed to operate efficiently over the pump range of Figure 2. With TPU parasitic losses due to bearings, seals, oil pump, and shaft windage, the design output power of 525 hp was chosen. The design specifications and the resulting design parameters and flow conditions are given in Table III.

The turbine design is determined largely by computer analysis because of the many calculations required to optimize performance. Based on design specifications and generalized correlations of specific speed and specific diameter, various cases are first analyzed with the Reaction Turbine Design Program to optimize efficiency and power as a function of velocity ratio and reaction fraction, other parameters being fixed. Then nozzle, wheel and blade geometries are optimized to determine details of the turbine configuration. The Reaction Turbine Off-Design Program is then used to determine off-design performance curves for a fixed geometry. This general procedure is iterated, using scaling relationships as a guide, until satisfactory performance is obtained over the range of interest.

Auxiliary computer calculations were required to relate turbine exhaust conditions to the condensing pressure and ambient wet bulb temperature on the surface.

Figure 4 shows the predicted performance curves and indicates the reference operating conditions corresponding to an inlet pressure of 570 psia.

Figure 5 shows the variation of turbine efficiency with velocity ratio, which is approximately proportional to pump flow rate. Again the higher efficiencies are biased toward the higher pump flow rates.

Fig. 6 indicates how R-114 flow rate varies with turbine inlet pressure.

Figure 7 shows the relationship of condensing temperature with turbine power and inlet pressure. The condensing temperature is determined by the condenser heat load and the ambient wet bulb temperature. The corresponding condensing pressure and the pressure drop in the exhaust return line determine the turbine exhaust pressure which in turn influences the turbine inlet pressure for a given flow rate and inlet temperature. Since the heat load that the evaporative condensers are capable of rejecting is a function of condensing temperature and ambient wet bulb temperature, an analysis was made to determine the relationship of actual test site wet bulb temperatures to turbine performance.

Figure 8 shows the predicted performance of a Baltimore Aircoil condenser unit, two of which will be available at the test site. The curves estimated by Barber-Nichols were used for the analysis.

Figure 9 shows the relationship between turbine power, inlet pressure, and wet bulb temperature. The range of practical interest is indicated by the condensing temperatures of 85°F and 105°F. The reference condition of 525 hp and 570 psia (and 90°F condensing temperature) requires a wet bulb temperature of less than 50°F. Thus one condenser unit is not sufficient to meet the design criterion of 75°F. However, the conditions under which one unit would be sufficient are indicated.

Figure 10 presents the same data for two condenser units as Figure 9 does for one unit. The reference conditions are seen to correspond to the specified wet bulb temperature of 75°F.

Figure 11 shows statistical data for wet bulb temperatures near the test site. The 75°F criterion was chosen because the condensers can handle the design heat load for 79% of the time during the summer months.

Laboratory tests will be performed on the turbine components to determine actual turbine performance curves. During these tests, measurements on a Baltimore Aircoil condenser will be made to refine the curves on Figure 8.

4.0 BEARING SELECTION

Hydrodynamic shaft bearings are more suitable for this TPU application than anti-friction bearings. However, to reduce development cost and to minimize the risk of unpredictable early failure, ball bearings were chosen since they were found to have adequate life for the field test. The disadvantage of ball bearings at high speed and high loads is limited life due to fatigue and the possibility of inadequate lubrication. Therefore, the analysis of the ball bearings revolved about these two factors.

Angular contact ball bearings require an axial preload to prevent centrifugal forces on the orbiting balls from creating a difference between the inner-race contact angle and the outer-race contact angle that is large enough to cause

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SUBJECT
TURBINE POWER VERSUS
INLET PRESSURE AND
CONDENSING TEMP.

FIGURE 7

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SUBJECT
PERFORMANCE OF
BALTIMORE AIR COIL
EVAPORATIVE CONDENSER

FIGURE 8

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SUBJECT
TURBINE PERFORMANCE
VS AMBIENT WET
BULB TEMPERATURE

FIGURE 9

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SUBJECT
TURBINE PERFORMANCE
VS AMBIENT WET
BULB TEMPERATURE

FIGURE 10

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SUBJECT

WET BULB TEMPS AT
EL CENTRO, CALIF.

FIGURE 11

excessive skidding of the balls on the outer race. Excessive skidding greatly decreases bearing life. Since any load decreases bearing life, it is desirable to minimize the preload.

Because of the theoretical difficulty of accurately predicting minimum preload for oil immersion, SKF was contracted by Sperry to conduct a detailed computer analysis. The results are shown in Figure 12 for size 212 bearings, with a 30° contact angle, arranged in a tandem duplex. A minimum load of about 800 lbs is indicated for a speed of 13,750 rpm. This is 21% less than the load predicted by the formula presented in Reference 1. Because of the complicated behaviour of the bearing for loads less than 1000 pounds, laboratory tests will be made to verify the minimum preload for the TPU bearings, and for the actual internally spring-preloaded bearing arrangement.

The selection of the bearing size to be used was based on comparative lifetime calculations. Figure 13 shows a comparison between 212 and 310 sizes. The 310 size gave a predicted life greater than any other standard bearing, and was therefore the one chosen for the TPU.

The life adjustment factor α_{23} accounts for the quality of materials and lubrication. The value used for Figure 13 was four. This is based on published SKF predictive data, use of consumable electrode vacuum melt (CEVM) M-50 steel, and the viscosity characteristics of Mobil SHC 629 oil.

Figure 14 gives the properties of the lubricant selected for the TPU, and Figure 15 presents the variation of α_{23} with temperature. Oil temperature at the bearings is estimated to be around 300°F with the present oil cooler design. Consequently, the L_{10} life for the TPU bearings at 10,500 rpm is estimated as 44,000 hours, as indicated in Figure 13. The calculations were done as described in Reference 1. The bearing specifications are given in Table IV.

5.0 MECHANICAL DESIGN

The TPU design is shown in Drawing GOP-900. The following discusses the main design considerations.

5.1 LUBRICATION SYSTEM

The TPU cavity will be filled with the oil described in Figure 14. It will be piped from a supply system on the surface that degasses it before use. The inlet tube is shown on the top left of the TPU drawing, and the outlet tube is on the lower left. The oil circulates through the TPU, lubricating the bearings and seals, and removing the heat created by parasitic losses. The inlet and outlet tubes connect to the oil cooler, which is an annular heat exchanger that filters the oil and transfers the heat to the turbine exhaust gas. The oil in the TPU cavity is maintained by the surface supply

CAGE SPEED/SHAFT SPEED

0.43
0.42
0.41
0.40
0.39
0.38
0.37

0

200

400

600

800

1000

1200

1400

1600

THRUST LOAD (LBS)

13,750 RPM

11,000 RPM

7212 Angular Contact Bearing
= 30° (Nominal)

Mobil SHC 629 Lubricant
250°F Operating Temperature

Figure 12: Sperry Vickers Pump Bearing Skid Analysis

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DATE
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SUBJECT

COMPARISON OF LIFE-
TIMES FOR 212 AND
310 BALL BEARINGS

FIGURE 13

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GOP

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DATE
11-6-79

SUBJECT

TPU OIL VISCOSITY

FIGURE 14

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PROJECT
GOP

BY

DATE
11-6-79

SUBJECT
BEARING LIFE FACTOR
VERSUS TEMPERATURE

FATIGUE LIFE ADJUSTMENT FACTOR FOR
BEARING MATERIALS AND LUBRICATION, K_{23}

- A-50 CEVM STEEL
- 310 BEARING S
- MOBIL SCH 629 OIL
- SKF DATA

10
8
6
4
2
N
0

150 200 250 300 350 400
OIL TEMPERATURE AT BEARINGS, °F

ESTIMATED
OPERATING
POINT

FIGURE 15

TABLE IV

BALL BEARING SPECIFICATIONS

Size	7310
Contact Angle	30° Nominal
Race Material	M-50 (CEVM)
Ball Material	M-50 (CEVM)
Cage Material	Silicon-Iron Bronze, Silver Plated
Class	5 or Better
Matched Machining	Tandem Duplex

system at a higher pressure than the brine to prevent brine leakage. The oil is circulated by a small impeller attached to the shaft between the two sets of bearings.

The heat load to the oil cooler was determined by estimating the individual parasitic losses associated with the bearings, seals, shaft windage, and oil pump. Heat conducted from the brine through the TPU was also included. The result is shown in Figure 16. The estimated maximum heat load, corresponding to an operating temperature of 300°F, is 26,000 BTU/hr.

For adequate heat removal in the oil cooler at the worst-case condition (250°F exhaust temperature at 75% of design power and 400 psia turbine inlet pressure), 5 gpm of oil flow is required. The corresponding pressure drop in the cooler and TPU were estimated, resulting in the oil impeller being sized for 30 psi rise.

Shaft windage losses are minimized by having a relatively small (0.030 in) clearance between shaft and housing. Oil pump work was also minimized by making the oil flow rate and pressure drop as low as practical. Equal oil flow to each pair of bearings is established by balancing the flow resistance with two drilled orifices.

5.2 CRITICAL SPEEDS

The critical speeds of the TPU were analyzed using a detailed computer analysis. The geometry is as shown in the TPU drawing. Bearing stiffness was treated as a parameter.

The results of the first two criticals are shown in Figure 17. The two curves are associated with the overhung turbine wheel and pump impeller. The third critical is shown in Figure 18, which is associated with bearing stiffness. The critical speed of the impeller exceeds design speed by more than 50%, a safe margin.

5.3 SEALS

5.3.1 Static Seals

Double O-ring seals will be used for all interfaces between the oil (300-350°F), the brine (350°F), and the R-114 (170-330°F). A good choice for the oil and water seals and a fair choice for the R-114 seals is Kalrez, made by Dupont. Its disadvantage is a cost of approximately \$2000 per TPU set. Other candidates are Viton and EPDM for R-114, EPDM for water and Viton for oil. During the laboratory testing, inexpensive Viton and EPDM can be used and replaced as necessary. Sperry will recommend the candidates for final selection, based on the results of compatibility tests.

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

GDP

BY

DATE

10-8-79

SUBJECT

MAXIMUM HEAT LOAD
TO TPU OIL COOLER

FIGURE 16

FIGURE 17
CRITICAL SPEED
GOP TURBO-PUMP-UNIT

5.3.2 Dynamic Seals

The shaft seals will separate the oil and incoming brine at the pump end (approximately 800 psid) and the oil and turbine exhaust at the turbine end (approximately 1160 psid). Tandem seals will be used on both ends. The mating rings and the nose pieces will probably be made of silicon carbide, and the seal carrier made of 15-5 stainless steel. The design and testing of the seals is being performed by Sperry Vickers.

5.4 MATERIALS

All of the TPU parts will be made of 15-5 PH stainless steel, including the TPU housing, pump and turbine diffusers, shaft, bearing spacers, bearing housing, pump impeller, turbine wheel, turbine nozzles, and load bolts. Armco 15-5 PH is a high-quality, chromium-based stainless steel which has high strength, excellent corrosion resistance, hardness, notch-toughness, and is easy to heat treat (hold at temperature and air cool). Its high transverse strength is advantageous to turbomachinery applications. It was selected over the more common and very similar 17-4 PH stainless because of superior stress-corrosion cracking. Tables V and VI give minimum 15-5 PH properties for various heat treatments. The treatment to be used for the TPU is H1025.

The turbine and pump diffusers, which also form part of the housing, will be sand cast and heat treated before machining. The rest of the parts will be machined from standard stock.

5.5 STRESS ANALYSIS

5.5.1 TPU Loads

Figure 19 is a load diagram of the TPU that shows the location of the various loads. Table VII gives the value of each surface load and the associated surface area and pressure. The pressures are those for the deadhead condition defined in Figure 1. Also shown are the bolt loads and the maximum thrust loads for the impeller and turbine. These loads were used for the stress analyses discussed below.

5.5.2 Shaft

The maximum torque τ occurs at zero speed, and is 2.0 times the torque at design speed:

$$\begin{aligned}\tau &= 2(63,025) \frac{P}{N} = \frac{126,050(525)}{10,500} \\ &= 6303 \text{ in} \cdot \text{lb}\end{aligned}$$

where P is the turbine horsepower, and N is the speed in rpm.

Table V
Minimum Properties
Longitudinal Direction — Intermediate Location
Air Melted (up to 8" section) or Consumable Electrode Vacuum Arc Remelted
(up to 12" section)

	Condition						
	H 900	H 925	H 1025	H 1075	H 1100	H 1150	H 1150-M
Ultimate Tensile Strength - psi	190,000	170,000	155,000	145,000	140,000	135,000	115,000
0.2% Yield Strength - psi	170,000	155,000	145,000	125,000	115,000	105,000	75,000
Elongation, % in 2" or 4 x D	10.0	10.0	12.0	13.0	14.0	16.0	18.0
Reduction of Area - %	35.0	38.0	45.0	45.0	45.0	50.0	55.0
Hardness							
Brinell	388/448	375/438	331/401	302/375	311/364	277/352	255/293
Rockwell	C 40/47	C 38/45	C 35/42	C 31/39	C 32/38	C 28/37	C 24/30
Impact, Charpy V-Notch, ft-lbs	•	5	15	20	25	30	55

• Minimum impact properties cannot be accepted in this condition.

Table VI
Minimum Properties
Transverse Direction
Consumable Electrode Vacuum Arc Remelted (up to 12" section)

	Condition						
	H 900	H 925	H 1025	H 1075	H 1100	H 1150	H 1150-M
Ultimate Tensile Strength - psi	190,000	170,000	155,000	145,000	140,000	135,000	115,000
0.2% Yield Strength - psi	170,000	155,000	145,000	125,000	115,000	105,000	75,000
Elongation, % in 2" or 4 x D	6.0	7.0	8.0	9.0	10.0	11.0	14.0
Reduction of Area - %	15.0	20.0	27.0	28.0	29.0	30.0	35.0
Hardness							
Brinell	388/448	375/438	331/401	302/375	311/364	277/352	255/293
Rockwell	C 40/47	C 38/45	C 35/42	C 31/39	C 32/38	C 28/37	C 24/30
Impact, Charpy V-Notch, ft-lbs Intermediate Location	•	•	10	15	15	20	35

• Minimum impact properties cannot be accepted in this condition.

FIGURE 19

TABLE VII

TPU LOADS

LOCATION	AREA, IN ²	PRESSURE, PSIG	LOAD, LBS
1	99	980	97,020
2	17	980	16,660
3	8	980	7,840
4	99	1775	175,725
5	38	1775	67,450
6	72	1775	127,800
7	112	1775	198,800
8	71	1775	126,025
9	19	1200	22,800
10	5	1800	9,000
11	36	1200	43,200
12	19	1200	22,800
13	13	1200	15,600
14	23	980	22,540
15	17	1775	30,175
16	9	1775	15,975
17	45	1775	79,875
18	72	1775	127,800
19	25	1200	30,000
20	36	1200	43,200
21	45	1775	79,875
22	24	1775	42,600
23	67	50	3,350
24	13	1200	15,600
25	13	75	975
26	22	1200	26,400
27	13	75	975
28	22	75	1,650
29	9	75	675
30	46	1775	81,650
A	-	-	108,195 tension
B	-	-	92,960
C	-	-	4,305 tension
D	-	-	1,500
E	-	-	0
F	-	-	1,500 tension
G	-	-	15,300 tension
H	-	-	130,100 tension
I	-	-	-1,500

The maximum stress is at the turbine keyway. The shear stress σ_s is

$$\sigma_s = \frac{16K\tau}{\pi D^3} = \frac{16(1.8)(6303)}{\pi (0.875)^3} = 86,251 \text{ psi}$$

where K is the stress concentration factor at the keyway, and D is the shaft diameter in inches.

The corresponding tensile stress σ_t is

$$\sigma_t = \frac{4F_t}{\pi D^2} = \frac{4(1500)}{\pi (0.875)^2} = 2,495 \text{ psi}$$

where F_t is the turbine thrust.

The maximum combined shear stress is therefore

$$\sigma_s \text{ max} = \sqrt{\frac{1}{4} \sigma_t^2 + \sigma_s^2} = 86,260 \text{ psi}$$

The shaft key shear stress is

$$\sigma_s = \frac{2\tau}{DdL} = \frac{2(6303)}{(0.875)(0.25)(0.938)} = 61,435 \text{ psi}$$

where d and L are the diameter and length of the cylindrical key.

The key and keyway bearing stress σ_b is

$$\sigma_b = 2\sigma_s = 122,870 \text{ psi}$$

5.5.3 Turbine Rotor

The stress due to rotation and a 13°F per inch radial temperature gradient was calculated by computer program for the actual rotor profile. The maximum tangential stress is at the hub keyway, which for a stress concentration factor of 1.3, is 17,980 psi. The maximum radial stress is at the thinnest section, and is due to a combination of rotational forces and bending caused by the pressure differential across the rotor that results in the 1500 pounds of turbine thrust. The maximum radial stress is

$$\sigma_r \text{ max} = \sigma_r + \sigma_t = 8608 + 5878 = 14,486 \text{ psi}$$

where σ_r is the radial stress due to rotation, and σ_t is the tensile stress due to bending.

5.5.4 Pipe String Load on TPU

The static load F_s of the pipe string is the weight of a 16-inch (O. D.) pipe with a 5/8 inch wall, less the weight of the 56.1 lb/ft³ brine displaced by the pipe:

$$F_s = (W_p - W_B)L_p = (104-78)2180 = 56,680 \text{ lb}$$

where W_p is the pipe weight in lb/ft, W_B is the weight of displaced brine in lb/ft, and L_p is the length of pipe in feet.

The dynamic load F_d , assuming the TPU bottoms out against the polished-bore receptacle (PBR) while being lowered at one ft/sec, is

$$\begin{aligned} F_d &= \sigma A = \rho CVA/g \\ &= 0.28 (200,000)(12)(30.8)/386 = 53,620 \text{ lb} \end{aligned}$$

where σ is the compressive stress in the pipe, A is the area of the pipe wall, ρ is the pipe density in lb/in^3 , C is the speed of sound in the pipe in inches/sec, v is pipe velocity at impact in inches/sec, and g is the acceleration of gravity.

The total load is

$$F = F_s + F_d = 110,300 \text{ lb}$$

and is transmitted from the threaded coupling, through the TPU, to the contact between the TPU and the PBR.

This load results in a compressive stress of approximately 5000 psi in the main diffuser vanes for the pump. If the PBR has a flared entrance so the pump housing impacts a flat, sloping surface, the local compressive stress in the housing will not exceed 126,600 psi. Even if the PBR has a sharp edge at the impact point, local permanent deformation occurs in the TPU to a depth of only 0.016 inches.

5.5.5 Fasteners

The bolt and screw loads are determined by summing the appropriate surface loads shown in Figure 19.

Load A

Eleven 1/2 - 20 15-5 PH bolts are used. Load bolts are normally preloaded to 80% of yield to eliminate cyclic loading, which corresponds in this case to 204,000 pounds. This is 89% greater than the applied load of 108,195 pounds.

Load B

These bolts are sized to be weaker than the connections at A, C or H so that, in the event the packer binds when the TPU is pulled from the well and failure occurs, it will occur at B.

Nine 1/2 - 20 15-5 PH bolts are used, giving 82% of the strength of the bolts at A. The bolts at A and B will be machined from the same stock, and specimens will be tested to failure to ensure acceptably uniform strength.

An 80% preload of 166,940 pounds, less the applied load of 92,960 pounds, gives 73,980 pounds required by the packer to reverse the bolt load. A packer load of 130,100 is required to fail the bolts. Initial laboratory tests on the first packer design indicate the packer friction in an unpolished pipe is approximately 10,000 pounds.

Load C

Eighteen 1/2 - 20 15-5 PH bolts are used, which are stronger than the flange.

Therefore the bolts will be preloaded to 80% of the flange strength. The flange design strength is 165,000 pounds and its ultimate strength is 282,000 pounds.

Load D

The turbine thrust is determined by computer analysis, and is due to the pressure differential across the rotor of approximately 25 psi. The thrust is 1500 pounds.

Load E

Pump-out vanes on the back of the pump impeller are designed to balance the combined impeller and turbine thrusts so that the net applied load on the shaft bearings is zero at design conditions. Therefore, at design conditions the net pump thrust balances the turbine thrust and is 1500 pounds. However, the maximum stress in the TPU is at zero speed when the pump thrust is zero; this is the value shown in Table VII.

Load F

Six 10-24 cap screws are used, resulting in a load of 250 pounds per screw and a safety factor of 7.8.

Load G

Ten 10-24 cap screws are used, resulting in a load of 1530 pounds per screw and a safety factor of 1.3.

Load H

The maximum pipe string load on the TPU threaded coupling was shown in Section 5.5.4 to be 110,300 pounds compression. The maximum tensile load on the coupling would be the failure load of the load bolts at B; i.e., 130,100 pounds. This results in a maximum shear stress on the threads of 2000 psi.

5.5.6 Main Flange

The flange was analyzed per the ASME Boiler Code. For a safety factor of 1.6 relative to the yield strength, the flange can safely carry a tensile load of 165,000 pounds, which is 27% greater than the load required to fail the fuse bolts at B.

REFERENCES

- (1) Barber-Nichols' Memorandum, "GOP TPU Design", dated May 29, 1979.

GEOTHERMAL PROGRAMS
WORK & INFORMATION RELEASE

DATE: December 29, 1980

SRC FILE NO. 03.04.111 SUPERSEDES _____

DOCUMENT NO. _____ SUPERSEDES _____

	INFORMATION	ACTION REQ'D	ACKNOWLEDGE RECEIPT	ACKNOWLEDGEM'T DATE
TO: RGI				
B/N				
MUSTANG	✓			
SPERRY-V				
SRC				

SRC DISTRI'BEN: ✓ W. McBee (Central File), ✓ H. Matthews, ✓ K. Hules, C. Harvey, R. Mara,
 ✓ W. Kent, B. Toekes File, H. Fettig, C. McHale, D. Gordon,
 ✓ Test Site

DESCRIPTION:

PUMP SYSTEM PERFORMANCE & DOWNWELL ORIFICE SIZING

Memo from B/N, dated 12/16/80

 Barna Toekes, SRC
 System Engineering Manager
 Geothermal Programs

COMMENTS BY ADDRESSEE:

Change in ☐ SCOPE, ☐ COST, ☐ SCHEDULE

OTHER COMMENTS

MEMO

December 16, 1980

TO: Barna Toekes, Sperry Research Center

FROM: R. W. Langley *RWL*

SUBJECT: 1) Pump System Off-Design Performance
2) Downwell Orifices for R-114 Feed Line

cc: Claude Harvey, SRC
Karst Postma, MEC

Attachment 1 is a summary of an analysis of the off-design performance of the pump system that predicts the brine pressure at the wellhead as a function brine and R-114 flow rates and TPU speed.

Attachment 2 gives the sizes of the orifice plates that restrict the R-114 feed to maintain a liquid-filled downcomer. Table 1 gives the orifice plate dimensions for a corresponding range of R-114 flow rates and feed pump pressures. The orifices, to be fabricated by Mustang Engineering, were sized to be housed in the same configuration as the body of the Vickers pressure control valve that the orifices will replace.

ATTACHMENT 1

WELLHEAD BRINE PRESSURE ANALYSIS

SUMMARY

Wellhead brine pressure was predicted for the Sperry stand-alone pump system as a function of R-114 flow rate and brine flow rate. The results are presented in Figure 1.

DISCUSSION

Throughout the analysis, a 70°F wetbulb temperature and a 119°F surface exhaust stream temperature were assumed. Also, assuming an oversized heat exchanger, the turbine inlet temperature T_o was held constant at 330°F for all flow conditions. Turbine inlet pressure P_o was varied from 250 to 650 psia. Additional assumptions and the analytical procedure, as outlined in Figure 2, are discussed below.

A turbine inlet pressure P_o was specified which, assuming a 330°F turbine inlet temperature and using a choked flow model, established an R-114 flow rate $\dot{m}_{R-114}$. Figure 3 summarizes predicted R-114 flow rate as a function of P_o .

Knowing $\dot{m}_{R-114}$ and T_{wb} , the condensing temperature T_c was determined by using the predicted performance for one Baltimore condenser shown in Figure 4. Assuming saturated condenser inlet and exit conditions and a 70°F T_{wb} , the heat load divided by the latent heat at T_c determines the condenser R-114 flow rate. Since two condensers are required, the total R-114 flow rate is equal to twice the flow rate through one condenser. Thus the relationship between the total R-114 flow rate and the condensing temperature determined from predicted condenser performance establishes the condensing temperature. The condensing pressure is then equal to the saturation pressure at T_c .

Condensing pressure P_c , surface equipment pressure drop $\Delta P_{s.e.}$, and exhaust column static head ΔP_g determines the turbine exhaust pressure. The frictional pressure drop in the exhaust stream was neglected because a preliminary analysis showed it was small relative to the static pressure. The surface equipment pressure drop was assumed to be proportional to the R-114 flow rate squared and inversely proportional to the average density. At a flow rate of 38 lbm/sec and an average density of 1.377 lbm/sec, ΔP_{se} was assumed to be 6.8 psid. The static head was determined using ideal gas relations and assuming an isentropic process. The equations used and resulting turbine exhaust pressures are summarized in Figure 5.

Knowing P_e and specifying a turbine speed from a range of 6,000 to 12,000 rpm, the isentropic head $\Delta h'$ and the velocity ratio U/C_o were evaluated. The turbine efficiency η_t was then obtained from Barber-Nichols turbine performance data of 9/23/80 summarized in Figure 6 and the turbine horsepower was calculated. Figure 7 shows the predicted turbine power as functions of speed and inlet pressure.

The turbine horsepower minus mechanical losses determines pump horsepower. Mechanical losses were assumed to be proportional to the speed squared and were 15 hp at 10,500 rpm. Knowing the pump horsepower and speed, the pressure rise through the pump and the brine flow rate were obtained using the pump performance curves shown in Figure 8 and pump similarity laws. These performance curves were developed using Barber-Nichols pump performance data of 10/3/80, see Figure 9.

Finally, the wellhead pressure P_{wh} was determined by summing the shut-in wellhead pressure, assumed to be 130 psia, and the pump pressure rise and subtracting the drawdown for a PI of 2.4 gpm/psi and the friction drop ΔP flow. It was assumed that the friction drop was proportional to the brine flow rate squared and at 1050 gpm its value was 25 psid.

The results of these calculations are summarized in Figure 1.

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

SUBJECT

BY

C.M.

DATE

11/17/80

FIGURE 1

BARBER-NICHOLS ENGINEERING
11/17/80 C.M.

Specify inlet brine temperature: $T_{w,0}$

Specify turbine inlet conditions: T_0, P_0

Choked Flow

from flowrate, $\dot{w}_{2,IN}$

$$\dot{w}_{2,IN} = \dot{w}_c = 2.0$$

$$\dot{w}_c = \frac{Q_{CONDENSER}}{\Delta T_c}; Q_{COND} = f(T_c, T_{w,0})$$

T_c

$$P_{EXHAUST} = P_c + \Delta P_{S.E.} + \Delta P_{STATIC}$$

$$P_c = \text{CONDENSING PRESSURE} = f(T_c)$$

$$\Delta P_{S.E.} = \text{SURFACE EQUIPMENT} \propto \frac{\dot{w}_{2,IN}^2}{E}$$

$$\Delta P_{STATIC} = \left(\frac{\Delta h}{T_c} + 1 \right)^{\frac{1}{\gamma-1}} P_1 - P_1$$

$$P_1 = P_c + \Delta P_{S.E.}; T_1 = 119^\circ\text{F (ASSUMED CONSTANT)}$$

$P_{EXHAUST}$

$\Delta h'$

Specify RPM

$$U = \frac{\pi \times 7.054 \times \text{RPM}}{60}; C = 229.8 \sqrt{\Delta h'}$$

η_T

$$HP_T = \dot{w}_{2,IN} \Delta h' \eta_T$$

HP_T

$$HP_P = HP_T - 15 \left(\frac{\text{RPM}}{10,500} \right)^2$$

HP_P

$\dot{w}_{2,OUT}, \Delta P_{PUMP}$

$$P_{SURFACE} = P_{SURFACE \text{ SHUT IN}} - (GPM)(P.I.) + \Delta P_{PUMP} - \Delta P_{FLOW}$$

$P_{SURFACE}$

$$P_{SURFACE \text{ SHUT IN}} = 130 \text{ PSIG}$$

$$P.I. = \frac{1}{2.4} \left(\frac{GPM}{1050} \right)^2$$

$$\Delta P_{FLOW} = 25 \left(\frac{GPM}{1050} \right)^2$$

GOR = 0.51

OFF-DESIGN

SURFACE WELLHEAD PRESSURE VS BRINE FLOWRATE

8/11/80 (C.M.)

FIGURE 2

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

GCP-651

BY

C.M.

DATE

10/28/80

SUBJECT

COP STAND ALONE
W_{21W} VS P₀

FIGURE 3

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

GOP-651

BY

C.M.

DATE

7/16/80

SUBJECT

CONDENSER TEMP VS HEAT LOAD
CONSTANT WET BULB TEMP LINES

$$Q = Q_c \frac{h_{sat}(T_c) - h_{sat}(T_{wb})}{h_{sat}(90) - h_{sat}(40)} = 0.003 \times 10^6 [h_{sat}(T_c) - h_{sat}(T_{wb})]$$

FIGURE 4

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

GDF-GSI

BY

3 m

DATE

9/19/83

SUBJECT

TURBINE EXHAUST PRESSURE
VS TURBINE FLOW RATE

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

60P-651

BY C.N.

DATE

11/28/80

SUBJECT

~~GEAR STAND ALONE~~
~~WATER PUMP SERVICE~~
PRESSURE

TURBINE OUTPUT (HP)

FIGURE 7

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

601-651

BY

C.R.

DATE

11/13/80

SUBJECT

PUMP PERFORMANCE
DATA

$QPM = 1,000$

$$Q_p = 0.50 \left(\frac{1,000}{10,500} \right) = 400$$

ATTACHMENT 2

ORIFICE PLATES FOR THE SPERRY STAND-ALONE PUMP SYSTEM

December 12, 1980

SUMMARY:

To maintain an all-liquid downcomer for the Sperry stand-alone pump system, a sequential set of thin-plate, square-edged orifice plates was sized. Seven orifice plates are required to cover an R-114 flow range of 15 to 60 lbm/sec. The operating ranges of consecutive orifices overlap by 5 percent. The size and operating range of each orifice is presented in Table 1.

DISCUSSION:

The useful operating range of each orifice is limited by the available surface pump pressure. The pump pressure required for a given flow rate and orifice size is determined by

$$\Delta P_P = P_O + \Delta P_{ORF} + \Delta P_{fHXR} + \Delta P_{fFP} - \Delta P_{gFP} - \Delta P_{gHXR} - P_S$$

where:

ΔP_P = pressure rise in surface pump, psid

P_O = turbine inlet pressure, psia

ΔP_{ORF} = pressure drop across orifice, psid

ΔP_{fHXR} = friction pressure drop in heat exchanger, psid

ΔP_{fFP} = friction pressure drop in feed pump, psid

ΔP_{gHXR} = gravity pressure rise in heat exchanger

ΔP_{gFP} = gravity pressure rise in feed pipe

P_S = inlet pump pressure.

The orifice pressure drop ΔP_{ORF} is determined by

$$\Delta P_{ORF} = \frac{G^2}{2 g_c \rho K^2 A_o^2 144}$$

where:

- $\dot{m}$ = R-114 flow rate, lbm/sec
 g_c = conversion factor, 32.17 lbm/sec
 ρ = upstream orifice density, lbm/Ft³
 k = velocity coefficient; = CF, ratio
 c = discharge coefficients, ratio
 f = velocity of approach factor; = $\frac{1}{\sqrt{1-\beta^4}}$, ratio
 β = ratio of diameters; = $\frac{d}{D}$, ratio
 d = orifice diameter, in.
 D = pipe diameter, in.
 A_o = orifice area, Ft²

Assuming insufficient length after the orifice to reestablish a fully developed velocity profile, a pressure recovery factor was not included in the orifice pressure drop equation. Discharge coefficients C were obtained from ASME's Fluid Meters: Their Theory and Application, Part II.

The calculations were made assuming the following conditions:

- 1) Constant turbine inlet temperature of 330°F;
- 2) 40 psia pump inlet pressure corresponding to a 90°F condensing temperature;
- 3) Constant gravity pressure rise of 992 and 18 psid in feed pipe and heat exchanger respectively; and
- 4) Constant upstream orifice density of 80.2 lbm/Ft³.

A summary of the results is presented in Table 1.

TABLE 1

ORIFICE	1	2	3	4	5	6	7
$\dot{m}$ (lb _m /sec)	15.0 - 18.5	17.5 - 21.5	20.5 - 25.5	24.0 - 30.5	29 - 37.5	35.5 - 47.0	44.5 - 60
d (in.)	0.440	0.482	0.531	0.587	0.665	0.763	0.895
β	0.293	0.321	0.354	0.391	0.443	0.509	0.597
K	0.6000	0.6019	0.6047	0.6084	0.6153	0.6265	0.6507
ΔP_{port} (psid)	754 - 1146	708 - 1068	654 - 1011	593 - 957	514 - 859	428 - 750	329 - 591
ΔP_{pump} (psid)	2.6 - 4.0	3.6 - 5.5	5.0 - 7.5	7.0 - 11.0	10.0 - 16.5	15.0 - 26.0	23.5 - 42.5
ΔP_{eff} (psid)	6.0 - 8.5	7.7 - 11.0	10.0 - 15.0	13.0 - 20.5	18.5 - 29.5	27.0 - 44.0	42.5 - 77
P_b (psia)	290 - 335	335 - 400	385 - 460	440 - 525	510 - 595	580 - 670	655 - 735
ΔP_{pump} (psid)	2.2 - 4.64	4.1 - 4.35	3.5 - 4.44	2.5 - 4.63	2.1 - 4.50	0.0 - 4.40	0.4 - 3.98

APPENDIX E

TURBINE-PUMP UNIT

TEST PROGRAMS

BARBER-NICHOLS

ENGINEERING

GEOTHERMAL ORGANIC PUMP
COMPONENT TESTING

AUGUST 19, 1981

PREPARED FOR:
SPERRY RESEARCH CENTER
100 North Road
Sudbury, MA 01776

1.0 INTRODUCTION

The high reliability requirement of the stand-alone Geothermal Organic Pump (GOP) system mandates that thorough laboratory testing of components be conducted. High confidence in the Turbo-Pump-Unit (TPU) performance must be established before it is subjected to downwell service, where there is no access for maintenance or repair.

A complete TPU test program was conducted at Barber-Nichols' facility from August 1979 to March 1981. The first of three phases of the program involved assembling critical subcomponents into special fixtures to test performance and to obtain data needed to finalize the designs. By testing separately, each subcomponent could be fully instrumented to obtain the data needed to map that component's performance. Hardware modification, if necessary, was simplified due to ease of access to components when they were tested separately. Tests were conducted on the following TPU subassemblies:

- a) Ball bearings and lube system
- b) Centrifugal pump section
- c) Turbine section
- d) Shaft seals

The second test phase involved performance mapping of the fully assembled TPU. A Rankine cycle facility was constructed at the Barber-Nichols plant for simulated downwell testing.

The final phase of TPU testing was a series of endurance runs designed to shake down the unit and to establish confidence in the system.

During commencement of the TPU test program another series of tests were conducted on a few other downwell components of the stand-alone system. A series of simulated well tests were conducted on the packer cups (which form the TPU-to-well seal). Also tested were the 4.5 inch sliding seal and the exhaust transition seal. The evaporative condensers which are to be used in the stand-alone system were tested to verify manufacturer's data.

Included in the report is a description of a test for two-phase flow pressure drop in an exhaust riser.

2.0 BALL BEARING TEST

Bearing longevity is a crucial element of trouble-free down-well TPU operation. Maximizing the bearing life by experimentally optimizing bearing preload was the overlying goal of the bearing test program.

The fatigue life of a rolling element bearing varies inversely with the cube of the axial load, which is comprised of the applied load and the preload. Thus, the amount of preload on a bearing should be minimized in order to obtain maximum life. There exists a minimum preload, however, below which a "skidding" condition occurs due to loss of rolling contact between the balls and the raceways. Skidding is reported to reduce drastically the life of a bearing. For optimum bearing life, preload should be set at the lowest level that insures adequate thrust for non-skidding operating conditions. The bearing test program was carried out to define experimentally the ball bearing skidding behavior unique to the TPU system and, thus, establish confidence in a safe minimum preload.

The test plan was to operate the bearings first at 80% of design speed, taking all data in increments ranging from high preload to very low preload, thus mapping a curve of retainer/race speed ratio versus preload. This procedure was then repeated for design speed and 20% over design speed.

2.1 BALL BEARING TEST APPARATUS

The test apparatus was designed to operate TPU bearings in an environment identical to that of the TPU, with the ability to vary preload while monitoring important parameters. Figure 2.1 is a layout of the ball bearing test rig, and Figure 2.2 is a layout of the test stand. The rig was designed to accept swing-pad hydrodynamic bearings as well as ball bearings.

2.2 BALL BEARING TEST INSTRUMENTATION

Preload thrust to the test bearing was supplied and measured by a strain gauge load bolt which exhibits varying electrical resistance proportional to the quantity of tension in its shank. Shaft rotational speed was measured with a magnetic pickup device. Rotational speed of the bearing retainer was measured with a proximity probe. The probe generates a magnetic field and reads interruptions in that field created by eleven steel screws set in the bronze retainer. Skidding

FIGURE 2.2
BALL BEARING TEST STAND

was detected by monitoring changes in the ratio between the retainer speed and the inner race (shaft) speed. Other parameters measured were: oil temperature, race temperature, oil flow rate, oil inlet and outlet pressures, and shaft torque. These data provided additional information such as oil circulation pump performance and bearing power consumption.

2.3 BALL BEARING TEST RESULTS

The results of the skid test program are presented in Figure 2.3. Using the three skidding curves obtained, an empirical correlation between minimum safe preload and shaft speed was calculated to be:

$$\text{Minimum preload (lbs)} = 4.355 (10^{-6}) \times \text{speed}^2 \text{ (rpm)}$$

The above equation is very slightly arbitrary in the sense that there is no distinct knee in the skidding versus preload curve (Figure 2.3) to assign a value where "skidding occurs". The stars indicate where the minimum safe preloads calculated by the above equation occur on the curves. These points lie near the maximum value of speed ratio for each curve.

It should be noted on the curves in Figure 2.3 that the speed ratio increases with decreasing preload before it starts dropping. We believe this increase is due to slight straining of the bearing materials under high thrust loads which increases the contact angle. When thrust is reduced, the contact angle reduces and the speed ratio increases slightly.

Other correlations were obtained during testing. Figure 2.4 shows curves of retainer/race speed ratio versus hydraulic pressure of lubricating oil. It is seen from these curves that there is no significant change in skidding characteristics due to variations in pressure. The vertical displacement between the curves is due to differing bearing temperatures from test to test. After the oil pressure test above was completed, a test was conducted to determine the effect of temperature on the retainer/race speed ratio during no-skid conditions. It was found that this ratio increases with bearing temperature, probably due to changes in bearing contact angle with thermal growth. This behavior explains the non-closure of the curves in Figure 2.3, since the test bearing always heated up during the course of a test run.

Before the data presented in this paper was generated, certain problems were encountered and corrected. For example, during one series of tests skidding rates were greatly affected by oil pressure. This test was halted by a failed shaft seal. When the seal was replaced and the unit was back on line, testing showed no sensitivity of the skidding rates to oil pressure. It is our opinion that poor de-aeration of the oil was the cause of the sensitivity to pressure. This opinion is based on careful scrutiny of the test unit and an improved method in filling the machine with air-free oil after the seal fix. There has been no testing to date, however, to verify the cause of the pressure sensitivity.

3.0 PUMP TEST

The initial pump testing involved an investigation of the pump's performance characteristics and the thrust loads. Then a series of tests compared the performance of several different impeller configurations to achieve the optimum pump performance and to achieve thrust balance with the turbine. Impeller parameters that were varied during these tests included impeller diameter, impeller clearances, and pump-out vane diameter.

3.1 PUMP TEST APPARATUS

The TPU pump section components were designed into a test assembly as shown in Figure 3.1. The input power to the impeller was provided by a gasoline engine coupled to the pump through a step-up gearbox. The design input power to the pump is 485 Hp at 10,500 rpm. However, since pump performance is correlated accurately over a wide range of conditions (by the dimensionless quantities of head coefficient and flow ratio), the available engine power of 150 Hp was sufficient to evaluate pump performance.

The pump end of the assembly mounted into a flow loop designed for it and for the TPU test. The flow loop (Figure 3.2) was designed to simulate well conditions. Water in the loop entered the pump suction at 350°F and 400 psig. At 10,500 rpm, the design pump flow rate is 1000 gpm, and the head rise is 1390 ft (600 psid). Downstream of the pump discharge the flow was regulated by a hand-operated flow control valve, metered with an ASME orifice, and returned to the pump suction. Water temperature was maintained by bypassing some of the flow to a 300 ton cooling tower. The accumulator tank was half full of water and half full of pressurized nitrogen, which was

FIGURE 3.1
PUMP TEST ASSEMBLY

FIGURE 3.2

**GOP-TPU
STAND ALONE PUMP
TEST UNIT
FLOW LOOP**

Barber-Nichols Engineering
SEPT. 20, 1980
GKR

used for controlling the pump inlet pressure and providing expansion room for the heated water.

3.2 PUMP TEST INSTRUMENTATION

The pump test was instrumented to monitor pump inlet pressure, exit pressure, exit temperature, water flow rate, output speed and torque of the gasoline engine, and the axial thrust developed by the impeller. All pressure measurements were made with Bourdon tube pressure gauges calibrated using instruments traceable to the National Bureau of Standards. Temperatures were measured with immersion-type bimetallic thermocouples. Water flow rate was measured as pressure drop across a square-edged orifice constructed and installed according to the ASME Power Piping Code. Shaft speed was monitored with a magnetic pickup device. A Stuska hydraulic torque meter measured the reaction force of the trunion-mounted gearbox housing for power calculations. The axial thrust load was measured using three strain gauge-type load cells calibrated statically in dead weight tests.

3.3 PUMP TEST RESULTS

Data for the selected pump design as tested separately and as part of the TPU is presented in Table 3.1. Comparison testing of several impeller diameters, pump-out vane configurations, and impeller clearances led to the selected pump impeller design. The selected impeller is 6.72 inches in diameter. Design pump impeller front clearance is 0.015 inch. Figure 3.3 shows performance versus flow rate.

Table 3.1 offers a point-by-point comparison of the predicted and measured pump performance. Since neither the pump test assembly nor the TPU test assembly were capable of producing the design turbine output power of 525 Hp, pump performance was reduced to a dimensionless quantity called flow ratio, which is the measured flow rate divided by measured speed over the ratio of design flow rate to design speed. Data presented in Table 3.1 is for the design flow ratio of 1.00. An examination of the performance parameters yields the following results.

1. Input head coefficient, ψ_{in} (defined in Figure 5.1) is slightly greater than predicted for both the individually tested pump and the pump tested in the TPU. The biggest part of this difference is due to impeller diameter, since ψ_{in} varies inversely with diameter squared. Using an impeller diameter of 6.72 inches yields a ψ_{in} of 0.717 versus the predicted 0.661 with a 7.00 inch

COMPARISON OF DESIGN PUMP SPECIFICATIONS AND PARAMETERS TO MEASURED HARDWARE PERFORMANCE

DESIGN SPECIFICATIONS

BRINE FLOW RATE	1050 GPM
MAX. HEAD RISE	1600 FT.
EFFICIENCY GOAL	75%
DESIGN SPEED	10,500 RPM

PARAMETERS

	<u>PREDICTED PERFORMANCE</u>	<u>AS TESTED INDIVIDUALLY</u>	<u>AS TESTED IN THE TPU</u>
IMPELLER DIAM.	7.00 IN	6.72 IN	6.72 IN
EYE DIAMETER	3.50 IN	3.50 IN	3.50 IN
HUE DIAMETER	1.25 IN	1.25 IN	1.25 IN
SPEED	10,500 RPM	6,000 RPM	9,000 RPM
SUCT. SPECIFIC SPEED (GPM UNITS)	2334	2330	N.A.
NPSH	767 FT	292 FT = MIN. MEASURED	
INPUT PUMP POWER	485 HP	104 HP	325 HP
FLOW RATE	1050 GPM	600 GPM	900 GPM
HEAD RISE	1600 FT	520 FT	1135 FT
HEAD COEF. η_{OUT}	0.502	0.540	0.525
THRUST VANE LOSS	21 HP	N.A.	N.A.
EFFICIENCY	76%	76.4%	72.0%
HEAD COEF. η_{IN}	0.661	0.707	0.729

NOTE: DATA PRESENTED FOR COMPARISON AT A DESIGN
FLOW RATIO (Q/Q_D) OF 1.00.

MEASURED PUMP PERFORMANCE DATA (INDIVIDUAL TEST DATA)

FIGURE 3.3

PUMP PARAMETERS:

IMP. DIAM: 6.72 IN.
FRONT CLEAR: 0.015 IN.
DESIGN FLOW (Q_D): 1050 gpm
DESIGN SPEED (N_D): 17500 rpm

NOTE: MEASURED SPEED
5000 - 6000 rpm

HEAD COEFFICIENT & EFFICIENCY

1.40
1.20
1.00
0.80
0.60
0.40
0.20
0

$$\eta_{out} = \frac{\Delta H (ft) \rho g Q}{P_r}$$

$$\eta_{in} = \frac{H_{PUMP} (550) \rho g Q}{G_{HP}} ; \text{ INPUT HORSEPOWER MEASURED DIRECTLY }$$

$$\eta_p = \eta_{out} / \eta_{in}$$

0.20 0.40 0.60 0.80 1.00

FLOW RATIO Q/Q_D

Birben-Michale
10/3/80 GMR

impeller diameter.

2. Output head coefficient, ψ_{out} , (defined in Figure 5.1) is also greater than predicted. Again, most of this difference is due to the change in impeller diameter. Head coefficient is also inversely proportional to impeller diameter. Reflecting this, the "predicted" ψ_{out} should be 0.545 instead of 0.502.

3. Pump efficiency, η_p , is defined as the ratio of output head coefficient to input head coefficient. The difference between the predicted efficiency and the efficiency measured in testing is not directly affected by the change in pump diameter. However, the efficiency of the pump as tested separately is slightly greater than predicted because the input head coefficient is lower, resulting in a greater mass flow rate for a given input horsepower. The lower efficiency measured in TPU testing reflects an uncertainty in fluid density in the pump loop due to an unknown amount of entrained air in the system. The result is a lowering of the real fluid density as compared with the saturation density of water at measured pump inlet temperatures. Because of this uncertainty, downwell pump performance is expected to be slightly better than the measured performance.

4.0 TURBINE TEST

Separate turbine testing concentrated on mapping the turbine performance over a wide range of operating conditions. There were no major modifications to the hardware as a result of these tests. As in the pump tests, turbine performance is correlated in terms of dimensionless parameters.

4.1 TURBINE TEST APPARATUS

The turbine test assembly was designed to approximate closely actual TPU operating conditions while allowing full instrumentation and providing access to the turbine components for any necessary modifications. A layout of this assembly is shown in Figure 4.1. The turbine wheel was mounted on a common shaft with a water brake dynamometer shown on the left side of the figure. The TPU nozzle block and turbine housing were incorporated such that the turbine section details of the test assembly were identical to those of the TPU.

A binary Rankine cycle turbine test facility was constructed at the Barber-Nichols plant to provide R-114 vapor for the turbine.

FIGURE 4.1
TURBINE TEST ASSEMBLY

Figure 4.2 shows a schematic of the facility. The water side of the system starts from a holding tank from which the water is pumped to an oil-fired boiler. Saturated steam from the boiler flows through a flow control valve and into the main R-114 heat exchangers. The steam condensate exits these heaters as saturated liquid, then passes through a liquid level control tank and into a series of heat exchangers that cool the water and preheat the R-114. The liquid level control tank assures that the heat exchangers are full of water in order for subcooling to take place. The water returns to the holding tank from the subcoolers.

On the R-114 side of the system, gravity feeds the refrigerant from the hotwell (R-114 reservoir) into two centrifugal pumps which boost its pressure up to 1200 psig and provide a maximum flow of 200 gpm. Beyond the pump discharge, flow is controlled by a hand-operated plug valve and metered through an ASME orifice. Then the R-114 flows through the preheating heat exchangers (steam condensate subcoolers) and then through the main heat exchangers. The R-114 enters the turbine inlet (design temperature of 330°F and pressure of 450 psia), expands across the turbine, and then is condensed in an evaporative condenser. Liquid (90°F) R-114 returns from the condenser to the hotwell.

Control of the turbine inlet temperature is achieved by adjusting the pressure of steam entering the main heat exchangers. The condensing pressure in the heat exchangers uniquely determines the condensing temperature of the steam, which in turn is related to the R-114 temperature leaving the heat exchangers. Turbine speed is controlled via adjustable water brake drag. Turbine inlet pressure and flow rate are controlled with a hand valve (HCV10).

4.2 TURBINE TEST INSTRUMENTATION

The turbine test instrumentation measures turbine inlet fluid conditions (temperature, pressure and flow rate), turbine exit fluid conditions (temperature and pressures), turbine shaft speed, torque, and rotor axial thrust loads. The instruments are similar to those used on the pump test apparatus. The trunion-mounted water brake housing is instrumented with a hydraulic force gauge to provide a cross-check of turbine output power.

4.3 TURBINE TEST RESULTS

The turbine test results are presented in Table 4.1. The turbine

COMPARISON OF DESIGN TURBINE SPECIFICATIONS AND PARAMETERS TO MEASURED HARDWARE PERFORMANCE.

DESIGN SPECIFICATIONS

WORKING FLUID	R-114
INLET TEMPERATURE	330°F
AMB. WET BULB TEMP.	75°F
SPEED	10,500 RPM
EFF. GOAL	70%
OUTPUT POWER	525 HP
SPECIFIC SPEED (CFS UNITS)	51.3

PARAMETERS

	PREDICTED PERFORM.	AS TESTED INDIVIDUALLY	AS TESTED IN THE TPU
MEAN WHEEL DIAM.	7.884 IN.	7.884 IN.	7.884 IN
UM/C ₀	0.454	0.477	0.475
TURBINE PRESS RATIO	8.9	7.5	7.8
OUTPUT POWER	525 HP	420 HP	420 HP
EFFICIENCY	76%	77%	78%
CONDENSING TEMP	90°F	85°F	85°F
AMB. WET BULB TEMP.	75°F	45°F	45°F

FLOW CONDITIONS

EXHAUST PRESS.	64.2 PSIA	60 PSIA	58 PSIA
EXHAUST TEMP	171°F	170°F	170°F
ISENTROPIC HEAD	12.6 BTU/LB	10.4 BTU/LB	10.5 BTU/LB
R-114 FLOW RATE	38.0 LB/SEC	30 LB/SEC	30 LB/SEC
INLET PRESS.	570 PSIA	450 PSIA	450 PSIA

design was not altered from the original hardware configuration, so the only consideration in this comparison is the difference between measured and predicted performance.

Turbine performance can be compared by using two methods, a comparison of output power at design pressure ratio or a comparison of efficiency at design velocity ratio. Turbine performance is presented as efficiency versus velocity ratio because design pressure ratio could not be achieved in the limits of the test apparatus. Examination of the pertinent parameters results in the following comparison.

Velocity ratio, U_m/C_o , is the ratio of the mean wheel diameter velocity, which is a function of turbine geometry, to the isentropic spouting velocity of the fluid. Since the measured U_m is identical to the predicted, the measured C_o is less than predicted. Isentropic velocity is directly proportional to the square root of isentropic head; therefore, for a given speed the measured isentropic head is less. Turbine efficiency, the ratio of actual turbine head to isentropic head, increases, so the turbine performance is slightly better than predicted. Figure 4.3 shows performance versus speed.

5.0 TPU PERFORMANCE TEST

Performance testing of the TPU spanned a range of operating conditions bounded by the limitations of the test assembly equipment. The common test plan for all runs was to set the turbine fluid inlet conditions (flow rate, temperature and pressure) and vary the speed of the turbine by changing pump flow ratio. The only parameters held during the tests were the pump inlet temperature and pressure and the turbine inlet temperature. These were maintained at 350°F, 400 psig and 330°F, respectively.

Turbine conditions were varied according to the following plan. R-114 vapor flow rate was varied from 8 lbs/sec to 40 lbs/sec. Turbine inlet pressure, limited by the boiler output capacity, ranged from 200 psig to 520 psig. Turbine speeds ranged from a maximum of 11,000 rpm down to 4500 rpm, at various input powers. A range of flow ratios from 0.2 to 1.8 was examined.

5.1 TPU TEST APPARATUS

The same Rankine cycle facility used for the turbine test was used for the TPU test. the facility was designed such that the turbine test assembly could be disconnected from the turbine inlet and exhaust

FIGURE 4.3

MEASURED TURBINE
PERFORMANCE DATA
(INDIVIDUAL TEST DATA)

flanges, and the TPU could be connected in the same place.

The pump flow loop used for the pump test was also used for the TPU test. The TPU was totally immersed in water in a straight section of the loop to match closely the downwell environment.

A lubrication support system was incorporated into the TPU assembly. Oil was pumped out of the TPU by the internal, shaft-mounted centrifugal pump at a rate of 2 to 5 gpm and a pressure of 20 to 40 psid. The oil flowed through an air-cooled heat exchanger which dropped the temperature by 50°F, through a filter and a flow orifice, then back to the TPU.

During early testing, oil pressure was supplied by an air-driven, positive-displacement pump. Pressure was kept 50 psi higher than the pump inlet during testing. A later design change in the TPU allowed for automatic pressurization of the oil by tapping into pump pressure. A reservoir was added which contained half oil (on top) and half water. The oil side was tapped into the oil system and the water side was tapped into a section of the impeller backplate that was 50 psid higher pressure than the pump inlet. Thus the lube oil pressure was automatically controlled to 50 psi higher than the pump inlet pressure. The positive-displacement pump was used only for adding makeup oil.

5.2 TPU TEST INSTRUMENTATION

The test assembly was instrumented to measure turbine output shaft speed, turbine inlet temperature and pressure, turbine exhaust temperature and pressure, and R-114 feed pump exit temperature, pressure and flow rate. The pump/water loop had instrumentation to monitor pump inlet pressure, exit pressure, water temperature, flow rate, and the water level in the accumulator tank. The lube system was instrumented for pressure, temperature, flow rate, and leakage. The hardware used to instrument the TPU test was identical to that used on the pump and turbine tests.

5.3 TPU PERFORMANCE TEST RESULTS

Performance data for TPU testing is presented in Figures 5.1 and 5.2. These curves represent data taken during the last four hours of 368 hours of total running time. This test demonstrated that there was no degradation of performance with time or due to various TPU modifications.

GOP-TTU PUMP HEAD-FLOW
DATA NORMALIZED FOR A 10HP PARASITIC LOAD
AT 9000 RPM.

FIGURE 5.1

$$\eta_{OUT} = \frac{\Delta H (10) \rho Q}{W_P}$$

$$\eta_{IN} = \frac{W_{IN} (550) \rho Q}{W_P}$$

$$\eta = \eta_{OUT} / \eta_{IN}$$

INPUT W_P BASED ON MEASURED TURBINE
OUTPUT W_P AND CALL. PARASITIC
LOSSES PROPORTIONAL TO SPEED

Barber-Nichols
5/27/81 GKA

BARBER-NICHOLS
ENGINEERING CO.

PROJECT
GOF - TPL

BY
GKR

DATE
4/14/61

SUBJECT
TURBINE EFFICIENCY
VERSUS VELOCITY RATIO

FIGURE 5.2

$$U_M = \text{MEAN BLADE VELOCITY} = \frac{\pi D N}{60}$$

$$C_0 = \text{SPOUTING VELOCITY} = \sqrt{2gH}$$

$$H = \text{ISENTROPIC HEAD BTU/LB}$$

$$D = \text{MEAN WHEEL DIAM. (INCHES)} = 7.854 \text{ IN.}$$

$$N = \text{SPEED, RPM}$$

$$J = 778 \text{ FT-LB/BTU}$$

$$g = 32.2 \text{ FT/SEC}^2$$

The performance curve for the pump as tested in the TPU is shown in Figure 5.1. The curve shows input and output head coefficients and pump efficiency plotted versus flow ratio. The maximum efficiency of 0.72 occurs at a flow ratio equal to 1.00 (design flow ratio). Design output head coefficient is 0.525 and design input head coefficient is 0.729. Fig. 5.1, based on TPU losses, agrees closely with Fig. 3.3.

The data plot can be used either to compare or to predict pump field performance, depending on how much data is available. If only pump speed and flow rate are known, all other pump performance parameters can be calculated using the plotted data by the method given in the following example.

Example 1 Given speed, N_m , rpm, and flow rate, Q_m , gpm, find flow ratio = QN_D/NQ_D where $N_D = 10,500$ rpm, $Q_D = 1050$ rpm.

1. Graphically determine input and output head coefficient using flow ratio.
2. Knowing output head coefficient, find pump head rise in feet: $H = \psi_{out} \times N^2 \times 2.67 \times 10^{-5}$
3. Knowing input head coefficient, calculate input horsepower required: $Hp_{in} = \psi_{in} N^2 Q \rho \times 1.08 \times 10^{-10}$; ρ is the brine density at the pump inlet in lb/ft^3 .
4. Knowing input power, the turbine output power can be calculated. Turbine output power is equal to pump input power plus the TPU parasitic losses. Assuming the parasitic losses vary linearly with speed and knowing the measured value of 10hp at 9000 rpm, the equation for turbine output horsepower is:

$$Hp_{out} = Hp_{in} + 10 \times (9000/N)$$

The turbine characteristic curve, presented in Figure 5.2 can be used to verify turbine performance. If only speed and R-114 mass flow rate are known, the curve can be used in conjunction with turbine output power as calculated above to estimate efficiency, as in the following example. Fig. 5.2 agrees closely with Figure 4.3.

Example 2 Given speed, N , and R-114 flow rate, in lb/sec , and knowing turbine output power, find the turbine efficiency.

1. Determine ΔH , across the turbine in Btu/lb:

$$\Delta H = (Hp_{out} \times 550) / (\dot{m} \times 778)$$

2. Estimate isentropic head, $\Delta H' = \Delta H / \eta_T$
3. Calculate mean wheel velocity,

$$U_m = \frac{DN}{720} = 0.034 N$$
4. Using the estimated $\Delta H'$, find isentropic spouting velocity, $C_o = \sqrt{2g_C \Delta H' \times 778} = 223.84 \Delta H'$
5. Graphically determine turbine efficiency using the velocity ratio U_m / C_o .
6. If η_T as estimated does not match the turbine efficiency found above, use the graphically determined η_T and return to step 1 above. Continue iterating on turbine efficiency until satisfactory convergence is achieved, making result consisted with Figures 5.1 and 5.2.

6.0 TPU ENDURANCE TESTING

The TPU endurance tests were designed to identify unforeseen problems and to evaluate reliability during long-term operation. They covered 344 hours of the total 368 hours of test time logged on the TPU. The tests were broken down into two 100-hour endurance runs, two 36-hour seal tests, two 24-hour seal/lube system tests, one 8-hour instrument checkout, and 16 hours of miscellaneous tests.

Several design modifications were made during the test program resulting from anomalies that occurred during operation.

A pump housing diffuser vane failed from fatigue early in the test program. The vanes on this (the first) diffuser housing were welded to the pump housing, and one of the welds partially sheared, leaving the vane to vibrate until failure. The problem was solved when the failed housing was replaced with a production housing in which the vanes are integrally cast in place.

During a physical inspection of the TPU after the first 100 hour run, it was found that the bearing outer races had "walked" around in the bearing housing during operation. This relative motion between race and sleeve left visible marks in the sleeve. If the observed wear is projected over 20,000 hours of operation, it is conceivable that enough clearance could develop to allow detrimental shaft vibration. A design change was therefore made which added tool steel inserts (Rockwell-C50) to the bearing housing where the bearing races are located. Subsequent inspections revealed no signs of wear in this location.

A clogged oil filter caused a low lubrication flow rate which led to an overheat condition in the TPU during one test. The routine teardown inspection which followed this test showed no significant damage. The heat coloration of the steel indicated that the areas around the seal housings were exposed to the highest temperature. As a preventative measure, modifications to the seal housings were made to improve cooling in the event of a similar overheat in service.

Unusual vibration forced a shutdown late in the endurance testing program. Inspection exposed a hairline crack along a fillet near the pump end of the shaft. A stress concentration at the fillet yielded the material with fatigue over 75% of the cross-section of the shaft. A stress analysis was conducted and a new shaft was fabricated with appropriately conservative fillet radii to lower the stress concentrations.

12.0 SHAFT SEAL TESTS

At the conceptual stages of the TPU program, it was intended that either Dyna-Seal or Borg-Warner shaft seals would be designed into the TPU. Before TPU testing began, Dyna-Seals were installed in the bearing test rig to facilitate the bearing tests. The seal cartridges failed under pressure so Dyna-Seals were dropped from further consideration. Although Borg-Warner seals successfully passed a test program at the Sperry Research Center, there were two drawbacks. First, the package size of the cartridge assembly was too large for installing them in tandem in the envelope constraints of the TPU. Secondly, there was low confidence in the reliability of reverse pressure sealing.

With this background, Barber-Nichols entered a program in mid-1980 to design a set of shaft seals for the TPU. The areas that were addressed are as follows:

- a) Materials
- b) Face design
- c) Pressure balancing
- d) Spring loading
- e) Secondary seal design
- f) Cooling

The B/N seals are a radial face design in which the nose is static and is spring-loaded against a mating ring which rotates with the shaft.

The original test approach was to install seal prototypes into the TPU and monitor leakage while conducting other TPU tests. After two incidents of seal failure, a new approach was undertaken in which seals were tested in a fixture that featured quick setup and disassembly and allowed closer observation of the seals during operation. Typically, promising seal designs that were proven in these tests were then installed into the TPU for testing.

12.1 SEAL TEST APPARATUS

Two different devices were used for testing and determining performance of the shaft seal designs. Besides using the TPU as a test bed, the bearing test rig (described in Section 2.1) was set up to test Barber-Nichols' seals. Only single seals could be tested in the bearing test rig. This unit provided pressurized, temperature-controlled oil to the inboard sides of the seals. The outboard sides, however, were exposed to room temperature air. The bearing test rig provided a good test bed for general seal development in which parts could be changed out quickly.

12.2 SEAL TEST INSTRUMENTATION

A number of instrumentation devices were added to the TPU and to the bearing test rig to facilitate data collection of shaft seal parameters. Devices were added to the bearing test rig for collecting oil that has leaked past the seals. This effluent is directed into graduated cylinders for measurement.

A pressure tap was added to the pump back-plate of the TPU for measuring brine pressure outboard of the pump side shaft seal. Bimetal thermocouples were installed in the lubrication lines such that oil temperature could be measured immediately in and out of the TPU housing, thus giving an indication of seal temperature.

Two methods were used to monitor shaft seal leakage in the TPU. Early in the test program, makeup oil was pumped into the TPU from an accumulator with a positive displacement hydraulic pump. By keeping the system pressure constant by adding oil, the oil lost through the seals was equally offset by oil pumped out of the accumulator. Thus, monitoring the accumulator level gave a measurement of oil lost through seals. Later in the program, an oil pressurization system identical to the downwell accumulator was installed. With this system a dual fluid, oil over brine, accumulator automatically kept the oil pressure 50 psi higher than the brine pressure. Oil leakage was indicated by measuring the level change of the line between the oil and the brine through the sightglass in the accumulator.

12.3 SEAL PROGRAM RESULTS

All of the seals tested use silicon carbide for both the seal nose and mating ring. An Invar ring was added to the OD of the mating ring to improve the tensile strength under centrifugal loading.

A mechanical design change was made after an October 1980 failure in which a spacer pinched a mating ring from hydraulic pressure. The change added a snap ring to captivate the spacer.

A number of iterations of secondary seal design were experienced. At first, standard O-ring seals and glands were used. Common materials such as Viton, Buna-N, and ethylene propylene were disregarded as a result of SRC tests showing incompatibility with one or more of the TPU fluids. An early material tested in B/N seals was Kalrez. During a November test, a mating ring fractured due to swelling of the elastomer. Also tried were teflon-coated Viton O-rings. These seals failed when the teflon coating separated from the Viton and extruded out of the gland.

The remainder of secondary seal development centered around spring-energized seals made of fiber-reinforced teflon. Initially Bal-Seals were tried. These performed satisfactorily, but have some drawbacks. First, the reliability is questioned, since manufacturing tolerances are a significant percentage of the dimensions of the small cross-section seals. This is a problem in all spring-energized seals of the same small cross-section (which is dictated by TPU package design constraints). Secondly, the wire spring used by Bal-Seal has a very small footprint on the teflon material, which can "Brinell" into the plastic. During one shaft seal test an overtemperature to 450°F had just that result, i.e. the spring pushed into the plastic. It is possible that a flat ribbon spring may not have yielded the teflon in that case. Another brand, Omni-Seal, which has ribbon springs, was tested.

It was difficult to determine the performance of spring-energized seals because it is difficult to distinguish whether leakage observed in a test is from the primary seal, the secondaries, or both. Some major design requirements, however, were verified which will improve the reliability. One is the surface finish of the gland. It must be below 16 microinches, and preferably under 8. Another requirement is freedom of installation. The seals should not be bent or bowed on installation. A design change was carried through on the TPU shaft seal assembly to eliminate having to bend the secondary seals for installing in ID grooves of the seal nose secondary.

Seal face design and testing covered three basic areas:

1) face width; 2) wear pads; and 3) pressure balancing. Three face widths were tested. They were 0.050 inch, 0.068 inch, and 0.100 inch. Three bearing rig tests of the 0.050 inch, pressure-balanced, face-seal style (December 1980), of 18, 15, and 2 hours each, had dynamic leak rates of 0.6, 0.3, and 1.5 cc/hr, respectively. These seals were also tested in the TPU for 11.5 hours with undetectable leakage. In February, a bearing rig test of over 400 hours was run with a 0.100 inch face seal partially balanced on top, and a 0.068 inch face, unbalanced seal, on the bottom. The average leakage was 1.5 cc/hr on top and 9 cc/hr on the bottom. It is evident from these and other tests that narrower, balanced face designs seal more effectively than wider unbalanced designs. The disadvantage of narrower faces, such as the 0.050 inch design, is that they have higher wear rates than the wider faces, such as the 0.100 inch design. Also, the narrower faces are more susceptible to failure from chipping or other damage to the face.

An early seal design utilized wear pads. When tested, this seal leaked in excess of 100 cc/hr due to hydrodynamic lifting of the pads. Rather than enter a series of development tests to optimize the wear pads, all further design approaches utilized no pads.

The wear rate of non-wear-pad seal faces was brought up to an acceptable level by reducing the preload spring force to 15 pounds, from 25 pounds. Also, a design change made on the TPU to improve oil cooling passages to the seal cavities resulted in an improved seal wear rate, presumably due to lowered operating temperature of the seals. After an 840 hour test run on the bearing test rig, the 0.068 inch face design had less than 10 microinches of wear.

TPU testing during the first four months of 1981 showed erratic shaft seal leakage rates while running, but very low leakage when static. In May it was discovered that the TPU shaft had been slowly failing by fatigue. It was judged that high speed imbalance of the cracked shaft was causing the seal problems.

The defect in the shaft was corrected and an 11-hour TPU test was run with the same seal configurations as had been previously used. The total leakage for both sets of seals was an acceptable 5 cc/hr.

The TPU was shipped to Mustang Engineering Company. All seals in the unit are of the 0.050 inch face design with 15 pound preload springs, except the outer pump-side set, which are the 0.100 inch face design with a 25 pound preload spring. The wider face is used on the outer pump side due to possible abrasion from brine contaminants.

APPENDIX F

GRAVITY HEAD TURBINE-PUMP UNIT
SIZING STUDY

BARBER-NICHOLS **ENGINEERING****MEMORANDUM**

July 30, 1980

TO: K. E. Nichols

FROM: H. Johnson *HL*

SUBJECT: Gravity Head TPU Sizing Study

OBJECTIVE: The objectives of this study were to determine the optimum speed match of the turbine and pump with respect to the combined efficiency and to compare the required turbine and pump diameters to the space available. These were to be done for two operating conditions: with and without brine reinjection pressure (w/ ΔP -REIJ and w/o ΔP -REIJ) supplied by the downhole pump.

APPROACH: A thorough study of commercial "turbine" type pumps was made to determine a reasonable expected efficiency as a function of specific speed. Pumps were chosen as near the design capacity as possible and scaled to the design flow rate. Data from Stepanoff - "Centrifugal and Axial Flow Pumps" was also included to determine the final efficiency curve. An efficiency curve for Francis Turbines as a function of specific speed was constructed by scaling down from data in Daugherty-Franzini - "Fluid Mechanics with Engineering Applications." The Francis type turbine offered the highest potential efficiency at the level of specific speeds examined in the study. Thermodynamic cycle calculations were carried out far enough to determine the properties through the turbine and pump for the calculation of specific speeds. The thermodynamic cycle computer program has been in a state of development so the calculations for this study have been updated from time to time accordingly.

RESULTS: The turbine and pump efficiency curves are shown on Curve Sheet 2. The specific speed has been computed using volume rate of flow in $\text{ft}^3 \cdot \text{s}^{-1}$ in both cases.

The thermodynamic data for the turbine follows:

	<u>w/o ΔP-REIJ</u>		<u>w/ ΔP-REIJ</u>	
$T_{t \text{ in}}$	<u>330.</u> °F	<u>320.</u>	<u>330.</u>	<u>320.</u>
$\dot{m}_{R114}$	493. lbm·s ⁻¹	523.	493.	523.
$P_{t \text{ in}}$	900. lbf·in ⁻²	900.	900.	900.
Δh_t	.598 Btu·lbm ⁻¹	.564	1.176	1.109
$(h_{t \text{ in}} - h_{\text{out}})_s$	.748 Btu·lbm ⁻¹	.705	1.470	1.386
P_{out}	686. lbf·in ⁻²	685.	503.	504.
H_s	582 ft.	549.	1144.	1079.

The pump data follows:

$\dot{V}$	1000 gal·min ⁻¹	1000	1000	1000
H	1270. ft	1270.	2500.	2500.
ρ_{in}	55.6 lbm·ft ⁻³	55.6	55.6	55.6

The efficiency, size, speed data is tabulated below and shown on Curve Sheet 1. Specific Speed $N_s = \frac{N \sqrt{\dot{V}}}{H^{1/4}}$ where N is the rotational speed in rev·min⁻¹, $\dot{V}$ is the turbine discharge or pump inlet volume rate of flow in ft³·s⁻¹, and H is the turbine isentropic head or the pump discharge head in ft.

w/o ΔP -REIJ

<u>N</u>	<u>N_s</u>	<u>η_T</u>	<u>N_{sp}</u>	<u>η_P</u>	<u>$\eta_{T \times P}$</u>
6000	160	.820	42.1	.698	.572
7000	186	.817	49.1	.718	.587
8000	213	.811	56.2	.734	.595
10000	266.	.792	70.2	.757	.600

w/ ΔP -REIJ

6000	107.	.810	25.3	.60?	.49?
7000	125.	.816	29.6	.63?	.51?
8000	143.	.8195	33.8	.663	.543
10000	178.	.8185	42.2	.698	.571

ANALYSIS: The efficiency product $\eta_t \times \eta_p$ w/o ΔP -REIJ peaks at a level of 0.60 at a speed of 10,000 rpm whereas the same speed results in an efficiency product of .57 w/ ΔP -REIJ but is still increasing with increasing speed. This is shown on Curve Sheet 1. The basic reason for the efficiency variation is shown on Curve Sheet 2 where T and P component efficiencies are shown as a function of specific speed N_s with the 10,000 speed points spotted in. The pump requirements force low N_s operation unless the rotational speed is relatively high. If the speed is increased to improve pump efficiency, the turbine specific speed goes to the high end where the turbine efficiency decreases. For these reasons, the speed of 10,000 rpm splits the opposing turbine and pump optimum specific speed effects quite well for the case w/o ΔP -REIJ. There is also considerable incentive to use a speed of 10,000 rpm because the Stand-Alone pump has been developed for that speed and would only require minor trimming to be used for the Gravity Head TPU.

There would be an expected increase in TPU life by using a lower speed. A 40% reduction to 6000 rpm would cost 3 product efficiency points or approximately 5%. This speed however would force the pump to operate at a lower specific speed where the efficiency is lower and not as certain.

The design of the pump w/ ΔP -REIJ would indicate a rotational speed requirement greater than 10,000 rpm in order to obtain a more acceptable product efficiency by raising the pump specific speed. This would require a turbine specific speed which might be approaching too high a level for satisfactory Francis design. Providing the reinjection pressure with the down-hole TPU also has the disadvantage that a major function is located downhole which does not have to be downhole.

TIP DIAMETERS of the rotors are shown on Curve Sheet 1. There is no problem in packaging the TPU to meet the well-hole size limitation until the speed is lowered to the 6000 rpm range for the design w/ ΔP -REIJ.

cc: Jim Lobach
 Bob Langley
 Hugh Matthews, Sperry
 Barna Toekes, Sperry
 Warren McBee, Sperry

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

GOP 656

BY

M

DATE

7-28-80

SUBJECT

Gravity Head TPU
Efficiency Study

7-28-8

(67c)

TPU DIAMETERS & EFFICIENCIES vs SPEED

CURVE SHEET 1

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

GOP 656

BY

RL

DATE

7-28-80

SUBJECT

Gravity Head T.P.
Efficiency Study.

7-28-80

67a

TURBINE & PUMP EFFICIENCIES vs N_s

CURVE SHEET 2

APPENDIX G

MICRO MOTION MASS FLOW METER

MICRO MOTION MASS FLOW METER

General Information

The Micro Motion mass flow meter is a non-intrusive meter designed to measure the mass flow rate of any fluid, liquid, gas or slurry. It measures actual mass flow rate (lbs/min, gms/sec, etc.) independent of density, viscosity, temperature, pressure or other flow parameters.

The sensor is a rigid "U" shaped tube that is magnetically vibrated about an axis passing through the open end of the "U". The "U" tube is excited and forced to oscillate at its natural frequency (approx. 50Hz) depending upon the size of the tube and the density of the material within the tube. Although the natural frequency of the "U" tube changes as the density of the material is varied, the mass flow measurement is not determined by measuring this frequency.

Flow in the sensor tube causes a Coriolis/gyroscopic twist that is measured optically using two photo transistors. The mass flow rate is determined by measuring the "U" tube twist angle on the downward vibration stroke and by comparing this twist angle to that of the upward stroke. In a no-flow condition the twist angle will be constant for both the upward and downward vibration strokes. If flow exists in the sensor tube, the tube will twist in opposite directions during the upward and downward vibration strokes. It is this difference angle that is measured optically and can be shown* to be a linear measurement of mass flow rate. This mass flow rate measurement is unaffected by two-phase flow or by changes in the effective tube diameter due to flow material deposition.

*ASME Paper 77-WA/FM-4 dated August, 1977

The mass flow meter offers the user excellent reliability since there are no electrodes to wear out, no bearings to deteriorate, no critical flow stream geometry and no requirement with respect to upstream/downstream piping.

The meter operates from either 115v AC or 18-30v DC and incorporates a slide switch to change input modes. The output of the meter is provided in either or both of two output formats (1) analog output current or (2) a frequency output proportional to flow rate. The analog output is compatible with any 4-20 (0-20)ma control system and the frequency output has been designed to be compatible with any type of logic.

As an additional output, the meter provides the user with a square wave output signal at the pipe vibration frequency. Should the user desire to monitor the relative density of the flowing fluid, he may do so by monitoring the period of this square wave frequency.

Mass Flow Measurement by Micro Motion

The flow sensor consists of a U-shaped tube which is vibrated at its natural frequency. The angular velocity of the vibrating tube in combination with the mass velocity of the flowing fluid causes the tube to twist. The amount of twist is measured with magnetic position detectors, producing a linear signal which is proportional to the mass flow rate of every particle passing through the U-shaped tube. The output of the meter is linearly proportional to mass flow rate and virtually unaffected by variations in fluid properties.

Physical Characteristics

Remote Flow Indicator D10-RT

The Micro Motion Model D10-RT is a digital flow indicator designed for use with Micro Motion Model C mass flow meters.

It converts the frequency of the meter output to a 4-digit flow rate indication, and a resettable, 7-digit totalizer displays the total flow. This compact and rugged instrument has a NEMA IV enclosure and can be operated on either AC or DC power.

It may be set on a table top, mounted on a wall, or installed in a panel.

General Specifications

Accuracy	±0.4% of rate, ± zero stability*
Fluid Temperature Range	-67°F to 257°F (-55°C to 125°C)
Ambient Temperature Range	-40°F to 185°F (-40°C to 85°C)
Wetted Parts	316L stainless steel
Output Signal	
Analog:	0-20 mA or 4-20 mA, 500 Ω max load 0-5 VDC or 1-5 VDC, 100 kΩ min input
Frequency:	Max output adjustable 3-15,000 Hz Solid state switch, 30 VDC, 1 amp capability
Response Time	Adjustable 0.1 to 1.1 sec
Power	110 VAC, 220 VAC ± 10% or 24 VDC ± 20%, ≈8 watts

*Zero set at process temp. ± 20°F

Detail Specifications

	<i>Flow Range lb/min (kg/min)</i>	<i>Zero Stability lb/min (kg/min)</i>	<i>Max Line Pressure Psi (bars)</i>	<i>Process Connections</i>	<i>Shipping Weight lb (kg)</i>	<i>Case Dimensions in (mm)</i>
Model C6	0-2.2 (0-1.0)	.0002 (.00009)	2000 (138)	¼" FNPT	11 (5)	Models C6, C12 and C25 8.63 (219) L 8.0 (203) W 3.48 (88) H
Model C12	0-13.5 (0-6.1)	.002 (.0009)	2000 (138)	¼" FNPT	11 (5)	
Model C25	0-90 (0-41)	.01 (.0045)	1000 (69)	¼" FNPT	11 (5)	
Model C50	0-440 (0-200)	.05 (.02)	2000 (138)	½" Flange*	53 (24)	Models C50 and C100 19.5 (495) L 15.0 (381) W 6.3 (100) H
Model C100	0-1600 (0-726)	.15 (.07)	1500 (103)	1" Flange*	53 (24)	
Model C150	0-2020 (0-916)	.2 (.09)	1500 (103)	1½" Flange*	175 (79)	Models C150 and C200 31.0 (787) L 21.5 (546) W 8.28 (210) H
Model C200	0-3200 (0-1450)	.3 (.14)	750 (52)	2" Flange*	185 (84)	

*Flanges are ASA 150 lb raised face slip ring

Options

Consult your local representative for information about the following options:

Expanded temperature range
 Remote electronics
 Corrosion resistant tubes
 Abrasion resistant tubes
 Expanded pressure ratings
 Other process fittings
 Intrinsically safe design, C6 - C200
 (UL-Approved Models C12 - C200)

APPENDIX H

DOUBLE TUBESHEET LEAK TESTS

DOUBLE TUBESHEET LEAK TESTS

MUSTANG ENGINEERING COMPANY
13230 CAMBRIDGE STREET
SANTA FE SPRINGS, CALIFORNIA 90670
JULY 25, 1980

DOUBLE TUBESHEET TEST FIXTURES

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ABSTRACT
DOUBLE TUBESHEET SEAL EVALUATION

ROLLED TUBES: Of all the tubes expanded by rolling none leaked when tested by a Halogen Leak Tester or by Hydraulic Testing at a minimum of 1500 psig with cold water before torturing.

WELDED TUBES: No leaks were found.

FLARED TUBES: These tubes were prepared as deliberate leakers. All leaked on Halogen Test.

EPOXY SEALED FLARED TUBES: One test unit epoxy coated and clamped did not leak on Halogen Test, but would not stand hydraulic pressure.

One test unit where epoxy was injected did not leak under repeated temperature cycling from ambient to 350°F and pressuring with hot water to a minimum of 1500 psi.

FURMANITE SEALED TUBES: Three units were sealed with Furmanite injected hot and under pressure (5000 psig).

- (a) One fixture with rolled tubes which had withstood the Halogen and Hydro Test failed on pressure and temperature cycling. Furmanite and steam issued from the injection holes, at at least 3000 psig.
- (b) The second fixture with rolled tubes showed no water (steam) leak at 3000 plus pressure, but Furmanite extruded through injection holes.
- (c) A third fixture (flared tubes) with deliberate leakers were sealed with Furmanite and withstood the pressure and temperature cycle torture test. Furmanite was extruded through the injection holes.

CONCLUSION

Either material epoxy (Scotch-Weld #2214) or Furmanites EPDM (F79E) will seal leaking tubes. Epoxy can be made to surround the tube in the tubesheet more completely than Furmanite, but Furmanite does penetrate the annular space to some degree.

Apparently Furmanite's compound expands more with temperature as it extruded back out the injection holes on heating. These holes would be plugged in the actual exchanger. The compounding of one Furmanite sample was bad (crumbly) and had to be expelled.

Epoxy (Scotch Weld #2214) did not adhere to the tubesheets as advertised, but these sheets were not degreased. The Furmanite did not adhere to the steel.

RECOMMENDATION

Provided proper procedures are used as regards tubesheet spacing, cleanliness, curing and injection either material may be selected. Epoxy has an advantage of lower overall cost. The flexibility and coefficient of expansion of the Furmanite may be of advantage. If Furmanite is used better quality control of the compound is required.

For minimum cost and maximum reliability, standard fit holes and grooves should be used for the tubesheets in the actual exchanger. The tubes should be welded to the tubesheets and then rolled.

DOUBLE TUBESHEET LEAK TESTS

OBJECTIVE

The objective of this test was to determine the best method of attaching the heat exchanger tubes to the tubesheets for the unit to be used in the Geothermal Project. By fabricating small test fixtures, each with a different tube to tubesheet joint, and testing them for leaks, the most suitable configuration could be selected. Another objective was to determine whether Furmanite or epoxy could be used to seal the tubesheet joints if necessary.

APPARATUS

- 1) Test Fixtures (see Appendix A)
- 2) Imperial Eastman System Analyzer - used for charging and pressure testing with R-22.
- 3) Allied Chemical refrigerant R-22, 15#
- 4) TIF Model 5000 Automatic Halogen Leak Detector, 3 PPM sensitivity - features audible beeper to detect R-22 leaks
- 5) Air Driven Sprague Water Pump for pressurizing test fixtures to 1500 psi.
- 6) Chromalox Model Series 3804 temperature controller - temperature range 0-400°F; used with contactor and heater to cure Furmanite and epoxy, also for torture testing.
- 7) Furnas magnetic contactor, catalog #41NB30BF
- 8) Chromalox tubular band heater, Catalog #HBT-75, 240V, 1275W, 7 1/2" barrel diameter
- 9) Furmanite EPDM F79-E Leak Sealing Compound - injected into some of the test fixtures to seal leaks.
- 10) Scotch - Weld Structural Adhesive #2214 - one part epoxy also used to seal leaks in test fixtures.

PROCEDURE

1. Eight (8) test fixtures were prepared according to procedures given in Appendix A.
2. The rolled tube fixtures were pressure tested with R-22 using the system analyzer connected to two fixtures simultaneously. Each unit was tested for a minimum of eight hours. The halogen leak detector was used to check for leaks. The flared tube fixtures (deliberate leakers) were also tested briefly as a check.
3. The rolled tube fixtures were filled with water, pressurized to 1500 psi, and checked for external water leakage or pressure loss.

4. Furmanite compound was injected between the tubesheets of those fixtures which had leaked in the previous tests, except for #6 and #7, which required epoxy. Fixture #2, which had not leaked, was also injected. The injection was done by Furmanite personnel using their own high pressure injection equipment. Before injecting, the test fixtures, Furmanite and injection unit were preheated to 300°F in an electric kiln. The Furmanite was injected at a pressure of about 5000 psi, then cured for one hour at 300°F (Fixture #2 cured overnight at 250°F) using the electric band heaters.
5. Epoxy was injected in fixture #7 using a hand screw type tool, then cured overnight at 250°F using the electric band heaters.
6. Epoxy was applied between the tubesheets of Fixture #6, then clamped together and cured for 20 minutes at 300°F in the electric kiln. No ring was used.
7. All sealed units were retested (R-22 and high pressure hydrostatic) and checked for leaks.
8. Sealed units (2, 4-1, 4-2, 7) were torture tested as follows:

Fixture #2: (a) It was filled with water, pressurized to 1500 psi, then heated to 350°F. The tubes were vibrated by striking with a hammer handle and the unit checked for leaks. It was then allowed to cool to room temperature.

(b) It was heated to 350°F while maintaining a pressure of 1500 psi (bleeding off excess). The tubes were vibrated as in part (a) and the unit checked for leaks. It was then allowed to cool to room temperature.

(c) Part (b) was repeated.

Fixture 4-1 & 7: (a) They were filled with water, heated to 350°F, then pressurized to 1500 psi. The tubes were vibrated by striking with a hammer handle and the units checked for leaks. They were then allowed to cool to room temperature.

(b) They were pressurized to 1500 psi, then heated to 350°F. The tubes were vibrated as in part (a) and the units checked for leaks. They were allowed to cool to room temperature.

Fixture 4-1 & 7 (c) The units were heated to 350°F while maintaining a pressure of 1500 psi (bleeding off excess). The tubes were vibrated as in parts (a) and (b) and the units checked for leaks. They were allowed to cool to room temperature.

Fixture 4-2:

- (a) It was filled with water, pressurized to 1500 psi, then heated to 350°F. The tubes were vibrated by striking with a hammer handle and the unit checked for leaks. It was then allowed to cool.
 - (b) It was heated to 350°F while maintaining a pressure of 1500 psi (bleeding off excess). The tubes were vibrated as in part (a) and the unit checked for leaks. It was then allowed to cool to room temperature.
 - (c) Part (b) was repeated.
9. A comparison was made between the inside diameters of the tubes which were rolled, then welded, and those which were welded, then rolled.
10. The sealed units were cut with a bandsaw between the tubesheets in order to observe penetration of the sealant.

DOUBLE TUBESHEET LEAK TESTS

	<u>BEFORE SEALING</u>		<u>AFTER SEALING</u>				
	<u>FREON</u>	<u>HYDRO</u>	<u>FREON</u>	<u>HYDRO</u>	<u>TORTURE</u>		
	<u>TEST</u>	<u>TEST</u>	<u>RETEST</u>	<u>RETEST</u>	<u>TEST</u>		
					a)	b)	c)
2 (std. fit holes, rolled tubes) ¹	OK	OK	OK	OK	LEAKS*	OK*	OK
3 (close fit holes, rolled tubes)	OK	OK	NONE	NONE		NONE	
4-1 (std. fit & grooved, rolled tubes) ¹	OK	OK	OK	OK	OK	OK*	OK
4-2 (std. fit & grooved, flared tubes) ¹	LEAKS	NONE	OK	LEAKS	OK*	OK*	OK
4-3 (std. fit & grooved, welded & rolled tubes)	OK	OK	NONE	NONE	NONE		
5 (combination holes, rolled tubes)	OK	OK	NONE	NONE	NONE		
6 (std. fit & counter- bored, flared tubes) ²	LEAKS	NONE	OK	LEAKS	NONE		
7 (std. fit, counter- bored & grooved holes, flared tubes) ³	LEAKS	NONE	OK	OK	OK	OK	OK

1: Injected with Furmanite

2: Epoxy Applied Between Tubesheets

3: Injected with Epoxy

*Furmanite Extruded from Filler Holes

DISCUSSION

Epoxy appears to be very effective when injected, although it is unsatisfactory when merely applied between tubesheets and clamped. When injected, it fills gaps and voids, and upon curing becomes a hard, homogeneous solid. However, being inelastic, there is the possibility of it separating from the tubesheets under repeated thermal stress. Inspection of the cut open tubesheets revealed that the epoxy did not adhere well to the uncleaned metal; this problem may be minimized if the tubesheets are thoroughly cleaned prior to injecting the epoxy.

Furmanite has the advantage of being elastic. However, in the tests, some problems were noted. The two fixtures which were subjected to pressures of 3000 psi or more while hot experienced extrusion of the Furmanite from the filler holes. From inspection of the cut open tubesheets, the Furmanite loss did not appear to cause any leaks. However, upon subsequent reheating of those test fixtures at 1500 psi Furmanite continued to extrude. In addition, it was observed that the Furmanite adhered poorly to the tubesheets. This was expected as Furmanite is not intended to be an adhesive.

Either standard or close fit holes will produce a leakproof joint with rolled tubes; however, standard holes pose fewer problems in assembly. Grooving the holes will provide an extra measure of safety and they are easily made. The same applies to welding the ends of the tubes to the tubesheet.

Concerning procedure #9, it was found that for standard fit holes, there was no significant difference in I.D. for tubes that were first rolled, then welded and those that were welded, then rolled. For close fit holes, rolling, then welding produced a significantly larger I.D. than with the reverse procedure. Also, it was observed that welding followed by rolling produced a neater joint.

DOUBLE TUBESHEET ASSEMBLY PROCEDURE

1. Make eight test devices.
2. Use 2" 600# ANSI flanges for tubesheets with seven 1/2" tubes each.
3. Drill six types of holes:
 - a. Standard fit, $0.510 \pm 0.002 - 0.004$ without grooves.
 - b. Close fit, 0.503 ± 0.002 without grooves.
 - c. Standard fit with standard grooves
 - d. Close fit with standard grooves
 - e. Standard fit with counterbore, 0.573" dia. x 0.1875" deep without grooves.
 - f. Standard fit with counterbore and standard grooves.
4. Drill tubesheets as follows:
 - a. One all close fit holes (Tubes to be rolled). Fixture #3
 - b. One all standard fit holes (Tubes to be rolled). Fixture #2
 - c. Three all standard fit and grooved (Tubes to be rolled on one, flared on one, and welded, then rolled on one). Fixture #4-1, 4-2, 4-3
 - d. One with four standard fit holes and three close fit holes, two of each grooved (Tubes to be rolled). Fixture #5
 - e. One all standard fit and counterbored holes (Tubes to be flared, secondary tubesheet to be left loose, epoxy to be applied, and tubesheets to be clamped). Fixture #6
 - f. One all standard fit, counterbored and grooved holes (Tubes to be flared and epoxy to be injected). Fixture #7
5. Prepare 56 tubes, 0.5" O.D. x 0.402" I.D. x 3'0" long. Material spec. SA-214. Plug weld one end shut all tubes. Flare ends of 21 tubes.
6. Roll 28 tubes to 7% - 8% wall reduction with a torque limiting air motor. Measure deformation with dial internal calipers. Determine torque required. Roll additional 7 tubes to the same after welding them to tubesheet. Do not roll flared-end tubes. Test tubes, after rolling, with tubesheet leak tester under full vacuum.
7. Assemble three tubesheets with all flared tubes (see 4.c, 4.e, and 4.f). Assemble all other tubesheets with rolled tubes, but one with welding prior to rolling.

Mustang Engineering Co.

13230 CAMBRIDGE STREET
SANTA FE SPRINGS, CA. 90670

DR'N. SW

REVISIONS

CHK.

NO.

DATE

BY

SCALE HALF

1

2/27/80

C.F.

DATE 2-15-80

2

3/29/80

SW

APP'D.

DOUBLE TUBE -
SHEET TEST
DEVICE

JOB NO.

DWG. NO.

7975

A-1100

DRILL (3) .503 $\pm$.002
DIA. HOLES
SPECIAL CLOSE FIT
GROOVE ONE

DRILL (4) .510 $\pm$.002
DIA. HOLES
STANDARD FIT
GROOVE (2)

(8) REAM $49/64$ " Φ
& TAP $7/8$ " X 9 N.C.

Mustang Engineering Co.

13230 CAMBRIDGE STREET
SANTA FE SPRINGS, CA. 90670

DR'N.	REVISIONS			
CHK.	NO.	DATE	BY	
HALE SCALE	1	3/8/80	SW	
DATE				
APP'D.				

DOUBLE TUBE-
SHEET TEST DEVICE
DRILLING PATTERN

JOB NO.	DWG. NO.
7975	A-1101

NOTES:

1. - ONE REQ'D W/ GROOVES & OUTER RING.
2. - ONE REQ'D W/O GROOVES & OUTER RING.

A-A

Mustang Engineering Co.

13230 CAMBRIDGE STREET
SANTA FE SPRINGS, CA. 90670

DR'N.C.F.

REVISIONS

CHK.

NO.

DATE

BY

SCALE HALF

1

3/24/80

SW

DATE

2/29/80

APP'D.

DOUBLE TUBE-SHEET TEST DEVICE

JOB NO.

DWG. NO.

7975

A-1102

Mustang Engineering Co.

13230 CAMBRIDGE STREET
SANTA FE SPRINGS, CA. 90670

DR'N. C.F.

REVISIONS

CHK.

NO.

DATE

BY

SCALE HALF

1

3/20/80

SW

DATE

3/29/80

APP'D.

DOUBLE TUBE -
SHEET TEST DEVICE
DRILLING PATTERN

JOB NO.

DWG. NO.

7975

A-1103

Furmanite Injected Tubesheet

Furmanite Injected Tubesheet

Furmanite Injected Tubesheet

Epoxy Injected Tubesheet

Freon Test Setup

Hydrostatic Test Setup

Torture Test Setup

HEATER WIRING DIAGRAM

June 19, 1980

FIXTURES

FREON TEST

2

No leaks (Rolled)

10:50 AM

3

No leaks (Rolled)

10:45 AM

4 - 1

No leaks (Rolled)

10:35 AM

4 - 2

Leaks at Tube to Tubesheet Joints
and Injection Holes (Flared Only)

1:30 PM

4 - 3

No leaks (Rolled)

10:40 AM

5

No leaks (Rolled)

1:25 PM

6

Leaks at Tube to Tubesheet Joints
and Injection Holes (Flared)

1:35 PM

7

Leaks at Tubes to Tubesheet Joints
and Injection Holes (Flared)

1:40 PM

FIXTUREHYDRO TEST - 1500 PSI

2	(6/23/80)	No Leaks (15 Minutes)
3	(6/23/80)	No Leaks (15 Minutes)
4 - 1	(6/20/80)	No Leaks (20 Minutes)
4 - 3	(6/23/80)	No Leaks (45 Minutes)
5	(6/24/80)	No Leaks (20 Minutes)

June 20, 1980

FREON TEST

FIXTURE

FIXTURE

TIME 2 (Std. fit holes, rolled tubes) 3 (Close fit holes, rolled tubes)

9:45 AM	No Leaks	No Leaks
10:15 AM	No Leaks	No Leaks
10:45 AM	No Leaks	No Leaks
11:15 AM	No Leaks	No Leaks
11:45 AM	No Leaks	No Leaks
12:45 PM	No Leaks	No Leaks
1:45 PM	No Leaks	No Leaks
2:45 PM	No Leaks	No Leaks
3:45 PM	No Leaks	No Leaks
5:00 PM	No Leaks	No Leaks
8:00 AM (6/23)	No Leaks	No Leaks

June 23, 1980

FREON TEST

FIXTURE

FIXTURE

TIME 4-1 (Std. fit & grooved holes,
rolled tubes)

5 (Comb. holes, rolled tubes)

10:30 AM	No Leaks	No Leaks
11:00 AM	No Leaks	No Leaks
11:30 AM	No Leaks	No Leaks
12:00 PM	No Leaks	No Leaks
1:00 PM	No Leaks	No Leaks
2:00 PM	No Leaks	No Leaks
3:00 PM	No Leaks	No Leaks
4:00 PM	No Leaks	No Leaks
5:00 PM	No Leaks	No Leaks
8:00 AM (6/24)	No Leaks	No Leaks

June 24, 1980

FREON TEST

FIXTURE

TIME

4 - 3 (Std. fit & grooved holes, tubes welded & rolled)

8:15 AM	No Leaks
8:45 AM	No Leaks
9:15 AM	No Leaks
9:45 AM	No Leaks
10:15 AM	No Leaks
10:45 AM	No Leaks
11:15 AM	No Leaks
11:45 AM	No Leaks
1:00 PM	No Leaks
2:00 PM	No Leaks
3:00 PM	No Leaks
4:00 PM	No Leaks
5:00 PM	No Leaks
8:00 AM (6/25)	No Leaks

June 24, 1980

FREON RETEST (EPOXY APPLIED)

FIXTURE

TIME

#6 (Std. fit & counterbored holes, flared tubes)

Tubesheets epoxied together (Scotch Weld 2214)
Cured for 20 Minutes at 300⁰F in electric kiln.

8:30 AM (6/25)	No Leaks
9:00 AM	No Leaks
9:30 AM	No Leaks
10:00 AM	No Leaks
10:30 AM	No Leaks
11:00 AM	No Leaks
11:30 AM	No Leaks
12:00 PM	No Leaks
1:00 PM	No Leaks
2:00 PM	No Leaks
3:00 PM	No Leaks
4:00 PM	No Leaks
5:00 PM	No Leaks
8:00 AM (6/26)	No Leaks

Hydro Retest (Epoxy Applied)

June 26, 1980

800 PSI loss in one minute
Water Leaks at tube to tubesheet joint and between tubesheets.

June 26, 1980

<u>TIME</u>	<u>FIXTURE</u>	<u>FIXTURE</u>
	4-1 (rolled tubes)	4-2 (Std. fit & grooved holes, flared tubes)

FREON RETEST (FURMANITE INJECTED)

5:00 PM	No Leaks	No Leaks
8:15 AM (6/27)	No Leaks	No Leaks

June 27, 1980

HYDRO RETEST (FURMANITE INJECTED)

(6/30)	No Leaks	Water Leaks at Tube to Tubesheet Joint, 100 PSI Drop in 10 Minutes
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June 27, 1980

FREON RETEST (EPOXY INJECTED)

<u>TIME</u>	<u>FIXTURE #7</u>
10:30 AM	No Leaks
11:00 AM	No Leaks
11:30 AM	No Leaks
12:00 PM	No Leaks
8:00 AM (6/30)	No Leaks

June 30, 1980

FREON RETEST (FURMANITE INJECTED)

FIXTURE

<u>TIME</u>	#2 (Std. fit holes, rolled tubes)
2:00 PM	No Leaks
2:30 PM	No Leaks
3:00 PM	No Leaks
3:30 PM	No Leaks
4:00 PM	No Leaks
4:30 PM	No Leaks
5:00 PM	No Leaks
8:00 AM (7/1)	No Leaks

HYDRO RETEST (FURMANITE INJECTED)

FIXTURE #2

(7/1/80)

No Leaks

FIXTURE #7

No Leaks

TORTURE TEST

Fixtures 4-1 & 7 (6/30/80 - 7/1/80)

2 & 4-2 (7/2/80)

- 4-1
 - a) No leaks
 - b) Excessive Pressure (Over 3000 psi) caused some extrusion of Furmanite
 - c) No leaks
- 7:
 - a) No leaks
 - b) No leaks
 - c) No leaks
- 2:
 - a) Excessive pressure (over 3000 psi) caused extrusion of Furmanite from filler holes and some loss of steam
 - b) No leaks - some extrusion of Furmanite
 - c) No leaks
- 4-2
 - a) Excessive pressure (over 3000 psi) caused some extrusion of Furmanite
 - b) No leaks - some extrusion of Furmanite
 - c) No leaks

July 3, 1980

Tubes rolled, then welded

close fit holes .449 I.D.

std. fit holes .418 I.D.

Tubes welded, then rolled

close fit hole .412 - .413 I.D.

std. fit holes .419 - .420 I.D.