Upstream H_2S Removal From Geothermal Steam

AP-2100
Research Project 1197-2

Final Report, November 1981
Work Completed, March 1981

Prepared by

COURY AND ASSOCIATES, INC.
7625 West 5th Avenue
Lakewood, Colorado 80226

Project Manager
G. Coury

Prepared for

Electric Power Research Institute
3412 Hillview Avenue
Palo Alto, California 94304

EPRI Project Manager
E. E. Hughes

Geothermal Power Systems Program
Advanced Power Systems Division

ORDERING INFORMATION

Requests for copies of this report should be directed to Research Reports Center (RRC), Box 50490, Palo Alto, CA 94303, (415) 965-4081. There is no charge for reports requested by EPRI member utilities and affiliates, contributing nonmembers, U.S. utility associations, U.S. government agencies (federal, state, and local), media, and foreign organizations with which EPRI has an information exchange agreement. On request, RRC will send a catalog of EPRI reports.

NOTICE

This report was prepared by the organization(s) named below as an account of work sponsored by the Electric Power Research Institute, Inc. (EPRI). Neither EPRI, members of EPRI, the organization(s) named below, nor any person acting on behalf of any of them: (a) makes any warranty, express or implied, with respect to the use of any information, apparatus, method, or process disclosed in this report or that such use may not infringe privately owned rights; or (b) assumes any liabilities with respect to the use of, or for damages resulting from the use of, any information, apparatus, method, or process disclosed in this report.

Prepared by
Coury and Associates, Inc.
Lakewood, Colorado

DISCLAIMER

ABSTRACT

The purpose of this project was to evaluate a new heat exchanger process as a method for removing hydrogen sulfide (H_2S) gas from geothermal steam upstream of a power plant turbine. The process utilizes a heat exchanger to condense geothermal steam so that noncondensable gases (including H_2S) can be removed in the form of a concentrated vent stream. Ultimate disposal of the removed H_2S gas may then be accomplished by use of other processes such as the commercially available Stretford process. The clean condensate is reevaporated on the other side of the heat exchanger using the heat removed from the condensing geothermal steam. The necessary heat transfer is induced by maintaining a slight pressure difference, and consequently a slight temperature difference, between the two sides of the heat exchanger.

Evaluation of this condensing and reboiling process was performed primarily through the testing of a small-scale 14 m^2 (150 ft^2) vertical tube evaporator heat exchanger at The Geysers Power Plant in northern California. The field test results demonstrated H_2S removal rates consistently better than 90 percent, with an average removal rate of 94 percent. In addition, the removal rate for all noncondensable gases is about 98 percent. Heat transfer rates were high enough to indicate acceptable economics for application of the process on a commercial scale. The report also includes an evaluation of the cost and performance of various configurations of the system, and presents design and cost estimates for a 2.5 MWe and a 55 MWe unit.

EPRI PERSPECTIVE

PROJECT DESCRIPTION

This project (RP1197-2) evaluated a new method for removing both H_2S and other noncondensable gases from geothermal steam upstream of the turbine. The process involved condensing and reboiling geothermal steam in a heat exchanger. The most important part of the work was the design, construction, testing, and analysis of a small experimental unit. This test unit had a nominal capacity of 1000-lb/hr steam flow, which is equivalent to about 50 kW(e). It was operated at The Geysers power plant, with cooperation and support from Pacific Gas and Electric Company (PG&E).

PROJECT OBJECTIVES

The primary objective of the project was to measure the H_2S removal capability of the upstream reboiler process. Greater than 90% removal was expected. Additional objectives were to measure the heat transfer coefficient in the heat exchanger, to evaluate alternative design options, and to estimate technical and economic features of large units, units in the 2- to 55-MW(e) size range.

PROJECT RESULTS

Average H_2S removal was 94% with a standard deviation of 2%, which was based on 38 measurements. These measurements covered a range of steam conditions and operating parameters, but the dependence of H_2S removal performance on either could not be discerned definitely from the test data.

The measured heat transfer coefficient had an average value of 3300 $W/m^2/^oC$ (580 $Btu/h/ft^2/^oF$), with a standard deviation of 15%. The actual heat transfer coefficient is thought to be higher because the measured values were rendered inaccurate by a leak discovered at the conclusion of the test program. The leak biased the measured values toward the low side, but the size of the error is not known.

Performance and cost estimates were made for three condenser-reboiler systems sized for a 55-MW(e) plant. These included the design tested and two alternative design configurations. Capital cost of a 55-MW(e) system of the type tested (a vertical tube evaporator with smooth tubes) was estimated at about $8 million (1979 dollars). This includes about $2.5 million for a Stretford plant to process the H_2S in the vent gas into elemental sulfur. The two alternative designs (vertical doubly fluted tubes and a horizontal spray evaporator) did not appear to offer significant cost reduction. Power consumed by this abatement technology is estimated to be 2 to 3% of the plant output as compared to a plant with no H_2S abatement system.

These results led to the following conclusions: (1) The performance of this technology, when combined with a Stretford unit, can meet requirements for over 90% H_2S removal. (2) Capital cost and power consumption can be competitive with other abatement systems. (3) The system has the advantages of removing all noncondensable gases (not just H_2S), removing gases upstream of the turbine, being relatively simple, and having a low requirement for support personnel.

As a consequence of the findings of this project, development and further testing of the upstream condenser-reboiler process is continuing. PG&E and EPRI are planning a pilot plant, sized at about 45,000-lb/hr steam flow or about 2.5-MW(e) equivalent, for evaluation at The Geysers power plant in 1982 and 1983. EPRI is planning to test the experimental unit at a flashed-steam (hydrothermal) geothermal site and to measure performance over a wider range of conditions relevant to hydrothermal applications.

Evan E. Hughes, Project Manager
Advanced Power Systems Division

ACKNOWLEDGMENTS

The Pacific Gas & Electric Company contributed to the success of this project by providing a suitable place to test the experimental unit and providing support while the unit was on site.

CONTENTS

Section		Page
1	INTRODUCTION	1-1
	Background: H_2S and Noncondensable Gas Control and Heat Exchanger Process	1-1
	Report Structure	1-4
	References	1-5
2	PROCESS DESCRIPTION	2-1
	General Heat Exchanger Process Discussion	2-1
	Critical Economic Factors	2-4
	Chemical and Physical Processes Determining Level of H_2S Removal	2-6
	Equilibrium Calculations	2-8
	Kinetic Effects	2-9
	References	2-13
3	PERFORMANCE OF SMALL-SCALE TEST UNIT AT THE GEYSERS POWER PLANT	3-1
	Test Objectives	3-1
	Design and Operation of Test Unit	3-1
	Test Program and Major Results	3-8
	Removal of H_2S and Other Noncondensable Gases	3-12
	Heat Transfer Performance	3-16
	Mass and Energy Balances	3-17
	Operational Characteristics and Response to Transients	3-19
	Performance of Equipment and Materials	3-20
	References	3-23
4	EVALUATION OF ADVANCED HEAT TRANSFER DESIGNS	4-1
	Introduction	4-1
	Vertical Tube Evaporator With Fluted Tubes	4-2
	Horizontal Tube, Spray-Film Evaporator	4-4
	Comparison of Predicted Performance and Costs	4-7
	References	4-9

5	2.5-MW PILOT PLANT	5-1
	Test Program Objectives	5-1
	System Description	5-1
	Design Basis	5-3
	Utility Requirements	5-6
	Equipment Lists and Specifications	5-7
	Estimated Pilot Plant Costs	5-7
	Schedule -- Erection of Pilot Plant and Test Program	5-8
6	APPLICATIONS AND COMMERCIAL-SCALE COST ESTIMATES	6-1
	Applications	6-1
	Estimated Costs for Commercial-Scale (55-MW) System	6-4
	Estimated Power Loss Due to Upstream Heat Exchanger	6-8
	Design Factors Affecting Capital Equipment Costs	6-12
	H_2S Disposal Alternatives	6-14
APPENDIX A	TEMPERATURE AND STEAM COMPOSITION OF SELECTED HYDROTHERMAL AREAS	A-1
APPENDIX B	TEST UNIT PERFORMANCE DATA	B-1
APPENDIX C	DATA COLLECTION AND REDUCTION	C-1
APPENDIX D	H_2S REMOVAL ERROR ANALYSIS	D-1
APPENDIX E	TRANSIENT TEST DESCRIPTIONS AND DATA	E-1
APPENDIX F	MASS AND ENERGY BALANCE DIAGRAMS	F-1
APPENDIX G	SPECIAL LABORATORY ANALYSIS FOR AMMONIA, BORON, AND CHLORIDE	G-1
APPENDIX H	2.5-MW PILOT PLANT MAJOR EQUIPMENT LIST	H-1
APPENDIX I	2.5-MW PILOT PLANT FIRST- AND SECOND-STAGE HEAT EXCHANGER SPECIFICATIONS	I-1
APPENDIX J	2.5-MW PILOT PLANT VENT GAS CONDENSER SPECIFICATION SHEET	J-1
APPENDIX K	2.5-MW PILOT PLANT FIRST-STAGE PUMP SPECIFICATION SHEET	K-1
APPENDIX L	2.5-MW PILOT PLANT SECOND-STAGE PUMP SPECIFICATION SHEET	L-1
APPENDIX M	2.5-MW PILOT PLANT FIRST-STAGE CONDENSATE TRANSFER TANK SPECIFICATION SHEET	M-1
APPENDIX N	2.5-MW PILOT PLANT SECOND-STAGE CONDENSATE TRANSFER TANK SPECIFICATION SHEET	N-1
APPENDIX O	2.5-MW PILOT PLANT CONTROL VALVE LIST	O-1
APPENDIX P	2.5-MW PILOT PLANT MANUAL VALVE LIST	P-1
APPENDIX Q	2.5-MW PILOT PLANT LINE LIST	Q-1

APPENDIX R	2.5-MW PILOT PLANT GENERAL PIPING SYSTEM SPECIFICATIONS	R-1
APPENDIX S	2.5-MW PILOT PLANT INSTRUMENTATION LIST	S-1
APPENDIX T	2.5-MW PILOT PLANT ADDITIONAL COMPONENT LIST	T-1
APPENDIX U	2.5-MW PILOT PLANT INTERFACE LIST	U-1

ILLUSTRATIONS

Figure		Page
2-1	Heat Exchanger Process Vertical Tube Evaporator With Baffled Shellside Configuration	2-2
2-2	Predicted H_2S Removal at 98% Condensation	2-10
2-3	Predicted H_2S Removal at 90% Condensation	2-11
3-1	Heat Exchanger Process Test Unit at Unit 7 of The Geysers Power Plant	3-2
3-2	Test Unit Piping and Instrumentation Diagram	3-4
3-3	Heat Exchanger Process Test Unit Installation at Unit 7 of The Geysers Power Plant	3-5
3-4	Test Unit Performance: H_2S Removal Versus Vent Rate	3-14
3-5	Test Unit Performance: H_2S Removal Versus Temperature Difference	3-15
3-6	Test Unit Performance: Coefficient of Heat Transfer Versus Vent Rate	3-18
4-1	Cross Section of a Doubly Fluted Heat Exchanger Tube	4-2
4-2	Horizontal Tube Evaporator Configuration	4-6
5-1	P&ID, Heat Exchanger H_2S Removal Process, 2.5-MW Pilot Plant	5-2
5-2	Process Flow Diagram	5-4
5-3	Schedule - 2.5-MW Pilot Plant	5-9
6-1	Process Flow Diagram, Commercial-Scale Heat Exchanger Process H_2S Abatement System	6-2
6-2	Comparison of Power Production Using Heat Exchanger Process as a Function of ΔT and Vent Rate	6-11
6-3	Comparison of Heat Exchanger Process Capital Costs as a Function of Temperature Difference (ΔT) and Heat Transfer Coefficient (HTC) in First-Stage Heat Exchanger	6-13
F-1	Mass and Energy Balance Sheet for Baseline Test Run 12/28/79	F-2
F-2	Mass and Energy Balance Sheet for H_2S Injection Run 1/18/80	F-3
F-3	Mass and Energy Balance Sheet for H_2S Injection Test Run 1/22/80	F-4
F-4	Mass and Energy Balance Sheet for NH_3 Injection Test Run 1/23/80	F-5
F-5	Mass and Energy Balance Sheet for H_2S and NH_3 Injection Test Run 1/24/80	F-6
F-6	Mass and Energy Balance Sheet for Baseline Test Run 1/25/80	F-7
I-1	General Configuration of the First- and Second-Stage Heat Exchangers	I-2
I-2	Tube Bundle Configuration for First- and Second-Stage Heat Exchangers	I-4
I-3	Typical Heat Exchanger Nozzle Arrangement	I-8

I-4	Nozzle Orientation	I-9
J-1	Vent Gas Condenser Configuration	J-2
M-1	First-Stage Condensate Transfer Tank Configuration	M-2
N-1	Second-Stage Condensate Transfer Tank Configuration	N-2

TABLES

Table		Page
1-1	Steam Compositions at The Geysers Geothermal Field	1-3
3-1	Test Program Chronology	3-9
4-1	Heat Transfer Performance of Doubly Fluted Tubes	4-3
4-2	Comparison of Predicted Heat Exchanger Performance and Costs	4-8
5-1	Appendix Identification for 2.5-MW Pilot Plant Equipment Lists and Specifications	5-7
5-2	Summary of 2.5-MW Pilot Plant Costs	5-8
6-1	List of Major Equipment, Heat Exchanger H_2S Removal Process, for a 55-MW Geothermal Facility	6-5
6-2	Overall H_2S Abatement System Cost Summary in 1979 Dollars	6-7
D-1	Comparison of Effects on Measured H_2S Removal Values	D-1
D-2	H_2S Concentration Reduction Errors due to Chemistry Analysis Techniques	D-3
E-1	Transient Test No. 1: Startup Ramp	E-2
I-1	Heat Exchanger Dimensions	I-5
I-2	Mist Eliminator Specifications	I-6
I-3	Heat Exchanger Nozzle List--First Stage	I-10
I-4	Heat Exchanger Nozzle List--Second Stage	I-13

SUMMARY

BACKGROUND

In 1974, the State of California imposed stringent environmental standards on hydrogen sulfide (H_2S) emissions from geothermal power plants. This contributed to a delay in the licensing of new plants and none were licensed between 1974 and 1979. During this period, the Pacific Gas and Electric Company (PG&E), owner and operator of The Geysers power plant project in Northern California, conducted an intense program to find suitable H_2S abatement technology for their geothermal plants.

From this effort, PG&E selected one set of abatement technology for new plants and another for existing plants. The new plant approach was to change the plant design from barometric condensers to surface condensers to minimize H_2S solution in the steam condensate. Primary treatment of the off-gas is accomplished by means of a Stretford process, but secondary treatment of the steam condensate is still necessary. An iron catalyst method of H_2S treatment in the steam condensate (cooling water) has been installed at several of the older plants.

Both of these methods treat the problem downstream of the turbine. In the short run these abatement methods can meet or exceed applicable standards; however, the cost is high and they have introduced new operating and maintenance problems. Also, since these methods cannot treat raw steam that must be vented occasionally from the geothermal wells in conjunction with power plant outages, these methods may prove insufficient for long term full field development.

As a result of the early efforts, a consensus emerged that removal of the H_2S upstream of the turbine, if the technology could be developed, would have a number of operational, environmental and possibly cost advantages. The Department of Energy and PG&E funded work on an upstream copper sulfate scrubber concept. Recognizing that the H_2S problem would not be unique to The Geysers and that second generation power plants should have upstream abatement technology available to them, EPRI awarded a contract to Coury and Associates in 1978 to study the feasiblity of removing H_2S upstream of the turbine by means of condensing and reboiling the geothermal steam.

OBJECTIVE

The objective of this project was to evaluate the technical feasibility of removing H_2S and other non-condensable gases from geothermal steam upstream of the turbine by means of a condensing and reboiling process, and to estimate the cost of commercial scale facilities. Technical feasibility hinged on two factors: H_2S removal efficiency and operability with little power loss. The project included process analysis, hardware and test equipment design, installation, field test, data analysis, scale-up and cost estimation.

PROCESS AND HARDWARE DESIGN

Based on mathematical modeling, it was estimated that better than 90 percent of the H_2S could be removed by the condense/reboil process over a wide range of steam conditions including H_2S, CO_2 and NH_3 concentrations. For the typical plant at The Geysers, it was estimated that 95 percent or more could be removed from the steam. It was also estimated that H_2S removal efficiency would increase with decreasing NH_3 and increasing CO_2 concentrations.

In this process geothermal steam, including its non-condensable gases, enters one side of a heat exchanger where all is condensed, except a small fraction that is vented, as a carrier, to remove the non-condensable gases. The vent steam is suitable for disposal of the H_2S, either by treatment in a secondary process such as a Stretford unit which converts the H_2S to elemental sulfur or by reinjection back into the formation. The condensate is reduced in pressure, which reduces its temperature relative to the incoming steam, and is transferred to the opposite side of the heat exchanger. The temperature difference so created causes the condensate to revaporize to produce clean steam for the turbine, thus providing a continuous condensing and reboiling effect.

The rate of removal of gases is determined by how much of each gas dissolves in the liquid phase as the entering steam condenses. The amount of gas absorbed at equilibrium and therefore the H_2S removal efficiency is controlled by three factors: (1) the partial pressure of the gas in the vapor phase; (2) the mass ratio of vapor to liquid in contact with each other; and (3) the pH of the liquid solution. The partial pressure of the gas depends on the amount of the gas present and the total pressure of the system. The mass ratio of vapor to liquid depends on the amount that is condensed; this ratio is a function of the vent rate, because more steam is condensed as less steam is vented. The pH of the liquid solution depends on the dissociation of the gases after they dissolve into the liquid phase. The amount of dissociation is determined by the appropriate equilibrium constants, which are a function of temperature, and by the concentration of the various gases in the steam. Thus, the major variables that affect gas removal rates are temperature, pressure, gas composition, and the percent of inlet steam vented.

The parameters that control cost are: (1) the amount of H_2S and total non-condensable gas to be removed; (2) the heat transfer area required; (3) the power production penalty for the loss of steam in the vent streams; (4) the power production penalty for the drop in pressure of the clean steam; and (5) the credit due to increase in power production caused by the reduction in steam flow requirements for operation of the vacuum system.

Based on these considerations, a 0.113 kg/s (900 lb/h) experimental unit was designed and erected at Unit 7 of the PG&E facilities at The Geysers geothermal area in California. The unit was a vertical tube falling-film evaporator consisting of 50 tubes with a total surface area of 14 m^2 (150 ft^2). The inlet steam was condensed on the shellside and transferred to the tubeside sump through a condensate transfer tank. A recirculating condensate pump was used to transfer the condensate to the top of the exchanger to provide flow through the tubes. Test began in March, 1979.

RESULTS

Field Test

Over 1000 hours of test time were accumulated on the experimental unit. The incoming geothermal steam had an H_2S concentration of about 240 ppm. Tests were run with unaltered steam to establish performance under various control conditions and to investigate transient response. Performance with higher concentration of H_2S was measured by injecting H_2S into the incoming steam upstream of the test unit to achieve concentrations up to 800 ppm.

For all test conditions, including H_2S concentrations from 134 to 800 ppm, vent rates from less than 1 to 17 percent, and heat exchanger temperature differences from 3 to 27°C (5 to 49°F), and within the limitations of measurement accuracy, H_2S removal (H_2S in outgoing steam divided by H_2S in incoming steam) was over 90 percent in 98 percent of the runs and averaged 94 percent during the test program. Total non-condensable gas removal was on the order of 98 percent with CO_2 accounting for most of the non-condensables. It was not possible to monitor all the factors affecting H_2S removal, especially the NH_3 content of the incoming steam; however, comparison of the removal rates obtained with the theoretical predictions showed that the agreement was within the range expected for the range of steam compositions and vent rates experienced during the tests. The H_2S removal did show an increase with increasing vent rate, but not quite as much as expected.

The heat transfer coefficient for all tests average 3268 $W/(m^2 \cdot {}^oC)$ {576 $Btu/(h \cdot ft^2 \cdot {}^oF)$}, which was somewhat less than expected but within the range of values found for commercial heat exchanger operation. The heat transfer coefficients were calculated based on the amount of steam condensed as measured by the rate of condensate transferred through the condensate transfer tank. However, it was discovered, during inspection of the unit after the tests had been completed, that there was a significant amount of leakage between the tubesheet and the sump. Such leakage would result in a measured value of the condensation rate that is less than what actually occurred, which in turn would result in a lower calculated value of the heat transfer coefficients than was actually experienced. Fouling of the tubes did not appear to be a significant cause for the lower than expected values, since a chemical cleaning of the unit was conducted and did not result in improved performance. Corrosion of the unit was negligible. There was evidence of some initial pitting of a welded connection, but analysis showed it was caused by an improper weld material.

These results established the technical feasibility of this technology to meet or exceed H_2S emission standards, providing similar performance can be achieved after scale-up to commercial size.

In summary this process should be able to achieve over 90 percent removal of H_2S and other non-condensable gases, operate at wellhead pressures and temperatures, and operate without chemical additives to the main steam. This makes it suitable for operation upstream of a turbine. Upstream removal of H_2S and other non-condensable gases has several advantages over processes which remove H_2S downstream of the turbine. These include: (1) the steam within the turbine is cleaner and less corrosive which should result in increased turbine reliability; (2) H_2S cannot get into the turbine condensate where it could require difficult liquid phase treatment to meet plant H_2S emissions requirements; (3) the removal of non-condensables ahead of the turbine reduces the steam requirements for the steam jet air ejectors which control the vacuum in the main condenser; and (4) steam can be vented through the upstream unit, as a stacking operation, when the power plant is not operating, thus avoiding the necessity to close down wells during such periods.

System Cost

The total cost for this type of H_2S abatement system for a 55-MW generating unit was estimated to be \$8.2 million including a Stretford unit. (Cost estimates were made in 1979 dollars.) This cost is based on processing 139 kg/s (1.1×10^6 lb/h) of steam in a two-stage heat exchanger system. The first stage produces clean steam to run the turbine. The vent from the first stage is fed to the second stage where a lower pressure clean steam is produced to run the steam jet air ejectors. The vent from the second stage is sent to a Stretford unit where the H_2S is converted to elemental sulfur. Total operational and maintenance costs, including annualized capital charges at 18.5 percent, would contribute 4.4 mills/kWh to the electrical busbar costs.

These costs do not include any penalty for lost power production due to the steam lost in the vent stream and the lower pressure of the clean steam. On the other hand, these costs also do not include any credit for the reduction in steam requirements for the steam jet air ejector system or for the possible increase in turbine efficiency. These items would have to be assessed on an individual basis for each application of the process. For The Geysers, the net power loss would be about 2 percent. In a case where the vent rate required for operating the heat exchanger process is less than the steam jet air ejector requirements, there could be a net power increase.

Alternate Design Considerations

In order to further optimize the design of the heat exchanger process, a preliminary design was completed for a two-stage, 2.5-MW pilot unit based on removing 95 percent of the incoming H_2S from steam with a composition typical of that found at The Geysers. It is estimated that such a unit would cost \$1,900,000 and would take 14 months to design and construct. A test program of 12 to 16 months would provide comprehensive design data for: (1) H_2S removal, non-condensables removal and heat transfer rates under various operating conditions; (2) design features necessary for best system performance; (3) equipment serviceability under unattended operation; (4) response to transient and upset conditions; and (5) operating and capital costs of commercial-scale applications.

Two alternate design options were studied for this process, doubly fluted tubes instead of smooth tubes in a vertical evaporator, and the use of a horizontal tube evaporator instead of a vertical tube evaporator. Both options increase the condensing heat transfer coefficient by decreasing the thickness of the condensate layer. For the doubly fluted tubes, however, the length of the heat conduction path through the tube wall is increased and results in a larger wall resistance. The net effect, due to low thermal conductivity of the stainless steel or titanium tubes normally used with geothermal steam, is that the overall heat transfer coefficient is not increased sufficiently with doubly fluted tubes to offer much advantage over smooth tubes. The horizontal tube design does have better overall heat transfer characteristics, however, and it does appear to warrant further investigation as a means of reducing equipment size and capital costs.

Application to Other Geothermal Fields

Although the test data are specific to The Geysers, the process has application to hydrothermal systems which generate steam by flashing geothermal liquid. Most locations using a flash system are expected to produce steam at a temperature and pressure comparable to conditions at The Geysers. Although there are wide variations in the H_2S and non-condensable gas concentrations among the various hydrothermal resources, the advantages of an upstream process listed earlier still apply. The differences in steam composition will only result in a slight change in the optimal operating condition.

Section 1

INTRODUCTION

This report presents the results of recently completed work evaluating a heat exchanger process that removes hydrogen sulfide (H_2S) gas from geothermal steam by condensing and reboiling the steam upstream of a power plant turbine. The project consisted primarily of the design and field testing of an experimental unit. It also included analytical and engineering studies related to process design optimization, future larger-scale demonstration testing, and commercial-scale applications and cost estimates.

BACKGROUND: H_2S AND NONCONDENSABLE GAS CONTROL AND HEAT EXCHANGER PROCESS

Geothermal steam is produced as a source of industrial process heat in many parts of the world. The pressurized steam often contains a variety of other gases that may include carbon dioxide, ammonia, nitrogen, hydrogen, hydrocarbons, and hydrogen sulfide. After the steam is cooled, for example, by expansion in a turbine for production of electricity, it may be either condensed or discharged directly to the atmosphere. The various gases are partially or completely liberated to the atmosphere, either in a vapor stream directly emitted from the condenser or during later processing of the condensate from the condenser.

The H_2S in the geothermal steam causes two types of problems: (1) general environmental problems related to the release of H_2S to the atmosphere; and (2) equipment damage due to the corrosive effects of high H_2S concentrations in both the geothermal steam feeding the turbines and the ambient environment of the power plant facility. At low levels of concentration, this emission causes an odor nuisance problem for the nearby areas. At higher levels of concentration, this H_2S emission may have a toxic effect on people and may be damaging to the environment. State and local government agencies have imposed H_2S emission limitations that are difficult to meet with current technology. Current and potential future regulations on H_2S emission can affect the future plans for geothermal development. The maintenance cost of power plant units is directly affected by equipment corrosion and deterioration caused by the H_2S present in both the steam and atmosphere. H_2S is particularly detrimental to low alloy steels and copper components in electrical equipment. The presence of H_2S in geothermal steam may also contribute to the failure of high stress materials such as turbine components.

The presence of other noncondensables directly affects the capital cost of new plant units since some of the power plant components, such as the condensers and vacuum systems, must be oversized to accommodate the noncondensables loading. The use of steam or power to drive vacuum systems to remove noncondensables from the condenser constitute an additional cost by decreasing the amount of energy available for sale.

These problems provide considerable incentive for developing a process that minimizes H_2S emissions at geothermal power plants. Such a process is considerably more valuable if it removes H_2S from geothermal steam upstream of the power plant turbines. Additional benefits could be gained if this process also removed most of the total noncondensables loading from the geothermal steam. The Pacific Gas and Electric Company (PG&E), U.S. Department of Energy (DOE), the Electric Power Research Institute (EPRI), and others have sponsored several research projects specifically aimed at removing H_2S from geothermal steam or liquid. These include absorption into copper sulfate and similar solutions, direct oxidation by adding oxygen to a liquid H_2S-bearing stream, the addition of iron to cooling tower basins, and others. Better processes are still being sought, due to deficiencies in processes investigated to date.

The purpose of this project was to evaluate a heat exchanger process which removes H_2S gas from geothermal steam. The primary component of the heat exchanger process is a heat exchanger which condenses geothermal well steam and reevaporates this steam after most of the H_2S and other noncondensables present in the geothermal steam have been removed via a small vent stream. The heat extracted during the condensing process is used to reevaporate the condensed steam. The vent stream containing most of the H_2S and other noncondensables is treated by another process for ultimate H_2S disposal such as by the commercially proven Stretford process. The heat exchanger process utilizes a slight pressure and temperature drop in the throughput steam as driving forces. Minimal ancillary equipment and control functions are required.

The specific application investigated in this report is for H_2S removal upstream of a geothermal power plant at The Geysers. This is a dry steam geothermal resource located north of San Francisco, California. The Geysers field produces saturated to slightly superheated steam. Usual line pressures range approximately from 700 to 800 kPa (100 to 120 psi) gauge during operations. The shut-in and transient line pressures at the facilities may rise to about 1034 kPa (150 psi) gauge. The steam contains a variety of noncondensable gases; there are also other volatilized species such as boric acid, mercury, and copper in trace amounts. The gas compositions may vary considerably from well to well. The nominal gas concentrations are presented in Table 1-1.

Table 1-1

STEAM COMPOSITIONS AT THE GEYSERS GEOTHERMAL FIELD

Component	Average Concentration (ppm)	Range (ppm)
CO_2	3000	300 - 6000
H_2S	220	70 - 570
NH_3	100	10 - 330
CH_4	200	
H_2	50	
N_2	50	
B	20	
Total	3640	

The process evaluation was based on measurements at The Geysers in cooperation with PG&E who owns The Geysers Power Plant, a commercial geothermal power-generating facility that has been operating for twenty years. Even though the test data are specific to The Geysers, this process has application at all geothermal resources in which steam is being generated directly from the geothermal fluid, including both dry steam and flashing steam (hydrothermal) locations. Appendix A shows typical conditions for some selected geothermal areas. All of the sites shown are hydrothermal locations except for The Geysers, which is a dry steam field. Most systems using a flash system are expected to produce steam of a comparable temperature and pressure as that at The Geysers, except in some cases the steam may be at lower temperatures due to lower reservoir temperatures. Variations in the noncondensable gas loadings and composition of the steam must be considered in optimizing the heat exchanger design at each location. The heat exchanger process has the potential to provide upstream H_2S abatement and upstream removal of noncondensable gases to all geothermal locations.

REPORT STRUCTURE

Section 2, PROCESS DESCRIPTION, describes the fundamental principles through which the heat exchanger process removes H_2S and other noncondensables from geothermal steam.

Section 3, PERFORMANCE OF A SMALL-SCALE TEST UNIT AT THE GEYSERS POWER PLANT, presents the results of field tests completed in 1980 demonstrating the heat exchanger process using a 14-m^2 (150-ft^2) heat exchanger test unit at Unit 7 of The Geysers Power Plant. This section includes a discussion of the test objectives, a description of the test unit, and a detailed discussion of the test results. Support material for this section is included in Appendices B through G.

Section 4, EVALUATION OF ADVANCED HEAT TRANSFER DESIGNS, discusses two design options for improving heat transfer in the heat exchanger. The two options are: a vertical tube evaporator using fluted tubes and a horizontal spray film evaporator. The expected overall heat transfer coefficient and expected costs for the two options are compared to those for the base case design--a vertical tube evaporator with smooth tubes.

Section 5, 2.5-MW PILOT PLANT, presents a conceptual design of a larger-scale pilot plant. The 2.5-MW pilot plant design package includes the system description, piping and instrumentation drawings, pilot plant design criteria, process flow diagrams, estimated equipment costs, and a proposed test plan summary. The equipment and component lists and specifications are included in Appendices H through U.

Section 6, APPLICATIONS AND COMMERCIAL-SCALE COST ESTIMATES, discusses applications at both dry steam and hydrothermal locations and presents a conceptual design scheme for a particular commercial-scale (55-MW) system with estimated costs. The direct and indirect cost effects of various design and operating parameters are also investigated.

REFERENCES

1. U.S. Department of Energy. Final Environmental Impact Statement, Geothermal Demonstration Program, 50 MW Power Plant, Baca Ranch, Sandoval and Rio Arriba Counties, New Mexico. January 1980. DOE/EIS-0049.

2. S. R. Cosner and J. A. Apps. A Compilation of Data on Fluids from Geothermal Resources in the United States. Lawrence Berkeley Laboratory, May 1978. LBL-5936.

3. P. H. Gudiksen et al. "Air Quality Assessment Studies of Geothermal Development in the Imperial Valley." In Transactions of the Geothermal Resources Council Annual Meeting, Vol. 2, Section 1, July 1978, pp. 235-236.

4. W. O. Jacobson. "Operational History of the Geothermal Loop Experimental Facility at the Salton Sea KGRA." In Transactions of the Geothermal Resources Council Annual Meeting, Vol. 2, Section 1, July 1978, pp. 325-326.

5. D. E. Robertson et al. "Chemical Characterization of Gases and Volatile Heavy Metals in Geothermal Effluents." In Transactions of the Geothermal Resources Council Annual Meeting, Vol. 2, Section 2, July 1978, pp. 579-582.

6. P. H. Gudiksen et al. The Potential Air Quality Impact of Geothermal Power Production in the Imperial Valley. Lawrence Livermore Laboratories, October 1979. UCRL-52797.

7. D. L. Ash, R. F. Dondanvill, and M. S. Gulati. Geothermal Reservoir Assessment, Cove Fort Sulfurdale Unit. U.S. Department of Energy, December 1979. DOE/ET/28405-1.

8. L. J. Garside. Nevada Bureau of Mines and Geology, Bulletin 91, 1979.

9. F. C. Brown. Preliminary Evaluation of the Copper Sulfate Process for Removal of Hydrogen Sulfide Over a Range of Geothermal Steam Conditions. Palo Alto, Calif.: Electric Power Research Institute, 1981. RP1197-3.

10. D. M. Thomas. "Water and Gas Chemistry from HGP-A Geothermal Wells: January 1980 Flow Test." In Transactions of the Geothermal Resources Council Annual Meeting, Vol. 4, September 1980.

11. B. H. Chen et al. "Progress Report on HGP-A Wellhead Generator Feasibility Project." In Transactions of the Geothermal Resources Council Annual Meeting, Vol. 4, September 1980.

12. B. H. Chen et al. "HGP-A Wellhead Generator Feasibility Project." In Transactions of the Geothermal Resources Council Annual Meeting, Vol. 3, September 1979.

13. D. Thomas and P. M. Kroopnick. "Isotopes and Gases in a Hawaiian Geothermal System: HGP-A." In Transactions of the Geothermal Resources Council Annual Meeting, Vol. 2, Section 2, July 1978, pp. 653-654.

14. A. L. Martinez. Public Service Company of New Mexico, unpublished data.

15. O. Weres et al. Resource Technology and Environment at The Geysers. Lawrence Berkeley Laboratory, June 1977. LBL 5231.

16. A. Manon et al. Extensive Geochemical Studies in the Geothermal Field of Cerro Prieto, Mexico. Lawrence Berkeley Laboratory, December 1979. LBL 7019.

17. C. B. Goranson and R. C. Schroeder. "Site Specific Geothermal Reservoir Engineering Activities at Lawrence Berkeley Laboratory." In Transactions of the Geothermal Resources Council Annual Meeting, Vol. 3, September 1979, pp. 265-268.

18. Coury and Associates, Inc. Geothermal Resource Investigation, East Mesa Test Site, Imperial Valley, California, Concluding Report. Bureau of Reclamation, September 1979.

19. Coury and Associates, Inc. Process for Removal of H_2S from Geothermal Steam, by Precipitation with Copper from Aqueous Solution, Preliminary Report. Denver, Colorado, June 1976.

Section 2

PROCESS DESCRIPTION

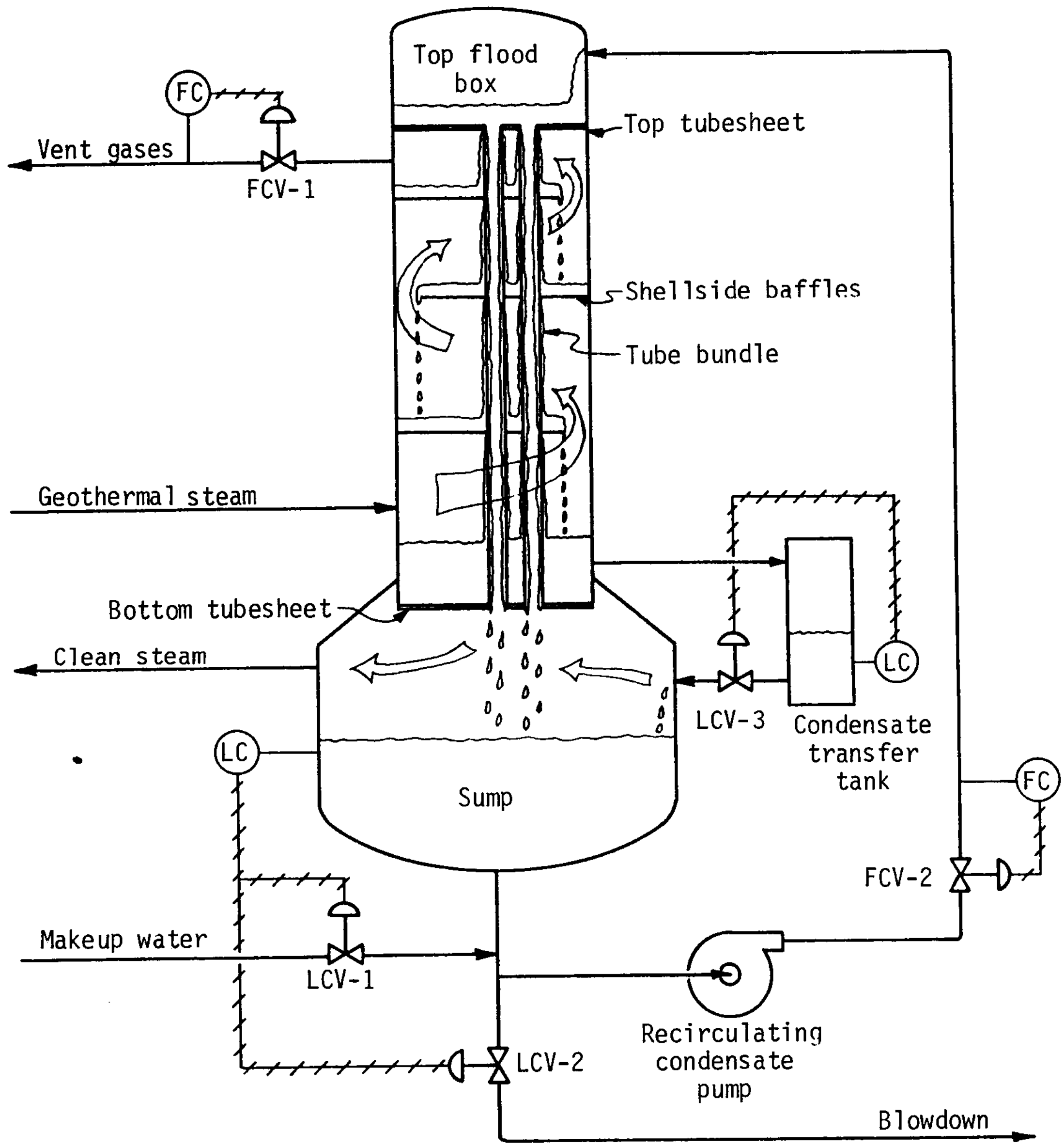
This section describes the process and discusses its underlying theory.

GENERAL HEAT EXCHANGER PROCESS DISCUSSION

The heat exchanger process is shown schematically in Figure 2-1. The tubeside is at a pressure and temperature slightly lower than the shellside. This temperature difference causes a heat transfer from the shellside to the tubeside resulting in steam condensing in the shellside and condensate evaporating in the tubeside. The incoming geothermal steam, directly from a well in the case of a vapor-dominated resource or from a vapor-liquid separator at hydrothermal locations, is almost completely condensed. The resulting condensate will dissolve some of the noncondensable gases contained in the steam, but about 98% of most of the gases, including CO_2, H_2 and N_2, will remain in the vent gas stream. Over a typical range of geothermal steam compositions and process operating conditions, 90 to 99% of the H_2S and about half of the NH_3 will remain in the vent stream. These estimates of gas removal fractions are based on calculations described later in this section and confirmed by test results presented in Section 3.

As the shellside condensate is transferred to the tubeside, a portion of it flashes to vapor due to the drop in pressure. The remaining unflashed condensate is then reevaporated as it circulates through the tubes. The total resulting tubeside vapor is the clean steam that leaves the heat exchanger. This clean steam would be supplied to the turbine in a geothermal power generation application.

Because about 98% of the major noncondensable gas components in the geothermal steam have been removed from the clean steam, as have essentially all of the light gases such as hydrogen and methane, the load on a power plant condenser vacuum system will be significantly reduced, and the quantity of steam that must bypass the turbine to run the vacuum system can be reduced. Accordingly, more steam is available for the production of electric power. In addition, any solid particles originally present in the geothermal steam will either remain with the vent gases, or they will fall out in



FC - Flow controller

LC - Level controller

FCV - Flow control valve

LCV - Level control valve

Figure 2-1. Heat Exchanger Process Vertical Tube Evaporator With Baffled Shellside Configuration.

the condensate and may be removed by filtration. This particulate removal, in addition to the more significant removal of corrosive H_2S gas, will contribute to savings in the cost of maintenance and replacement of power plant equipment and components.

The vent gas mass flow rate depends on the amount of noncondensable gases originally present in the geothermal steam. Calculations based on generalized conditions at The Geysers indicate that the vent gas stream would contain something in the range of one to four percent of the initial geothermal steam when the inert gas content is in the range of 2000 to 6000 ppm. However, the gain in the steam flow to the turbine, which represents an amount of steam that would otherwise be consumed in the vacuum system to remove noncondensables, helps to compensate for this loss of steam in the vent gas.

Thus, the final overall design will be strongly influenced by the total quantity of noncondensable gases in the geothermal steam, the composition of these gases, and the source steam pressure. Heat exchanger design would be optimized for each power plant according to the gas content existing in the untreated steam feed to each such unit.

The flow path of various streams can be followed by referring to Figure 2-1. Wellhead steam enters the shellside of the heat exchanger just above the bottom tubesheet and flows upward through the steam chest. The segmental baffles cause the steam to pass back and forth across the tubes, and thus provides a means to control the turbulence level within the vapor phase. The flow rate of the exiting stream from the shellside, the vent gas stream, is governed by control valve FCV-1. One to four percent of the inlet steam flow will be vented with the noncondensable gases. As mentioned earlier, more than 98% of all noncondensable gas will be isolated from the condensate and exit via the vent stream. The vent gas stream composition may range from less than 10% to more than 30% noncondensable gases. The condensate formed on the heat exchanger tubes flows down the tubes as a thin film and ultimately collects on the bottom tubesheet. As shown in Figure 2-1, a level controller operates a valve, LCV-3, to prevent flooding of the shellside space; the valve also serves to maintain a liquid-filled condensate transfer line so that the shellside steam can not flow directly to the tubeside of the heat exchanger.

The heat released by condensation of geothermal steam causes the evaporation of water within the tubes. Condensate from the sump is pumped to the top of the heat exchanger where it collects on the top tubesheet. Flow distributors direct the liquid into each tube so that a thin film of liquid flows down the inside tube surfaces. Only a fraction of the liquid flowing down the tube will evaporate during a single pass; thus, the condensate ultimately makes several passes through the tubes. The sump is sized to act as a feed reservoir for the pump. A level control on the sump ensures adequate positive suction head for the recirculation pump as well as protection against a high liquid level that could result in liquid carry-over with the clean steam. As shown in Figure 2-1, a flow controller would regulate the pump discharge valve, FCV-2.

Under normal operation, the heat exchanger will require a small makeup water stream to satisfy the enthalpy imbalance occurring when the higher-temperature wellhead steam is converted into lower-temperature clean steam. This makeup requirement would be decreased by heat loss from the system and increased by superheated inlet steam. For an adiabatic system with operating conditions typical of proposed commercial applications at The Geysers (as described in Section 6), the makeup requirement would be on the order of 1% or less of the net steam throughput if the inlet steam had no superheat.

Condensate blowdown may be required to purge chemical species, such as boric acid produced in the wellhead steam, which may become concentrated in the tubeside condensate. This would require additional clean makeup water to keep the system in balance.

CRITICAL ECONOMIC FACTORS

The following is a review of those factors that affect the cost of the process and the design considerations that must be analyzed in every application so as to minimize costs while attaining environmental goals. The economic aspects of the process are discussed in more detail in Section 6.

The cost factors to be considered include both capital and operating costs. These will be reviewed separately but, as will be seen, they are closely related. Capital costs can be related almost completely to the size of the heat exchanger as defined by its surface area. The required surface area (A) is directly proportional to the heat load (Q), and inversely proportional to the heat transfer coefficient (U) and the temperature driving force (ΔT), as expressed below:

$$A = \frac{Q}{U \Delta T} \qquad (2\text{-}1)$$

For a given application, the heat load is essentially fixed by the amount of steam required to supply the turbine. The U value, however, may be dependent on heat exchanger size and design. It is believed that higher U values can be approached in a larger heat exchanger than were attained in the test unit described in Section 3 because the turbulence level of the flowing steam can be better controlled in the larger units. If this can actually be achieved, the capital costs of commercial-scale units may be lower than indicated in Section 6. As shown in Section 4, the predicted commercial-scale U value is 4200 $W/(m^2 \cdot {}^oC)$ [740 $Btu/(h \cdot {}^oF \cdot ft^2)$], based on theory and extrapolated data from actual commercial-scale operation of vertical tube evaporators in other applications. A lower U value of 3400 $W/(m^2 \cdot {}^oC)$ [600 $Btu/h \cdot {}^oF \cdot ft^2)$] was used in the commercial-scale capital cost calculations in Section 6 because of the lower values experienced during the field tests discussed in Section 3.

With respect to ΔT, this can be specified in the design at any desired value, with all other factors held constant. Thus, if ΔT is doubled, the unit size will be cut in half with lower resultant capital costs; while if ΔT is cut in half, the unit size would double. The ΔT value selected, however, must reflect the results of an optimization study where operating costs are balanced against capital costs, since high ΔT values lead indirectly to high operating costs as is discussed below.

It should be made clear here that the direct operating costs associated with this H_2S removal process are quite low, amounting only to normal routine maintenance and operator surveillance. On the other hand, the H_2S removal process indirectly affects the overall electric generating system in various ways, some of which increase and others of which decrease the cost of making electricity. The three main factors indirectly affecting the overall plant operating costs are listed and then briefly discussed below. A more detailed analysis is presented in Section 6.

1. The heat exchanger ΔT reduces the turbine supply steam pressure.
2. The heat exchanger vent gas stream rejects steam that might have otherwise been available to the turbine.
3. The process provides a cleaner steam as turbine feed, thus reducing operating costs and potentially improving turbine reliability.

The lower temperature on the tubeside of the heat exchanger, as is necessary for causing heat transfer to occur, means that the steam in the line leading to the turbine is at a lower pressure than it would have been without the H_2S removal process. Thus, less electrical energy may be produced per mass unit of steam feed. In addition, the steam in the vent gas stream is steam that could have gone to the turbine, so that the total flow rate is reduced. These two effects can be considered as operating costs that result from the H_2S removal process, although they are not operating costs of the process itself.

On the other hand, some operating costs are reduced as a result of the H_2S removal process, and these cost reductions can mitigate or eliminate the cost increases mentioned above. First, the quantity of steam required to operate the vacuum system is reduced when the process is used; thus, more steam is available to produce electricity. Second, the steam entering the turbine is now clean, in that corrosive gases (H_2S and CO_2), solid debris, and boric acid have been removed. It can thus be expected that two benefits will result from the clean steam: there should be less downtime of the system for maintenance and the average pressure loss across the strainer (located in the steam line upstream of the turbine) should be reduced. Each of these effects would result in increased electric power production over a year.

In summary, these general factors control the cost of the system and its economic attractiveness as compared to alternative H_2S control systems. The actual impact of each factor will vary at each site.

CHEMICAL AND PHYSICAL PROCESSES DETERMINING LEVEL OF H_2S REMOVAL

As steam condenses on the heat transfer surface to form a pool or layer of condensate, there will be a tendency for part of the gases in the steam to dissolve in the liquid. The extent to which gases are dissolved in the liquid phase depends on the equilibrium relationships between the liquid and vapor. The amount of gases that actually are so transferred depends also on the kinetics of mass transfer between the vapor and liquid phases and on the kinetics of reaction in the liquid phase. These are discussed separately below.

The equilibrium of gases distributed between liquid and vapor can be described in a first approximation by the constant that appears in Henry's Law (Eq. 2-2):

$$P_g = K_H C_g \tag{2-2}$$

where P_g = partial pressure of gas

K_H = Henry's Law constant

C_g = concentration of gas dissolved in liquid

By that law, the amount of a specific gas that dissolves in liquid is proportional to the partial pressure of that gas in the vapor phase. The proportionality factor, or Henry's Law constant, which defines the equilibrium condition, increases as the system temperature increases. In practice, this process is complicated by the fact that the partial pressure of each gas changes as the vapor stream passes through the heat exchanger. As some of the gas dissolves, less of it remains in the vapor phase, lowering the partial pressure of that gas. On the other hand, this tendency is strongly overpowered by an opposing one that occurs because of steam condensation. From the inlet vapor phase, comprising more than 99% steam, between 96 and 99% of the water will be condensed. Therefore, the noncondensable gas concentrations and partial pressures in the vent gas will increase by a factor of 20 to 30 times. Towards the discharge end of the steam chest, the driving force for dissolution becomes many times greater than at the exchanger inlet because of this increase in partial pressure. The steam chest pressure drop, due to friction as the vapor passes over the tubes, is a relatively minor factor affecting the driving force for dissolution. The slightly lower total pressure causes a small percentage reduction in the partial pressure of each component gas.

Henry's Law is completely adequate to describe the behavior of the nonreactive gases present in geothermal steam, such as hydrogen, nitrogen, methane and other hydrocarbons. On the other hand, additional complex interactions between water and dissolved species must be considered for the acid or basic gases such as carbon dioxide, hydrogen sulfide, boric acid, and ammonia. With respect to H_2S, for example, Henry's Law applies only to the relation between the H_2S in the vapor phase and that dissolved H_2S which remains undissociated in the liquid phase. Complexity occurs because part of the dissolved H_2S (an acid gas) will dissociate successively to HS^- and $S^=$ ions, releasing a hydrogen ion at each step as follows:

$$H_2S = H^+ + HS^- \qquad (2\text{-}3)$$

$$HS^- = H^+ + S^= \qquad (2\text{-}4)$$

Dissolved carbon dioxide and boric acid will experience similar dissociation reactions in the liquid phase, while dissolved ammonia (a basic gas) reacts in an opposite manner to release the hydroxyl ion:

$$NH_3 + H_2O = NH_4^+ + OH^- \qquad (2\text{-}5)$$

Each of these reactions is governed by its own equilibrium constant, and the value of each constant is dependent upon the system temperature. The definition of the constant for the first dissociation step for H_2S is given in the following equation, where each term in parentheses refers to the concentration of that species in the aqueous phase:

$$K_{1-H} = \frac{(H^+)\ (HS^-)}{(H_2S)} \tag{2-6}$$

The interaction of these dissociation reactions is quite complex and strongly affects the amount of gas that actually dissolves in the liquid phase. Note that a common term for all the reactions is the pH of the solution as expressed by the concentration of the hydrogen ion. The prediction of the fraction of each gas that dissolves at equilibrium is achieved by simultaneous solution of the complete set of nonlinear equilibrium equations. This solution is accomplished by means of a computer because of the very large number of calculations involved.

EQUILIBRIUM CALCULATIONS

The general, qualitative interaction among these competing reactions can be described briefly. This discussion will clarify the importance of the pH value. As the acidity of the liquid solution increases, the amount of H_2S that can dissolve is decreased. Acidity is increased in part by the dissociation of H_2S itself. A much stronger factor for increasing acidity, however, is the dissociation of CO_2 because of the much higher concentration of CO_2 in the wellhead geothermal steam. Thus, H_2S removal will be improved (that is, H_2S dissolution in the condensate will be decreased) when larger amounts of CO_2 are present in the steam. Ammonia, on the other hand, acts in an opposite fashion by producing the hydroxyl ion as it dissociates, which partially neutralizes the acids formed by CO_2 and H_2S. This tendency of ammonia to increase the pH favors increased dissolution of the acid gases. Thus, if higher ammonia concentrations occur in the geothermal steam, larger quantities of both CO_2 and H_2S will be dissolved, and accordingly, the H_2S removal efficiency will be lessened. Since every well produces both different quantities of each gas as well as different ratios of individual gases, the performance of the heat exchanger process should be evaluated separately for each well or steam trunkline so that the most efficient heat exchanger design can be determined for each application. Some of the most important design considerations are flow rates and flow patterns on the condensing side of the heat exchanger and performance considerations such as the shellside to tubeside temperature difference.

On the basis of these principles, quantitative predictions were made for the rate of removal of H_2S that is possible with this system. These calculations cover parametrically the range of chemistries that can be expected in most geothermal steams, and the results are summarized in Figures 2-2 and 2-3. Each of these two figures includes four curves labeled A, B, C, and D that show the percentage removal of H_2S as a function of the H_2S concentration in geothermal steam. Curves A and B are for the case where no NH_3 is present and where the initial CO_2 content is 8000 and 3000 ppm, respectively. Curves C and D refer to the same concentration of CO_2 as in the first two curves, but with an initial NH_3 concentration equal to the initial H_2S concentration. Figure 2-2 is based on an inlet steam condensing rate of 98% (2% vent rate), and Figure 2-3 is based on an inlet steam condensing rate of 90% (10% vent rate). During the field tests at The Geysers described in Section 3, the vent rates ranged from 2 to 20% with most of the testing at about 5%.

As seen from these figures, when no NH_3 is present, almost 97% of the H_2S will be removed when 98% of the steam is condensed. On the other hand, when 90% is condensed, almost 99% of the H_2S will be removed. As was discussed above, high NH_3 concentrations will reduce the efficiency of H_2S removal. As shown on Figure 2-2 for 98% condensation of the steam, H_2S removal in the presence of NH_3 will be in the range of 91 to 96%. At the lower condensation rate of 90%, Figure 2-3 shows an H_2S removal rate of 95 to 98% in the presence of NH_3. At The Geysers, the NH_3 concentration ranges from 50% to almost 100% of the H_2S concentration. At most other geothermal fields, the NH_3 concentration is a much smaller fraction of the H_2S concentration. Values of these concentrations are indicated in Appendix A.

KINETIC EFFECTS

The above discussion relates only to the calculation of the amount of gases that will dissolve at equilibrium. The quantities that actually dissolve can be greatly affected by kinetic factors. As might be expected, there are counteracting kinetic effects with opposing tendencies. This is discussed here.

The first kinetic effect is related to the mass transfer rate of each gas from the bulk vapor phase to the vapor/liquid interface, and to the mass transfer of the dissolved gases from the interface into the bulk liquid phase. To the extent that these mass transfer rates are not very fast, the amount of dissolution of each gas will be reduced. Notably, the efficiency of H_2S removal from the clean steam will be increased as these mass transfer rates decrease. However, generalized calculations (1) have shown that the mass transfer rates are relatively high. Therefore, from the point of view of mass transfer kinetics, alone, it can be expected that equilibrium dissolution will be attained.

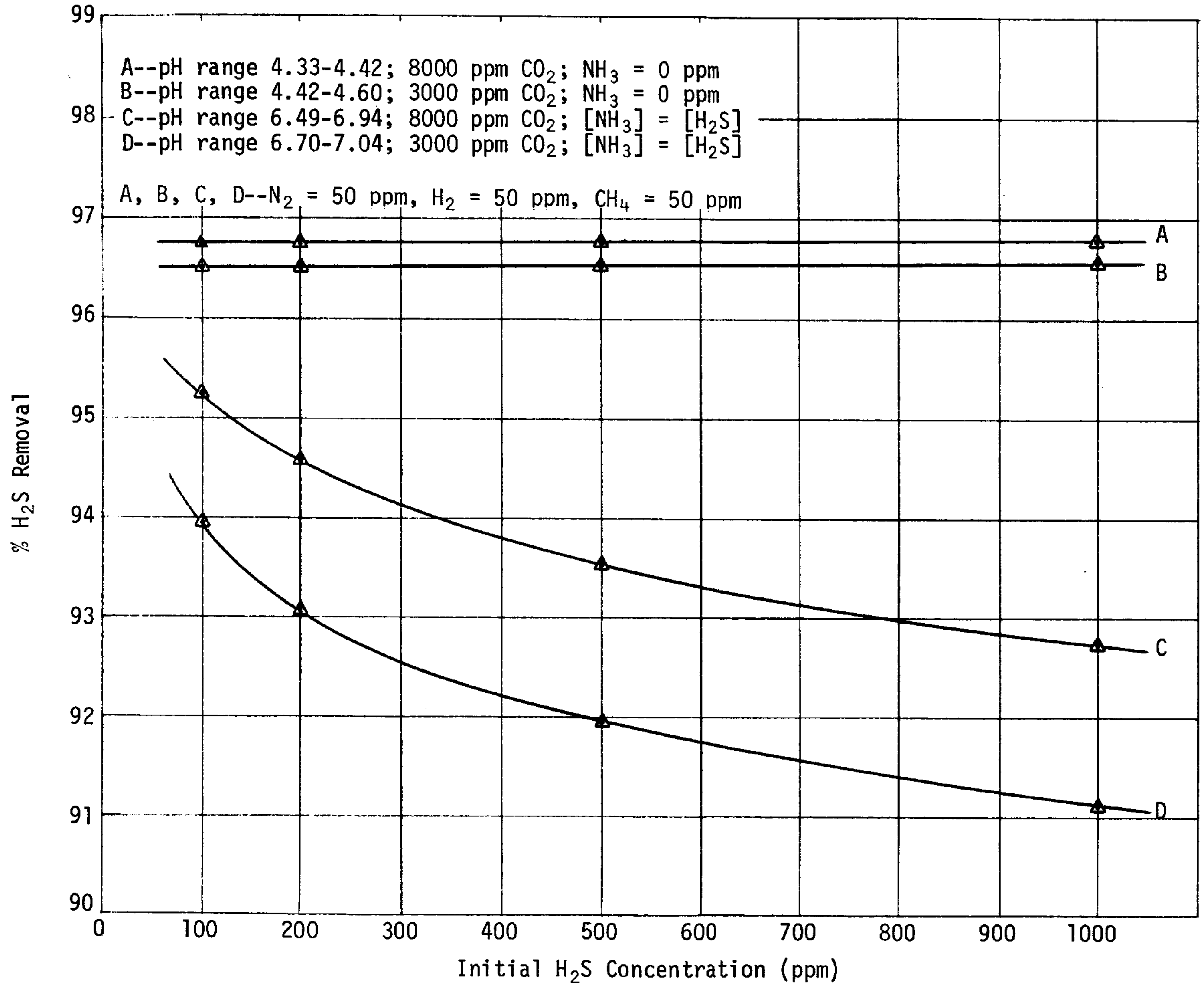


Figure 2-2. Predicted H_2S Removal at 98% Condensation (2% Vent Rate)

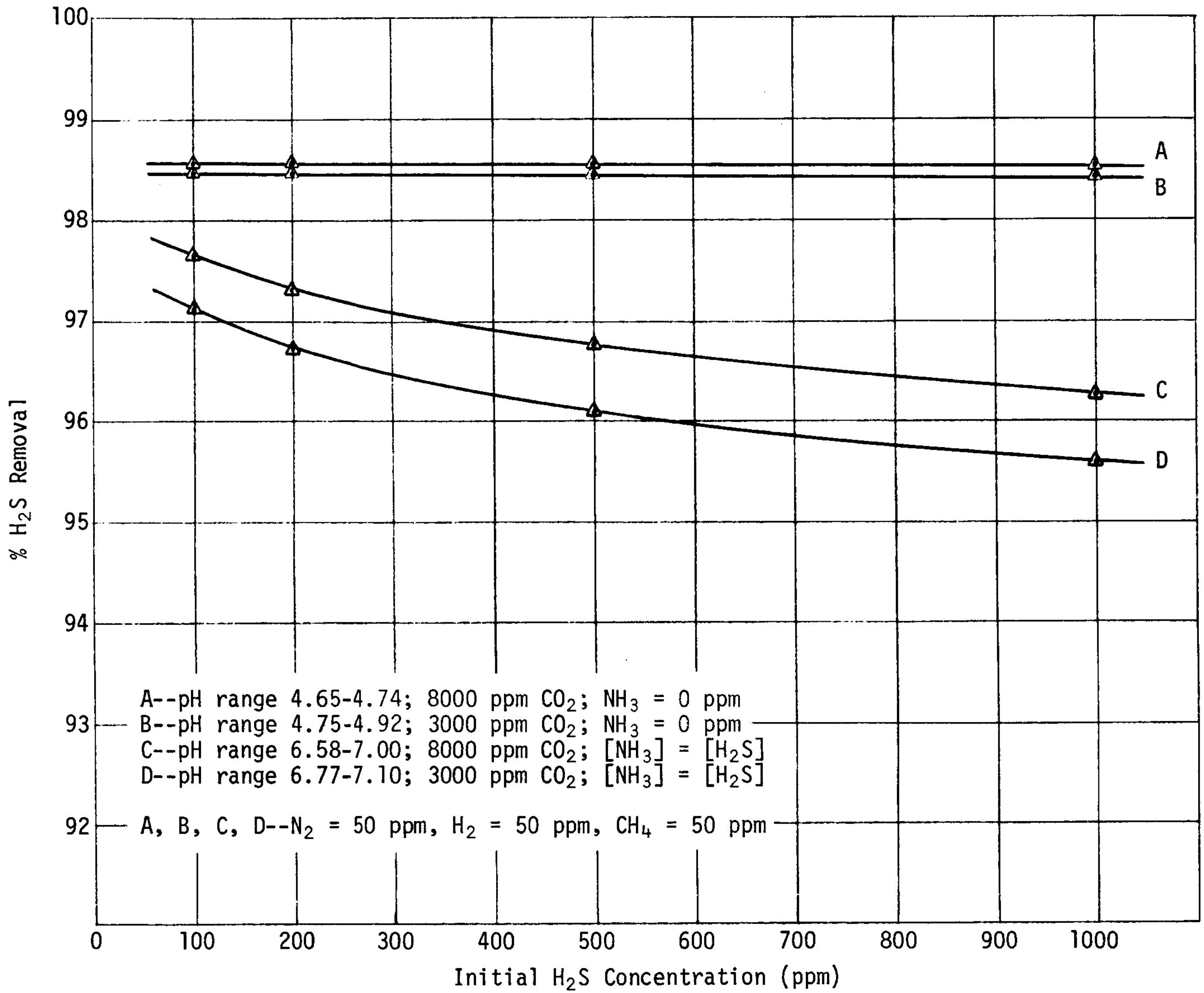


Figure 2-3. Predicted H_2S Removal at 90% Condensation (10% Vent Rate)

The second kinetic effect is related to the reaction rate of dissociation of the dissolved gas molecules. In particular, it has been suggested that the dissociation of dissolved CO_2, to produce acid and bicarbonate ion as shown below, is relatively slow.

$$CO_2 + H_2O = H^+ + HCO_3^- \tag{2-7}$$

The practical effect of slow dissociation of CO_2 is to decrease the amount of acidity due to CO_2, and thus raise the solution pH from the expected pH at equilibrium. Since the major formation of acid is due to CO_2 dissociation, as has been previously discussed, a slow rate for the dissociation of CO_2 would tend to increase the amount of H_2S dissolved. This reduces the efficiency of H_2S separation from the steam stream.

The pertinent reaction kinetic rates are not well quantified, and, in any case, the corresponding calculations would be beyond the scope of this study. We have, however, attempted to determine in a general way the effect of a reduced rate of dissociation of carbon dioxide. This was accomplished by making equilibrium calculations for hypothetical steam concentrations with smaller and smaller amounts of CO_2 than are actually present in the steam from The Geysers resource.

The logic behind this approach is the assumption that the amount of acidity that can be contributed by CO_2 is reduced when the initial presence of CO_2 is reduced, just as the CO_2 acidity would be reduced for normal amounts of CO_2 in the steam when the rate of dissociation is slow. The results of these calculations show that the efficiency of H_2S removal is not much decreased as the rate of CO_2 dissociation is decreased. In the worst case, where an infinitely low rate of CO_2 dissociation was simulated by assuming that geothermal steam contains no CO_2, H_2S removal efficiencies of over 90% were still found.

The nature of the kinetic effects and the general results that are expected if kinetic rates are so slow as to prevent the attainment of equilibrium have been discussed above. To be precisely quantitative as to the impact of kinetic effects on a full-size heat exchanger would require a considerable amount of work, both because of the complexity of the calculations and because of the need to determine what the kinetic rate constants actually are. In a more qualitative fashion, however, a general relationship can be developed to compare the magnitude of kinetic effects in a large, commercial-size system with these same effects in a small-scale test unit, such as the test unit discussed in Section 3. Such a comparison would then provide the basis for extrapolating the results measured in the small-scale unit.

To summarize the analyses of this type that have been made, the residence time of steam within the heat exchanger as a function of exchanger size will be reviewed. The residence time, T, can be qualitatively expressed as the ratio of the steam chest volume, V, to the volumetric flow rate of feed steam, F. The volume of the steam chest is proportional to the total volume of the tubes and, therefore, is proportionally related to the number of tubes, n, the tube diameter, d, and the length of tubes, L, as follows:

$$V \propto n\ L\ d^2 \qquad (2\text{-}8)$$

The surface area of the tubes, A, across which heat transfer takes place is proportional to these same variables:

$$A \propto n\ L\ d \qquad (2\text{-}9)$$

The surface area is also proportional to the steam flow rate because the steam density and latent heat, as well as the heat transfer coefficient and temperature driving force, will be relatively constant as the system size changes. That is:

$$A \propto F \qquad (2\text{-}10)$$

Combining these relationships, we have:

$$T \propto \frac{V}{F} \propto d \qquad (2\text{-}11)$$

Thus, the residence time will be about the same, regardless of the size of the heat exchanger, as long as the tube diameter is unchanged. Then, it could be expected that whatever effect kinetics may have with respect to how closely equilibrium is achieved can be directly determined in the small test unit. Accordingly, the same degree of H_2S removal that was achieved in the field, as reported in Section 3, can be expected to occur in a commercial-size unit.

REFERENCES

1. Calculations performed by Glenn Coury of Coury and Associates, Inc., 1976.

Section 3

PERFORMANCE OF SMALL-SCALE TEST UNIT AT THE GEYSERS POWER PLANT

A 12-month field test program with an accumulative run time of approximately 1000 hours was completed in January 1980 demonstrating the performance of a small-scale, 14-m^2 (150-ft^2) falling-film vertical tube evaporator heat exchanger. These field tests were conducted with the cooperation of Pacific Gas and Electric Company at Unit 7 of The Geysers Power Plant located north of San Francisco, California. The test unit heat exchanger was designed to condense approximately 0.113 kg/s (900 lb/h) of geothermal steam at a normal operating temperature drop across the heat exchanger of 5.6^oC (10^oF). A typical 55-MW power plant unit at The Geysers requires approximately 139 kg/s (1.1 x 10^6 lb/h) steam; therefore, the test unit heat exchanger was equivalent to about a 0.05-MW unit.

This section presents the objectives of the test program, a description of the test unit, a description of the test program, and a detailed presentation of the test results.

TEST OBJECTIVES

The test program was developed and conducted to achieve two objectives. These were:

1. Demonstrate the process' capability to remove at least 90% of the H_2S present in the incoming geothermal well steam.
2. Demonstrate the heat transfer performance of the falling-film vertical tube evaporator in the geothermal environment.

In addition to satisfying these two specific objectives, the test program was designed to provide performance data over a range of process parameters representing the anticipated normal operating and upset conditions typical of the intended full-scale geothermal H_2S abatement applications of the heat exchanger process.

DESIGN AND OPERATION OF TEST UNIT

General Description of Test Unit Installation

The photograph in Figure 3-1 shows the skid-mounted test unit installed at Unit 7 of The Geysers Power Plant. The basic components of the test unit included the heat exchanger--a 14-m^2 (150-ft^2) vertical tube evaporator; the recirculating condensate

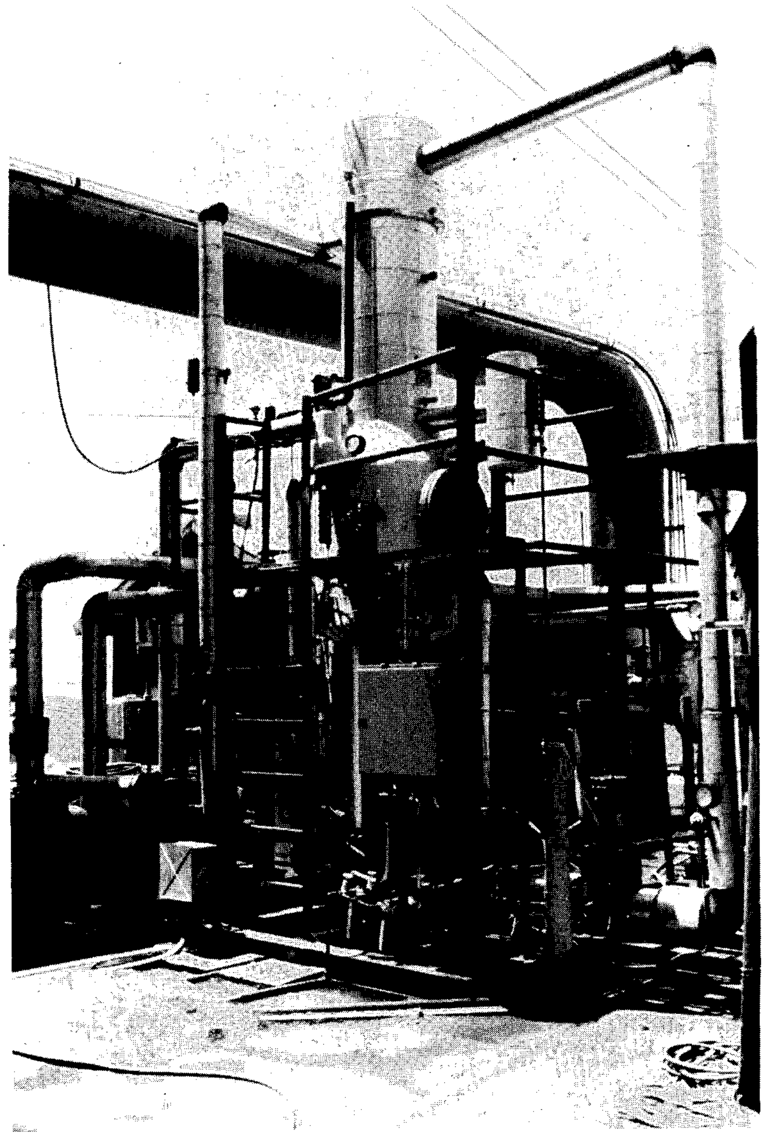


Figure 3-1. Heat Exchanger Process Test Unit at Unit 7 of The Geysers Power Plant

pump; the condensate transfer tank; the sample condenser and cooler; and the interconnecting piping, valves, controls, and instrumentation. These components are indicated in Figure 3-2, the piping and instrumentation diagram for the test unit.

As described in Section 2 and Figure 2-1, the process involves condensation of inlet steam on the shellside of the heat exchanger, removal of most H_2S and other noncondensable gases through a vent from the shellside, transfer of condensate from the tubesheet at the base of the shellside to the sump which is at a slightly lower pressure and temperature, recirculation of condensate from the sump to the top of the tubeside of the heat exchanger, reevaporation of the condensate on the tubeside, and discharge of the resulting clean steam out of the tubeside sump. Physical parameters (flow, pressure, temperature and liquid levels) were measured at the many points indicated by circles in Figure 3-2. Chemical constituents of the various flow streams were analyzed at various times after samples were extracted at the points indicated by squares. The sample cooler, drain and collection system, common to all sample points, is shown at the lower left of the figure.

Figure 3-3 shows how the test unit installation interfaced with the existing Unit 7 facility. Supply steam for the heat exchanger was provided by a side stream off the geothermal well steam supply to the Unit 7 power plant. The vent gas and clean steam streams from the test unit were combined into a single stream downstream of the test unit and then directed to the Unit 7 cooling tower basin. Utilities and drain provisions were also provided by the Unit 7 power plant. As shown in Figure 3-3, the steam load required by the test unit was less than 0.1% of the normal steam load for the Unit 7 turbine; therefore, the total steam consumption, and consequently the total H_2S emission to the atmosphere, were increased by less than 0.1% when the test unit was in operation. The test unit operated independently from the Unit 7 power plant, with the exception that Unit 7 had to be operating before the test unit could operate.

Test Unit Component Description

The test unit heat exchanger was a vertical tube evaporator with a baffled shellside configuration similar to that shown in Figure 2-1. The heat exchanger was constructed entirely of 304 stainless steel except for the heat transfer tubes which were titanium. The selection of titanium for the tubes was strictly based on convenience at the time of fabrication and was not necessary from corrosion or heat transfer considerations. A commercial unit would probably have 304 stainless steel tubes. Titanium has a coefficient of heat transfer about one-half of that of 304 stainless steel; however, with respect to the test results this difference is insignificant since the coefficients

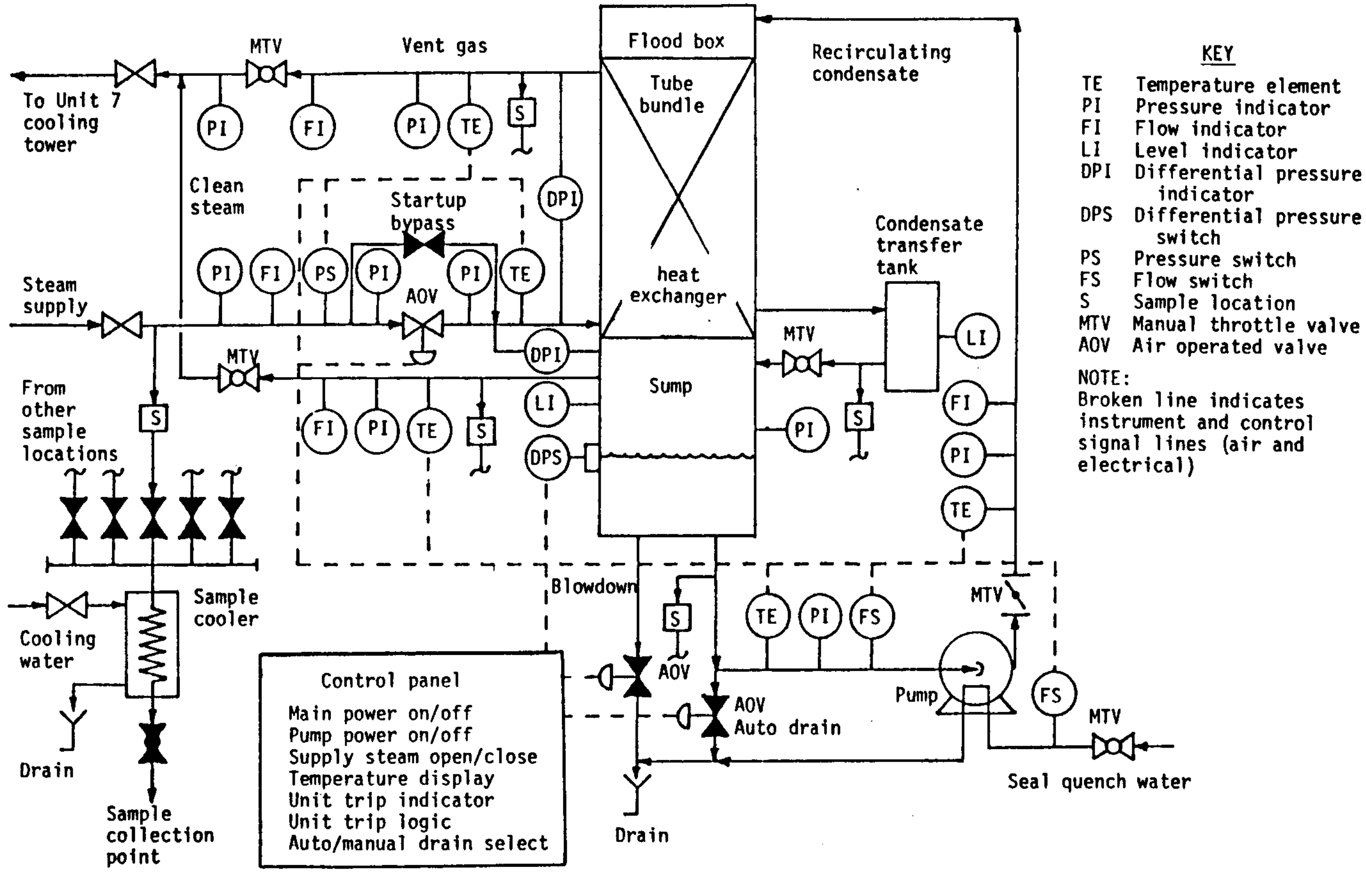
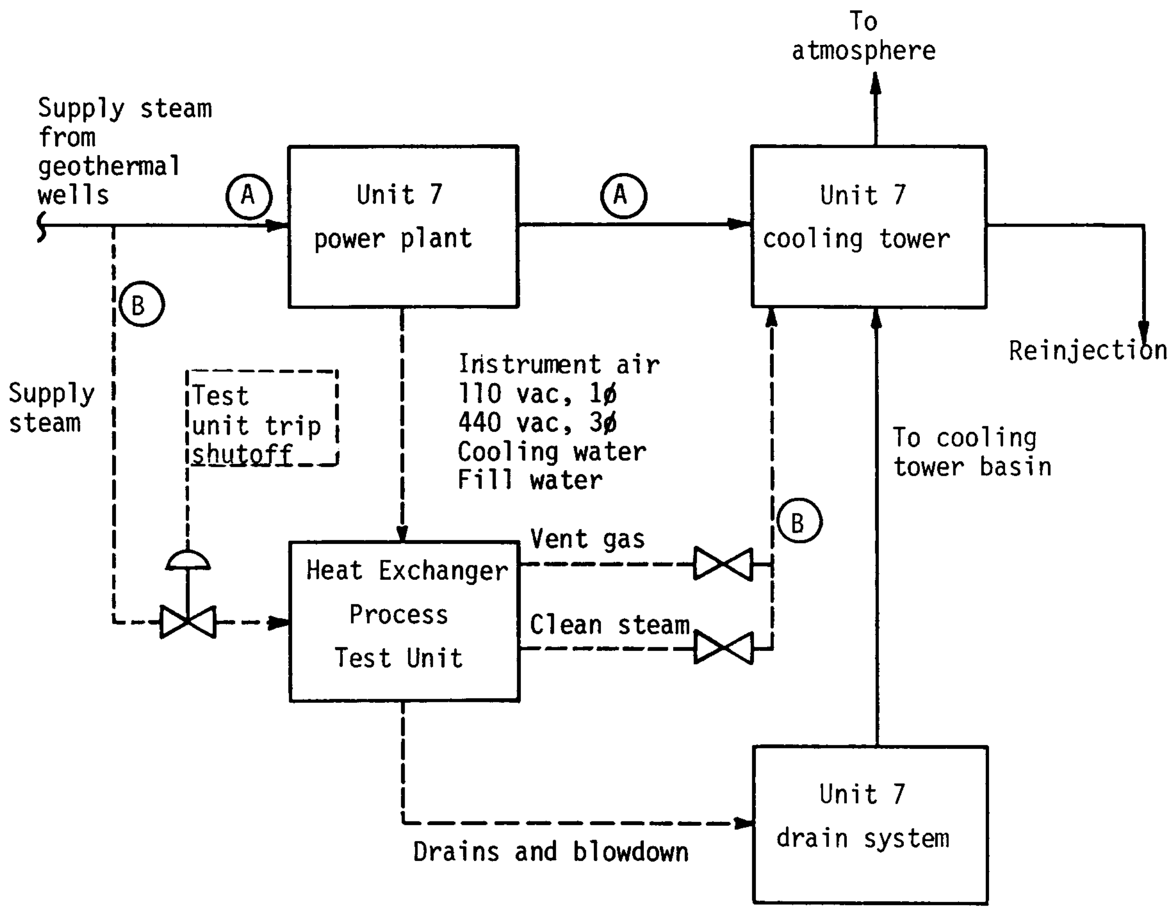


Figure 3-2. Test Unit Piping and Instrumentation Diagram



Typical normal flowrates: Stream A >126 kg/s (10^6 lb/h)
Stream B <0.1 kg/s (10^3 lb/h)
Stream B <0.1% of Stream A

——— Permanent Unit 7 installation

– – – – Temporary test unit connections

Figure 3-3. Heat Exchanger Process Test Unit Installation at Unit 7 of The Geysers Power Plant for the two materials (based on the tube wall thickness) are about 5 to 10 times the total heat transfer coefficients, including the film coefficients, experienced during the tests. The tube bundle was approximately 2 m (6 ft) long and 508 mm (20 in) in diameter, and contained 50 tubes with diameters of 50.8 mm (2.0 in). The total tube bundle surface area was approximately 14 m^2 (150 ft^2). The wall thickness of the tubes was 0.71 mm (0.028 in).

The recirculating condensate pump was an in-line vertical centrifugal pump with stainless steel internals and mechanical shaft seals which required a small, constant flow of cooling water. The pump was electrically driven by a 4-kW (5-hp) TEFC motor.

The condensate transfer tank was constructed of large-diameter 304 stainless steel pipe.

The sample condenser and cooler was a small heat exchanger with the sample stream flowing through stainless steel tubing inside the heat exchanger and cooling water flowing through the outside jacket of the heat exchanger. The purpose of the sample condenser and cooler was to condense the steam samples and cool all samples (both steam and condensate streams) so that samples could be taken at atmospheric pressure. All of the sample connections were manifolded to this single condenser and cooler with isolation valves allowing one process stream to be sampled at a time.

The interconnecting piping and valves were all 304 stainless steel, except for the flanges and the manual flow control valve on the recirculating condensate line which were made of carbon-steel; the selection of carbon-steel for these components was based on material availability in consideration of the test program schedule and the budget. The instrumentation and controls will be discussed in the following portions of this section.

Test Unit Design

Sizing Criteria. The test unit heat exchanger was designed to condense a nominal 0.113 kg/s (900 lb/h) of incoming saturated steam at 177^{o}C (350^{o}F) with a shellside to tubeside temperature difference of 5.6^{o}C (10^{o}F) and an assumed heat transfer coefficient of 3404 $W/(m^2 \cdot {}^{o}C)$ [600 $Btu/(h \cdot ft^2 \cdot {}^{o}F)$]. This value of the heat transfer coefficient was considered to be conservative based on earlier in-house studies of vertical tube evaporators without the noncondensable gas loadings typical of The Geysers steam.

The total system design was based on efficient operation with an incoming steam condensing rate ranging from 0.063 to 0.189 kg/s (500 to 1500 lb/h). Controls were sized to allow special tests with vent rates up to 0.025 kg/s (200 lb/h)--20% of the 0.126 kg/s (1000 lb/h) normal condensing rate. The tubeside recirculating condensate pump and piping were sized for a flow range of 0.1 to 0.3 l/s (1 to 4 gpm) per tube.

Operation, Controls, and Instrumentation. The following discussion includes a description of the overall operation and control of the test unit and a description of the test unit instrumentation as shown in Figure 3-2. The operation of the heat exchanger is simple and straightforward, with a minimal amount of automatic control requirements. A goal in the test unit design was to demonstrate this simplicity of operation. The control functions of the test unit for both steady-state operation and upset conditions were as follows (refer to Figure 3-2):

- Clean Steam Flow Rate--Set by a manual throttle valve; monitored by a rotameter. (This simulates the clean steam demand of the turbine, and would not be a control function on a full-scale heat exchanger.)
- Vent Gas Flow Rate--Set by a manual throttle valve; monitored by a rotameter.
- Recirculating Condensate Flow Rate--Set by a manual throttle valve; monitored by a flow meter.
- Condensate Transfer Tank Level--Set by a manual throttle valve; monitored by a level glass.
- Sump Level--Automatically controlled by a differential pressure level switch which opened the blowdown valve on high level and shut the blowdown valve on low level. (There were sufficient heat losses from the test unit to cause a slight condensate accumulation in the sump during normal operation; thus allowing automatic level control by occasional blowdown. A commercial-scale unit would be better insulated and would require a slight makeup to remain in thermodynamic equilibrium.)
- Pump-Seal Quench Water Flow Rate--Set by a manual throttle valve; monitored by a flow meter.
- Unit Shutdown Controls--Automatically closed air-operated valve on inlet steam line, and pump power cutoff; initiated by any one of six switches--low flow in recirculating condensate line; low flow in pump-seal quench water line; high-high sump level; low-low sump level; high inlet steam line pressure; and low inlet steam line pressure. In freezing weather, the unit trip function could be set to also open the system drain valve.

The test unit startup procedure further demonstrated the simplicity in operation.

The startup procedure consisted of the following sequence of manual operations:

- Establish proper sump level through fill connection nozzle; initial fill water was filtered auxiliary cooling water from the Unit 7 power plant
- Establish pump-seal quench water flow
- Start pump
- Set recirculating condensate flow rate
- Open manual inlet steam bypass valve to warm up unit and bring unit up to inlet steam pressure.
- Open air-operated inlet steam valve and close manual bypass valve
- Set vent gas flow rate
- Set clean steam flow rate
- Set condensate transfer tank level

The test unit was instrumented to allow visual monitoring of temperatures, pressures, differential pressures, flow rates, and liquid levels to assist in the manual operation and control of the test unit, and to provide performance data information as required by the test program. Some of these instruments were described in the preceding discussion of the test unit control functions. The following is a brief description of the remaining instruments.

Resistance thermometer probes (RTDs), were located in all of the system process lines. The signals from the RTDs were transmitted back to the control panel where a digital output of the temperatures was displayed. Pressure gauges were located throughout the test unit system. Manometers were included to monitor the differential pressures between the inlet steam and vent gas lines, and between the inlet steam line and the heat exchanger sump. An orifice/manometer flow indicator was located in the inlet steam line.

TEST PROGRAM AND MAJOR RESULTS

Test Program Description

Table 3-1 presents a brief chronological summary of the testing activities and other significant events that occurred during the 12-month test program and the equipment examination period that followed the test program.

Shakedown tests for the test unit were conducted in February 1979 and the actual data collection was initiated in March 1979. Testing operations and data collection

Table 3-1

TEST PROGRAM CHRONOLOGY

Time Period	
Feb.-Mar. 1979	Startup and shakedown tests
Mar.-Aug. 1979	Baseline testing
Aug.-Oct. 1979	Pump failure; shutdown to repair pump and motor and investigate corrosion noted inside heat exchanger sump.
Oct.-Dec. 1979	Continued baseline testing
Dec. 1979	Examined and chemically cleaned heat exchanger to determine if scale buildup was occurring
Dec. 1979-Jan. 1980	Postcleaning baseline tests; detailed chemistry analysis of process streams; transient tests; inlet-steam gas injection tests; special performance tests
Jan.-Dec. 1980	No further testing; extensive examination of test unit heat exchanger and equipment including: instrument calibration, examination of heat exchanger internals and tube bundle, and analysis of material specimens and deposit samples

continued up through January 1980. The first part of the test program was devoted to baseline testing with the purpose of establishing the general performance characteristics (H_2S removal and heat transfer coefficients) of the heat exchanger during extended steady-state operation, with only slight variations in the primary operating parameters of vent rate and temperature difference.

The baseline testing was interrupted when the recirculating condensate pump failed. During the resulting shutdown, corrosion pitting was discovered in the vicinity of one of the welds inside the heat exchanger vessel. The pump was repaired, and the corrosion problem was thoroughly examined by an outside consultant with the determination that it was not serious enough to affect the test program. Baseline testing continued but was interrupted again when the heat exchanger was chemically cleaned to determine if scale buildup was occurring and affecting the heat transfer performance. Operation following the chemical cleaning included additional baseline tests (to determine the effects of the chemical cleaning), detailed chemistry analysis of the process flow streams (to help complete component mass balances), special transient tests, artificial increases in inlet steam H_2S and NH_3 concentrations, and other special performance tests. The test program was terminated in

January 1980 with a total accumulative operation time of approximately 1000 hours. During the following months, extensive examinations of the test unit were performed including the calibration of the instruments, a thorough examination of the heat exchanger internals and tube bundle, and laboratory analysis of material specimens and deposits collected inside the heat exchanger at various locations.

Baseline Tests

During the 12-month test program the test unit accumulated a total operating time of approximately 1000 hours, with the bulk of that devoted to baseline testing and the remainder to special performance tests. The baseline tests, which are identified in Appendix B, "Test Unit Performance Data," were the most important part of the test program, consisting of relatively long continuous steady-state run periods (the longest estimated to be 288 hours) with performance parameters adjusted to simulate commercial-scale operation at The Geysers. The purpose of the baseline tests was to obtain enough data to adequately demonstrate the H_2S removal capability and heat transfer performance of the test unit heat exchanger, as well as to demonstrate general operating characteristics of the heat exchanger. During the baseline tests, the vent rate was generally maintained between 2% and 8% of the inlet steam flow rate, with an average vent rate of slightly less than 5%. The shellside to tubeside temperature difference (ΔT) was generally maintained between 3^oC and 5^oC (5^oF and 9^oF) with an average ΔT of about 4^oC (7^oF). The ΔT was controlled by regulating the clean steam flow rate. On a few occasions during the baseline tests, and also during some of the special tests described below, vent rates and ΔT values were outside of these ranges. The extreme ranges for these parameters during the test program were 1% to 20% and 2.7^oC to 12.1^oC (4.9^oF to 21.8^oF), for the vent rate and ΔT respectively.

Special Tests

In addition to the baseline tests, several special tests were conducted. These special tests, which were intended to demonstrate performance characteristics of the pilot plant heat exchanger beyond the principal objectives previously stated in this section, included high vent rate tests, high ΔP tests, gas injection tests, and transient tests. Special chemistry analysis during the gas injection tests included detailed analysis of stream compositions.

High Vent Rate Tests. These are represented by Data Set numbers 126, 129, 152, and 153 in Appendix B. The purpose of these tests was to note the effect on U values and H_2S removal of higher vent rates. During these four tests the vent rates ranged between 17% and 20% of the inlet flow rate. Due to technical difficulties, only three of the four tests had corresponding U values and only two had corresponding values of H_2S removal.

High ΔP Test. This test is represented by Data Set numbers 150 and 151 in Appendix B. The objective was to demonstrate the performance of the pilot plant heat exchanger, including its performance as a silencer, at conditions simulating a wellhead application during which the heat exchanger would discharge directly to the atmosphere after removing H_2S from the wellhead steam. Ideally, the test would be conducted with a shellside to tubeside pressure drop (ΔP) equal to the full inlet steam pressure, 758 to 827 kPa (110 to 120 psi); however, design limitations of the pilot plant only permitted testing at a lower ΔP. The test was run with a ΔP of about 365 kPa (53 psi) which corresponds to a ΔT of about 27°C (49°F) and a flow rate about five times greater than normal. The H_2S removal and heat transfer performance results of this test are not considered valid due to an unexplained water loss that occurred during the test; the sump level dropped rapidly, tripping the unit after about 20 minutes.

Gas Injection Tests. The objective of these tests was to demonstrate H_2S removal performance with significantly higher concentrations of H_2S and NH_3 and significantly different ratios of NH_3 to H_2S in the inlet steam. During these tests, various quantities of H_2S and NH_3 gas were injected into the inlet steam line, increasing the concentrations of these components up to four times their normal concentrations. The ratio of NH_3 concentration to H_2S concentration in the inlet steam line varied from 0.2 to 2.0 during these tests. Appendix B and Appendix F, "Mass and Energy Balance Diagrams," show the results of these tests.

Detailed component analyses of the major flow streams were completed during the gas injection tests and baseline tests conducted during the same time period. These analyses included the determination of the component concentrations of H_2S, NH_3, CO_2, and total noncondensables in the inlet steam, vent, and clean steam flow streams. The results are shown in Appendix F.

Transient Tests. These tests are described in Appendix E, "Transient Test Descriptions and Data." The objective of these tests was to demonstrate the performance of the pilot plant heat exchanger during both normal and abnormal transient conditions that might be experienced when installed upstream of a geothermal turbine generator facility similar to existing power plant units at The Geysers. The transient tests included:

- Startup Ramp--Simulated the normal startup load increase rate of a typical power plant turbine at The Geysers.
- Clean Steam Valve Sudden Close--Simulated a turbine trip.
- Clean Steam Valve Sudden Open--Simulated rapid load increase at the turbine.
- Vent Valve Sudden Close--Simulated sudden shutdown of equipment processing heat exchanger vent stream.
- Vent Valve Sudden Open--Simulated rapid increase in vent rate--may be caused by line rupture, control valve failure, etc.
- Inlet Steam Valve Sudden Close--Simulated sudden loss of well steam supply.
- Pump Trip--Simulated trip of condensate recirculation pump.

Major Test Results

Appendix B, "Test Unit Performance Data," presents a listing of the critical performance parameters for 153 data sets representing the estimated 1000 hours of operation. The critical parameters shown in Appendix B include the temperature difference (ΔT), the vent rate, the coefficient of heat transfer (U), and H_2S removal values.

The major results of the field tests are summarized below:

- The test unit accumulated approximately 1000 hours of operation.
- The H_2S removal rate averaged about 94% with a range of 87% to 98%. Only one measured point was below 90%.
- The coefficient of heat transfer (U) averaged about 3268 $W/(m^2 \cdot {}^oC)$ [576 $Btu/(h \cdot ft^2 \cdot {}^oF)$] with a range of 1889 to 4471 $W/(m^2 \cdot {}^oC)$[333 to 788 $Btu/(h \cdot ft^2 \cdot {}^oF)$]. All measured U values are thought to be conservative.
- The test unit demonstrated very simple and predictable operating characteristics.

REMOVAL OF H_2S AND OTHER NONCONDENSABLE GASES

H_2S removal data for the field tests are shown in Appendix B, "Test Unit Performance Data." The H_2S removal rates shown are based on the following equation:

$$\% \; H_2S \text{ removal} = \left[1 - \left(\frac{\text{Clean steam } H_2S \text{ ppm}}{\text{Inlet steam } H_2S \text{ ppm}}\right)\right] \times 100 \qquad (3\text{-}1)$$

Appendix F, "Mass and Energy Balance Diagrams," shows detailed noncondensable gas concentrations of the various flow streams for four gas injection cases and two baseline cases.

The techniques used to measure H_2S and the other noncondensable gas concentrations in the various streams are discussed in Appendix C, "Data Collection and Reduction."

All but one of the 46 H_2S removal data points shown in Appendix B were greater than 90%. The measurements ranged from 98.1% to 87.3 % (the one point lower than 90%), with an average of 94.0% and a standard deviation of 2.1%. Figures 3-4 and 3-5 show plots of H_2S removal versus vent rate and ΔT. Although no conclusive correlation is shown between the H_2S removal rate and ΔT (no direct correlation is expected based on theory), these figures do indicate that the H_2S removal rate is dependent on the vent rate, increasing as the vent rate is increased, as predicted by theory. As seen in Figure 3-4, however, the linear curve fit of the data gives values slightly less than theoretical values based on average conditions at The Geysers, with this difference in percent removal values ranging from about 1 at a vent rate of 1% to about 3 at a vent rate of 10%. Information provided from Appendix D, "H_2S Removal Error Analysis," indicates that the expected variations in percent H_2S removal values due to fluctuations in inlet steam concentrations of H_2S and NH_3 expected at The Geysers range from 0.5 to 2, and that the expected error in percent H_2S removal values due to chemistry analysis techniques ranges from 1 to 4. Error bands of ± 1 and ± 4 are indicated in Figure 3-4 and, as can be seen, most of the data points and the predicted values are inside the ± 4 band.

During the high vent rate tests (Data Sets 126 and 129 in Appendix B), the H_2S removal rates were approximately 97% which, when compared with the overall average value of 94% and the standard deviation of 2.1%, supports the theory that the H_2S removal is enhanced by higher vent rates.

Appendix F shows detailed analyses of H_2S and other noncondensable gases in the various flow streams for six specific cases: four gas injection test cases during which the inlet steam composition was modified by injecting H_2S and NH_3, and two baseline test cases during this same general time period (the means for performing these complete analyses were not available during other periods of the test program). With each of the six cases in Appendix F, the measured H_2S removal rates are compared with the predicted removal rates based on interpolation of Figures 2-2 and 2-3 for the measured inlet H_2S, NH_3, and CO_2 concentrations and the measured vent rates for each case. The measured percent H_2S removal values ranged from 0 to 5 less than the predicted percent removal values. The inlet ratio of NH_3 concentration to H_2S concentration (in terms of ppm) ranged from 0.2 to 2.0; however, no conclusive correlation between the H_2S removal rate and this ratio can be seen in the data. Theory predicts that the H_2S removal rate is very dependent on this ratio. In Appendix F, the H_2S mass balance closures ranged from -35% to +37%. These closure values can

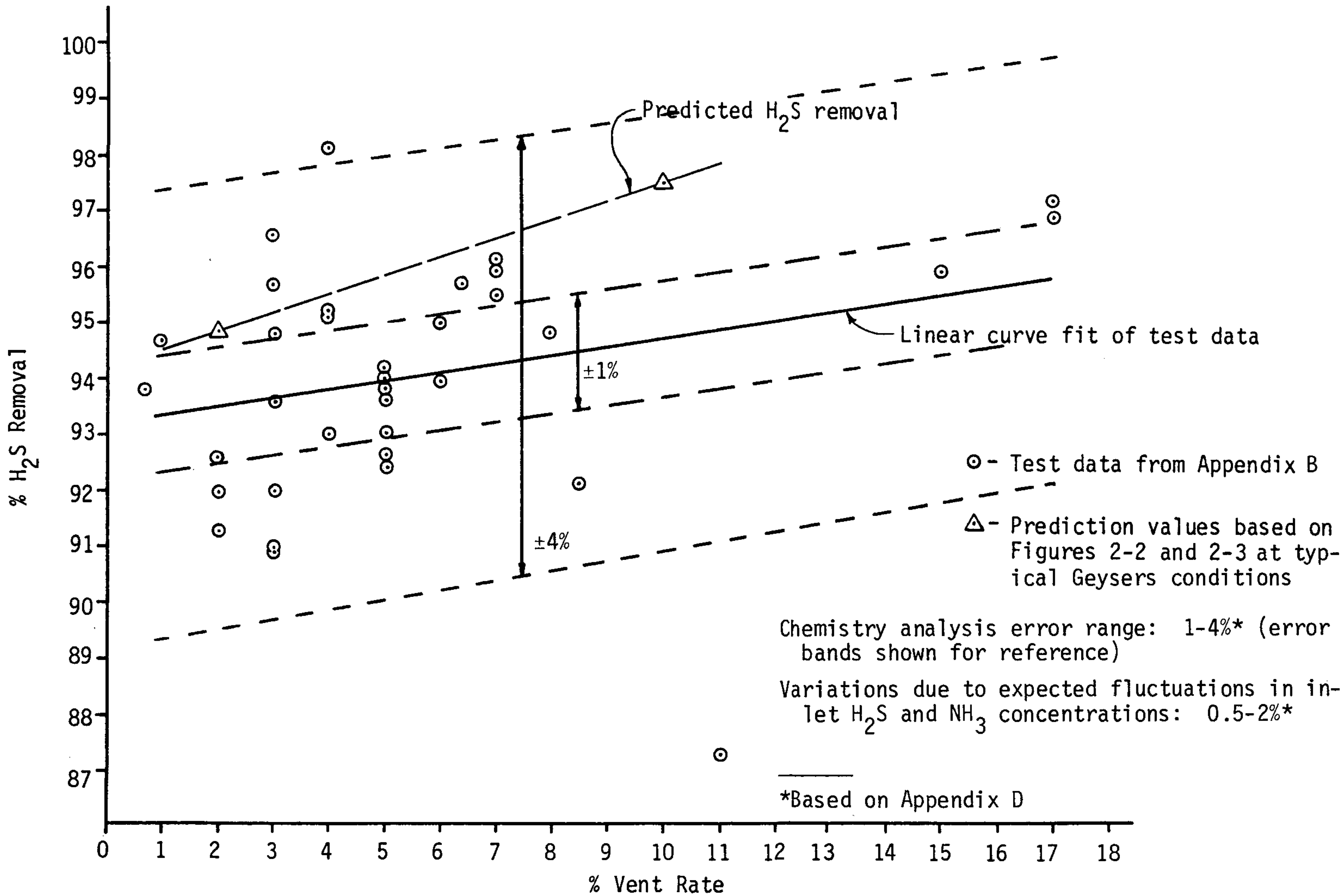


Figure 3-4. Test Unit Performance: H_2S Removal Versus Vent Rate

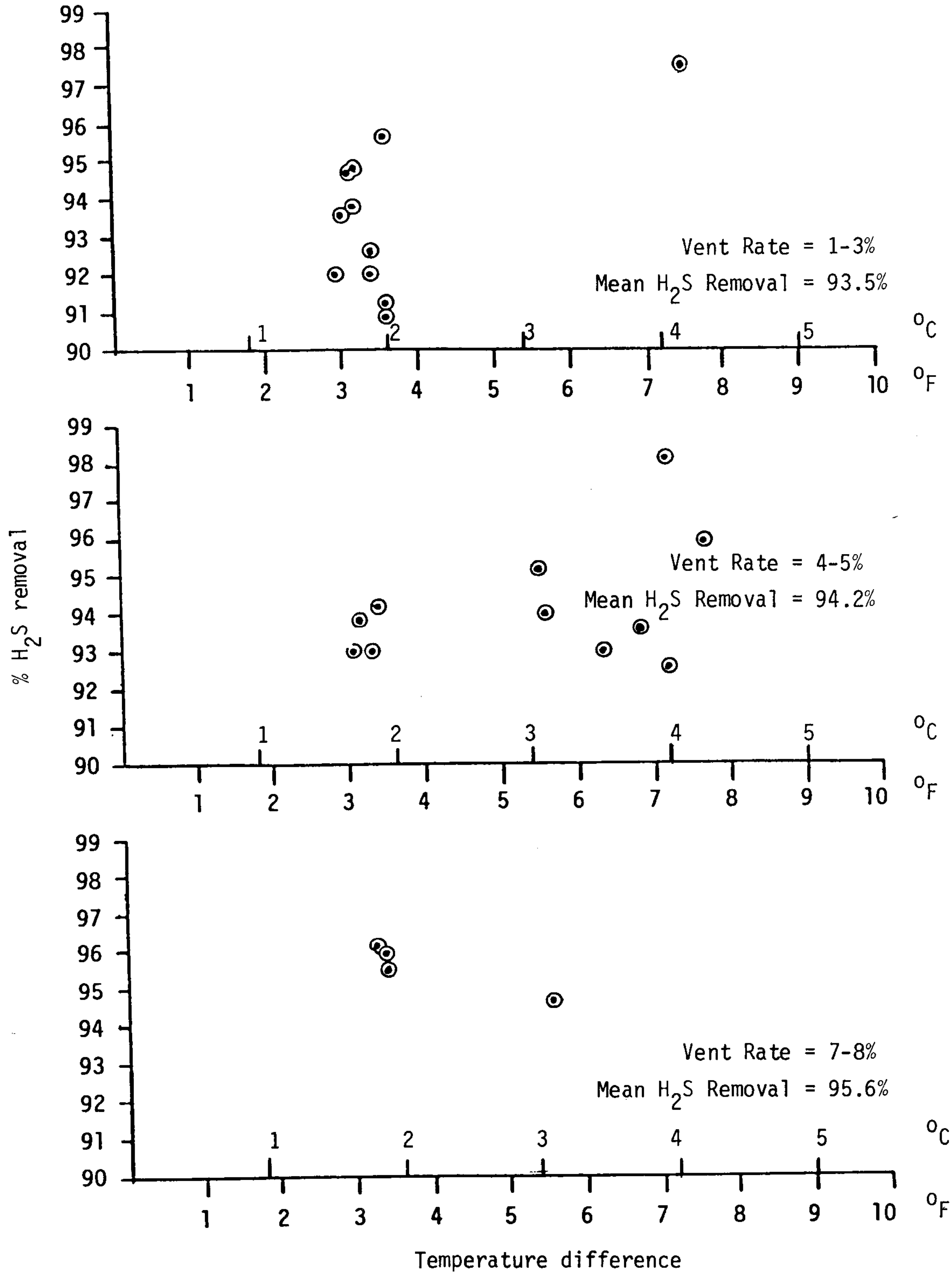


Figure 3-5. Test Unit Performance: H_2S Removal Versus Temperature Difference

probably be explained by the fact that they depend on the water mass balance closures, which were as much as 26% off, and also due to the awkward combination of chemistry analysis techniques that had to be used for determining the relatively high H_2S concentrations in the vent stream, as discussed in Appendix C.

Appendix F also presents data related to removal of other noncondensables. The measured removal of NH_3 ranged from 0% to 60%. The removal rate of the total noncondensables, which is mostly CO_2, could not be measured directly because the clean steam concentrations were not detectable (the required gas bubbles, as described in Appendix C, could not be collected). Because no bubbles could be seen and because the calculated mass flow rates of the total noncondensables in the vent stream were greater than in the inlet stream, it appears that almost all of the noncondensables were removed. The estimated removal is 98 percent.

The most significant aspects of the experimental results with respect to H_2S removal are that throughout the field tests the H_2S removal rate was greater than 90% (except for one data point) and that most of the H_2S removal data were significantly greater than 90%, averaging 94%. With respect to removal of other noncondensables, the removal rates were very high for CO_2, approaching 100%; however, the measured removal of NH_3 was quite variable.

HEAT TRANSFER PERFORMANCE

The coefficient of heat transfer (U) values calculated from field measurements are tabulated in Appendix B. This heat transfer coefficient is defined as follows:

$$U = \frac{Q}{A\Delta T} \qquad (3\text{-}2)$$

where Q = heat transfer rate, shellside to tubeside
A = tube surface area
ΔT = shellside to tubeside temperature difference

Appendix C describes the method used for calculating the U values from the field data. These values, shown as 148 data points, ranged from 1889 to 4471 $W/(m^2 \cdot {}^oC)$ [333 to 788 $Btu/(h \cdot ft^2 \cdot {}^oF)$], with an average value of 3268 $W/(m^2 \cdot {}^oC)$ [576 $Btu/(h \cdot ft^2 \cdot {}^oF)$] and a standard deviation of 482 $W/(m^2 \cdot {}^oC)$ [85 $Btu/(h \cdot ft^2 \cdot {}^oF)$]. As noted in Appendix B, the relatively low U values measured during test run numbers 150 and 151 were not included above because of extreme water mass balance problems experienced during these runs.

The measured U values are shown plotted with respect to vent rate and ΔT in Figure 3-6. As can be seen in this figure, correlations between the measured U values and the vent rate and ΔT cannot be obviously shown from the field data. Intuitively, the U value would be expected to increase as either the vent rate or ΔT was increased due to a decrease in the noncondensable blanketing effect, either by purging the shellside of the heat exchanger or by increasing the turbulence on the shellside because of the higher flow rates associated with the higher ΔT.

During the high vent rate tests, with vent rates approximately four times the normal rate (Data Sets 126, 129, 152, and 153 in Appendix B), the resulting U values were very close to the overall average U value, further showing a lack of correlation between vent rates and measured U values.

Throughout the test program these values were consistently lower than predicted values as shown in Table 4-2 of Section 4. The formation of films or scale deposits on the heat transfer surfaces was initially suspected to be the reason for the lower test data values. The test unit heat exchanger was chemically cleaned to determine if film or scale formation was causing the lower calculated U values. When comparing the calculated U values before and after chemical cleaning, no conclusive difference could be seen.

Examinations of the test unit revealed several deficiencies in the heat exchanger design and condition that would cause the calculated U values to be significantly lower than the actual U values. The most notable deficiency was the inappropriate design of the lower tubesheet gasket which would allow considerable leakage of condensate from the shellside to the tubeside. This leakage would not affect the actual heat exchanger performance with respect to H_2S removal or heat transfer but would affect the flow rate measurements used in calculating the heat exchanger U value (see Appendix C) in a conservative manner so that the measured U values shown in Appendix B are probably lower than the actual U values.

The most important result with respect to heat transfer performance is that field data are now available for calculation of the heat transfer coefficient of a vertical tube evaporator in this geothermal application and these calculations are likely conservative.

MASS AND ENERGY BALANCES

Mass and energy balances for six select cases are shown in Appendix F. These include both baseline tests and special tests during which the inlet steam composition was modified by injecting H_2S and NH_3.

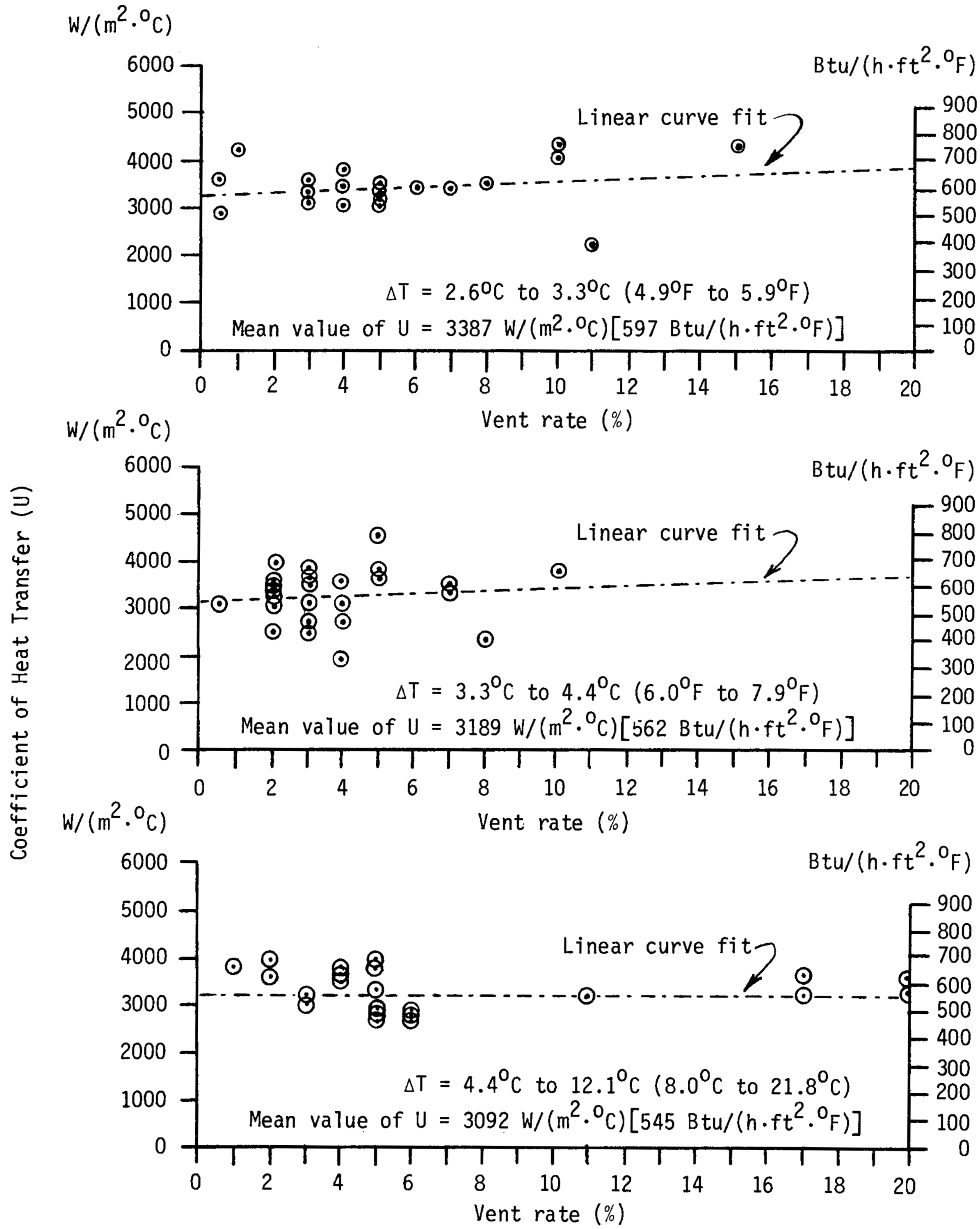


Figure 3-6. Test Unit Performance: Coefficient of Heat Transfer Versus Vent Rate

The diagrams include total mass balance, H_2S mass balance, and heat balance information in addition to detailed analysis of various streams for other noncondensables including NH_3, CO_2, and total noncondensables. The H_2S mass balances and the information on the other noncondensables were discussed earlier in this section. A discussion of the total mass balance and energy balance closures is presented below.

For the six cases the total mass balance closures ranged from a net system loss of 26% to a net system gain of 8%. This can be best explained by limited accuracies in flow measurement methods and instrumentation. The energy balance closures ranged from a net system loss of 23% to a net system gain of 9%. The energy balance is dependent on the total mass balance; therefore, the same degree of lack of closure is expected.

As discussed earlier in this section, it is believed that the measured heat transfer coefficient (U) values are low because of significant internal leakage across the bottom tubesheet. This leakage was visually noted during an examination of the heat exchanger discussed later in this section. The measured U value was based on the measured flow rate from the shellside to the tubeside through the condensate transfer line. Any leakage across the bottom tubesheet would give a false low measurement of the condensate transfer rate, thus indicating a measured U value lower than the actual U value. In completing independent shellside and tubeside mass balances (not shown on the diagrams) the indicated error of the measured condensate transfer flow rate ranged from 31% too low to 11% too high.

The implied error in the condensate transfer rate for most of the cases was not consistent for the independent shellside and tubeside mass balances. Therefore, the mass balances shown in Appendix F do not conclusively support the argument that the measured U values are conservative; however, this is believed to be so based on the visual observations discussed later in this section.

OPERATIONAL CHARACTERISTICS AND RESPONSE TO TRANSIENTS

As predicted, the operation of the test unit proved to be very simple. The only function automatically controlled was the sump level; all other control functions were manually set. A stable steady-state operating condition was easily obtained within a few minutes when changing the control parameters of the pilot plant (vent and clean steam flow rates) within the intended operating ranges of the pilot plant. Cold startup required a warm-up period between 30 minutes and an hour, after which time the desired steady-state operating condition was obtained within a few minutes. The pilot plant would easily accept normal fluctuations in inlet steam pressures and temperatures. As previously discussed, the baseline tests included several periods of continuous operation lasting for several days with operator attention during only a few hours a day.

The test unit did demonstrate an unusual operating characteristic during the high ΔP test. As previously discussed, the sump level dropped very rapidly during this test. The test unit was operating at a flow rate of about five times the normal rate during this test and it is possible that the condensate transfer line could not keep up with the condensing rate.

One objective of the high ΔP test was to evaluate the test unit's performance as a silencer. The only observation that can be made with respect to the performance as a silencer is that, at both a normal operating ΔP of about 110 kPa (16 psi) and the high ΔP of 365 kPa (53 psi) experienced during the test, the noise generated inside the heat exchanger could not be heard above existing operational noise at Unit 7.

During all of the transient tests, as described in Appendix E, the test unit performed very well and did not demonstrate any transient characteristics that would be detrimental to the heat exchanger operation or to power plant operation when installed upstream of a turbine generator unit. The heat exchanger responded smoothly and maintained stable operation during all tests except the Clean Steam Valve Sudden Open test; during this test, there was a false indication of high sump level (although the sump level had not actually changed significantly) which caused the test unit to trip. This type of false indication is a common phenomenon in power plant applications when vessels containing saturated liquid are rapidly depressurized. This problem can be corrected by appropriately sizing the sump with respect to rise rates and capacity and by incorporating time delays in the high level switches.

PERFORMANCE OF EQUIPMENT AND MATERIALS

Corrosion

Inspections of the test unit revealed no evidence of corrosion on the 304 SS (stainless steel) components of the heat exchanger vessel or the titanium tubes, with one exception. This exception involved pitting corrosion, or weld decay, which was discovered at one weld location inside the sump. A detailed analysis (1) of this corrosion attributed the problem to improper welding procedures. Specifically, 316 SS weld material was used to weld the 304 SS components. All other stainless steel welds inside the vessel show no signs of corrosion.

The localized pitting corrosion did not affect the integrity of the vessel during the test program. It was recognized, however, that if there were a slow progression of this pitting, it could cause problems if the vessel was used for an extended period of time. Accordingly, the vessel was inspected again about a year later. No additional corrosion was found.

As expected, the carbon steel piping components of the test unit showed some evidence of minor corrosion (such as rust forming in the recirculation line); however, no serious corrosion of the carbon steel piping components was experienced.

Tube Fouling

Early in the test program, a visual inspection of the heat exchanger tubes indicated a dirt-like film on the outside of the tubes. Baseline testing was continued. Then, after performance had been established by several months of testing, the heat exchanger was chemically cleaned using hot solutions of $KMnO_4$ and NaOH followed by citric acid. The purpose of the cleaning was to see if the measured heat transfer coefficients would be higher after cleaning, thus indicating the presence of significant scaling on the tube surfaces before cleaning. This cleaning procedure was chosen on the basis of laboratory tests that were made using a very small amount of deposit scraped from the outside of the tubes and from inside of the vent line. The chemical cleaning resulted in no conclusive change in measured heat transfer coefficients; thus, the existence of significant tube fouling did not appear to exist prior to the chemical cleaning.

A few months after the conclusion of the test program, the tube bundle was removed from the heat exchanger and specimens of the tubes were taken for analysis. The only visual evidence of film deposits on the tube walls was the existence of what appeared to be a very thin blue "film" on the outside of the tubes and thin brown "film" on the inside. These "films" and other minor deposits collected from the tube bundle and from inside the heat exchanger were analyzed by an outside laboratory (2) with the following results:

- The blue "film" on the outside of the tube specimen was too thin to analyze. It was explained as probably being heat discoloration.
- The brown "film" inside the tube specimen was found to be less than 1000 angstroms thick and was identified as an organic material. This is probably the result of the deterioration of the plastic flow distributor, as discussed below.
- The collected deposits were shown to be various corrosion and oxidation products of iron and sulfur, probably occurring in the carbon

steel steam lines upstream of the test unit, being caused by the occasional exposure to oxygen when the system is open to atmosphere.

Flow Distributors

Two types of plastic flow distributors were used in the tops of the heat exchanger tubes during the test program. The first distributor material used was nylon, which completely disintegrated early in the test program. The nylon distributors were replaced with distributors made of a high temperature plastic, Riton. These lasted throughout the remaining portions of the test program without showing any signs of significant deterioration. The Riton distributors did become embrittled by the exposure, but did not appear damaged in any way that would affect their normal function.

Gaskets and Seals

Several types of elastomer materials, including Viton, were used as gaskets and internal seals. All of the elastomer materials seemed to be unsuitable for the application, probably due to the elevated temperatures. In future applications, alternative seal designs will be used to resolve this problem.

The most significant seal problem was at the lower tubesheet. Leakage at this location resulted in some condensate not flowing through the transfer line where the condensing rate measurement was made. As a result, the calculated heat transfer coefficient values were lower than the actual values. Thus, the heat transfer performance measurements of the test unit are thought to be conservative; that is, lower than actually achieved.

Tube Bundle Deficiencies

Early in the test program it was discovered that 2 of the 50 tubes were crushed, probably due to excessive pressure during the hydrostatic pressure tests. The test program was continued with the crushed tubes, with the understanding that the heat transfer performance might be adversely affected. The heat transfer coefficient would be reduced by 4% if these tubes provided no usable heat transfer area.

Examinations of the heat exchanger after the conclusion of the test program indicated that significant leakage was occurring between the tubes and lower tubesheet. This leakage would increase the effect of the lower tubesheet seal leakage, as previously discussed, causing the calculated heat transfer coefficient to be lower than the actual value.

Pump Failure

The test program was interrupted by the failure of the recirculating condensate pump and motor. Two independent problems were involved: failure of the pump seal, and water intrusion into the pump motor. Both of these problems were independent of the specific process application and could have been prevented by better equipment specification and selection.

Chemistry Analysis for Future Selection of Materials

To aid in the selection of materials for subsequent systems, periodic pH analysis and special laboratory analyses for various species in the various flow streams were conducted. The resulting pH ranges were as follows:

- Inlet steam pH--6.0 to 7.1
- Vent pH--6.3 to 7.6
- Clean steam pH--8.8 to 9.4
- Condensate transfer pH--8.5 to 9.0
- Recirculating condensate pH--7.1 to 7.5

Appendix G, "Special Laboratory Analysis for Ammonia, Boron, and Chloride," presents the results of special analyses conducted for these species in various flow streams of the test unit.

REFERENCES

1. Boeing Material Laboratories. Report to Coury and Associates. Seattle, Washington, October 1979.

2. Coors Spectro-Chemical Laboratory. Report to Coury and Associates. Golden, Colorado, December 1980.

Section 4

EVALUATION OF ADVANCED HEAT TRANSFER DESIGNS

INTRODUCTION

The heat exchanger type used in the test unit at The Geysers, as discussed in Section 3, is the vertical tube falling-film evaporator (VTE) with smooth tubes. Since the heat exchanger represents as much as 75% of the total capital equipment cost for the process, it is desirable to consider alternative heat transfer designs which could reduce the heat exchanger size and cost. Two other designs have been considered: (1) a falling-film VTE with doubly fluted tubes, and (2) a horizontal tube evaporator (HTE) with smooth tubes.

The purpose of this section is to compare the predicted performance and cost of the fluted-tube VTE unit and the HTE unit with those for a VTE with smooth tubes. A literature review was made of heat exchanger design and performance data to estimate comparative overall heat transfer coefficients (U). Costs were then determined based upon information provided by manufacturers of tubing and heat exchangers.

Although results of this study are limited because of the lack of data on heat transfer coefficients at the condition of interest, some comparison of performance and costs has been possible. No significant cost improvement was seen in a VTE with fluted tubes over a VTE with smooth tubes. This is due to the high thermal conductivity of the stainless steel and titanium tubes that are used with geothermal steam, which in turn result in large wall resistances for the fluted tubes. On the other hand, it appears that HTEs can achieve an increase in heat transfer coefficient of approximately 50%. However, the cost quotations received for an HTE unit were double those for comparable vertical tube exchangers. The high HTE costs can not be justified on the basis of equipment design or complexity and may be due solely to the previously noncompetitive supply situation; it is expected that these costs will decrease substantially because other manufacturers are beginning to supply HTE units.

VERTICAL TUBE EVAPORATOR WITH FLUTED TUBES

Doubly fluted tubes were developed by the desalination industry to increase the heat transfer coefficients over the smooth-tubes VTE unit. Doubly fluted tubes are fabricated with ridges both on the inside and outside tube surfaces. Although there are a number of different configurations, a common doubly fluted tube is shown in Figure 4-1.

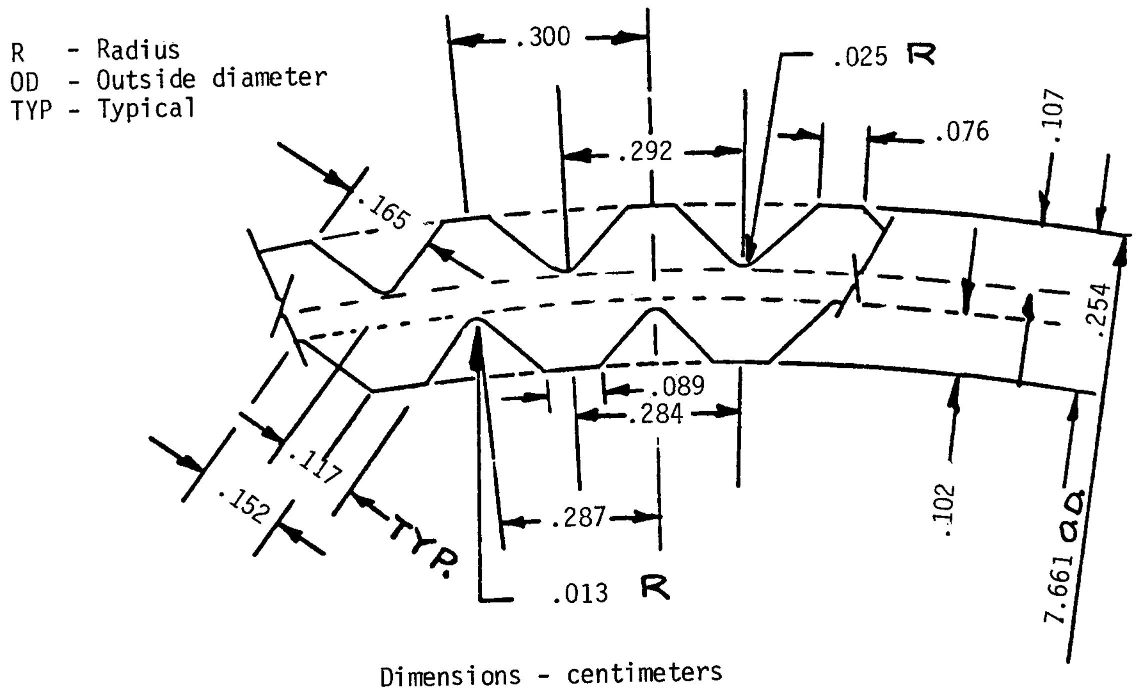


Figure 4-1. Cross Section of a Doubly Fluted Heat Exchanger Tube

Source: Reference 1.

The major advantages of the doubly fluted tubes are that the condensing heat transfer coefficient is greatly improved. This is due to surface tension effects that cause most of the condensate to flow through the channels, leaving the ridge area with a very thin condensate layer that has a very low resistance to heat transfer.

Extensive testing of heat transfer characteristics of doubly fluted tubes in VTE units has been conducted by the then Office of Saline Water (OSW), U.S. Department of the Interior. Most of their work was done in desalination applications with copper, aluminum/brass and 90:10 copper/nickel alloys (Table 4-1).

Table 4-1

HEAT TRANSFER PERFORMANCE OF DOUBLY FLUTED TUBES

Tube Material	Thermal Conductivity		Experimental U	
	$W/(m^2 \cdot {}^oC)$	$[Btu/(h \cdot ft^2 \cdot {}^oF)]$	$W/(m^2 \cdot {}^oC)$	$[Btu/(h \cdot ft^2 \cdot {}^oF)]$
Copper	1350	(238)	14,750	(2600)
Al/brass	330	(58)	10,500	(1850)
90:10 Cu/Ni	150	(27)	7,940	(1400)

Basis:

Fluid media -- seawater, steam

Tube flute profile -- ORNL #9/80/80

SFO* -- 1.35

Nominal 3" OD; L/D = 8

Nominal wall thickness -- 1.65 mm (0.065 in) before fluting
-- 1.24 mm (0.049 in) fluted

Evaporation temperature -- 71^oC (160^oF)

Condensing temperature -- 82^oC (180^oF)

*Surface Factor Outside--ratio of true outside surface area to surface area of comparable smooth tube with same OD

Source: References 1, 2.

The results indicated that the overall transfer coefficients for fluted tubes were on the order of 100% higher than for smooth tubes. The data also showed that the thermal conductivity of the tube wall had an important effect on heat transfer coefficients. This is mainly attributed to a longer heat transfer path through the wall of fluted tubes than for smooth tubes. Resistance to heat flow is therefore highly dependent on thermal conductivity.

The overall heat transfer coefficient for conditions of interest in applying this design to geothermal steam applications--177^oC (350^oF) steam with 304 SS materials of construction--can not be readily calculated because there are no adequate models or generalized correlations for evaluating fluted-tube film coefficients. However, extrapolations of data from lower-temperature tests can be used to compare the heat transfer performance of the two types of tubes at geothermal temperatures.

The basis for estimating U for the required condition is that the overall heat transfer coefficient is the same for different materials if the resistance of tube walls is neglected. A simplified equation for U can then be developed of the form:

$$\frac{1}{U} = \frac{1}{h^*} + \frac{\alpha x}{12k} \qquad (4\text{-}1)$$

where h^* = net cumulative film coefficient accounting for heat transfer through liquid film on both sides of the tube wall

αx = effective wall thickness for a fluted tube of actual thickness x

k = thermal conductivity of the tube wall

If it is assumed that the effective wall thickness of all tube materials is the same, then by curve fitting heat transfer data for the three materials in Table 4-1, a net film coefficient, h^*, can be calculated. To extrapolate h^* to 177°C (350°F) a design correlation of U versus temperature data presented by Coury (2) for 90:10 Cu/Ni alloy is used. Based on this approach, U for 304 SS at 177°C (350°F) is estimated to be about 4426 $W/(m^2 \cdot {}^\circ C)$ [780 $Btu/(h \cdot ft^2 \cdot {}^\circ F)$] for 51-mm (2-in) OD tubes.

The cost basis for VTEs with fluted tubes can be developed by combining smooth-tube VTE costs with the incremental cost required for fabricating fluted tubes. Costs in 1979 dollars for falling-film VTEs is $193.77 per sq m ($18 per sq ft)* surface area based upon construction of large units using 304 SS. Fluted tubes of 304 SS construction cost approximately $4.92 per m ($1.50 per ft)* more than smooth tubes of 304 SS. Therefore, for 51-mm (2-in) OD tubes, the cost of fluting adds $32.29 per sq m ($3 per sq ft) to the basic heat exchanger cost, yielding a total estimated cost of $226.06 per sq m ($21/sq ft).

HORIZONTAL TUBE, SPRAY-FILM EVAPORATOR

In the HTE spray-film unit, the geothermal steam is introduced on the tubeside and condensate on the shellside. The condensate would be sprayed over the outside of

*Personal communications: Resources Conservation Company, Grob Tube Company, and Aqua-Chem, Inc.

the tubes, and the steam would condense within the tubes and flow out of the ends. Figure 4-2 shows an HTE configuration for the heat exchanger process.

The major advantage of the HTE is that the heat transfer coefficient is significantly improved over a smooth tube VTE design even while using smooth tubes in the horizontal unit. The primary gain is due to the improved condensing side coefficient. This occurs because of a reduced overall film thickness as the condensate that is formed collects along the bottom of the tube, leaving a thin film over the rest of the tube surface.

Most of the published information on HTEs was developed under OSW research and development programs. The work was done for typical desalination applications with temperatures in the range of 38^{o} to $127^{o}C$ (100^{o} to $260^{o}F$). Although data indicate that very high heat transfer coefficients may be attainable, there is some question as to how to apply these results to commercial applications. The HTE performance is very sensitive to such factors as tube spacing and bundle design. However, definitive design and test data on these parameters are not available.

To estimate a heat transfer coefficient for the spray-film HTE at conditions of interest for geothermal steam, a method of approximation was used similar to that for the doubly fluted tube case. Experimental data presented by Coury (2) show that expected values of U for an HTE with 90:10 copper/nickel tubes 76-mm (3-in) OD, 0.48-mm (0.019-in) wall thickness at $177^{o}C$ ($350^{o}F$) is 9078 $W/(m^2 \cdot {}^{o}C)$ [1600 $Btu/(h \cdot ft^2 \cdot {}^{o}F)$]. Adjusting this heat transfer coefficient to account for the wall resistance differences for 304 SS tubes [51-mm (2-in) OD, 1.24-mm (0.049-in) wall thickness], a new U value for 304 SS at $177^{o}C$ ($350^{o}F$) is calculated to be 6241 $W/(m^2 \cdot {}^{o}C)$ [1100 $Btu/(h \cdot ft^2 \cdot {}^{o}F)$].

For HTE smooth tube units constructed of 304 SS in the size range of interest, fabricated costs were estimated to be $114.64 per sq m ($52 per sq ft)*. This relatively high cost in comparison to the cost for smooth tube VTE units may be due, in part, to design differences. For instance, the HTE requires greater spacing between tubes to minimize pressure loss as the vapor is disengaged from the tube bundle, and also requires more space for spray nozzles or distributors. However, a three-fold increase in cost over VTE units can not be explained.

---

*Personal communications: Resources Conservation Company, Grob Tube Company, and Aqua-Chem, Inc.

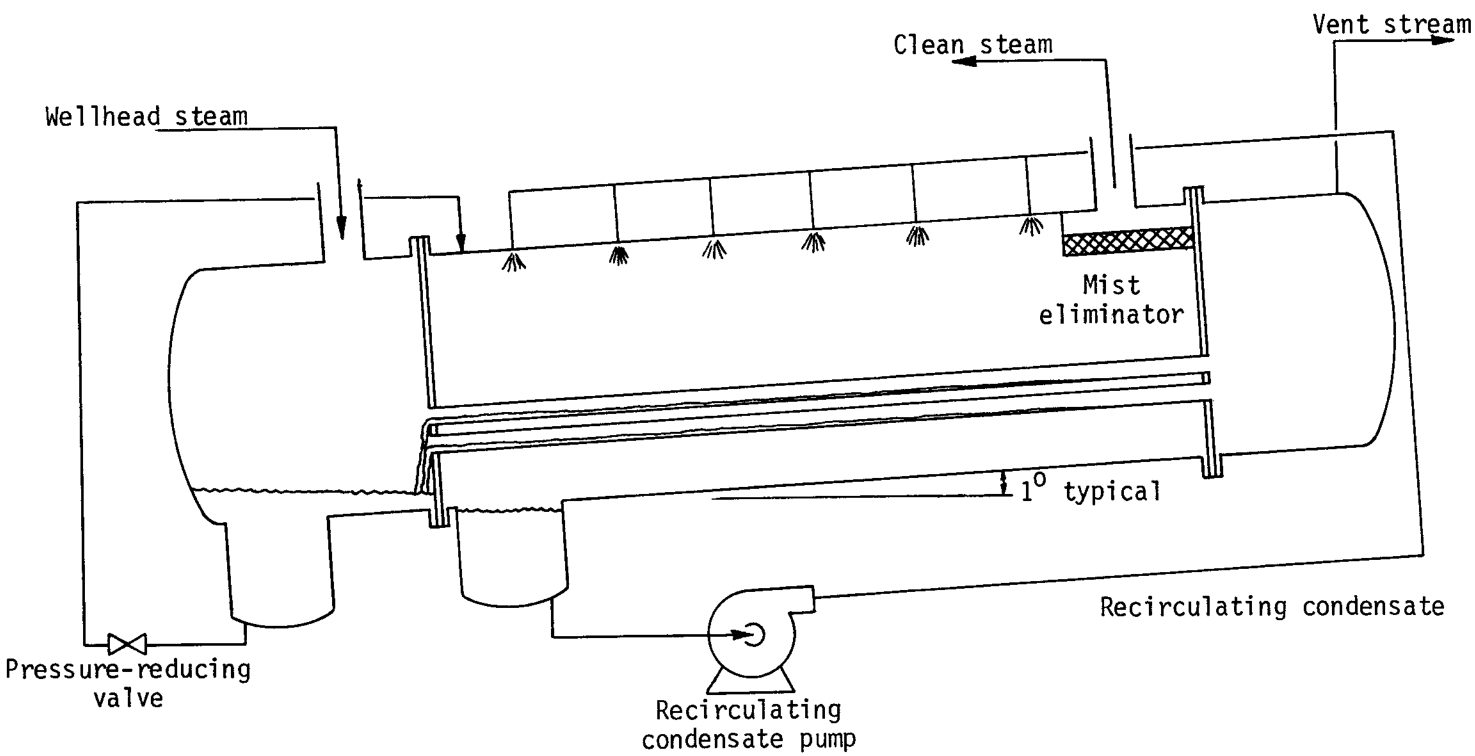


Figure 4-2. Horizontal Tube Evaporator Configuration

COMPARISON OF PREDICTED PERFORMANCE AND COSTS

The application of the heat exchanger process to a 55-MW geothermal power plant will provide the basis for comparing the alternative heat exchanger designs. The process will be assumed to treat saturated steam at 177°C (350°F) and a maximum pressure of 1034 kPa (150 psig). The required heat transfer area for the heat exchangers will be based on condensing 139 kg/s (1.1×10^6 lb/h) of steam at a geothermal steam/clean steam temperature difference of 5.6°C (10°F). Tube dimensions in all exchanger designs are 51 mm (2 in) OD with a 1.24-mm (0.049-in) wall thickness. The material of construction is 304 SS. The three designs compared will be the VTE with smooth tubes, VTE with doubly fluted tubes and the HTE with smooth tubes.

Table 4-2 shows a comparison of calculated heat transfer areas and costs for the above conditions. The heat transfer coefficients therein do not strictly take into account the effects of fouling and noncondensable gases. Rather, these coefficients and other data shown in the table should be considered as relative information useful only for the purpose of making comparisons between different heat exchanger designs.

The comparison of data between VTE units with smooth tubes and those with fluted tubes indicates that there is very little difference in performance or cost within the level of accuracy of this estimate. The anticipated improvement in overall U for fluted tubes was minimized by the high thermal conductivity of 304 SS which resulted in large tube wall resistances for the fluted tube. The same conclusions apply to titanium--another acceptable tube material for this application--since its conductivity is about the same as that of 304 SS.

The HTE smooth tube design appears to be significantly different in both performance and cost when compared to VTE units. The required heat transfer area for an HTE unit is about two-thirds of that for the vertical tube exchangers. Costs for the horizontal exchanger, on the other hand, are about double those for the VTE units. These large cost differences are due to the high estimated costs per m^2 of heat transfer area.

Table 4-2

COMPARISON OF PREDICTED HEAT EXCHANGER PERFORMANCE AND COSTS

Unit	Tubes	Overall Heat Transfer Coefficient $\frac{W}{m^2 \cdot {}^{o}C}$	$\left(\frac{Btu}{h \cdot ft^2 \cdot {}^{o}F}\right)$	Total Heat Surface Area $10^3\ m^2$	$(10^3\ ft^2)$	Heat Exchanger Cost $\frac{\$}{m^2}$	$\left(\frac{\$}{ft^2}\right)$	Total Heat Exchanger Costs $\$10^6$
VTE	Smooth	4199	(740)	11.0	(120)	194	(18)	2.2
VTE	Fluted	4426	(780)	10.5	(110)	226	(21)	2.3
HTE	Smooth	6241	(1100)	8.2	(80)	560	(52)	4.2

Basis:

1. Steam conditions: 139 kg/s (1.1×10^6 lb/h) steam condensed
   177°C (350°F) saturated steam

2. U for VTE smooth tubes estimated from experimental data for 90:10 copper/nickel tubes (1, 2) corrected for differences between heat transfer tube wall resistance in 304 SS and 90:10 copper/nickel.

3. Total heat exchanger surface area calculated from $A = Q/U/\Delta T$

$$\text{where } Q^* = m\Delta H_{vap}^{**} = 139 \text{ kg/s} \times 2.02 \times 10^6 \frac{W \cdot s}{kg}$$

$$= 2.81 \times 10^8 \text{ W}$$

U = overall heat transfer coefficient as defined in table above

$\Delta T = 5.6^oC\ (10^oF)$

---

* $$Q = m\Delta H_{vap} = \frac{1 \times 10^6 \text{ lb}}{h} \times \frac{871 \text{ Btu}}{16}$$

$$= 871 \times 10^6 \text{ Btu/h}$$

** m = mass flow rate

ΔH_{vap} = latent heat of vaporization

REFERENCES

1. A. E. Dukler, L. C. Elliott, and A. L. Farber. Distillation Plant Data Book. Houston, Texas: Houston Research, Inc. for Office of Saline Water, January 1971. Contract No. 14-01-0001-2099.

2. G. E. Coury. Heat Exchanger Design for Desalination Plants (Review of the OSW/R&D Program). ORNL, December 1977.

Section 5

2.5-MW PILOT PLANT

This section presents the suggested design for a 2.5-MW pilot plant. The major parts of the section include the objectives of the pilot plant test program; a brief system description of the pilot plant; the pilot plant design basis; the plant utility requirements; equipment lists and specifications; estimated pilot plant costs; and a proposed schedule for erection of the pilot plant and the pilot plant test program. Data from this pilot plant will provide the design basis for subsequent commercial-scale systems.

Due to the nature of the material presented in this section and its supporting appendices, much of the data related to design specifications are shown in English units only.

TEST PROGRAM OBJECTIVES

The objectives of the pilot plant test program are to provide comprehensive design data for:

- H_2S removal, noncondensables removal and heat transfer under various operating conditions
- Design features necessary for best system performance
- Equipment serviceability under unattended operation
- Response to transient and upset conditions
- Operating and capital costs of commercial-scale applications

SYSTEM DESCRIPTION

The pilot plant as depicted in Figure 5-1 will be a two-stage system similar to the commercial-scale design shown in Section 6. It will consist of two heat exchanger units and a vent gas condenser operated in series.

The first-stage heat exchanger unit includes a shell-and-tube heat exchanger, a condensate transfer tank, a heat exchanger sump, and two recirculating condensate pumps. Wellhead steam enters the shellside of the heat exchanger. Clean steam is discharged

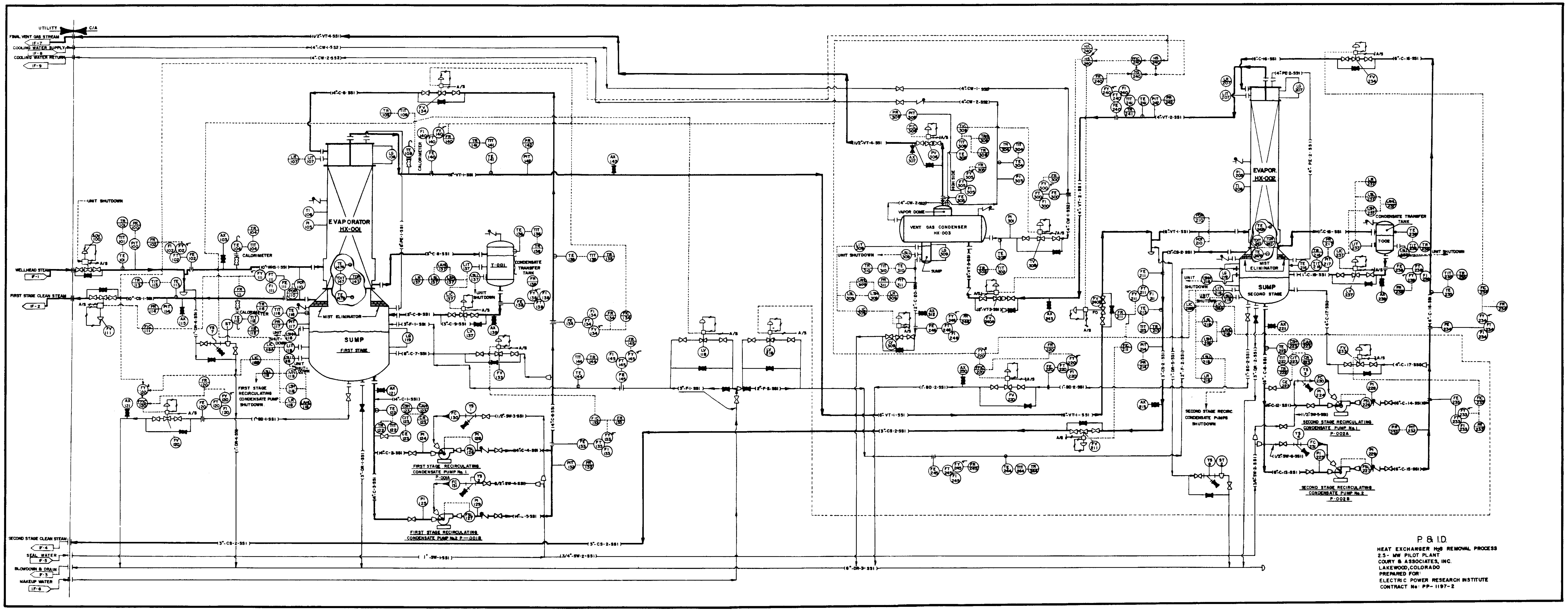


Figure 5-1. P&ID, Heat Exchanger H_2S Removal Process, 2.5-MW Pilot Plant

from the sump. The vent stream exits from the shellside of the exchanger and goes to the second stage. The heat exchanger operating principles are discussed in detail in Section 2.

The second-stage heat exchanger unit also includes a shell-and-tube heat exchanger, a condensate transfer tank, a heat exchanger sump, and two recirculating condensate pumps. The equipment is basically identical to that of the first-stage heat exchanger except for its size. The second-stage heat exchanger functions like that of the first stage, except that it operates with different vent rates and produces a "clean" steam of slightly lower pressure and possibly slightly higher H_2S concentrations.

The vent gas condenser is a shell-and-tube heat exchanger which uses cooling water on the tubeside. The condenser cools the vent stream from the second-stage heat exchanger to 49°C (120°F) resulting in two effluent streams: a liquid condensate stream and a gas stream which contains most of the total system incoming noncondensables.

Utility requirements for the pilot plant include makeup water, cooling water, drains for discharging blowdown and vent condenser condensate, pump seal water, electricity, and instrument air.

DESIGN BASIS

A process flow diagram for the pilot plant is shown in Figure 5-2. The process design is based on the following criteria:

- Proposed location--The Geysers Power Plant in northern California
- Environmental conditions:

  --Altitude - 983 m (3225 ft) MSL

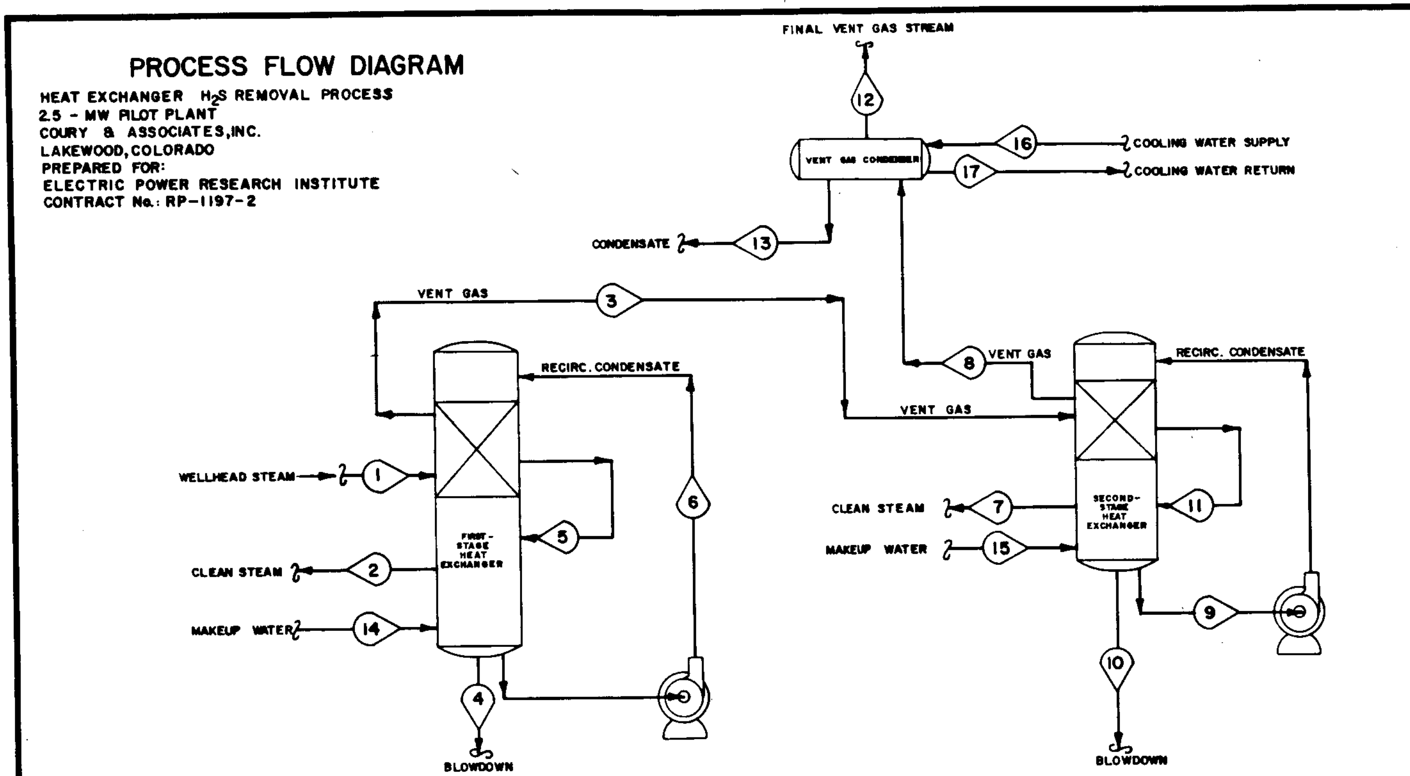
  --Barometric pressure - 89.6 kPa (26.48 in, Hg - 13.0 psia)

  --Maximum outdoor temperature - 49°C (120°F)

  --Minimum outdoor temperature - -18°C (0°F)

  --Seismic loading - 0.2 g horizontal, 0.13 g vertical

  --Special problems - very corrosive atmosphere, significant concentrations of H_2S combined with frequent moisture precipitation



† Based on maximum recirculation rate.

STREAM No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
TEMP. °F	347	337	346	337	347	337	336	344	336	336	346	120	120	70	70	85	135
PRESSURE, p.s.i.a.	129	113	127	113	129	113	112	125	112	112	127	18	18	113	112	63	151
G.P.M.	—	—	—	1.11	112	3,000†	—	—	426 †	0.03	2.55	—	2.44	1.06	0.02	54.0	54.6
LBS per HOUR	52,600	50,000	2,630	500	50,000	1,350,000†	1,140	1,490	191,000†	11.4	1,140	280	1,210	530	11.4	26,900	26,900
H_2S CONC., PPM.	250	12.5	—	—	—	—	26.3	—	—	—	—	40,200	1,030	—	—	—	—
TOT. NC, WT. %	0.5	0.01	—	—	—	—	0.01	—	—	—	—	96	—	—	—	—	—

Figure 5-2. Process Flow Diagram

- Process design basis (based on proposed 55-MW commercial application as defined in Section 6, with flow rates scaled down as required by ratio of 2.5:55)

  Wellhead steam

  --Temperature - 175°C (347°F)

  --Pressure - 889 kPa (129 psia)

  --H_2S concentration - 250 ppm

  --NH_3 concentration - 125 ppm

  --Total noncondensable concentration - 0.5%

  Clean steam from first-stage heat exchanger

  --Temperature - 169°C (337°F)

  --Pressure - 779 kPa (113 psia)

  --Quality - dry saturated steam

  --H_2S concentration - 12.5 ppm (based on 95% reduction of inlet concentration)

  --Total noncondensable concentration - 0.01%

  --Full load flow rate - 6.3 kg/s (50,000 lb/h)

  Clean steam from second-stage heat exchanger

  --Minimum pressure - 710 kPa (103 psia)

  --Total flow rate - 0.14 kg/s (1136 lb/h)

  --Total noncondensable concentration - 0.01%

  --Quality - dry saturated steam

  Vent stream from second-stage heat exchanger

  --H_2S concentration - as required to assure a minimum total system H_2S removal of 95%

  Vent stream from vent gas condenser

  --Maximum pressure - 124 kPa (18 psia)

  --Maximum temperature - 49°C (120°F)

  --Stream composition - primarily noncondensables with large concentrations of H_2S and some water vapor

Utility cooling water

--Temperature - 29°C (85°F)

--Maximum pressure drop through vent gas condenser - 83 kPa (12 psia)

Heat transfer coefficients

--First-stage heat exchanger - 2800 $W/(m^2 \cdot {}^\circ C)$ [500 $Btu/(h \cdot ft^2 \cdot {}^\circ F)$]

--Second-stage heat exchanger - 1100 $W/(m^2 \cdot {}^\circ C)$ [200 $Btu/(h \cdot ft^2 \cdot {}^\circ F)$]

Condensate recirculating rates

--Maximum - 0.13 l/s/tube (2 gpm/tube)

--Normal - 0.05 l/s/tube (3/4 gpm/tube)

--Minimum - 0.03 l/s/tube (1/2 gpm/tube)

Heat exchanger ΔT's

--5.6°C (10°F) at full load

Heat exchanger blowdown rates

--Maximum of 1% of clean steam flow rates

Vent stream from first-stage heat exchangers

--Total flow rate - 5% of well steam flow rate

The major instrumentation and control elements for the pilot plant are presented in the piping and instrumentation drawing shown in Figure 5-1. The plant has been instrumented to provide fully automatic operation and control, and to demonstrate all the test objectives stated earlier. A data collection system is also provided and includes a data logger and strip chart recorders to facilitate data collection and analysis. The major control concepts for the process heat exchangers are discussed in Section 2. The basic control requirements for the vent gas condenser consist of sump level control, pressure control on the effluent gas vent line, and temperature control on the cooling water line.

UTILITY REQUIREMENTS

Utility requirements for the 2.5-MW pilot plant are summarized as follows:

- Makeup water - 0.07 l/s (1.1 gpm)
- Cooling water - 3.8 l/s (60 gpm)
- Seal water - 0.25 l/s (4 gpm)

- Power - 40 kW, 480 V, 60 Hz
- Instrument air - 0.013 m^3/s (27 scfm)

EQUIPMENT LISTS AND SPECIFICATIONS

Equipment requirements for the pilot plant are presented in the appendices shown in Table 5-1. Equipment design conditions, applicable standards and codes, and materials of construction are given in these appendices.

ESTIMATED PILOT PLANT COSTS

Table 5-2 contains a summary of the cost estimates for the 2.5-MW pilot plant. Equipment estimates are based on September 1980 cost information supplied by vendors and fabricators. Costs for engineering and construction are preliminary estimates based on installation of similar pilot plant systems requiring a high degree of instrumentation.

Table 5-1

APPENDIX IDENTIFICATION FOR 2.5-MW PILOT PLANT
EQUIPMENT LISTS AND SPECIFICATIONS

Equipment Item	Appendix
Major equipment list	H
First- and second-stage heat exchanger specifications	I
Vent gas condenser specifications	J
First-stage recirculating condensate pump specification	K
Second-stage recirculating condensate pump specification	L
First-stage condensate transfer tank specification	M
Second-stage condensate transfer tank specification	N
Control valve list	O
Manual valve list	P
Line list	Q
General piping system specification	R
Instrument list	S
Additional components list	T
Interface list	U

Table 5-2

SUMMARY OF 2.5-MW PILOT PLANT COSTS

Item	Cost
First-stage heat exchanger	$ 130,000
Second-stage heat exchanger	18,800
Vent gas condenser	30,000
First-stage condensate transfer tank	2,800
Second-stage condensate transfer tank	1,200
First-stage recirculating condensate pumps	4,800
Second-stage recirculating condensate pumps	800
Motor control center	15,000
Manual valves	156,000
Control building	10,200
Instrumentation and control (including valves)	153,000
Data collection system	150,000
Sampling condensers	17,400
Other equipment and materials	19,500
Estimated freight cost	43,200
Equipment subtotal	$ 752,700
Engineering	600,000
Construction	550,000
Total*	$1,900,000

*Rounded to nearest $100,000

SCHEDULE -- ERECTION OF PILOT PLANT AND TEST PROGRAM

A schedule for design, procurement, construction, and startup of the pilot plant and the pilot plant test program is shown in Figure 5-3. It is estimated that the plant will take 14 months to erect. Subsequent testing is expected to require 12 to 16 months.

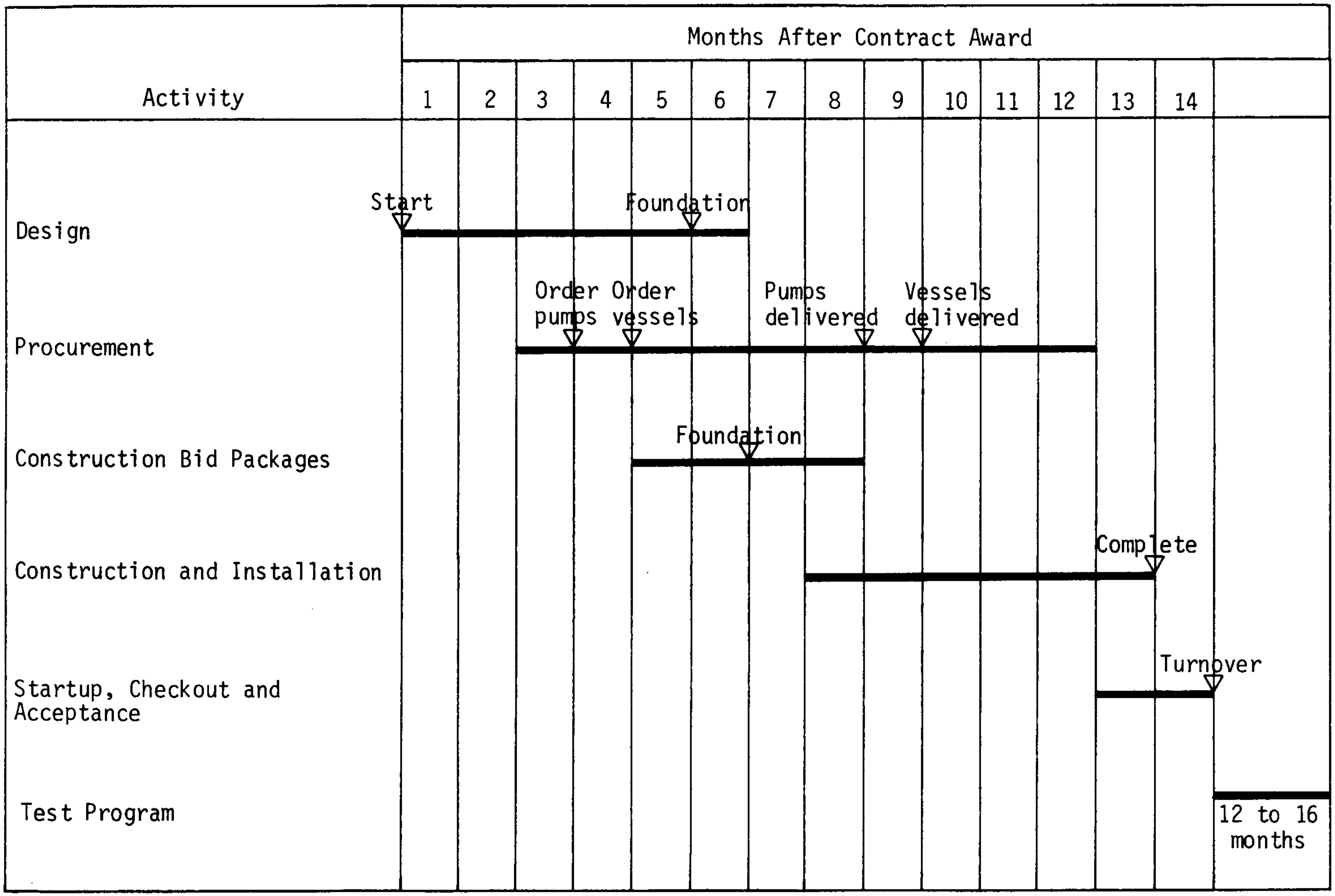


Figure 5-3. Schedule - 2.5-MW Pilot Plant

Section 6

APPLICATIONS AND COMMERCIAL-SCALE COST ESTIMATES

This section discusses applications of the heat exchanger process, with respect to both vapor dominated and hydrothermal applications. A conceptual design of a total H_2S abatement system is presented as a cost model for determining estimated equipment and operating costs of commercial-scale applications, based on a 55-MW geothermal power plant. This section also reviews the potential gains and losses in steam use efficiency, due to the heat exchanger process, and system design variations and options affecting the system costs.

The information presented in this section provides a basis for comparing this process with other H_2S abatement options. In completing an accurate evaluation with respect to other options, both quantitative and qualitative considerations must be included, such as capital and operating costs, system complexity, effects on power plant reliability, effectiveness of H_2S removal and other potential advantages such as stacking applications.

APPLICATIONS

Much of the discussion throughout this report, including this section, is based on applications of the heat exchanger process at The Geysers, both as an H_2S removal system upstream of a power plant turbine and to provide H_2S removal and silencing capabilities while stacking wells directly to atmosphere. Figure 6-1 is an example of a total H_2S abatement system that would be very appropriate at The Geysers, and other locations as discussed later below. This total H_2S abatement system is the basis for the commercial-scale cost estimates presented later in this section. The following is a brief description of the system shown in Figure 6-1.

> This system consists of a two-stage heat exchanger process for removing H_2S and other noncondensables and a Stretford plant for disposal of the removed H_2S.
>
> Geothermal steam enters the first-stage heat exchanger unit and is separated into clean steam and a small vent gas stream. The clean steam is sent to the turbine and the vent gas goes to the second stage. Blowdown from and makeup to the first-stage sump are controlled to limit the buildup of various chemical species in the tubeside condensate.

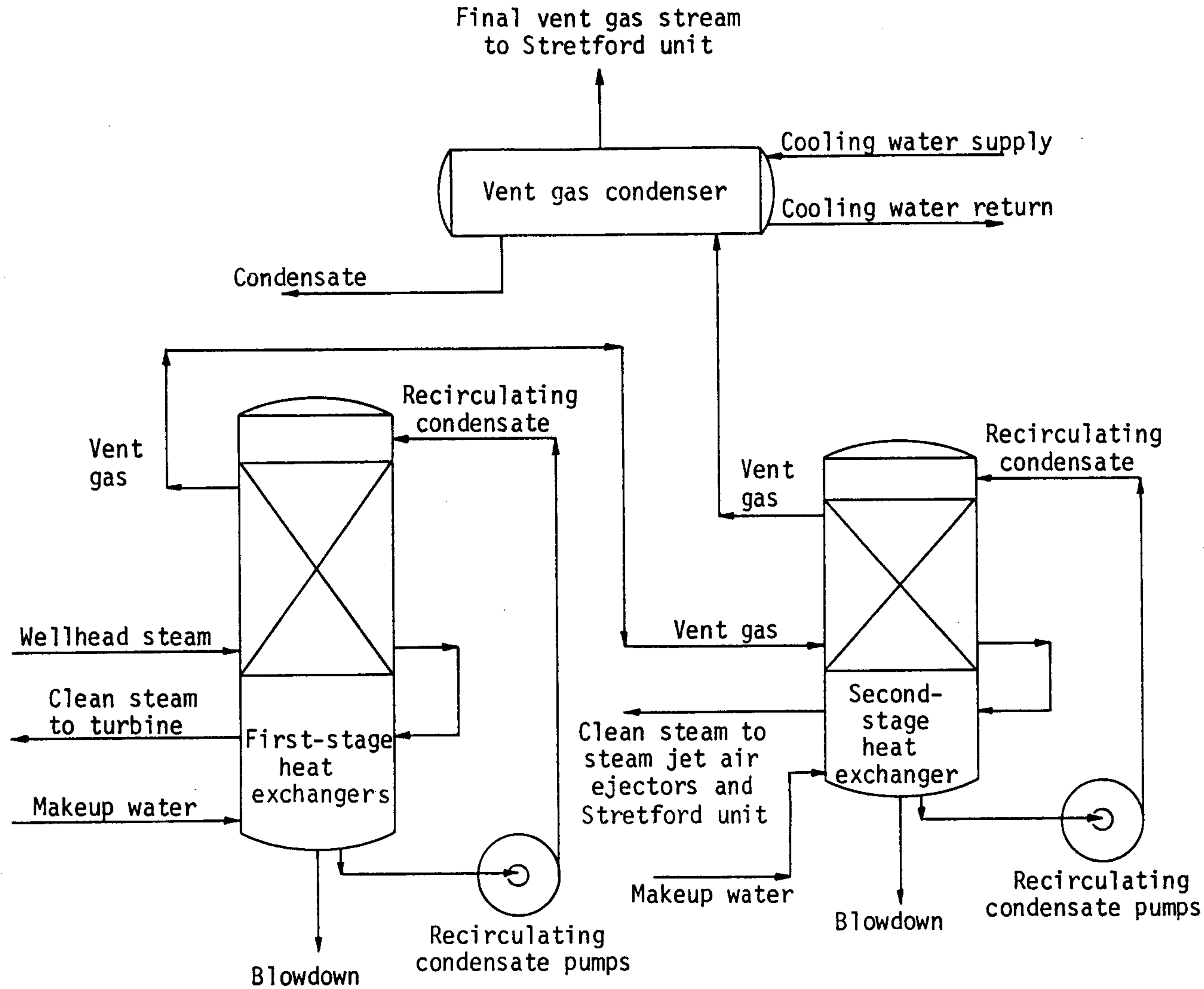


Figure 6-1. Process Flow Diagram, Commercial-Scale Heat Exchanger Process H_2S Abatement System

In a manner similar to that of the first stage, the stream entering the second stage is also separated into clean steam and a vent stream. Clean steam from the second stage is used to supply the after-turbine condenser vacuum system and the Stretford process. Vent gas from the second-stage heat exchanger goes to the vent condenser. The second-stage sump also has provisions for blowdown and makeup.

The vent condenser cools the second-stage vent gas down to temperatures required for discharge to a Stretford unit, normally around 49°C (120°F). The condensate formed in the condenser is injected into disposal wells or discarded by some other means.

The Geysers is a vapor dominated geothermal resource; that is, dry steam is produced from the geothermal wells and can be provided directly to the power plant turbines. Many of the geothermal locations in the world are liquid dominated (hydrothermal) resources. At these locations, steam can be produced to drive power plant turbines by partially flashing the geothermal brine, either in the producing well or in surface flash tanks. In hydrothermal applications, the heat exchanger process would be located on the steam supply to the turbine, downstream of the liquid-vapor separator, to remove H_2S and other noncondensables before they enter the turbine. Also, as in the case of the The Geysers, the system could be used to allow stacking of the flashed steam to keep the hydrothermal wells flowing during periods of brief plant shutdowns.

Most of the proposed hydrothermal power plant applications have operating pressures and temperatures similar to the steam conditions at The Geysers (see Appendix A). On this basis, the design and operation of the heat exchanger process would be very similar at both hydrothermal locations and vapor dominated fields such as The Geysers, with the understanding that the design considerations at each location must take into account differences in noncondensable gas concentrations. Accordingly, the commercial-scale cost estimates presented in this section should be equally appropriate for both hydrothermal and vapor dominated resource applications. A major exception might be that at a hydrothermal location the disposal of the H_2S removed by the process may possibly be accomplished by mixing the vent stream with the relatively large quantities of unflashed brine that is typically disposed of by reinjection into the ground. Thus, a hydrothermal application of the heat exchanger process might be simpler and have lower costs than the system shown in Figure 6-1 and used as the cost model in this section, since the ultimate H_2S disposal system, such as a Stretford plant, may not be needed.

ESTIMATED COSTS FOR COMMERCIAL-SCALE (55-MW) SYSTEM

The system design and associated cost estimates are based on the following design criteria:

- Steam. Steam flow is 139 kg/s (1.1 million lb/h). The steam contains 220 ppm of H_2S gas and a noncondensable gas content of 5000 ppm. The heat exchanger process removes a net of 95% of the H_2S from the steam in the two-stage heat exchanger configuration. The trunkline steam is supplied at 177°C (350°F) and 931 kPa (135 psi) absolute, saturated. The concentrated steam feed to the second-stage heat exchanger has a dew-point temperature of 172°C (341°F) and a total pressure of 910 kPa (132 psi) absolute. The saturation temperature of the second-stage vent stream, at 889 kPa (129 psi) absolute, is 167°C (333°F).

- Stream Factor. The generating plant is assumed to be on-line 8000 h/yr for a stream factor of 91%.

- Heat Exchangers. The first-stage heat exchangers convert 95% of the inlet steam to clean steam, using a 5% vent rate. The second-stage heat exchanger condenses about 50% of the remaining steam in the vent stream from the first-stage heat exchanger, for an overall 98% level of condensation of main inlet steam flow. The second-stage vent is cooled to 49°C (120°F) in the vent condenser and sent to the Stretford unit.

- Sulfur Production. Based on the steam conditions given above, the Stretford unit will produce 2.5 t/d of sulfur.

A list of the major equipment for the heat exchanger process is given in Table 6-1.

The capital cost for the heat exchanger process equipment is estimated at $5.6 million. Based on vendor quotes, a 2.5 t/d Stretford unit cost is $2.6 million, giving a total abatement system cost of $8.2 million. A summary of the capital cost estimate is given in Table 6-2. All costs are based on 1979 dollars.

Annual operating and maintenance costs were estimated at 2% of capital costs for the heat exchanger process and 10% of capital costs for the Stretford unit. The Stretford factor of 10% is based on vendor information and the heat exchanger factor of 2% is based on the fact that the recirculating pump is the only moving part in the system and that little personnel attention to the process should be required. Total direct annual operating costs were $425,000 or 1.0 mill/kWh. With annualized capital charges of 18.5%, the total operating and capital costs are $1,945,000 or 4.5 mills/kWh. The overall process summary costs are presented in Table 6-2.

Table 6-1

LIST OF MAJOR EQUIPMENT, HEAT EXCHANGER H_2S REMOVAL PROCESS, FOR A 55-MW GEOTHERMAL FACILITY

HEAT EXCHANGER

- First-stage heat exchangers
  - Number of heat exchangers -- 3
  - Capacity -- 33-1/3%/heat exchanger: 100% load = 55 MW - 139 kg/s (1.1×10^6 lb/h)
  - Tube surface area -- 4812 sq m (51,800 sq ft)/heat exchanger
  - Tube bundle height -- 11 m (37 ft)
  - Tube bundle diameter -- 3 m (11 ft)
  - Design -- radial flow vertical tube evaporator
  - Material -- 304 SS for tubes, shell, and piping

- Second-stage heat exchanger
  - Number of heat exchangers -- 1
  - Capacity -- 100% load = 3.5 kg/s (2.8×10^4 lb/h)
  - Tube surface area -- 338 m^2 (3638 ft^2)
  - Tube bundle height -- 5.9 m (19.5 ft)
  - Tube bundle diameter -- 1 m (4 ft)
  - Design -- radial flow vertical tube evaporator
  - Material -- 304 SS for tubes, shell, and piping

- Vent condenser
  - Number of heat exchangers -- 1
  - Capacity -- 100% load = 3.4 kg/s (2.7×10^4 lb/h)
  - Tube surface area -- 344 m^2 (3700 ft^2)
  - Tube bundle length -- 5.9 m (19.5 ft)
  - Tube bundle diameter -- 1 m (4 ft)
  - Design -- horizontal
  - Material -- 304 SS for tubes, shell, and piping

Table 6-1 (continued)

PUMPS

- First-stage circulation pumps

  Number of pumps -- 4

  Configuration -- one battery of four 33-1/3% load pumps common to all three first-stage heat exchangers; three pumps normally operating with one standby; each pump capable of meeting requirements of one heat exchanger

  Flow capacity -- 255 l/s (4050 gpm)/pump (based on 0.09 l/s (1.5 gpm)/heat exchanger tube)

  Pump head -- 16 m (52 ft)

  Pump power -- 48 kW (64 hp)/pump

  Materials -- SS and other corrosion-resistant materials

  Type -- electric-driven horizontal centrifugal

- Second-stage circulation pumps

  Number of pumps -- 2

  Configuration -- two 100% load pumps serving the single second-stage heat exchanger; one pump normally operating with one standby

  Flow capacity -- 33.8 l/s (535.5 gpm)/pump (based on 0.1 l/s (1.5 gpm)/heat exchanger tube)

  Pump head -- 10 m (32 ft)

  Pump power -- 3.9 kW (5.2 hp)/pump

  Materials -- SS and other corrosion-resistant materials

  Type -- electric-driven horizontal centrifugal

Table 6-2

OVERALL H_2S ABATEMENT SYSTEM COST SUMMARY IN 1979 DOLLARS

Bases:

139 kg/s (1.1×10^6 lb/h) steam feed

55-MW generating capacity

220 ppm H_2S in steam feed with 95% removal

8000 h/yr stream time

HTC = 3404 $W/(m^2 \cdot {}^oC)$ [600 $Btu/(h \cdot ft^2 \cdot {}^oF)$] -- first stage

Annualized capital charges -- 18.5% of total plant cost

Annual O&M cost ratios:

- H_2S removal -- 2% of removal process plant cost
- H_2S disposal -- 10% of disposal process plant cost

Capital Investment:

Heat exchangers	$2,900,000
Pumps	100,000
Piping, valves, controls, insulation	900,000
Major equipment cost	3,900,000
Construction @ 20% of major equipment cost	780,000
Subtotal	$4,680,000
Engineering and fees @ 20%	940,000
Total capital cost -- heat exchanger process	5,620,000
Stretford unit	2,600,000
Total capital cost H_2S abatement system	$8,220,000

Annual Cost of Investment:	$1,520,000

Annual Operating Costs:

Power @ 4.5¢/kWh	$ 53,000
Operating and maintenance (heat exchanger process)	112,000
Operating and maintenance (Stretford unit)	260,000
Total	$ 425,000
Operating costs (mills/kWh)	1.0
Total annual capital and operating costs	$1,945,000
Total annual capital and operating costs (mills/kWh)	4.4

ESTIMATED POWER LOSS DUE TO UPSTREAM HEAT EXCHANGER

The heat exchanger process could result in a slight loss in power production because of the vented steam and the lower pressure of the steam which goes to the turbine. However, since the process removes all of the noncondensable gases ahead of the turbine, the demands of the steam jet air ejector system are reduced and enough clean steam can be obtained from the second-stage heat exchanger to drive the ejectors. Only the air which enters the condenser from deentrainment from the cooling water and from air in leakage must be removed. Since, without an upstream process, some of the wellhead steam has to be used to drive the ejectors, the potential power which can be produced per unit of wellhead steam must take this into account. Typically at The Geysers, about 5% of the wellhead steam is used to drive the ejectors. Calculations show that only about 2% of the total steam would be required for the ejectors if the heat exchanger process is used upstream.

The amount and condition of the steam going to the turbine per mass unit of steam delivered to the heat exchanger process depend on the vent rate and ΔT of the first-stage exchanger. As the vent rate increases, the amount of steam available to the turbine decreases. As the ΔT increases, the temperature and pressure of the clean steam decreases so that less power can be derived per unit of steam.

In order to compare the power production between a generating unit which uses the heat exchanger process and one that does not, it is necessary to do a thermodynamic analysis of the two cases. The following derivation was used to calculate the theoretical power production per unit of steam entering the turbine and is based on:

1. Steam entering turbine is saturated with enthalpy, H_i, and entropy, S_i.
2. Steam leaving turbine is part vapor with enthalpy, H_{OV}, and entropy, S_{OV}, and part liquid with enthalpy, $H_{O\ell}$, and entropy, $S_{O\ell}$.
3. Maximum theoretical power is for an isentropic process, $\Delta S = 0$.
4. Power production = ΔH x flow to turbine

Let L = weight fraction liquid exiting turbine

V = weight fraction vapor exiting turbine

H_O = enthalpy of total fluid exiting turbine

S_O = entropy of total fluid exiting turbine

$$L + V = 1.0 \qquad (6\text{-}1)$$

$$H_o = LH_{o\ell} + V\,H_{ov} \qquad (6\text{-}2)$$

$$S_o = LS_{o\ell} + V\,S_{ov} \qquad (6\text{-}3)$$

but $$S_o = S_i \text{ for } \Delta S = 0.0 \qquad (6\text{-}4)$$

$$\therefore \quad LS_{o\ell} + (1 - L)\,S_{ov} = S_i \qquad (6\text{-}5)$$

$$L = \frac{S_{ov} - S_i}{S_{ov} - S_{o\ell}} \qquad (6\text{-}6)$$

The procedure is to calculate L from entropy data and then calculate overall outlet enthalpy. The difference in enthalpy between inlet and outlet is the maximum power production per unit of steam. For example, using the flow conditions shown in Figure 5-2 in Section 5, and assuming: (1) that without the heat exchanger process, 5% of the entering steam would be used to supply the steam jet air ejectors; and (2) the turbine exit temperature is 49°C (120°F), the following conditions prevail. For simplicity in using standard steam tables, all calculations are given in English units. Values are for a pilot plant of 2.5 MW(e) equivalent.

	Without Heat Exchanger	With Heat Exchanger
Overall steam flow, lb/h	52,600	52,600
Steam flow to turbine, lb/h	49,970	50,000
Steam flow to gas ejectors, lb/h	2,630	1,140*
Inlet temperature, °F	347**	337
Inlet pressure, psia	129	113
Inlet enthalpy, H_i, Btu/lb	1191.7	1189.4
Inlet entropy, S_i, Btu/lb·°F	1.5815	1.5924
Exit liquid enthalpy, Btu/lb	87.92	87.92
Exit vapor enthalpy, Btu/lb	1113.7	1113.7
Exit liquid entropy, Btu/lb·°F	0.1645	0.1645
Exit vapor entropy, Btu/lb·°F	1.9339	1.9339
Exchanger ΔT, °F	none	10

*Note: The steam to drive ejectors is clean steam produced in the second stage from the 2,600 lb/h vent stream out of the first stage.
**Figure 6-2 below is based on 350°F, rather than 347°F, inlet temperature.

Without heat exchanger:

$$L = \frac{1.9339 - 1.5815}{1.9339 - 0.1645} = 0.1992 \tag{6-7}$$

$$H_o = (0.1992)(87.92) + (1.0 - 0.1992)(1113.7)$$
$$= 909.4 \text{ Btu/lb} \tag{6-8}$$

$$\Delta H = 1191.7 - 909.4 = 282.3 \text{ Btu/lb} \tag{6-9}$$

$$\text{Total power} = (49{,}970)(282.3)/3413 = 4133 \text{ kW} \tag{6-10}$$

With heat exchanger:

$$L = \frac{1.9339 - 1.5924}{1.9339 - 0.1645} = 0.1930 \tag{6-11}$$

$$H_o = (0.1930)(87.92) + (1.0 - 0.1930)(1113.7)$$
$$= 915.7 \text{ Btu/lb} \tag{6-12}$$

$$\Delta H = 1189.4 - 915.7 = 273.7 \text{ Btu/lb} \tag{6-13}$$

$$\text{Total power} = (50{,}000)(273.7)/3413 = 4009 \text{ kW} \tag{6-14}$$

$$\frac{\text{Power with heat exchanger}}{\text{Power without heat exchanger}} = \frac{4009 \text{ kW}}{4133 \text{ kW}} \times 100 = 97.0\% \tag{6-15}$$

Similar calculations were done for other ΔT's and other vent rates. The results are presented in Figure 6-2 which shows the relative power produced by the steam from the heat exchanger process versus using 177^{o}C (350^{o}F) saturated wellhead steam directly. The figure is based on 95% of the wellhead steam going to the turbine and 5% going to the ejectors for the case without the heat exchanger process. If ejector requirements are different, then a different set of curves would apply. For simplification, the figure also assumes no pressure losses before the steam enters the turbine. In actual operations, other pressure losses upstream of the turbine, from the throttling valve for instance, change the thermodynamic properties of the steam and the amount of power which can be produced. The nature of these pressure losses varies from plant to plant and, in some cases, they are quite large. These actual pressure losses must be taken into account in the design of an abatement system for a

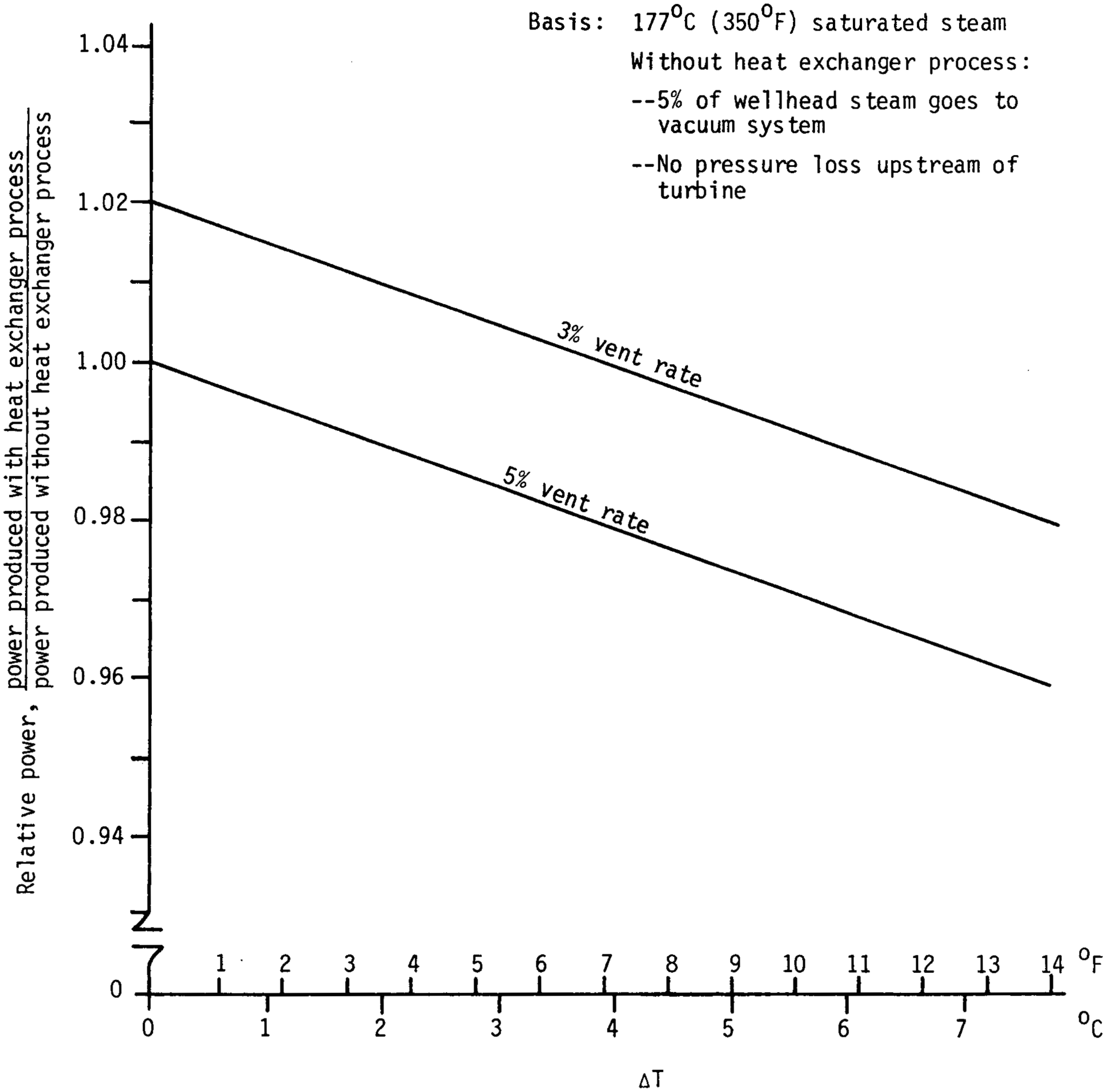


Figure 6-2. Comparison of Power Production Using Heat Exchanger Process as a Function of ΔT and Vent Rate

specific plant. When these losses are considered, the reduction in power output due to the heat exchanger process, as shown in Figure 6-2, may also be reduced.

The selection of the vent rate required is generally dependent on the steam composition and the degree of H_2S removal required. Wellhead steam with a high $H_2S:NH_3$ ratio can be cleaned with a lower vent rate. As can be seen from Figure 6-2, the vent rate has a large effect on relative power production; a rigorous economic analysis must take the composition of the steam into account. In general, higher removal requirements, which could be expected at high H_2S contents, will lead to higher vent rates and greater penalty for loss of power generated.

To summarize, the effect of the heat exchanger process on power production depends on the combined results of the design factors discussed above which will vary with each specific application. In a situation where noncondensable gases lead to a use of 5% of the steam to drive the air ejectors, the addition of the heat exchanger process could result in no net power loss at all and, in some special cases (low ΔT's and low vent rates), a net power increase might be conceivable.

DESIGN FACTORS AFFECTING CAPITAL EQUIPMENT COSTS

The amount of heat transfer area required has a substantial effect on equipment costs, affecting both the size of the heat exchanger required and the sizes of pumps, piping, and other auxiliary parts of the system. The required heat transfer area, A, is governed by the following relationship:

$$A = \frac{Q}{\Delta T \cdot HTC} \qquad (6\text{-}16)$$

where A = heat transfer area

Q = heat transfer rate

ΔT = temperature driving force

HTC = heat transfer coefficient (called U in previous sections)

Since the heat transfer rate, Q, is fixed by wellhead steam conditions and turbine steam requirements, the heat transfer area is chiefly a function of the ΔT and HTC. Varying the ΔT affects both the heat transfer area and power production. For example, reducing ΔT increases the heat exchanger area required but also increases power production. As a result, ΔT must be optimized by balancing capital cost against power production. Figure 6-3 shows the effect of changing ΔT on capital costs for a 55-MW system. The base case used in Figure 6-3 is the cost estimate developed in the

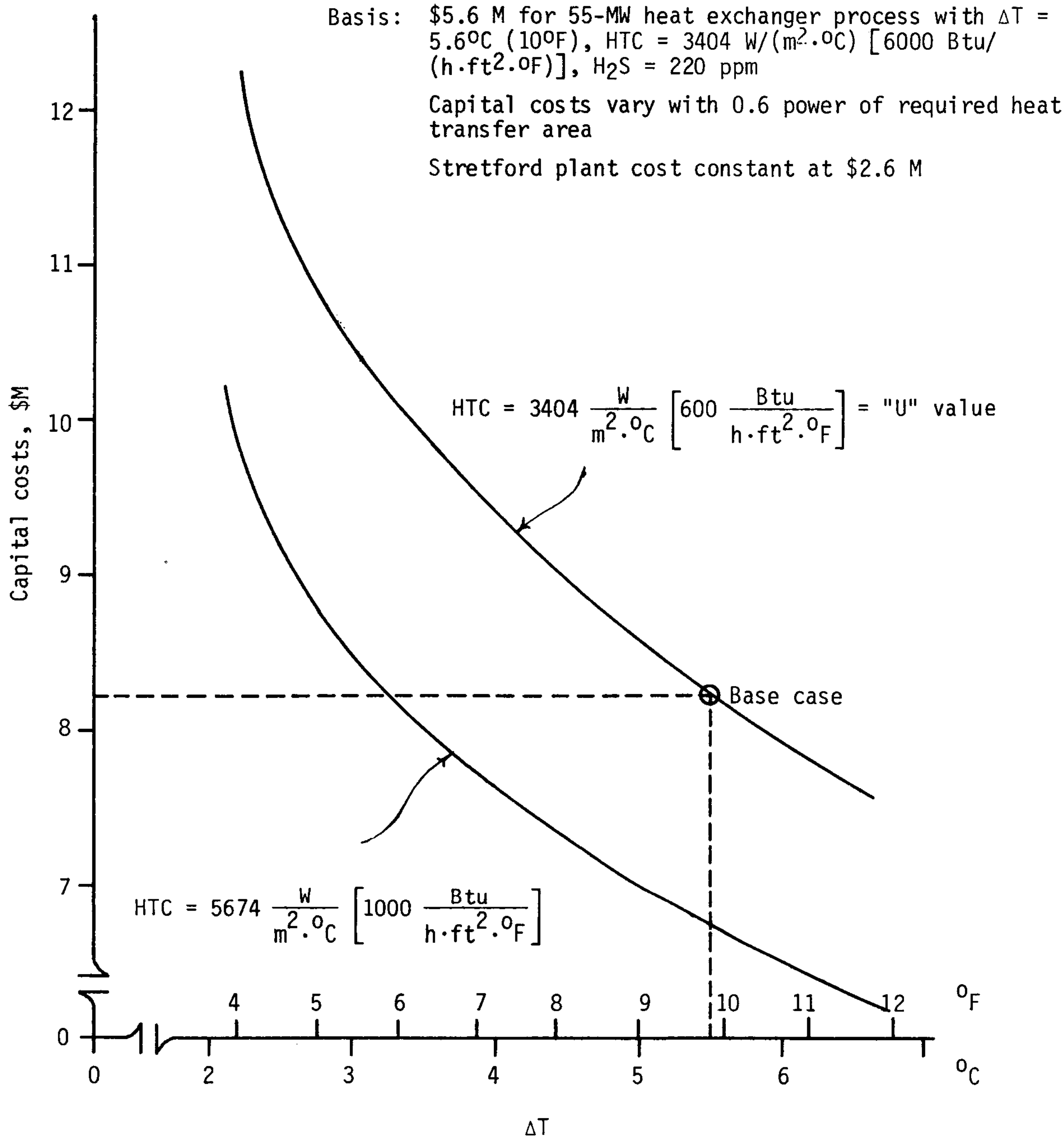


Figure 6-3. Comparison of Heat Exchanger Process Capital Costs as a Function of Temperature Difference (ΔT) and Heat Transfer Coefficient (HTC) in First-Stage Heat Exchanger

initial cost estimate subsection. A 0.6 power law dependence based on surface area is adopted based on normal process industry scale-up cost estimating techniques.

Changes in the heat transfer coefficient also affect the heat transfer area. The cost estimate developed was based on a HTC of 3404 $W/(m^2 \cdot {}^oC)$ [600 $Btu/(h \cdot ft^2 \cdot {}^oF)$]. This is considered a conservative estimate (Section 3, page 3-17). Problems with leakage in the test unit heat exchanger likely have caused calculated values of the heat transfer coefficient to be low. Different designs such as fluted tubes or a horizontal tube spray film exchanger (see Section 4) could provide higher heat transfer coefficients. For this reason, Figure 6-3 includes capital cost comparisons for design heat transfer values of 3404 and also for expected values of 5674 $W/(m^2 \cdot {}^oC)$ [600 and 1000 $Btu/(h \cdot ft^2 \cdot {}^oF)$].

It should be noted that Figures 6-2 and 6-3 should not be used for design purposes since they represent estimates for specific conditions. They do, however, indicate the nature of some of the design options which must be considered in optimizing the heat exchanger process equipment costs for a given set of conditions. Knowing the cost of power, load factor, equipment design life, and interest rate, a trade-off could be made to run the heat exchanger at whatever ΔT gives the lowest combination of capital and operating costs.

H_2S DISPOSAL ALTERNATIVES

The commercial cost estimates presented above are based on a Stretford unit being used for ultimate disposal of the removed H_2S. Under proper geologic conditions, however, one alternative to this approach is to reinject the high pressure H_2S-rich vent gas into an outlying geologic formation which has little or no interaction with the producing field. If this were done, the substantial capital and operating costs associated with the Stretford unit could be avoided.

Appendix A

TEMPERATURE AND STEAM COMPOSITION OF SELECTED HYDROTHERMAL AREAS

Location	Temperature Downhole/Flash (°C)	Steam Composition (ppm)					
		CO_2	H_2S	NH_3	H_2	N_2	CH_4
I. Dry Steam Field							
California--The Geysers Geothermal Field	177/177	3260	222	100/194	56	52	54-200
II. Hydrothermal Fields							
California--Imperial Valley							
Salton Sea KGRA	324/--	3000	2.5	35			10
Brawley KGRA	254/--						
Heber KGRA	174/--						
East Mesa KGRA	163/--	1920	1.0	5.1			30
Hawaii-Puna Area							
HGP-A	358/188	750-1000	700-900		5-10	100-200	
Nevada--Eureka County							
Beowawe Geysers	229/--	98 vol %[a]	1 vol %[a] (10-15)	1 vol %[a] (6-9)			
New Mexico							
Baca Ranch KGRA	291/169	30,000 (10,000-40,000)	250 (150-550)	1-6	1-4	40-100	1-6
Utah--Beaver County							
Fort Cove-Sulphurdale	166/--		10-50[b]				
Roosevelt Hot Springs	266/--						
Mexico							
Cerro Prieto Geothermal Field	--/176	8,500-45,000	1500-3000	80-100			

Note: Blanks in the table indicate that values are unknown or were not reported.

[a] % figures refer to noncondensable fraction. Numbers in parentheses are in ppm.

[b] This value was recorded during drilling and may not represent long-term operating conditions.

Appendix B
TEST UNIT PERFORMANCE DATA

Data Set No.	Date	Heat Transfer ΔT[a] °C	(°F)	Vent Rate[b] %	Heat Transfer Coef. - U[c] $W/(m^2 \cdot °C)$	$[Btu/(h \cdot ft^2 \cdot °F)]$	H_2S Conc. Reduction[d] %	Remarks
1	3/ 7/79	2.7	(4.9)	15 (f)	4250	(748)	95.9	Insulation incomplete through
2	3/14/79	2.7	(4.9)	10 (g)		--	--	March 1979
3	3/15/79	3.1	(5.6)	10 (g)	3975	(700)	--	
4	3/16/79	3.1	(5.6)	8 (f)	4240	(747)	94.7	
5	3/19/79	3.8	(6.8)	5 (f)	3750	(660)	93.6	Continuous operation began on
6	3/20/79	4.0	(7.2)	5 (f)	3500	(616)	92.6 (99.5)[h]	3/19/79
7	3/20/79	4.0	(7.2)	4 (f)	3450	(607)	98.1	
8	3/21/79	4.3	(7.7)	4 (f)	2970	(523)	95.9 (99.2)[h]	
9	3/22/79	4.2	(7.5)	3 (f)	2965	(522)	97.6	
10	5/16/79	3.3	(5.9)	--	4410	(777)	--	Start Baseline testing w/com-
11		3.2	(5.7)	--	4250	(749)	--	pleted insulation on 5/16/79
12	5/17/79	2.9	(5.2)	--	4015	(707)	97.0	Baseline test
13		3.5	(6.3)	--	3395	(598)	--	
14		3.2	(5.8)	--	3610	(636)	--	
15	5/18/79	3.5	(6.3)	--	3690	(650)	--	
16		3.5	(6.3)	--	3685	(649)	--	
17	5/21/79	3.2	(5.7)	--	3305	(582)	--	
18	5/22/79	3.7	(6.7)	--	3750	(660)	--	
19	5/23/79	3.7	(6.6)	--	4135	(728)	--	
20		3.7	(6.7)	--	3755	(661)	--	
21		4.6	(8.3)	--	3250	(572)	--	
22		5.5	(9.9)	--	4225	(744)	--	
23	5/24/79	3.5	(6.3)	10 (g)	3690	(650)	--	
24		3.8	(6.8)	--	3015	(531)	--	
25		4.8	(8.7)	2 (g)	3810	(671)	--	
26	5/25/79	3.1	(5.6)	--	3140	(553)	95.8	
27		3.4	(6.1)	7 (g)	3240	(571)	--	
28		3.4	(6.1)	--	5460	(962)	--	
29	5/29/79	4.8	(8.6)	--	3050	(537)	91.6	
30		3.3	(6.0)	3 (g)	3500	(616)	--	
31		4.1	(7.4)	--	2835	(499)	--	
32	5/30/79	3.8	(6.8)	--	3430	(604)	94.5	
33		3.7	(6.7)	--	3400	(599)	--	
34		4.2	(7.5)	--	3040	(535)	--	
35	5/31/79	3.6	(6.5)	--	3135	(552)	--	
36		6.9	(12.4)	--	2570	(453)	--	
37	6/ 1/79	6.6	(11.9)	--	2955	(520)	--	
38	6/ 5/79	4.1	(7.3)	--	3770	(664)	95.5	
39		5.0	(9.0)	--	2595	(457)	--	
40		4.6	(8.2)	--	3065	(540)	--	
41	6/ 6/79	3.0	(5.4)	5 (g)	3325	(586)	--	
42		3.1	(5.6)	6 (g)	3400	(599)	--	
43		3.1	(5.5)	--	3280	(578)	--	Baseline test

[a]See footnotes on page A-4

TEST UNIT PERFORMANCE DATA (continued)

Data Set No.	Date	Heat Transfer ΔT[a] °C	(°F)	Vent Rate[b] %	Heat Transfer Coef. - U[c] W/(m^2·°C)	[Btu/(h·ft^2·°F)]	H_2S Conc. Reduction[d] %	Remarks
44	6/ 7/79	3.7	(6.6)	--	3440	(606)	92.5	Baseline test
45		3.6	(6.5)	--	3490	(615)	--	
46	6/ 8/79	3.2	(5.7)	5 (g)	3290	(579)	93.8	
47		3.6	(6.4)	--	3275	(577)	--	
48		3.7	(6.7)	--	3640	(641)	--	
49	6/26/79	3.1	(5.6)	--	3520	(620)	90.8	
50	6/27/79	10.8	(19.5)	--	3405	(600)	95.0	
51	6/28/79	3.0	(5.4)	3 (g)	3345	(589)	93.6	
52		3.5	(6.3)	--	2720	(479)	--	
53		3.1	(5.6)	--	3440	(606)	--	
54	6/29/79	2.9	(5.2)	3 (g)	3475	(612)	92.0	
55		3.3	(5.9)	5 (g)	3060	(539)	--	
56		3.2	(5.7)	5 (g)	3170	(558)	--	
57	7/ 5/79	3.2	(5.8)	<1 (g)	2880	(507)	93.8	
58		3.4	(6.2)	--	3210	(565)	--	
59	7/ 6/79	3.1	(5.5)	--	3195	(636)	--	
60	7/ 9/79	3.2	(5.8)	--	3600	(634)	94.0	
61		3.4	(6.1)	2 (g)	3315	(584)	--	
62		3.2	(5.7)	--	3280	(578)	--	
63	7/10/79	3.2	(5.7)	--	3080	(542)	93.0	
64		3.1	(5.6)	--	3415	(601)	--	
65	7/11/79	4.1	(7.4)	--	2685	(473)	--	
66		3.7	(6.6)	--	2915	(513)	--	
67	7/12/79	3.1	(5.5)	5 (g)	3445	(607)	93.0	
68		3.7	(6.6)	3 (g)	2600	(458)	--	
69		3.1	(5.6)	--	3005	(529)	--	
70	7/13/79	3.9	(7.0)	--	3565	(628)	93.7	
71		3.9	(7.0)	3 (g)	3465	(610)	--	
72	7/16/79	3.9	(7.1)	--	3288	(579)	94.6	
73		3.5	(6.3)	2 (g)	3445	(607)	--	
74		3.4	(6.4)	--	3180	(560)	--	
75	7/17/79	3.2	(5.7)	--	3070	(541)	90.6	
76		3.2	(5.8)	--	2885	(508)	--	
77		3.1	(5.5)	--	3045	(536)	--	
78	7/19/79	3.1	(5.5)	4 (g)	2940	(518)	--	
79		3.6	(6.5)	4 (g)	2630	(463)	--	
80		3.0	(5.4)	3 (g)	3165	(557)	--	
81	7/20/79	3.1	(5.6)	4 (g)	3395	(598)	--	
82		3.3	(6.0)	--	3110	(548)	--	
83		4.3	(7.8)	--	3335	(587)	--	
84	7/24/79	3.1	(5.5)	11 (g)	2215	(390)	87.3	
85		3.7	(6.6)	4 (g)	1890	(333)	--	
86	7/26/79	3.3	(6.0)	2 (g)	3855	(679)	--	
87		3.5	(6.3)	2 (g)	3425	(603)	--	
88		3.4	(6.2)	3 (g)	3375	(594)	--	Baseline test

TEST UNIT PERFORMANCE DATA (continued)

Data Set No.	Date	Heat Transfer ΔT[a] °C	(°F)	Vent Rate[b] %	Heat Transfer Coef. - U[c] $W/(m^2 \cdot °C)$	$[Btu/(h \cdot ft^2 \cdot °F)]$	H_2S Conc. Reduction[d] %	Remarks
89	7/26/79	3.4	(6.1)	3 (g)	3430	(604)	--	Baseline test
90	7/27/79	3.6	(6.5)	3 (g)	2340	(412)	90.9	
91		3.5	(6.3)	2 (g)	2430	(428)	--	
92		3.8	(6.8)	8 (g)	2300	(405)	--	
93	7/30/79	3.4	(6.1)	--	3795	(668)	--	
94		3.3	(6.0)	3 (g)	2985	(526)	--	
95		3.2	(5.8)	3 (g)	3230	(569)	--	
96	8/ 6/79	3.2	(5.7)	<1 (g)	3475	(612)	--	
97	8/ 7/79	3.6	(6.5)	<1 (g)	3040	(535)	--	
98		3.4	(6.1)	3 (g)	3140	(553)	--	
99		3.3	(6.0)	--	3015	(531)	--	
100	8/ 8/79	3.6	(6.5)	2 (g)	2940	(518)	91.3	
101		3.5	(6.3)	2 (g)	3040	(535)	--	
102		3.7	(6.7)	2 (g)	3100	(546)	--	
103	8/ 9/79	3.9	(7.0)	2 (g)	2880	(507)	--	
104		3.4	(6.2)	2 (g)	3055	(538)	--	
		Mechanical Problems						
105	10/23/79	4.7	(8.5)	5 (g)	3810	(671)	--	
106		5.2	(9.3)	5 (g)	3680	(648)	--	
107	11/ 9/79	3.3	(5.9)	4 (g)	3675	(647)	93.0	
108	11/13/79	3.1	(5.6)	1 (g)	4060	(715)	94.7	
109		3.2	(5.7)	3 (g)	3225	(568)	94.8	
110	11/14/79	3.4	(6.2)	2 (g)	3000	(528)	92.0	
111		3.4	(6.1)	2 (g)	3110	(548)	92.6	
112	11/18/79	3.6	(6.5)	2 (g)	3540	(623)	--	
113		5.6	(10.1)	1 (g)	3680	(648)	--	
114		5.6	(10.1)	--	3600	(634)	--	
115	11/19/79	3.3	(6.0)	--	--	--	--	
116		5.4	(9.7)	2 (g)	3505	(617)	--	
		Chemical Cleaning						
117	12/19/79	3.4	(6.1)	5 (g)	4475	(788)	94.2	
118	12/20/79	3.6	(6.5)	3 (g)	3635	(640)	95.7	
119	12/21/79	4.8	(8.7)	--	2860	(504)	--	
120		4.4	(8.0)	--	2985	(526)	--	
121	12/27/79	3.4	(6.2)	7 (f)	3365	(593)	95.5	
122	12/28/79	3.4	(6.1)	7 (f)	3355	(591)	95.9	
123		3.3	(5.9)	7 (f)	3360	(592)	96.1	
124		3.2	(5.8)	8 (f)	3425	(603)	--	
125	1/ 8/80	5.5	(9.9) (i)	4 (f)	3540	(623)	95.2	Baseline test
126		--	--	17 (e)	--	--	97.1	High vent rate test
127	1/ 9/80	5.9	(10.6) (i)	4 (f)	3373	(594)	--	Baseline test

TEST UNIT PERFORMANCE DATA (continued)

Data Set No.	Date	Heat Transfer ΔT^a °C (°F)	Vent Rate[b] %	Heat Transfer Coef. - U^c W/(m²·°C)	[Btu/(h·ft²·°F)]	H_2S Conc. Reduction[d] %	Remarks
128	1/ 9/80	5.9 (10.5) (i)	4 (f)	3395	(598)	--	Baseline test
129		5.6 (10.0) (i)	17 (f)	3520	(620)	96.8	High vent rate test
130	1/10/80	5.2 (9.4) (i)	4 (f)	3700	(652)	--	Baseline test
		Transient Test					
131	1/18/80	6.1 (10.9)	5 (f)	2590	(456)	92.4	H_2S injection test
132		5.9 (10.6)	5 (f)	2690	(474)	--	H_2S injection test
133	1/21/80	5.7 (10.3)	5 (f)	3250	(572)	--	Baseline test
134		6.1 (10.9)	17 (f)	3180	(560)	--	Baseline test
135		10.8 (19.5)	11 (f)	3135	(552)	--	High flow rate test
136		12.1 (21.8)	3 (f)	2895	(510)	--	High flow rate test
137	1/22/80	5.4 (9.7)	6 (f)	2665	(469)	--	H_2S injection test
138		5.4 (9.8)	6 (f)	2620	(461)	94.0	H_2S injection test
139		5.4 (9.7)	6 (f)	2710	(477)	--	H_2S injection test
140		5.3 (9.5)	6 (f)	2765	(487)	--	Baseline test
141	1/23/80	6.3 (11.4)	5 (f)	2835	(499)	93.0	Baseline test
142		6.2 (11.2)	5 (f)	2755	(485)	--	NH_3 injection test
143		6.1 (10.9)	5 (f)	2755	(485)	--	NH_3 injection test
144	1/24/80	5.2 (9.4)	6 (f)	2760	(486)	95.0	H_2S and NH_3 injection test
145		5.1 (9.2)	6 (f)	2745	(483)	--	H_2S and NH_3 injection test
146		5.2 (9.3)	6 (f)	2610	(460)	--	H_2S and NH_3 injection test
147	1/25/80	5.7 (10.2)	5 (f)	2730	(481)	--	Last run date
148		5.5 (9.9)	5 (f)	2795	(492)	--	Baseline test
149		5.6 (10.0)	5 (f)	2725	(480)	94.0	Baseline test
150 (j)		26.9 (48.5)	12 (f)	1130	(199)	99.0	High ΔP test
151 (j)		27.4 (49.4)	11 (f)	1135	(200)	--	High ΔP test
152		5.3 (9.5)	20 (e)	3270	(576)	--	High vent test
153		4.9 (8.9)	20 (e)	3520	(620)	--	High vent test

[a] Measured temperature difference between shell- and tubeside of heat exchanger.

[b] Vent rate is shown as percent of inlet steam mass flow rate.

[c] Calculated heat transfer coefficient across tube surface area - based on measured shellside/tubeside temperature differences and measured condensate flow rate from shellside to tubeside.

[d] $100 \times \left(1 - \frac{\text{inlet steam } H_2S \text{ mass concentration}}{\text{clean steam } H_2S \text{ mass concentration}}\right)$

[e] Estimate.

[f] Based on rotameter reading.

[g] Based on measuring condensed vent stream.

[h] H_2S concentration reduction based on measuring H_2S concentration in condensate transfer stream instead of clean steam.

[i] Based on temperature measurements from PG&E installed instrumentation for transient tests; all other values based on RTD's.

[j] The high ΔP test data were not included in the overall statistical evaluation of the performance data because of unexplained water mass balance problems that occurred during these tests.

Appendix C

DATA COLLECTION AND REDUCTION

H_2S MEASUREMENT

Lower-Level Concentrations

A $AgNO_3$ titration method was used to measure the H_2S concentrations in the lower-level streams including the inlet steam, clean steam, condensate transfer, recirculating condensate, and blowdown streams. This is a standard analysis procedure used by PG&E at The Geysers. In this procedure, a steam or condensate sample is taken through a condenser/cooler at atmospheric pressure and collected in a bottle containing a solution of NH_4OH which captures essentially all of the H_2S present in both the liquid and noncondensable gas portions of the sample. The collected sample is then titrated with a $AgNO_3$ solution until a specific conductivity change is noted using a conductivity meter with a sulfide electrode. The amount of $AgNO_3$ required is an indication of the H_2S concentration in the sample.

Higher-Level Concentrations

A different procedure was required to measure the relatively higher H_2S concentrations in the vent stream. In this procedure (which is also a standard PG&E procedure at The Geysers), the sample is condensed in the condenser/cooler; however, only the resulting liquid stream is analyzed using the $AgNO_3$ titration method described above. The noncondensable gas portion of the condensed sample (which contains most of the H_2S in the total sample stream) is analyzed using the Tutwiler method in which the collected gas sample is mixed with a starch solution. Iodine is then titrated into the starch solution until a color change is noted. The amount of iodine required indicates the concentration of H_2S in the gas portion of the collected sample. The results of the $AgNO_3$ and Tutwiler analyses are then combined to get the total H_2S concentration in the vent stream.

Determining the H_2S Removal Value

The H_2S removal value, which is more accurately described as the percent reduction in H_2S concentration, was determined by the following equation:

$$R_{H_2S} = \left[1 - \left(\frac{H_2S_{C.S.}}{H_2S_{I.S.}}\right)\right] \times 100 \qquad \text{(C-1)}$$

where R_{H_2S} = percent reduction in H_2S concentration

$H_2S_{C.S.}$ = clean steam H_2S concentration

$H_2S_{I.S.}$ = inlet steam H_2S concentration

Uncertainties in flow measurements, chemical sampling and chemical analysis limited the accuracy of the sulfur mass balance. The standard deviation of 5 measurements of the total sulfur balance was 23 percent of the total sulfur mass flow. (See Appendix F.) Nevertheless, the H_2S removal efficiency, defined in C-1, was well-established at values above 90 percent by the 38 measurements of H_2S concentration in the outlet steam compared to the H_2S concentration in the inlet steam. These measurements, presented in Table D-2, were not affected by either the flow rate measurement difficulties or the problems of chemical sampling and analysis. These latter problems affected only the measurement of the high H_2S concentration in the vent stream. This stream has a major effect on the mass balance, because most of the sulfur flows out the vent, but H_2S concentration in the vent stream does not enter into the sulfur removal calculation.

NH_3 MEASUREMENT

An ion electrode technique was used to perform field analyses for NH_3 concentrations. The special electrode in this procedure develops an electrical potential which is proportional to the concentration of NH_3 present in the liquid sample being analyzed. This potential is measured with a voltmeter. Laboratory analyses for NH_3 were performed using the Kjeldahl procedure. Both of these techniques are standard analysis procedures for PG&E at The Geysers.

CO_2 AND TOTAL NONCONDENSABLES

The concentrations of CO_2 and total noncondensables were determined using procedures that are standard for PG&E at The Geysers. First, a cooled sample was collected in a liquid/gas separator and the volumetric ratio between the collected liquid and non-condensable gases was noted. The Orsat Analysis procedure was then used to determine the mass ratios of various species (CO_2 and H_2S combined, O_2, CH_4, H_2, and N_2) in the gas portion of the collected sample. With the Orsat procedure, the collected gas sample was selectively bubbled through various liquids which absorbed specific species. The increase in volume of each absorber liquid indicated the amount of the particular species being absorbed by that liquid. The information obtained from the liquid/gas ratio measurement and the Orsat analysis, when combined with the separate H_2S analyses obtained from previously discussed procedures, was used to calculate the mass concentrations of CO_2 and total noncondensables present in the gas portion of the collected sample.

HEAT TRANSFER COEFFICIENT CALCULATIONS

The resulting heat transfer coefficient (U) values were determined using the following equations:

$$U = \frac{Q}{A \Delta T} \tag{C-2}$$

$$Q = (\dot{M}_{C.T.})\, H_{fg} \tag{C-3}$$

$$\Delta T = T_{I.S.} - T_{C.S.} \tag{C-4}$$

where U = heat transfer coefficient

Q = heat transfer rate from the shellside to the tubeside of the heat exchanger

A = heat transfer surface area of the heat exchanger, 12.9 m^2 (138.5 ft^2)

$\dot{M}_{C.T.}$ = measured Condensate Transfer flow rate (which represents the condensing rate inside the shellside of the heat exchanger)

H_{fg} = latent heat of condensation of the Inlet Steam

$T_{I.S.}$ = measured temperature of the Inlet Steam

$T_{C.S.}$ = measured temperature of the Clean Steam

Appendix D

H_2S REMOVAL ERROR ANALYSIS

COMPARISON OF EFFECTS ON MEASURED H_2S REMOVAL VALUES

Table D-1 presents a comparison of the effects on the measured H_2S removal values due to variations in process parameters and errors associated with the chemistry analysis techniques used in measuring the H_2S concentrations.

Table D-1

COMPARISON OF EFFECTS ON MEASURED H_2S REMOVAL VALUES

Process Parameter or Analysis Error	Process Parameter Range or Error Assumptions	Predicted Variation of Measured H_2S Removal Values	Reference
Vent rate	2-10% of inlet flow rate	∿ ±3 to 4%	Figures 2-2 and 2-3
Inlet H_2S concentration	150-350 ppm	∿ ±0.5 to 1%	Figures 2-2 and 2-3
Inlet NH_3 concentration	50-100% of inlet H_2S concentration	∿ ±1 to 2%	Figures 2-2 and 2-3
Chemistry analysis error	dX = ±0.05X dY = ±5 ppm	∿ ±1 to 4%	Discussion in Appendix D and Table D-2

EVALUATION OF THE EFFECTS OF CHEMISTRY ANALYSIS ERRORS ON MEASURED H_2S REMOVAL VALUES

The reduction of H_2S concentration in the clean steam leaving the test unit heat exchanger and its expected range of error can be defined by the following equations:

$$R = \frac{(X \pm dX) - (Y \pm dY)}{(X \pm dX)} \tag{D-1}$$

$$dR = \sqrt{\left(\frac{Y dX}{X^2}\right)^2 + \left(\frac{dY}{X}\right)^2} \qquad (D-2)$$

where R = H_2S concentration reduction (100 x R = % reduction)

X = inlet steam H_2S concentration

Y = clean steam H_2S concentration

dX, dY = accuracy errors of chemistry analysis techniques

dR = resulting error of measured H_2S concentration reduction

The analysis techniques used to determine the inlet steam and clean steam H_2S concentrations are described in Appendix C. The error values used in this analysis are:

$dX \sim \pm 0.05X$

$dY \sim \pm 5$ ppm

These values are based on communications with PG&E personnel who are familiar with these techniques at The Geysers (1) and also on the standard deviation of the Y values shown in Table D-2. The PG&E personnel suggested very confidently that $dX \sim \pm 0.05X$ and less confidently that $dY \sim \pm 0.20Y$; this lack of confidence in the dY value is based on the limited field experience in analyzing lower concentrations of H_2S with this technique. The standard deviation for the Y values (σY) shown in Table D-2 is 5 ppm. Based on the lack of confidence in the dY value suggested by the PG&E personnel, dY was assumed to be equal to σY which is ±5 ppm.

Table D-2 shows the measured inlet steam and clean steam H_2S concentrations for several data sets and the resulting H_2S reduction values and predicted error ranges based on Eq. D-1 and D-2, and the assumed error values of dX = 0.05X and dY = 5 ppm. The mean values and standard deviations for X, Y, and R are also shown at the bottom of Table D-2.

REFERENCES

1. Telephone communication with Gary Sharp of PG&E on 7/31/80.

Table D-2

H_2S CONCENTRATION REDUCTION ERRORS DUE TO CHEMISTRY ANALYSIS TECHNIQUES

Run No.	Date	X Inlet Steam H_2S Concentration (ppm)	Y Clean Steam H_2S Concentration (ppm)	R H_2S Concentration Reduction (%)	dR H_2S Concentration Reduction Error (%)
1	3/ 7/79	261.8	12.6	95.2	± 1.9
4	3/16/79	267.0	15.5	94.2	± 1.9
5	3/19/79	265.1	17.8	93.3	± 1.9
6	3/20/79	264.5	20.8	92.1	± 1.9
7	3/20/79	258.8	5.2	98.0	± 1.9
8	3/21/79	265.6	11.5	95.7	± 1.9
9	3/22/79	260.1	6.5	97.5	± 1.9
12	5/17/79	375.0	11.1	97.0	± 1.3
26	5/25/79	273.8	11.5	95.8	± 1.8
29	5/29/79	260.4	21.9	91.6	± 2.0
32	5/30/79	277.5	15.3	94.5	± 1.8
38	6/ 5/79	267.0	12.0	95.5	± 1.9
44	6/ 7/79	260.0	19.5	92.5	± 2.0
46	6/ 8/79	170.0	10.5	93.8	± 3.0
49	6/26/79	151.2	13.9	90.8	± 3.3
50	6/27/79	156.9	7.3	95.3	± 3.2
51	6/28/79	245.0	15.6	93.6	± 2.1
54	6/29/79	213.4	16.9	92.1	± 2.4
57	7/ 5/79	247.5	15.2	93.9	± 2.0
60	7/ 9/79	240.2	14.3	94.0	± 2.1
63	7/10/79	227.5	15.9	93.0	± 2.2
67	7/12/79	267.0	18.6	93.0	± 1.9
70	7/13/79	254.6	15.8	93.8	± 2.0
72	7/16/79	254.9	13.8	94.6	± 2.0
75	7/17/79	243.0	22.8	90.6	± 2.1
84	7/24/79	227.1	28.8	87.3	± 2.3
90	7/27/79	243.0	22.0	90.9	± 2.1
100	8/ 8/79	253.1	22.0	91.3	± 2.0
107	11/ 9/79	233.3	16.4	93.0	± 2.2
108	11/13/79	200.6	11.2	94.4	± 2.5
111	11/14/79	187.8	14.8	92.1	± 2.7
117	12/19/79	134.0	8.0	94.0	± 3.7
118	12/20/79	214.0	9.2	95.7	± 2.3
121	12/27/79	219.0	9.9	95.5	± 2.3
122	12/28/79	198.6	8.1	95.9	± 2.5
123	12/28/79	205.0	7.9	96.1	± 2.4
125	1/ 8/80	272.0	13.0	95.2	± 1.9
149	1/25/80	309.5	18.2	94.1	± 1.6

$\bar{X}$ = 240.1 ppm σX = 18.6% = 44.7 ppm

$\bar{Y}$ = 14.5 ppm σY = 35.9% = 5.2 ppm

$\bar{R}$ = 93.9% σR = 2.2% (of 93.9%) = 2.1% (of inlet H_2S concentration)

Appendix E

TRANSIENT TEST DESCRIPTIONS AND DATA

TRANSIENT TEST NO. 1: STARTUP RAMP

Objective--To observe the response characteristics of the heat exchanger test unit during a power plant startup based on a maximum load increase rate of 5%/min (0 to 100% load in 20 min).

Pretest Conditions--Stabilized steady-state operation at the following conditions:

Inlet temperature - 175.9^{o}C (348.6^{o}F)

Clean steam temperature - 170.7^{o}C (339.3^{o}F)

ΔT - 5.2^{o}C (9.3^{o}F)

Clean steam flow - 0.101 kg/s (800 lb/h)

Vent rate - 0.0057 kg/s (45 lb/h)

Inlet steam - 0.133 kg/s (1058 lb/h)

Procedure--Closed clean steam valve, waited until system stabilized, then opened clean steam valve in small increments as shown in Table E-1.

Data--See Table E-1.

Observations--Heat exchanger responded smoothly, stabilizing essentially instantaneously at each load increase increment.

TRANSIENT TEST NO. 2: CLEAN STEAM VALVE SUDDEN CLOSE

Objective--To observe the transient response of the heat exchanger test unit during a simulated turbine-trip situation.

Pretest Conditions--Stabilized, steady-state operation at the following conditions:

Inlet temperature - 175.9^{o}C (348.6^{o}F)

Clean steam temperature - 170.3^{o}C (338.6^{o}F)

ΔT - 5.6^{o}C (10^{o}F)

Clean steam flow - 0.113 kg/s (900 lb/h)

Vent rate - 0.0057 kg/s (45 lb/h)

Inlet steam - 0.136 kg/s (1082 lb/h)

Line pressure - 827 kPa (120 psig)

Pump flow - 9.59 l/s (152 gpm)

Table E-1

TRANSIENT TEST NO. 1: STARTUP RAMP

Elapsed Time		T in		T out		ΔT		ΔP		P Sump		Clean Steam Flow			Inlet Steam Flow	
s	(min)	°C	(°F)	°C	(°F)	°C	(°F)	kPa	(psi)	kPa	(psig)	kg/s	(lb/h)	% Load	kg/s	(lb/h)
0	(0)	--	--	--	--	--	--	--	--	17.4	(120)	0.01	(100)	11	--	--
120	(2)	176.1	(349.0)	175.0	(347.0)	1.1	(2.0)	1.0	(7.0)	--	--	0.03	(200)	22	--	--
300	(5)	--	--	--	--	--	--	--	--	--	--	0.03	(250)	28	--	--
360	(6)	--	--	--	--	--	--	--	--	--	--	0.04	(300)	33	--	--
420	(7)	--	--	--	--	--	--	--	--	--	--	0.04	(350)	39	--	--
480	(8)	--	--	--	--	--	--	1.6	(11.0)	--	--	0.05	(400)	44	--	--
540	(9)	--	--	--	--	--	--	--	--	--	--	0.06	(450)	50	--	--
600	(10)	--	--	--	--	--	--	--	--	--	--	0.06	(500)	56	--	--
660	(11)	--	--	--	--	--	--	--	--	--	--	0.07	(550)	61	0.089	(704)
720	(12)	175.9	(348.7)	172.3	(342.2)	3.6	(6.5)	--	--	--	--	0.08	(600)	67	--	--
780	(13)	--	--	--	--	--	--	--	--	--	--	0.08	(650)	72	--	--
840	(14)	--	--	--	--	--	--	--	--	--	--	0.09	(700)	78	--	--
900	(15)	175.9	(348.7)	171.6	(340.8)	4.3	(7.9)	2.5	(17.0)	--	--	0.09	(750)	83	--	--
960	(16)	175.9	(348.7)	171.1	(340.0)	4.8	(8.7)	--	--	--	--	0.10	(800)	89	--	--
1020	(17)	175.9	(348.6)	170.9	(339.6)	5.0	(9.0)	--	--	--	--	0.11	(850)	94	--	--
1080	(18)	--	--	--	--	--	--	--	--	--	--	0.11	(900)	100	0.136	(1082)
1140	(19)	175.9	(348.6)	170.6	(339.0)	5.3	(9.6)	--	--	--	--	0.11	(900)	100	--	--
1320	(22)	175.9	(348.6)	170.3	(338.6)	5.6	(10.0)	>2.9	(>20.0)	15.4	(106)	0.11	(900)	100	--	--

Constant parameters: Pump flow = 9.59 l/s (152 gpm)
Vent rate = 0.0057 kg/s (45 lb/h)
Line pressure = 17.4 kPa (120 psig)

Procedure--Closed clean-steam valve very rapidly at start of test.

Data--

Elapsed Time (seconds)	Sump Pressure kPa	(psig)
0	731	(106)
60	786	(114)
120	814	(118)
180	827	(120)(line pressure)

Observations--There were no sudden changes in any of the test unit parameters. The sump (tubeside) pressure increased smoothly up to line pressure in about 180 seconds.

TRANSIENT TEST NO. 3: CLEAN-STEAM VALVE SUDDEN OPEN

Objectives--To observe the transient response of the heat exchanger test unit during an abnormal condition simulating a rapid increase in steam supply to the turbine.

Pretest Conditions--This test immediately followed Test No. 2, Clean-Steam Valve Sudden Close. The actual start time of Test No. 3 was 450 s (7.5 min) after starting Test No. 2. The clean-steam valve was fully closed at the start of Test No. 3.

Procedure--Opened clean-steam valve rapidly until unit tripped. Restarted unit by opening the inlet steam air operated valve immediately after the trip.

Data--

Elapsed Time (seconds)	Clean Steam Flow Rate kg/s	(lb/h)
0	0-0.095	0-750
30	0.095	750 (unit tripped)

Restarted unit immediately. Unit stabilized at a clean steam flow rate of 0.095 kg/s (750 lb/h) immediately after restart.

Observations--False high-level indication caused the unit to trip. The unit was restarted immediately without any additional problems and quickly reached stable, steady-state operation. Observed a rapid level rise in sump level glass just prior to trip.

TRANSIENT TEST NO. 4: VENT VALVE SUDDEN CLOSE

Objectives--To observe the transient response of the heat exchanger test unit during a sudden blockage of the vent flow.

Pretest Conditions--Stabilized, steady-state operation at the following conditions:

Inlet temperature - 175.8^{o}C (348.5^{o}F)

Clean steam temperature - 169.3^{o}C (336.7^{o}F)

ΔT - 6.6^{o}C (11.8^{o}F)

Clean steam flow - 0.11 kg/s (900 lb/h)

Vent rate - 0.0057 kg/s (45 lb/h)

Inlet steam flow - 0.145 kg/s (1148 lb/h)

Line pressure - 827 kPa (120 psig)

Pump flow - 9.59 l/s (152 gpm)

Procedure--Suddenly closed vent valve at start of test.

Data--

Elapsed Time (seconds)	Sump Pressure kPa	(psig)	Clean Steam Flow kg/s	(lb/h)
0	717	(104)	0.113	(900)
60	710	(103)	--	--
120	703	(102)	0.116	(920)
180	690	(100)	--	--
240	683	(99)	0.113	(900)
Stopped test				

Observations--There were no sudden changes in any of the test unit parameters. The sump pressure decreased slowly, indicating a gradual decrease in condensate transfer rate from the shell- to tubeside, indicating an accumulation of noncondsables in the shellside of the heat exchanger. The test was stopped after 240 s (4 min).

TRANSIENT TEST NO. 5: VENT VALVE SUDDEN OPEN

Objectives--To observe the transient response of the heat exchanger test unit during a sudden increase in vent flow rate.

Pretest Conditions--Stabilized, steady-state operation at the following conditions:

Inlet temperature - 175.8^{o}C (348.5^{o}F)

Clean steam temperature - 169.7^{o}C (337.5^{o}F)

ΔT - 6.1^{o}C (11.0^{o}F)

Clean steam flow - 0.113 kg/s (900 lb/h)

Vent rate - 0.0057 kg/s (45 lb/h)

Inlet steam flow - 0.152 kg/s (1209 lb/h)

Line pressure - 827 kPa (120 psig)

Pump flow - 9.59 l/s (152 gpm)

Procedure--Suddenly opened vent rate valve to full open position at start of test.

Data--

Elapsed Time (seconds)	Sump Pressure kPa	(psig)	Clean Steam Flow kg/s	(lb/h)	Inlet Flow kg/s	(lb/h)
0	717	(104)	0.113	(900)	0.152	(1209)
60	717	(104)	0.113	(900)	0.156	(1241)
120	724	(105)	--	--	--	--
180	--	--	--	--	--	--

Elapsed Time (seconds)	Inlet Temperature ^oC	(^oF)	Clean Steam Temperature ^oC	(^oF)	ΔT ^oC	(^oF)
0	175.8	(348.5)	169.7	(337.5)	6.1	(11.0)
60	--	--	--	--	--	--
120	175.8	(348.4)	170.3	(338.6)	5.4	(9.8)
180	--	--	--	--	--	--

Observations--There were no sudden changes in any of the test unit parameters. The unit started a smooth transition to a new steady-state operating condition. A slight increase in sump pressure was noted along with a decrease in ΔT.

TRANSIENT TEST NO. 6: INLET STEAM VALVE SUDDEN CLOSE

Objective--To observe the transient response of the heat exchanger test unit during a simulated interruption of the well steam supply.

Pretest Conditions--Stabilized, steady-state operation at the following conditions:

Inlet temperature - 175.9^oC (348.6^oF)

Clean steam temperature - 169.9^oC (337.9^oF)

ΔT - 5.9^oC (10.7^oF)

Clean steam flow - 0.113 kg/s (900 lb/h)

Vent rate - 0.0057 kg/s (45 lb/h)

Inlet steam flow - 0.141 kg/s (1120 lb/h)

Line pressure - 827 kPa (120 psig)

Pump flow - 9.59 l/s (152 gpm)

Procedure--Closed air-operated inlet steam valve at start of test.

Data--

Elapsed Time (seconds)	Clean Steam Flow kg/s	(lb/h)	Vent Flow kg/s	(lb/h)	Sump Pressure kPa	(psig)	Shell Pressure kPa	(psig)
0	0.113	(900)	0.0057	(45)	724	(105)	827	(120)
30	--	--	--	--	648	(94)	662	(96)
60	0.110	(875)	0.0044	(35)	--	--	--	--
90	--	--	--	--	--	--	--	--

Observations--There were no sudden changes in any of the test unit parameters. Both the shell- and tubeside started to gradually depressurize.

TRANSIENT TEST NO. 7: PUMP TRIP

Objective--To observe the transient response of the heat exchanger test unit during a sudden stoppage of condensate recirculation.

Pretest Conditions--Stabilized, steady-state operation at the following conditions:

- Inlet temperature - 175.9^{o}C (348.7^{o}F)
- Clean steam temperature - 170.2^{o}C (338.4^{o}F)
- ΔT - 5.7^{o}C (10.3^{o}F)
- Clean steam flow - 0.113 kg/s (900 lb/h)
- Vent rate - 0.0057 kg/s (45 lb/h)
- Inlet steam flow - no measurement due to loss of fluid in manometer
- Line pressure - 827 kPa (120 psig)
- Pump flow - 9.59 l/s (152 gpm)

Procedure--Opened 440 circuit breaker at start of test, thus stopping pump.

Data--

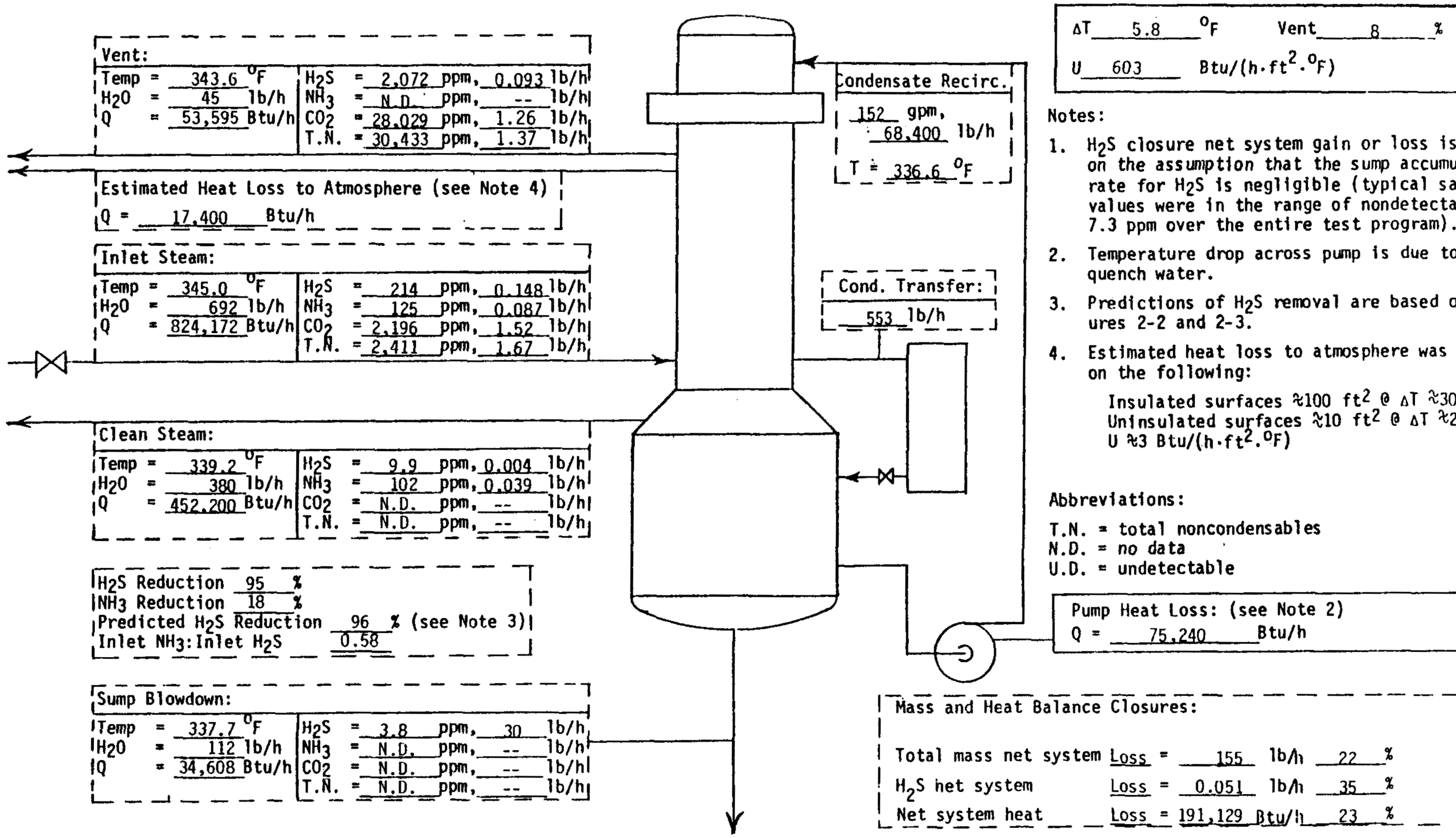
Elapsed Time (seconds)	Sump Pressure kPa	(psig)	Shell Pressure kPa	(psig)
0	724	(105)	827	(120)
15	690	(100)	827	(120)
60	621	(90)	827	(120)
Closed inlet steam valve				

Observation--There were no sudden changes in any of the test unit parameters. The sump pressure started dropping relatively quickly, but smoothly.

Appendix F

MASS AND ENERGY BALANCE DIAGRAMS

This appendix includes mass and energy balance diagrams for six selected cases from the field tests described in Section 3. For each case, total mass, H_2S mass, and energy balance closures are shown. Additional information is also presented showing NH_3, CO_2, and total noncondensable concentrations for various streams.



Notes:

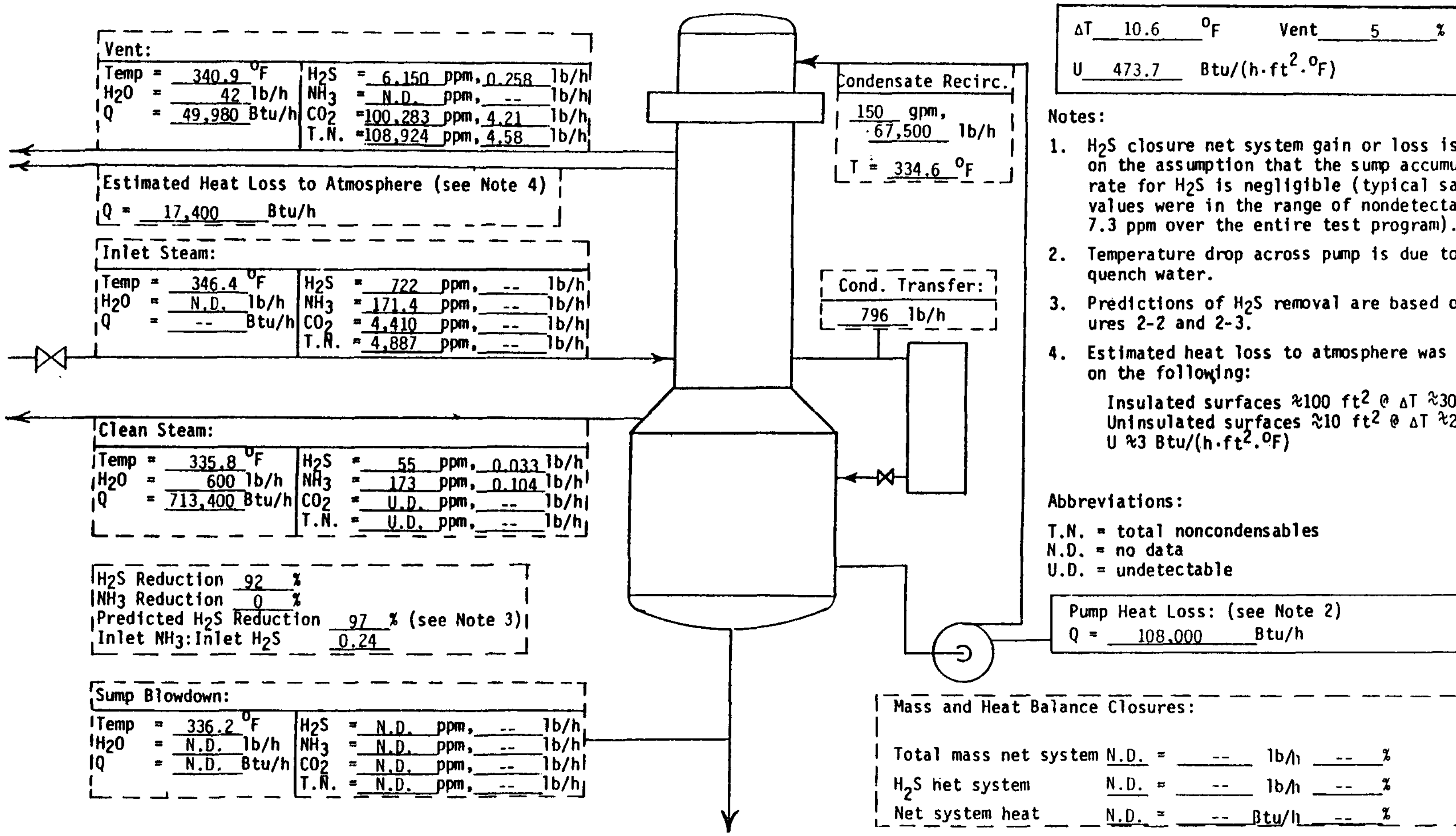
1. H_2S closure net system gain or loss is based on the assumption that the sump accumulation rate for H_2S is negligible (typical sampled values were in the range of nondetectable to 7.3 ppm over the entire test program).
2. Temperature drop across pump is due to seal quench water.
3. Predictions of H_2S removal are based on Figures 2-2 and 2-3.
4. Estimated heat loss to atmosphere was based on the following:

   Insulated surfaces ≈100 ft² @ ΔT ≈30°F
   Uninsulated surfaces ≈10 ft² @ ΔT ≈280°F
   U ≈3 Btu/(h·ft²·°F)

Abbreviations:

T.N. = total noncondensables
N.D. = no data
U.D. = undetectable

Figure F-1. Mass and Energy Balance Sheet for Baseline Test Run 12/28/79



Notes:

1. H_2S closure net system gain or loss is based on the assumption that the sump accumulation rate for H_2S is negligible (typical sampled values were in the range of nondetectable to 7.3 ppm over the entire test program).
2. Temperature drop across pump is due to seal quench water.
3. Predictions of H_2S removal are based on Figures 2-2 and 2-3.
4. Estimated heat loss to atmosphere was based on the following:

   Insulated surfaces ≈100 ft^2 @ ΔT ≈30°F
   Uninsulated surfaces ≈10 ft^2 @ ΔT ≈280°F
   U ≈3 Btu/(h·ft^2·°F)

Abbreviations:

T.N. = total noncondensables
N.D. = no data
U.D. = undetectable

Figure F-2. Mass and Energy Balance Sheet for H_2S Injection Run 1/18/80

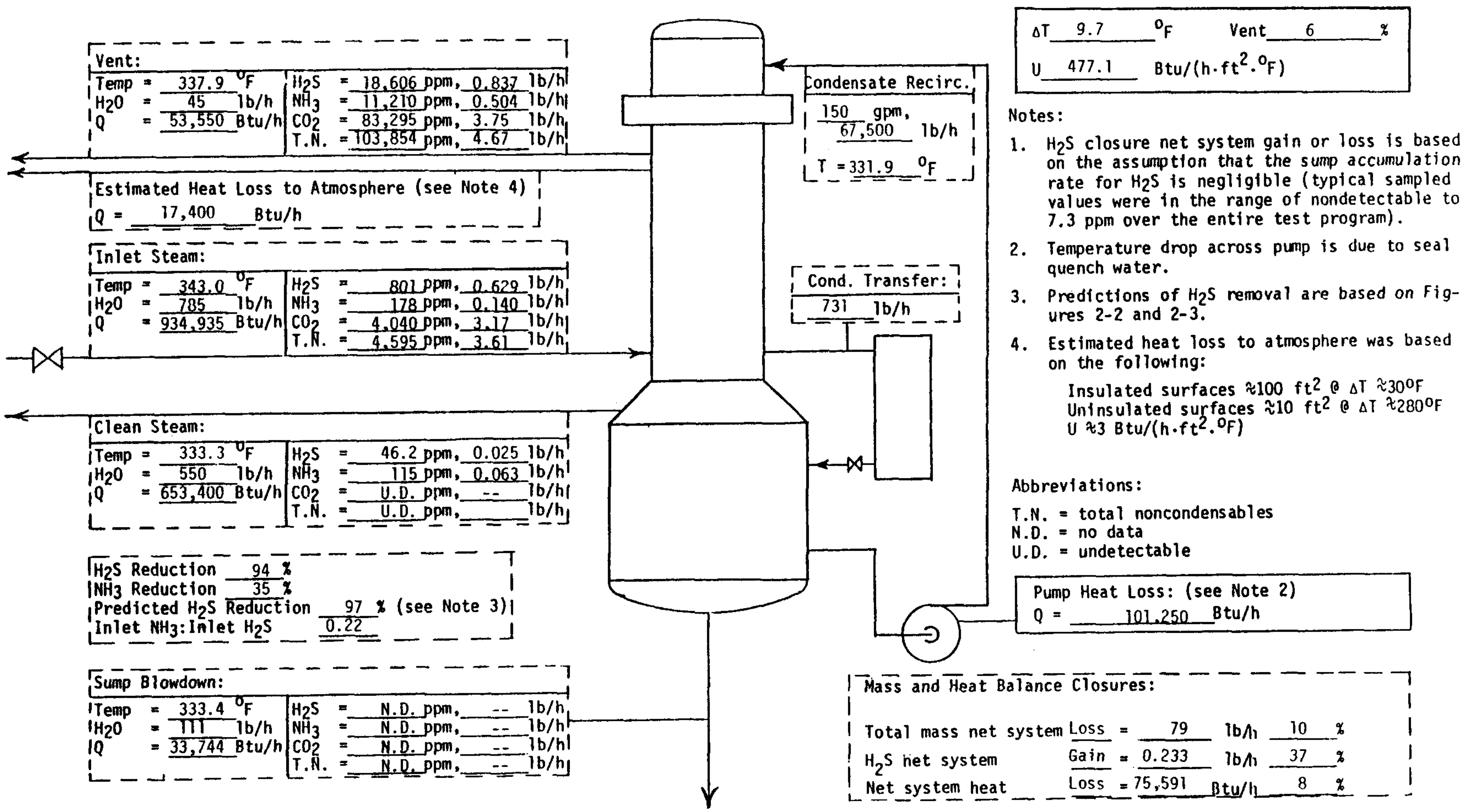
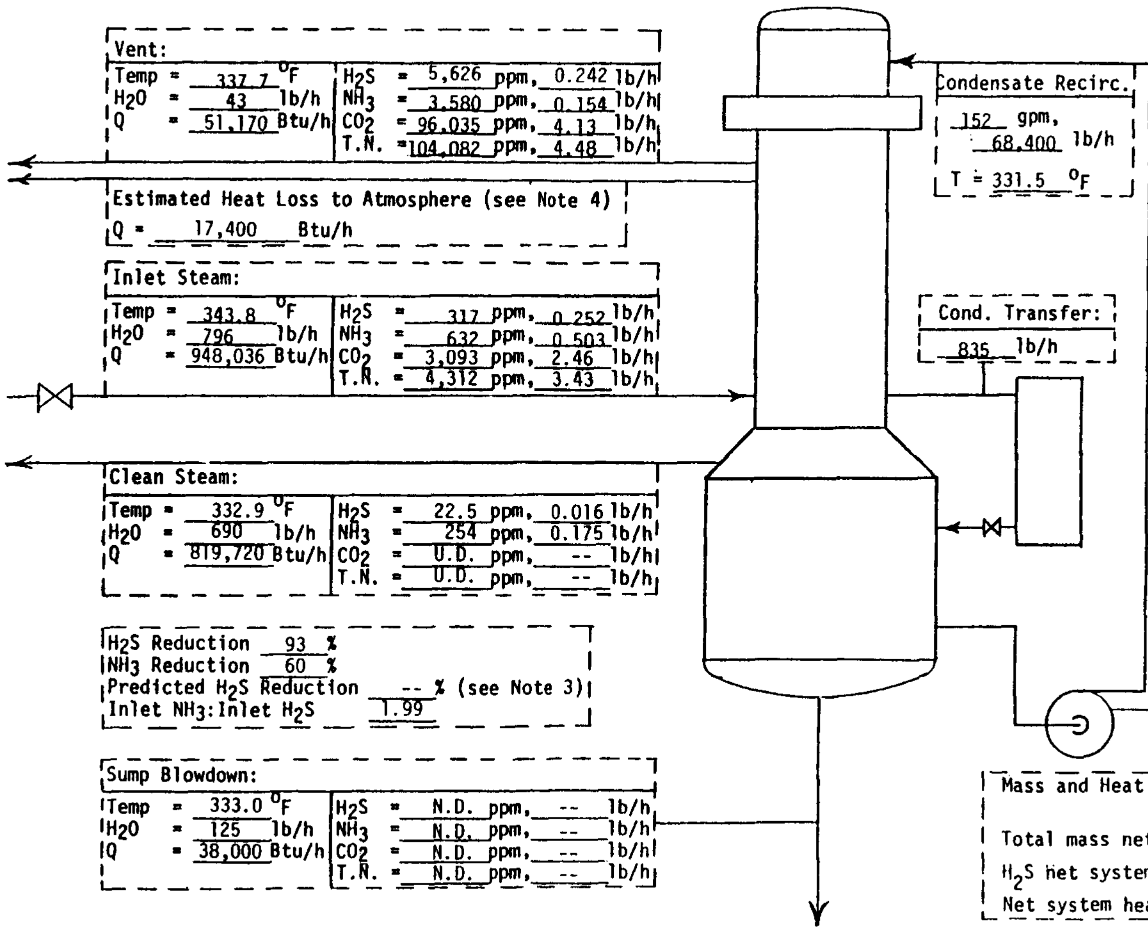


Figure F-3. Mass and Energy Balance Sheet for H_2S Injection Test Run 1/22/80



ΔT 10.9 °F	Vent 5 %
U 484.6 Btu/(h·ft²·°F)	

Notes:

1. H_2S closure net system gain or loss is based on the assumption that the sump accumulation rate for H_2S is negligible (typical sampled values were in the range of nondetectable to 7.3 ppm over the entire test program).
2. Temperature drop across pump is due to seal quench water.
3. Predictions of H_2S removal are based on Figures 2-2 and 2-3.
4. Estimated heat loss to atmosphere was based on the following:

   Insulated surfaces ≈100 ft² @ ΔT ≈30°F
   Uninsulated surfaces ≈10 ft² @ ΔT ≈280°F
   U ≈3 Btu/(h·ft²·°F)

Abbreviations:

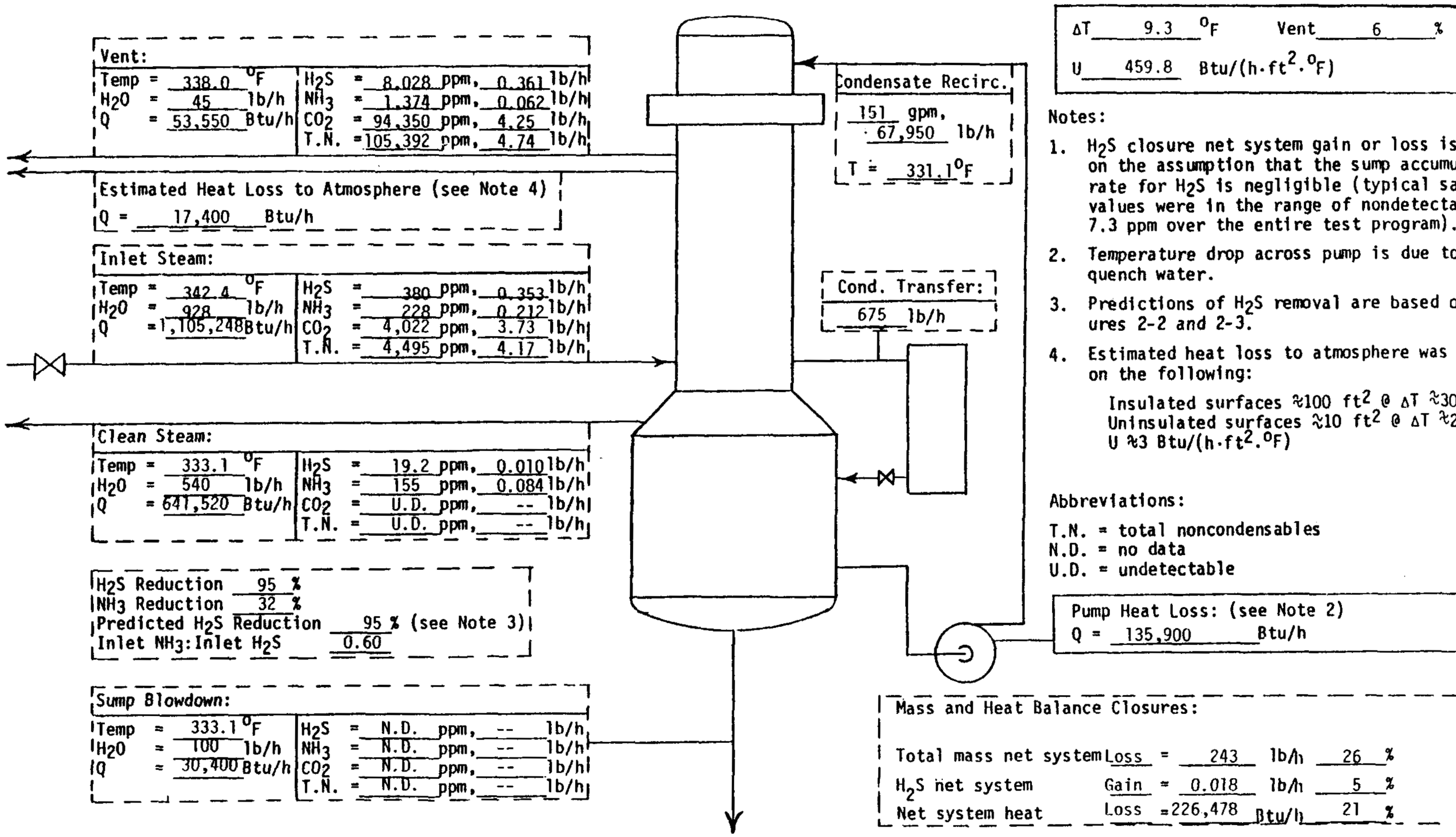
T.N. = total noncondensables
N.D. = no data
U.D. = undetectable

Pump Heat Loss: (see Note 2)
Q = 102,600 Btu/h

Mass and Heat Balance Closures:

Total mass net system	Gain =	62 lb/h	8 %
H_2S net system	Gain =	0.006 lb/h	2 %
Net system heat	Gain =	81,054 Btu/h	9 %

Figure F-4. Mass and Energy Balance Sheet for NH_3 Injection Test Run 1/23/80



Notes:

1. H_2S closure net system gain or loss is based on the assumption that the sump accumulation rate for H_2S is negligible (typical sampled values were in the range of nondetectable to 7.3 ppm over the entire test program).
2. Temperature drop across pump is due to seal quench water.
3. Predictions of H_2S removal are based on Figures 2-2 and 2-3.
4. Estimated heat loss to atmosphere was based on the following:

   Insulated surfaces ≈100 ft^2 @ ΔT ≈30°F
   Uninsulated surfaces ≈10 ft^2 @ ΔT ≈280°F
   U ≈3 Btu/(h·ft^2·°F)

Abbreviations:

T.N. = total noncondensables
N.D. = no data
U.D. = undetectable

Figure F-5. Mass and Energy Balance Sheet for H_2S and NH_3 Injection Test Run 1/24/80

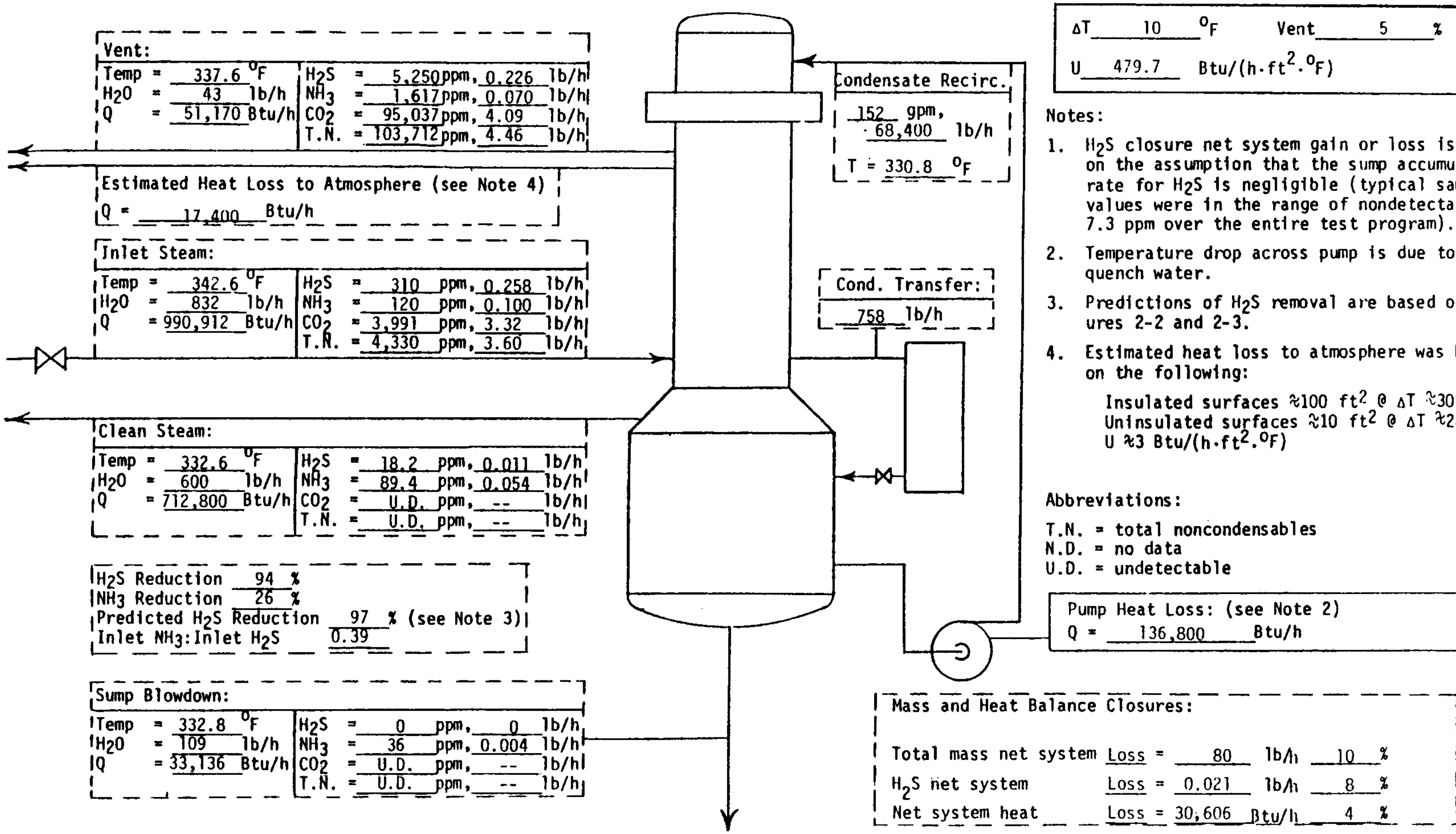


Figure F-6. Mass and Energy Balance Sheet for Baseline Test Run 1/25/80

Appendix G

SPECIAL LABORATORY ANALYSIS FOR AMMONIA, BORON, AND CHLORIDE*

Sample Identification	Ammonia (NH_3) mg/l	Boron (B) mg/l	Chloride (Cl^-) mg/l
Inlet, 12/20/79, 1400 h	120	16	2
Inlet, 12/21/79, 1040 h	92	23	3
Inlet, 12/27/79, 1400 h	133	13	5
Inlet, 12/28/79, 1000 h	130	17	5
Inlet, 1/25/80	--	--	1
Conds. trans., 12/20/79, 1400 h	--	120	1
Conds. trans., 12/21/79, 1040 h	--	41	3
Conds. trans., 12/27/79, 1400 h	--	22	<1
Conds. trans., 12/28/79, 1000 h	--	24	<1
Conds. trans., 1/25/80	--	--	4
Vent, 12/20/79, 1400 h	1680	2	29
Vent, 12/21/79, 1040 h	1140	1	54
Vent, 12/27/79, 1400 h	1008	2	1
Vent, 12/28/79, 1000 h	980	5	34
Vent, 1/25/80	--	--	5
Clean, 12/20/79, 1400 h	71	3	38
Clean, 12/21/79, 1040 h	55	3	20
Clean, 12/27/79, 1400 h	64	2	17
Clean, 12/28/79, 1000 h	55	2	3
Clean, 1/25/80	--	--	7
Blowdown, 12/20/79, 1400 h	--	133	43
Blowdown, 12/21/79, 1040 h	--	115	1
Blowdown, 12/27/79, 1400 h	--	98	2
Blowdown, 12/28/79, 1000 h	--	91	6
Blowdown, 1/25/80	--	--	1

*Performed by PG&E

Appendix H

2.5-MW PILOT PLANT MAJOR EQUIPMENT LIST

Item No.	Nomenclature	Size / Basis	Codes and Standards
HX-001	1st-Stage Heat Exchanger	7855 ft^2 / tube surface area	TEMA, ASME Sec. VIII
HX-002	2nd-Stage Heat Exchanger	445 ft^2 / tube surface area	TEMA, ASME Sec. VIII
HX-003	Vent Gas Condenser	1500 ft^2 / tube surface area	TEMA, ASME Sec. VIII
P-001A	1st-Stage Recirculating Condensate Pump No. 1	37 ft (TDH) 2300 gpm 24 hp / Pump design conditions	Pump - as specified Motor - NEMA
P-001B	1st-Stage Recirculating Condensate Pump No. 2	37 ft (TDH) 2300 gpm 24 hp / Pump design conditions	Pump - as specified Motor - NEMA
P-002A	2nd-Stage Recirculating Condensate Pump No. 1	22 ft (TDH) 330 gpm 2 hp / Pump design conditions	Pump - as specified Motor - NEMA
P-002B	2nd-Stage Recirculating Condensate Pump No. 2	22 ft (TDH) 330 gpm 2 hp / Pump design conditions	Pump - as specified Motor - NEMA
T-001	1st-Stage Condensate Transfer Tank (Pressure Vessel)	12.8 ft^3 / volume	ASME Sec. VIII
T-002	2nd-Stage Condensate Transfer Tank (Pressure Vessel)	0.40 ft^3 / volume	ASME Sec. VIII

Appendix I

2.5-MW PILOT PLANT FIRST- AND SECOND-STAGE HEAT EXCHANGER SPECIFICATIONS

PROCESS EQUIPMENT ITEM NUMBERS HX-001 AND HX-002

Scope

This specification includes the design requirements for one first-stage heat exchanger and one second-stage heat exchanger.

General Configuration

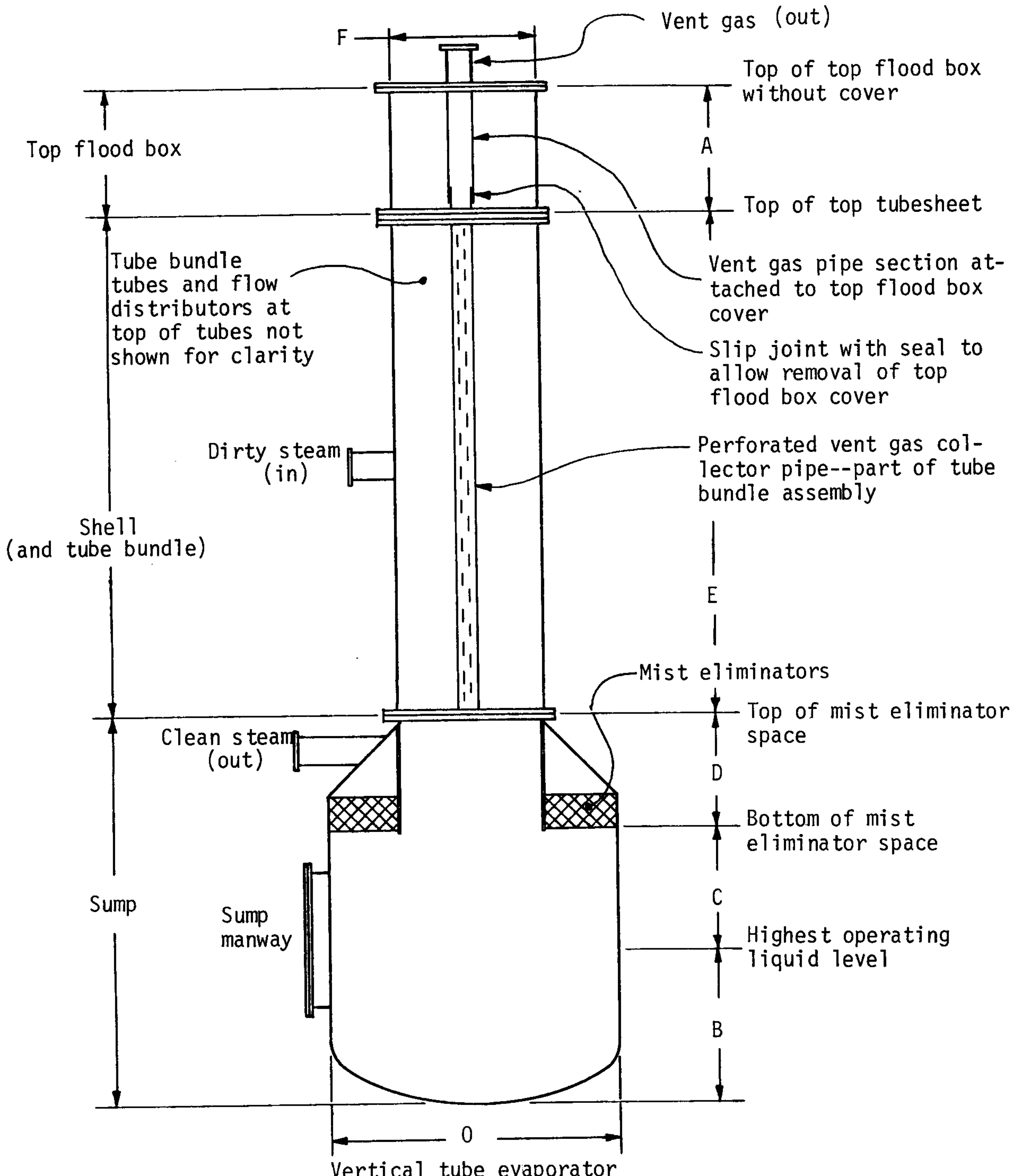
Each heat exchanger shall be a radial flow vertical tube evaporator with a general configuration as shown in Figure I-1. The primary components of each heat exchanger are identified as follows:

1. Top flood box
2. Shell
3. Sump
4. Tube bundle assembly
5. Mist eliminators
6. Flow distributors

The top flood box, shell, and sump shall be connected with flange connections. The support structure shall be designed and attached to the heat exchangers in a manner that allows the top flood box, shell, and sump to be separated and reconnected in the field without cutting or rewelding being required. The tube bundle assembly for each heat exchanger shall have a fixed top tubesheet and a floating bottom tubesheet. Each heat exchanger shall be designed so that the tube bundle assembly can be removed by two options: (1) pulling it vertically after removing the top flood box, or (2) pulling it horizontally after removing the shell and tube bundle as an assembled unit.

Flow distributors shall be located in the top of each tube and shall be accessible through the top flood box cover.

The mist eliminators shall be located inside the sump and shall be accessible for maintenance through the sump manway.



Note: Some of the nozzle connections and other details are omitted for clarity.

Figure I-1. General Configuration of the First- and Second-Stage Heat Exchangers

The heat exchanger dimensional requirements, and nozzle and connection requirements are specified in the following sections of this specification.

Codes and Standards

The heat exchangers shall be code-stamped in accordance with Section VIII, Division 1, of the ASME Boiler and Pressure Vessel Code (the Code). The heat exchangers shall also be designed in accordance with the Code so that full design temperatures and pressures can be experienced by the shellside of the heat exchangers when the flood box and sump manway cover are removed. The heat exchanger designs shall follow the guidelines of the Tubular Exchanger Manufacturers Association (TEMA) as much as possible within the constraints of this specification. All nozzle connection designs shall follow ANSI standards (i.e., flange and pipe thread dimensions).

Design Pressures and Temperatures

The heat exchangers shall be designed in accordance with the following conditions:

- Design temperature - 370^{0}F
- Design pressure - 150 psig

Tube Bundle Design

The tube bundles for each heat exchanger shall be designed based on the general configuration shown in Figures I-1 and I-2 and the dimensions listed in Table I-1. The tube bundle shall be designed for radial flow with an impingement plate directly in front of the shellside steam inlet nozzle as shown in Figure I-2. A perforated vent gas collector pipe shall be located in the center of the tube bundle. The overall heat exchanger design shall allow the collected condensate on top of the bottom tubesheet to be either drained completely during normal operation or periodically as required to eliminate the accumulation of potential corrosive materials.

Shell Design

The shell diameter and length shall be governed by the tube bundle geometry. The shell shall be designed so that the tube bundle can be totally enclosed and fully supported by the shell alone, and so that the seal between the top and bottom tubesheets and the shell is not dependent on the shell being connected to the sump and top flood box.

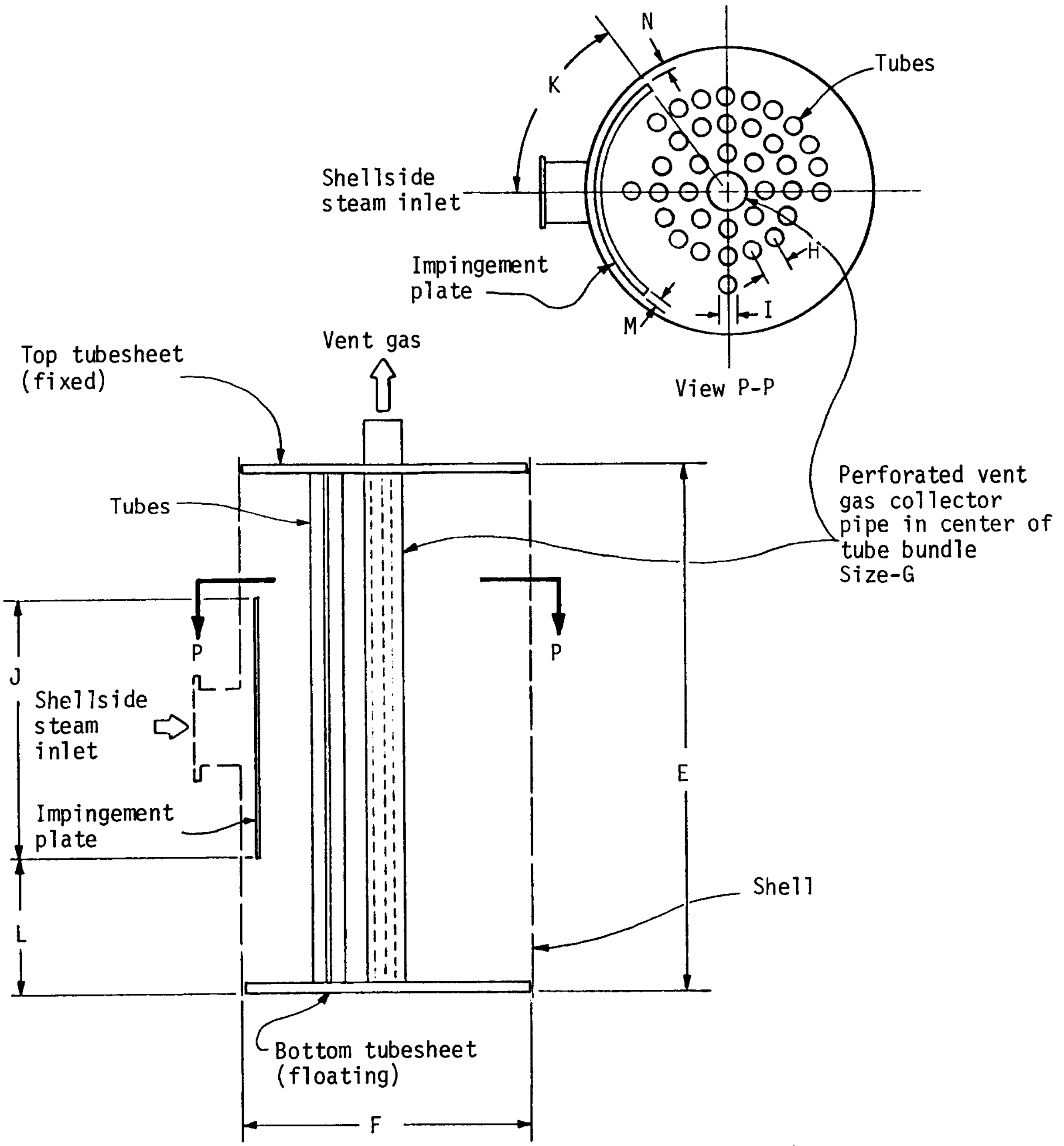


Figure I-2. Tube Bundle Configuration for First- and Second-Stage Heat Exchangers

Table I-1

HEAT EXCHANGER DIMENSIONS

(Refer to Figures I-1 and I-2)

Dimension	Description	First Stage	Second Stage
A	Top flood box length	36 in	18 in
B	Sump high liquid level	72 in	48 in
C	Minimum vapor space below mist eliminators	*	*
D	Distance from bottom of mist eliminators to top of sump	*	*
E	Tube bundle (and shell) length	240 in	96 in
F	Shell diameter (approximate)	60 in	24 in
G	Perforated vent gas collector pipe size	6 in	4 in
H	Tube pitch	1.25 in circular	1.25 in circular
I	Tube diameter	1.00 in	1.00 in
-	Number of tubes	1500	213
-	Tube wall thickness	As required	As required
J	Impingement plate height	120 in	48 in
K	Impingement plate extension beyond shellside steam inlet	45 degrees	45 degrees
L	Distance between bottom tubesheet and bottom of impingement plate	60 in	24 in
M	Impingement plate thickness	**	**
N	Distance between impingement plate and shell	2.0 in	1.0 in
O	Sump diameter	*	*

* As required for proper mist eliminator operation.

** Based on inlet steam velocity of 150 ft/s with some entrained particles, such as sand.

Top Flood Box Design

The top flood box shall be the same diameter as the shell for each heat exchanger and shall be designed with a removable cover.

Sump Design

The sump diameter for each heat exchanger shall be larger than the shell diameter as required for the mist eliminator geometry.

Mist Eliminators

Mist eliminators shall be of a wire mesh pad type and shall be located inside the sump of each heat exchanger. No part of the mist eliminators or their associated components shall be directly under the tube bundle. Design conditions are listed in Table I-2.

Table I-2

MIST ELIMINATOR SPECIFICATIONS

	First Stage	Second Stage
Maximum ΔP	2.0 in H_2O	2.0 in H_2O
Minimum steam flow rate	5000 lb/h	100 lb/h
Maximum steam flow rate	49,000 lb/h	1100 lb/h
Minimum steam temperature	330°F	321°F
Maximum steam temperature	342°F	342°F
Steam pressure	Saturation	Saturation
Efficiency	99%	99%

Flow Distributors

Flow distributors shall be located at the top of each tube inside the top flood box. The flow distributors shall be fabricated from 304 stainless steel, or other suitable materials compatible with the process conditions.

Nozzles, Manways, Access Ports, and View Ports

The typical configuration and orientation of the heat exchanger nozzles is shown in Figures I-3 and I-4. All nozzles and other items shall be located as shown in Tables I-3 and I-4. All manways and access covers shall be designed (hinged if required) so that one person can easily open and close each cover.

Gaskets and Seals

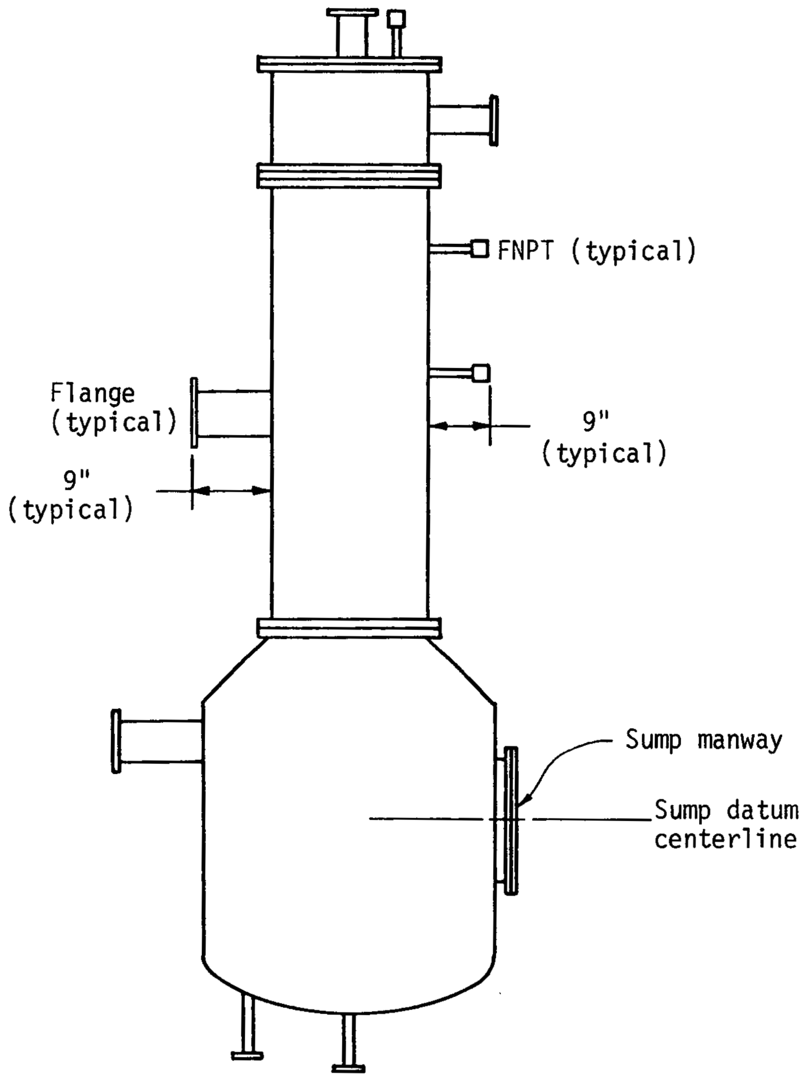
All fixed gaskets and seals shall be either spiral wound stainless-asbestos, self-energizing metal seals (such as metal o-rings, c-seals, omega seals, etc.), or elastomer materials proven to be suitable for the design temperatures and conditions. All fixed seals and gaskets shall be retained by a suitable retaining ring incorporated into the seal or gasket design or by containing the seal or gasket in grooved flange configuration.

Support Structure, Ladders, and Platforms

Each heat exchanger shall be provided with the necessary brackets and other devices for securing the heat exchangers to an appropriate support structure and for attaching ladders and platforms. The support structure, ladders, and platforms will be designed based on the total system requirements which will include, but will not be limited to, the two heat exchangers, pumps, control valves, piping, and instrumentation.

The heat exchanger's support system shall be designed based on all anticipated static and dynamic loads such as the component weights, wind loads, and seismic loads. The support system design shall be based on the heat exchangers and all other components being completely water filled.

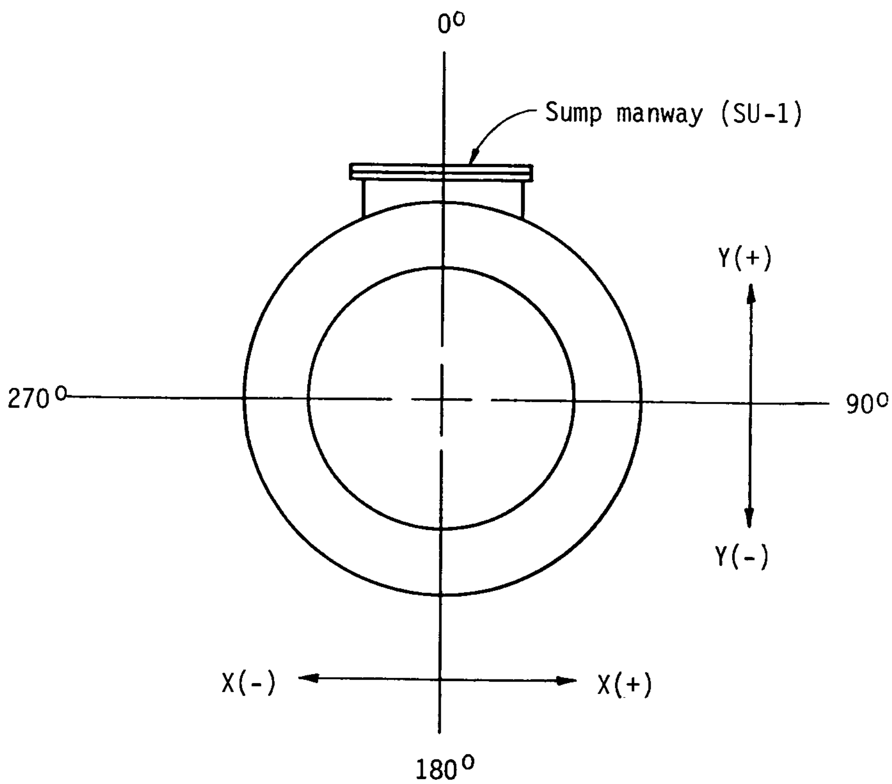
The ladders and platforms shall be placed so that access is provided to all connections, manways, and access ports on the heat exchangers as well as to control valves, instruments, and other items requiring operator attention for the total heat exchanger system. As a minimum requirement, platforms shall be placed at 10-ft elevation intervals completely around the heat exchangers. The heat exchanger platforms shall be designed so that a space of 12 in. is provided between the platforms and the heat exchanger vessels for insulation purposes.



Side view of heat exchanger

Notes:
1. All nozzle connections to be 9 inches long
2. All nozzle flanges to be ANSI Class 150 raised face
3. All FNPT nozzles per ANSI standards
4. For actual nozzle types and locations, refer to Figure I-4 and Tables I-3 and I-4

Figure I-3. Typical Heat Exchanger Nozzle Arrangement



Top view of heat exchanger

Elevation datums:

Sump	-- Sump manway centerline:	above manway - (+) below manway - (-)
Shell	-- Top of bottom tubesheet	
Top flood box	-- Top of top tubesheet	

Figure I-4. Nozzle Orientation

Table I-3

HEAT EXCHANGER NOZZLE LIST--FIRST STAGE

Nozzle Identi-fication	Location	Orientation	Description	Size (inches)	Type	Elevation (inches)	Rotation (degrees)	X (inches)	Y (inches)
SH-1	Shell	Horizontal	Wellhead steam	10	Flange	60	90	NA	NA
SU-2	Sump	Horizontal	Clean steam	10	Flange	As Required	90	NA	NA
FB-3	Flood box	Vertical	Vent gas	6	Flange	NA	NA	0	0
SU-3	Sump	Vertical	Blowdown	1	FNPT	NA	NA	0	0
SH-2	Shell	Horizontal	Condensate transfer	3	Flange	2	270	NA	NA
SU-4	Sump	Horizontal	Recirculating condensate	14	Flange	-78	90	NA	NA
FB-1	Flood box	Horizontal	Recirculating condensate	14	Flange	18	90	NA	NA
SU-5	Sump	Horizontal	Condensate transfer	3	Flange	0	270	NA	NA
SU-6	Sump	Horizontal	Initial fill	3	Flange	0	90	NA	NA
SU-7	Sump	Vertical	Drain	3	Flange	NA	NA	0	+18
SU-1	Sump	Horizontal	Manway	24	Flange	0	0	NA	NA
SU-8	Sump	Horizontal	Pump recircu-lation line	8	Flange	0	180	NA	NA
FB-2	Flood box	Vertical	Vent	6	Flange	NA	NA	0	-15
SU-9	Sump	Horizontal	Vent	6	Flange	12	180	NA	NA

Table I-3 (continued)

Nozzle Identi-fication	Location	Orientation	Description	Size (inches)	Type	Elevation (inches)	Rotation (degrees)	X (inches)	Y (inches)
FB-14	Flood box	Horizontal	Instrument	2	FNPT	6	0	NA	NA
FB-15	Flood box	Horizontal	Instrument	2	FNPT	30	0	NA	NA
FB-16	Flood box	Horizontal	Instrument	2	FNPT	6	180	NA	NA
FB-19	Flood box	Horizontal	Instrument	2	FNPT	30	270	NA	NA
SH-10	Shell	Horizontal	Instrument	2	FNPT	60	180	NA	NA
SH-11	Shell	Horizontal	Instrument	2	FNPT	70	180	NA	NA
SH-12	Shell	Horizontal	Instrument	2	FNPT	196	180	NA	NA
SU-15	Sump	Vertical	Instrument	2	FNPT	+12	135	NA	NA
SU-16	Sump	Horizontal	Instrument	2	FNPT	+12	170	NA	NA
SU-17	Sump	Horizontal	Instrument	2	FNPT	+12	200	NA	NA
SU-18	Sump	Horizontal	Instrument	2	FNPT	+12	60	NA	NA
SU-19	Sump	Horizontal	Instrument	2	FNPT	-12	60	NA	NA
SU-20	Sump	Horizontal	Instrument	2	FNPT	+12	240	NA	NA
SU-22	Sump	Horizontal	Instrument	2	FNPT	-36	240	NA	NA

Table I-3 (continued)

Nozzle Identi-fication	Location	Orientation	Description	Size (inches)	Type	Elevation (inches)	Rotation (degrees)	X (inches)	Y (inches)
SU-23	Sump	Horizontal	Instrument	2	FNPT	-60	240	NA	NA

NOTES:
1. Refer to Figures I-3 and I-4 for nozzle orientation
2. Sump manway centerline is located 90 inches above bottom of sump
3. NA-- Not Applicable
4. FNPT--Female National Pipe Thread

Table I-4

HEAT EXCHANGER NOZZLE LIST--SECOND STAGE

Nozzle Identification	Location	Orientation	Description	Size (inches)	Type	Elevation (inches)	Rotation (degrees)	X (inches)	Y (inches)
SU-2	Sump	Horizontal	Clean steam	3	Flange	As required	90	NA	NA
FB-3	Flood box	Vertical	Vent gas	4	Flange	NA	NA	0	0
SU-3	Sump	Horizontal	Recirculating condensate	6	Flange	-54	90	NA	NA
FB-1	Flood box	Horizontal	Recirculating condensate	6	Flange	9	90	NA	NA
SH-2	Shell	Horizontal	Condensate transfer	1	FNPT	2	270	NA	NA
SH-3	Shell	Horizontal	Inlet steam	6	Flange	24	90	NA	NA
SU-4	Sump	Horizontal	Condensate transfer	1	FNPT	0	270	NA	NA
SU-5	Sump	Horizontal	Initial fill	2	FNPT	0	90	NA	NA
SU-6	Sump	Vertical	Drain	3	FNPT	NA	NA	0	+12
SU-1	Sump	Horizontal	Manway	24	Flange	0	0	NA	NA
SH-10	Shell	Horizontal	Instrumentation	2	FNPT	80	180	NA	NA
SH-11	Shell	Horizontal	Instrumentation	2	FNPT	60	180	NA	NA
SH-12	Shell	Horizontal	Instrumentation	2	FNPT	70	180	NA	NA
SU-8	Sump	Vertical	Blowdown	1	FNPT	NA	NA	0	0

Table I-4 (continued)

Nozzle Identification	Location	Orientation	Description	Size (inches)	Type	Elevation (inches)	Rotation (degrees)	X (inches)	Y (inches)
SU-9	Sump	Horizontal	Vent	4	Flange	12	180	NA	NA
FB-2	Flood box	Vertical	Vent	4	FNPT	NA	NA	0	-12
FB-10	Flood box	Vertical	Vent	1	FNPT	NA	NA	0	+6
FB-14	Flood box	Horizontal	Instrumentation	2	FNPT	3	0	NA	NA
FB-15	Flood box	Horizontal	Instrumentation	2	FNPT	15	0	NA	NA
FB-16	Flood box	Horizontal	Instrumentation	2	FNPT	3	180	NA	NA
SU-7	Sump	Horizontal	Pump recirculation line	4	Flange	0	180	NA	NA
SU-12	Sump	Horizontal	Vent	4	FNPT	+12	135	NA	NA
SU-13	Sump	Horizontal	Instrumentation	2	FNPT	+12	170	NA	NA
SU-14	Sump	Horizontal	Instrumentation	2	FNPT	+12	200	NA	NA
SU-16	Sump	Horizontal	Instrumentation	2	FNPT	+12	60	NA	NA
SU-17	Sump	Horizontal	Instrumentation	2	FNPT	-12	60	NA	NA
SU-20	Sump	Horizontal	Instrumentation	2	FNPT	+12	240	NA	NA
SU-22	Sump	Horizontal	Instrumentation	2	FNPT	-27	240	NA	NA

Table I-4 (continued)

Nozzle Identification	Location	Orientation	Description	Size (inches)	Type	Elevation (inches)	Rotation (degrees)	X (inches)	Y (inches)
SU-23	Sump	Horizontal	Instrumentation	2	FNPT	-48	240	NA	NA

NOTES:
1. Refer to Figures I-3 and I-4 for nozzle orientation
2. Sump manway centerline is located 66 inches above bottom of sump
3. NA--Not Applicable
4. FNPT--Female National Pipe Thread

Materials of Construction

The heat exchangers shall be fabricated completely of 304 stainless steel except for bolting materials, flow distributors, and seals and gaskets. The bolting materials shall be selected based on considerations of strength requirements, corrosion, galling, and galvanic reactions. The seals, gaskets, and flow distributors shall be as previously specified in this specification.

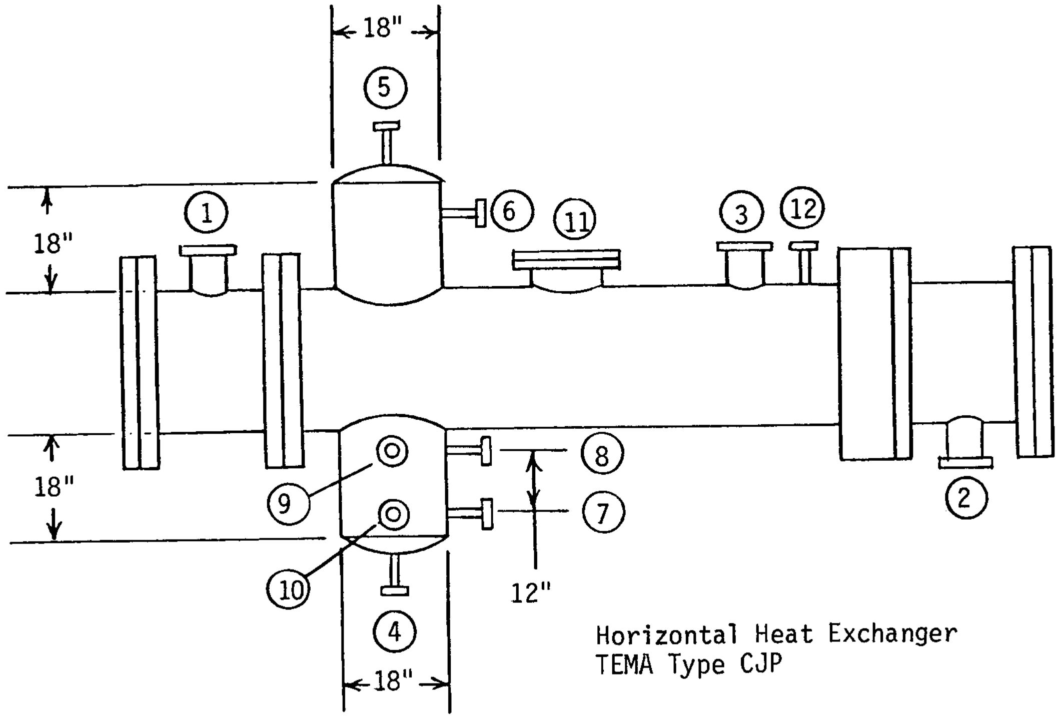
Appendix J

2.5-MW PILOT PLANT VENT GAS CONDENSER SPECIFICATION SHEET

Process Equipment Item Number:	HX-003	
TEMA Heat Exchanger Type:	CJP, horizontal	
Surface Area:	1490 ft^2	
Performance of Unit:	Shellside	Tubeside
Fluid description	2nd-stage vent gas[1]	Cooling water[2]
Fluid quantity, lb/h	1450	26,750
Water vapor, in/out	-/5	0/0
Steam	1338/-	-/-
Water	-/1333	26,750/26,750
Noncondensables	112/112	-/-
Temperature, °F, in/out	215/120	85/135
Density, lb/ft^3, in/out	0.038/62.4[3], 0.124[4]	-/-
Molecular weight, noncondensables	42	-
Inlet pressure, psig	5	12
Pressure drop allowed (calculated) psi	NA	2
Fouling resistance	0.0005	0.003
Heat exchanged:	1.33×10^6 Btu/h	
LMTD:	20°F[5]	
Heat transfer rate:	50 Btu/(h·ft^2·°F) (est.)	
Construction of One Shell:		
Design/test pressure, psig	150/code	150/code
Design temperature, °F	370	370
No. passes per shell	1	7
Corrosion allowance, in.	1/16	1/16
Gaskets	Metal or asbestos	Metal or asbestos
Tube type:	Smooth	
Material:	304 stainless steel	
Sketch showing connection size and rating:	See Figure J-1	

Standards and Code Requirements: TEMA, Class C, and ASME, Section VIII

[1]Consists of steam with following approximate noncondensable loadings: 32,900-134,700 ppm CO_2; 2450-9800 ppm H_2S; 200-2550 ppm NH_3; and 400-2550 ppm other (N_2, CH_4, H_2)
[2]Filtered cooling tower water
[3]Condensate
[4]Gas
[5]Based on 120°F on shellside



Nozzle No.	Description	Size	Type
1	Inlet, tubeside	4"	Flange, ANSI RF
2	Outlet, tubeside	4"	Flange, ANSI RF
3	Inlet, shellside	8"	Flange, ANSI RF
4	Condensate outlet, shellside	2"	Flange, ANSI RF
5	Gas outlet, shellside	1½"	Flange, ANSI RF
6	Instrument, shellside	1"	Flange, ANSI RF
7	Instrument, shellside	1"	Flange, ANSI RF
8	Instrument, shellside	1"	Flange, ANSI RF
9	Instrument	1"	Flange, ANSI RF
10	Instrument	1"	Flange, ANSI RF
11	Inspection Port, shellside	8"	Flange, ANSI RF
12	Vacuum breaker	1"	Flange, ANSI RF

Figure J-1. Vent Gas Condenser Configuration

Appendix K

2.5-MW PILOT PLANT FIRST-STAGE PUMP SPECIFICATION SHEET

Process Equipment Item Nos.: P-001A and P-001B

Type: Horizontal Centrifugal or Vertical Can

Design Conditions per Pump--Flow: 2300 gpm
TDH: 37 ft
Approx. Mechanical hp: 24

Piping System Design Conditions--Temperature: $370^{\circ}F$ Pressure: 150 psig

Available Net Positive Suction Head: 6 ft

Operating Fluid Temperature Range: $330^{\circ}F$ to $345^{\circ}F$

Fluid: Water with possible significant concentrations of H_2S, NH_3, boron, and chlorides

Accessories: Electric motor driver, coupling, base plates

Seals: Stuffing box design suitable for process conditions

Materials: All wetted parts of 316 stainless steel construction, all other parts of materials suitable for process conditions and environment

Motor Type: NEMA-TEFC of suitable construction for specified environment

Performance Curves: Certified performance curves are required

Appendix L

2.5-MW PILOT PLANT SECOND-STAGE PUMP SPECIFICATION SHEET

Process Equipment Item Nos.: P-002A and P-002B

Type: Horizontal Centrifugal or Vertical Can

Design Conditions per Pump--Flow: 330 gpm
TDH: 22 ft
Approx. Mechanical hp: 2

Piping System Design Conditions--Temperature: 370°F Pressure: 150 psig

Available Net Positive Suction Head: 6 ft

Operating Fluid Temperature Range: 320°F to 345°F

Fluid: Water with possible significant concentrations of H_2S, NH_3, boron, and chlorides

Accessories: Electric motor driver, coupling, base plates

Seals: Stuffing box design suitable for process conditions

Materials: All wetted parts of 316 stainless steel construction, all other parts of materials suitable for process conditions and environment

Motor Type: NEMA-TEFC of suitable construction for specified environment

Performance Curves: Certified performance curves are required

Appendix M

2.5-MW PILOT PLANT FIRST-STAGE CONDENSATE TRANSFER TANK SPECIFICATION SHEET

Process Equipment Item No.: T-001

Code Requirements: Code stamped in accordance with ASME Section VIII

Testing: Per the above code, will be witnessed

Materials of Construction: 304 stainless steel

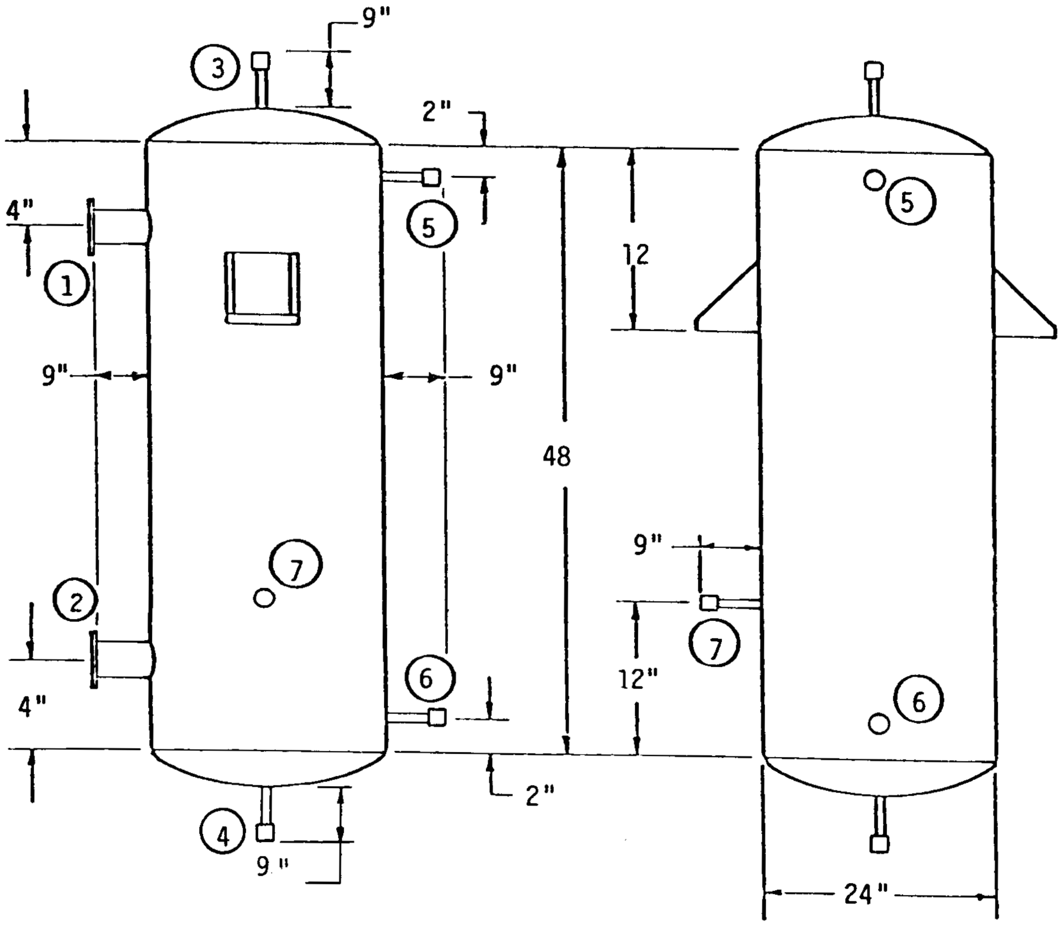
Fabrication: All welds to be heat-treated

Design Temperature: 370°F

Design Pressure: 150 psig

Fluid: Water with possible significant concentrations of H_2S, NH_3, and other species that might encourage stress corrosion

Corrosion Allowance: 1/16 in. minimum



Nozzle Number	Description	Size	Type
1	Inlet	3"	Flange, ANSI RF
2	Outlet	3"	Flange, ANSI RF
3	Vent	1"	FNPT
4	Drain	3/4"	FNPT
5-7	Instrument	3/4"	FNPT

Figure M-1. First-stage Condensate Transfer Tank Configuration

Appendix N

2.5-MW PILOT PLANT SECOND-STAGE CONDENSATE TRANSFER TANK SPECIFICATION SHEET

Process Equipment Item No.: T-002

Code Requirements: Code stamped in accordance with ASME Section VIII

Testing: Per the above code, will be witnessed

Materials for Construction: 304 stainless steel

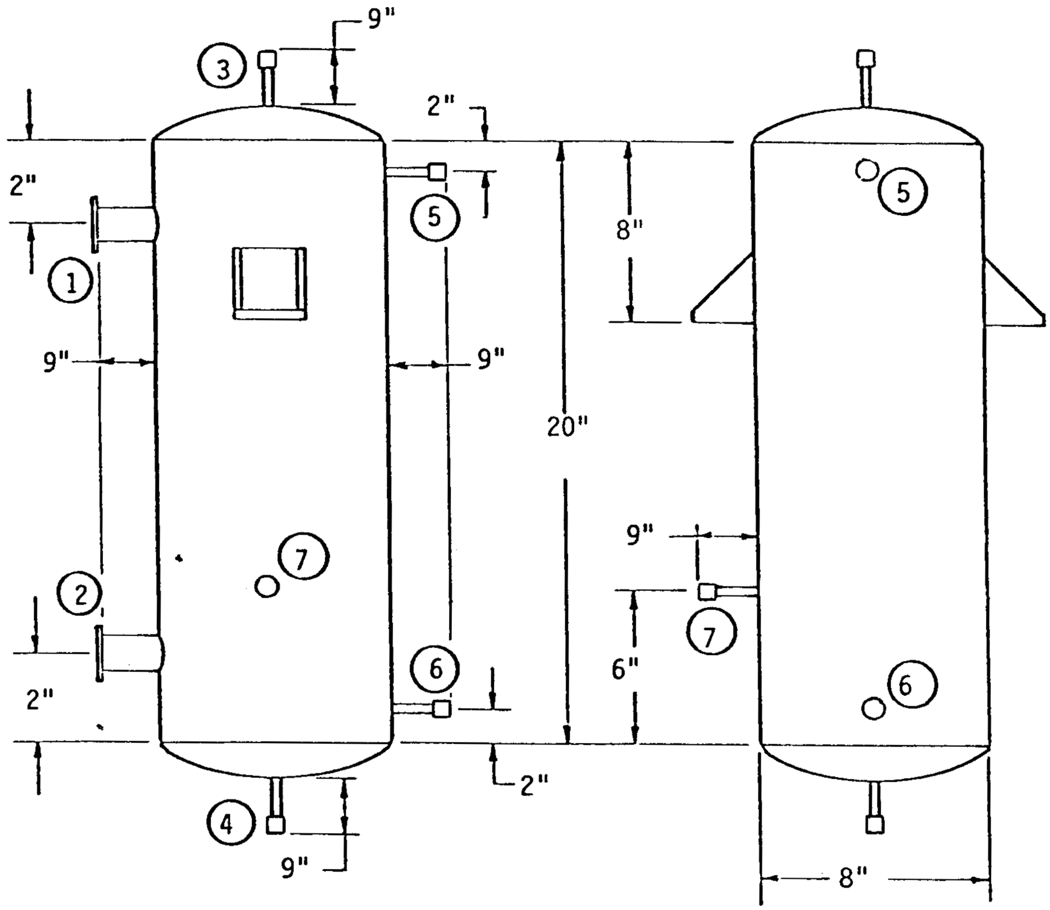
Fabrication: All welds to be heat-treated

Design Temperature: 370°F

Design Pressure: 150 psig

Fluid: Water with possible significant concentrations of H_2S, NH_3, and other species that might encourage stress corrosion

Corrosion Allowance: 1/16 in. minimum



Nozzle Number	Description	Size	Type
1	Inlet	1"	Flange, ANSI RF
2	Outlet	1"	Flange, ANSI RF
3	Vent	1"	MNPT
4	Drain	3/4"	MNPT
5-7	Instrument	3/4"	MNPT

Figure N-1. Second-stage Condensate Transfer Tank Configuration

Appendix O

2.5-MW PILOT PLANT CONTROL VALVE LIST

Valve Number	Line Number	Valve Type	Fail	Service	Upstream Temperature (°F)	Upstream Pressure (psia)	Design Flow	ΔP at Design Flow (psi)
AOV-100	10"-WHS-1-SS1	Globe	Closed	Steam	347	129	52,600 lb/h	3
PV-111	10"-CS-1-SS1	Globe	As is	Steam	337	113	50,000 lb/h	2
LV-118	3"-F-1-SS1	Globe	As is	Water	70	113	1.06 gpm	5
FV-120	1"-BD-1-SS1	Globe	Closed	Water	337	113	1.11 gpm	15
FV-133	8"-C-7-SS1	Globe	As is	Water	370	200	1,125 gpm	5
FV-134	14"-C-6-SS1	Butterfly	As is	Water	370	200	3,000 gpm	1
LV-137	3"-C-9-SS1	Globe	As is	Water	370	150	111.4 gpm	15
FV-211	3"-CS-2-SS1	Globe	Closed	Steam	370	150	1,100 lb/h	5
LV-218	2"-F-2-SS1	Globe	As is	Water	370	150	0.01 gpm	5
FV-220	1"-BD-2-SS1	Globe	As is	Water	370	150	0.01 gpm	15
FV-223	4"-C-17-SS1	Globe	As is	Water	370	200	160 gpm	15
FV-234	6"-C-16-SS1	Butterfly	As is	Water	370	200	426 gpm	1
LV-237	1"-C-19-SS1	Globe	As is	Water	370	150	2.6 gpm	15
FV-240A	8"-VT-3-SS1	Globe	As is	Steam	370	150	1,450 lb/h	15
FV-240B	6"-VT-1-SS1	Globe	Closed	Steam	370	150	2,600 lb/h	2
TV-300	4"-CW-1-SS2	Globe	Open	Water	To be determined		59.5 gpm	5
PV-306	1½"-VT-4-SS1	Globe	Closed	Gas	370	150	260 lb/h	2.5
LV-309	2"-C-20-SS1	Globe	As is	Water	370	150	2.3 gpm	2.5

Appendix P

2.5-MW PILOT PLANT MANUAL VALVE LIST

Type	Size (in.)	Quantity
Gate	14	6
	10	4
	8	3
	6	8
	4	7
	3	9
	2	4
	1 1/2	2
	1	12
Globe, throttling	12	1
	8	2
	6	2
	4	2
	3	3
	2	1
	1	4
	3/4	3
	1/2	3
Globe, tight shutoff	3	1
	2	1
	3/4	4
	1/2	122
Check, in-line	14	2
	8	1
	6	2
	4	1
Check, vacuum breakers	1	7

Appendix Q

2.5-MW PILOT PLANT LINE LIST

Line Identification Size/serv/no/ mat'l spec	Fluid (Phase)	Design Conditions					Operating Conditions			Insul. Spec. Basis*	Location	
		Design Flow		Specific Volume V (ft^3/lb)	Design Temp T (°F)	Design Press P (psia)	Velocity @ Design Flow (ft/s)	Oper. Temp. T (°F)	Oper. Press P (psia)		From	To
		M (lb/h)	Q (gpm)									
10"-WHS-1-SS1	Steam	52,600		3.47	347	129	92.5	335-350	113-140	C	IF1	HX001-SH1
10"-CS-1-SS1	Steam	50,000		3.94	337	113	99.9	330-342	103-121	C	HX001-SU2	PG&E-IF2
3"-CS-2-SS1	Steam	1,140		3.99	336	112	24.6	321-342	91-121	C	HX002-SU2	IF4
6"-PE-1-SS1	Steam	--		--	337	113	--	330-342	122-140	C	HX001-SU9	HX001-FB2
4"-PE-2-SS1	Steam	--		--	336	112	--	321-342	120-140	C	HX002-SU12	HX002-FB2
5"-VT-1-SS1	Gas	2,630		3.52	346	127	12.8	336-350	111-126	C	HX001-FB3	HX002-SH3
4"-VT-2-SS1	Gas	1,490		3.61	344	125	16.9	320-350	109-125	B	HX002-FB3	8"VT-3-SS1
3"-VT-3-SS1	Gas	1,490		16.31	344	25	19.4	330-350	15-25	B	4"VT-2-SS1	HX003-N3
1 1/2"-VT-4-SS1	Gas	280		22.2	120	18	122	95-120	18	B	HX003-N5	IF7
14"-C-1-SS1	Water		3,000		337	113	7.1	330-342	103-121	C	HX001-SU4	14"-C-2-SS1
14"-C-2-SS1	Water		2,250		337	113	5.3	330-342	103-121	C	14"-C-1-SS1	P001A(suction)
14"-C-3-SS1	Water		2,250		337	113	5.3	330-342	103-121	C	14"-C-1-SS1	P002A(suction)
14"-C-4-SS1	Water		2,250		337	130	5.3	330-342	122-140	C	P001A(discharge)	14"-C-6-SS1
14"-C-5-SS1	Water		2,250		337	130	5.3	330-342	122-140	C	P001B(discharge)	14"-C-6-SS1
14"-C-6-SS1	Water		3,000		337	130	7.1	330-342	122-140	C	14"-C-4-SS1	HX001-FB1
8"-C-7-SS1	Water		1,125		337	130	7.2	330-342	122-140	C	14"-C-6-SS1	HX001-SU8
3"-C-8-SS1	Water		112		347	129	4.8	333-350	107-130	C	HX001-SH3	T001-N1
3"-C-9-SS1	Water		112		347	129	4.8	333-350	107-130	C	T001-N2	HX001-SU5
6"-C-11-SS1	Water		426		336	112	4.7	320-342	109-121	C	HX002-SU3	6"-C-12-SS1
6"-C-12-SS1	Water		320		336	112	3.6	320-342	109-121	C	6"-C-11-SS1	P002A(suction)
6"-C-13-SS1	Water		320		336	112	3.6	320-342	109-121	C	6"-C-11-SS1	P002B(suction)
6"-C-14-SS1	Water		320		336	122	3.6	320-342	109-112	C	P002A(discharge)	6"-C-16-SS1
6"-C-15-SS1	Water		320		336	122	3.6	321-342	110-130	C	P002B(discharge)	6"-C-16-SS1
6"-C-16-SS1	Water		426		336	122	4.7	321-342	110-130	C	6"-C-14-SS1	HX003-N3
4"-C-17-SS1	Water		160		336	122	4.0	321-342	110-130	C	6"-C-16-SS1	4"HX002-SU7
1"-C-18-SS1	Water		2.6		346	127	1.0	336-350	111-130	C	HX002-SH2	T002-N1
1"-C-19-SS1	Water		2.6		346	127	1.0	336-350	111-130	C	T002-N2	HX002-SU4

Appendix Q (continued)

Line Identification Size/serv/no/ mat'l spec	Fluid (Phase)	Design Conditions					Operating Conditions			Insul. Spec. Basis *	Location	
		Design Flow		Specicic Volume V (ft^3/lb)	Design Temp T (°F)	Design Press P (psia)	Velocity @Design Flow (ft/s)	Oper. Temp. T (°F)	Oper. Press P (psia)		From	To
		M (lb/h)	Q (gpm)									
1"-C-20-SS1	Water		2.4		120	18	1.0	95-120	18	B	HX003-N4	6"-DR-3-SS1
4"-CW-1-SS2	Water		54.0		85	63	1.5	85	63	A	IF8	HX003-N1
4"-CW-2-SS2	Water		54.6		135	51	1.5	135	51	A	HX003-N2	IF9
1"-SW-1-SS1	Water		2		80-100	15-30	1.5	80	20	A	IF5	1/2"-SW-3-SS1
3/4"-SW-2-SS1	Water		1		80-100	15-30	1.2	80	20	A	1"-SW-1-SS1	1/2"-SW-5-SS1
1/2"-SW-3-SS1	Water		0.5		80-100	15-30	0.5	80	20	A	3/4"-SW-7-SS1	P001A(seal)
1/2"-SW-4-SS1	Water		0.5		80-100	15-30	0.5	80	20	A	3/4"-SW-7-SS1	P001B(seal)
1/2"-SW-5-SS1	Water		0.5		80-100	15-30	0.5	80	20	A	3/4"-SW-2-SS1	P002A(seal)
1/2"-SW-6-SS1	Water		0.5		80-100	15-30	0.5	80	20	A	3/4"-SW-2-SS1	P002B(seal)
3/4"-F-1-SS1	Water		1.06		70	113	0.05	70	100-115	C	IF6	HX001-SU6 2"-F-2-SS1
2"-F-2-SS1	Water		0.02		70	112	0.002	70	100-115	C	3"-F-1-SS1	HX002-SU5
1"-BD-1-SS1	Water		1.11		337	113	0.41	330-342	103-121	B	HX001-SU3	6"-DR-3-SS1
1"-BD-2-SS1	Water		0.03		336	112	0.01	328-340	101-119	B	HX002-SU3	6"-DR-3-SS1
3"-DR-1-SS1	Water		100		337	113	10.5	200-340	100-120	A	HX001-SU7	6"-DR-3-SS1
3"-DR-2-SS1	Water		100		336	112	10.5	200-340	100-120	A	HX002-SU6	6"-DR-3-SS1
6"-DR-3-SS1	Water		215		330	110	11.4	100-340	100-120	B	--	IF3
1"-DR-4-SS1	Water		10		347	129	3.1	335-350	110-140	A	10"-WHS-1-SS1	6"-DR-3-SS1
1"-DR-5-SS1	Water		5		346	127	1.5	335-350	110-140	A	6"-VT-1-SS1	6"-DR-3-SS1

* A--No insulation

B--Insulation thickness for personnel protection only

C--Insulation thickness for process efficiency

Appendix R

2.5-MW PILOT PLANT GENERAL PIPING SYSTEM SPECIFICATIONS

GENERAL

All detailed piping system specifications and components shall comply with the Power Piping Code ANSI B31.1. All piping, fittings, valve bodies, and other components shall be of commercial quality and design, suitable for outdoor power plant applications.

MATERIALS

All piping, fittings, valve bodies, and other piping components shall be type 304 stainless steel with the following specifications:

- Piping -- ASTM A312
- Forgings -- ASTM A182
- Bar stock -- ASTM A479
- Plate -- ASTM A240

All valves shall have stainless steel trim; all other valve wetted and nonwetted parts shall be of materials suitable for corrosive fluids and corrosive ambient conditions.

MECHANICAL CONNECTIONS

All mechanical piping connections shall be threaded connections for 2-inch and smaller pipe and flanged connections for pipe sizes larger than 2 inches. All thread and flange dimensions shall comply with the appropriate ANSI standards including the Power Piping Code (ANSI B31.1) and those standards that dictate dimensions and sizes. All flanges shall be raised-face flanges. All gaskets shall be spiral-wound stainless steel and asbestos with stainless steel backing rings.

WELDED CONNECTIONS

All welded piping connections shall be socket-weld for 2-inch and smaller pipe and butt-weld for pipe sizes larger than 2 inches. All socket dimensions, weld prep dimensions, welding specifications, and inspection requirements shall comply with the Power Piping Code (ANSI B31.1) and all other appropriate ANSI standards dictating dimensions, sizes, welding practices, and inspection requirements.

PIPE SCHEDULES

The minimum pipe schedule for all sizes shall be Schedule 40.

DESIGN CONDITIONS

- Pipe specification SS1: Service -- general process piping
  Pressure -- 150 psig
  Temperature -- 370°F

- Pipe specification SS2: Service -- low pressure, low temperature
  Pressure -- to be determined
  Temperature -- to be determined

Appendix S

2.5-MW PILOT PLANT INSTRUMENTATION LIST

Instrument Number	Measured Parameter	Components and Functions	Data Monitoring Channels
100	Wellhead steam pressure	Indicator/transmitter	1
101	Wellhead steam temperature	Temperature element Indicator/transmitter	1
102	Wellhead steam flow	Flow element Transmitter Indicator Integrator	1
104	Wellhead steam quality	Calorimeter Temperature element Indicator/transmitter	1
105	Shellside pressure	Indicator	0
106	Shellside temperature	Indicator	0
107	Flood box level	Indicator/transmitter	1
108	Flood box level	Level gauge	0
109	Vent steam quality	Calorimeter Temperature element Indicator/transmitter	1
110	ΔP; wellhead to clean steam	ΔP transmitter	1
111	Clean steam flow	Flow element Transmitter Indicator Integrator Indicator/controller	1
112	Clean steam quality	Calorimeter Temperature element Indicator/transmitter	1
113	Clean steam temperature	Temperature element Indicator/transmitter	1
114	Clean steam pressure	Indicator/transmitter	1

Appendix S (continued)

Instrument Number	Measured Parameter	Components and Functions	Data Monitoring Channels
116	Sump temperature	Temperature element Indicator/transmitter	1
117	Sump pressure	Indicator/transmitter	1
118	Sump level	Indicator/transmitter Indicator/controller (2) High-level sensor Low-level sensor High-level alarm High, high-level/unit shutdown Low, low-level/unit shutdown Low, low-level/pump shutdown	1
119	Sump level	Level gauge	0
120	Blowdown flow	Flow element Transmitter Indicator Integrator (2) Indicator/controller	1
122	Recirculating temperature	Temperature element Indicator/transmitter	1
123	Recirculating conductance	Conductance element Indicator/transmitter High-level sensor High-level alarm	1
124	Pump suction pressure	Indicator	0
125	Pump suction pressure	Indicator	0
126	Pump discharge flow	Low-flow sensor	0
127	Pump discharge flow	Low-flow sensor	0
128	Pump discharge pressure	Indicator	0
129	Pump discharge pressure	Indicator	0
130	Seal water flow	Controller	0
131	Seal water flow	Controller	0

Appendix S (continued)

Instrument Number	Measured Parameters	Components and Functions	Data Monitoring Channels
132	Recirculation pressure	Indicator/transmitter	1
133	Recirculation flow	Flow element Transmitter Indicator Integrator Indicator/controller	1
134	Recirculation flow	Flow element Transmitter Indicator Integrator Indicator/controller	1
135	Recirculation temperature	Temperature element Indicator/transmitter	1
136	Condensate temperature	Temperature element Indicator/transmitter	1
137	Condensate level	Indicator/transmitter Indicator/controller High-level alarm Low-level sensor High-level sensor Low, low-level/unit shutdown	1
138	Condensate flow	Flow element Transmitter Indicator Integrator	1
140	Vent steam flow	Flow element Transmitter Indicator Integrator	1
141	Vent steam temperature	Temperature element Indicator/transmitter	1
142	Vent steam pressure	Indicator/transmitter	1
205	Shellside pressure	Indicator	0
206	Shellside temperature	Indicator	0
207	Flood box level	Indicator/transmitter	1
208	Flood box level	Level gauge	0

Appendix S (continued)

Instrument Number	Measured Parameter	Components and Functions	Data Monitoring Channels
210	ΔP; inlet to clean steam	ΔP transmitter	1
211	Clean steam flow	Flow element Transmitter Indicator Integrator Indicator/controller	1
213	Clean steam temperature	Temperature element Indicator/transmitter	1
214	Clean steam pressure	Indicator/transmitter	1
216	Sump temperature	Temperature element Indicator/transmitter	1
217	Sump pressure	Indicator/transmitter	1
218	Sump level	Indicator/transmitter Indicator/controller (2) High-level sensor Low-level sensor High-level alarm High, high-level/unit shutdown Low, low-level/unit shutdown Low, low-level/pump shutdown	1
219	Sump level	Level gauge	0
220	Blowdown flow	Flow element Transmitter Indicator Integrator (2) Indicator/controller	1
222	Recirculating temperature	Temperature element Indicator/transmitter	1
223	Recirculating conductance	Conductance element Indicator/transmitter High-level sensor High-level alarm	1
224	Pump suction pressure	Indicator	0
225	Pump suction pressure	Indicator	0
226	Pump discharge flow	Low-flow sensor	0

Instrument Number	Measured Parameter	Components and Functions	Data Monitoring Channels
227	Pump discharge flow	Low-flow sensor	0
228	Pump discharge pressure	Indicator	0
229	Pump discharge pressure	Indicator	0
230	Seal water flow	Controller	0
231	Seal water flow	Controller	0
232	Recirculation pressure	Indicator/transmitter	1
233	Recirculation flow	Flow element Transmitter Indicator Integrator Indicator/controller	1
234	Recirculation flow	Flow element Transmitter Indicator Integrator Indicator/controller	1
235	Recirculation temperature	Temperature element Indicator/transmitter	1
236	Condensate temperature	Temperature element Indicator/transmitter	1
237	Condensate level	Indicator/transmitter Indicator/controller High-level alarm Low-level sensor High-level sensor Low, low-level/unit shutdown	1
238	Condensate flow	Flow element Transmitter Indicator Integrator	1
240	Vent steam flow	Flow element Transmitter Indicator Integrator Indicator controller (2) Switch (2)	1
241	Vent steam temperature	Temperature element Indicator/transmitter	1

Appendix S (continued)

Instrument Number	Measured Parameters	Components and Functions	Data Monitoring Channels
242	Vent steam pressure	Indicator/transmitter	1
300	Cooling water flow	Flow element Transmitter Indicator Integrator	1
301	Cooling water inlet pressure	Indicator	0
302	Cooling water inlet temperature	Temperature element Indicator/transmitter	1
303	Cooling water outlet pressure	Indicator	0
304	Cooling water outlet temperature	Temperature element Indicator/transmitter	1
305	Vent gas flow	Flow element Transmitter Indicator Integrator	1
306	Vent gas pressure	Indicator/transmitter Indicator/controller	1
308	Vent gas temperature	Temperature element Indicator/transmitter Indicator/controller High-temperature alarm	1
309	Condenser sump level	Indicator/transmitter Indicator/controller High-level sensor Low-level sensor High-level alarm Low, low sump level/unit shutdown Level gauge	1
310	Condensate temperature	Temperature element Indicator/transmitter	1
311	Condensate pressure	Indicator/transmitter	1

Appendix T

2.5-MW PILOT PLANT ADDITIONAL COMPONENT LIST

Item	Description
Distributors	304 stainless steel; one per tube in HX-001, HX-002; 1713 total
Sample stations (designation AX)	Provisions for collecting and cooling samples for chemical analysis; total of 12 sample lines
Strainers	316 stainless steel bodies and mesh, 4 each ½" and 2 each 1" (similar to Armstrong no. E2SW-½ and E2SW-1)
Steam trap	316 stainless steel, 2 each 1" (similar to Armstrong size 1013)
Structural steel	Steel required to support vessels, piping and all other components during operation and in the event of a seismic disturbance; also provide for OSHA-approved handrails and platforms to assist unit operators
Pipe hangers	As required to support piping under all operational and seismic conditions
Data collection system	Provide capability of monitoring 58 parameters and storing of critical system performance data
Motor control center	Control operation of 4 pump motors representing a load of 40 kW

Appendix U

2.5-MW PILOT PLANT INTERFACE LIST

Identifying Number	Description	Requirements
IF 1	Wellhead steam	347°F, 129 psia 250 ppm H_2S, 0.5% total noncondensables
IF 2	Clean steam to turbines	337°F, 113 psia 12.5 ppm H_2S, 0.01% total noncondensables 50,000 lb/h
IF 3	Blowdown, condensate and drain to cooling tower basin	335°F, 110 psia 45 gpm when draining both heat exchanger sumps
IF 4	Clean steam	335°F, 110 psia 0.01%, total noncondensables 1136 lb/h
IF 5	Seal water for pumps	Good quality water 80°F, 20 psia, 4 gpm
IF 6	Makeup water	Good quality water 340°F, 120 psia, 45 gpm for startup
IF 7	Vent gases to Stretford unit	120°F, 5 psig Primarily noncondensables and H_2S
IF 8	Cooling water to vent gas condenser	85°F, 50 psia 60 gpm
IF 9	Cooling water return from vent gas condenser	135°F, 48 psia 60 gpm
IF 10	Service water	For washing and cleaning
IF 11	Instrument air	100 psia Clean oil-free air, instrument quality
IF 12	Electrical power	440 vac, 3-phase 40 kW
IF 13	Electrical power	110 vac