DOE/ET/20444 -- T6

TRW
SYSTEMS GROUP

ONE SPACE PARK • REDONDO BEACH, CALIFORNIA

MASTER

CODE IDENT 11982

TITLE

1-MWe HEAT EXCHANGERS FOR OTEC
FINAL DESIGN REPORT

AC03-77ET 20444

DATE INITIAL ISSUE
19 JUNE 1980

NO.80.6806-JES-271

SUPERSEDING: _____

DISCLAIMER

This book was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

INITIAL ISSUE - SUBJECT TO REVISION

PREPARED BY: A. M. Sprouse
A. M. Sprouse

APPROVAL SIGNATURES:

A. M. Sprouse 6-24-80. J. E. Snyder 6-24-80.
A. M. Sprouse DATE J. E. Snyder DATE

DATE

DATE

DATE

DATE

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

MOW

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

TABLE OF CONTENTS

	<u>Page</u>
1. INTRODUCTION	1-1
2. SYSTEM DESIGN	2-1
2.1 System Description	2-1
2.2 System Modifications	2-1
3. EVAPORATOR AND ASSOCIATED SYSTEMS DESIGN	3-1
3.1 Evaporator System Description	3-1
3.2 Reflux Loop	3-1
3.3 Drain Tank	3-2
3.4 Phase Separator and Drain	3-2
3.5 Flexible Operation	3-2
3.6 Waterbox Design	3-3
4. CONDENSER DESCRIPTION	4-1
4.1 Condenser Description	4-1
4.2 Condenser Drainpots	4-1
5. INSTRUMENTATION	5-1
5.1 Instrumentation Description	5-1
5.2 Pressure and Differential Pressure Installations	5-1
5.3 Instrumented Tube Loops	5-1
6. MATERIALS AND PROCESSES	6-1
6.1 Materials and Processes Scope	6-1
6.2 Titanium Welding Program	6-1
6.3 Waterbox and Skirt Coating	6-1
6.4 Shroud Seals	6-1
7. REFERENCES	7-1
APPENDIX A - SYSTEM DESIGN	A-1
APPENDIX B - EVAPORATOR AND ASSOCIATED SYSTEMS DESIGN	B-1

TABLE OF CONTENTS (Continued)

	<u>Page</u>
APPENDIX C - CONDENSER DESIGN	C-1
APPENDIX D - INSTRUMENTATION DESIGN	D-1
APPENDIX E - MATERIALS AND PROCESSES DESIGN	E-1

1. INTRODUCTION

The 1 MWe Heat Exchangers for Ocean Thermal Energy Conversion (OTEC) Project has for its purpose the design and fabrication of large heat exchangers for deployment on an ocean platform. The heat exchangers, an evaporator and condenser, will be used to gain experience in the ocean environment. The equipment will be operated in sea state conditions typical of a real OTEC power plant. Seawater flow through the heat exchangers provides a realistic biofouling and corrosion environment.

Heat transfer performance of large heat exchangers will be determined under the realistic sea state conditions necessary to evaluate the potential of the OTEC system. Both heat exchangers are shell-and-tube type with ammonia as the working fluid. Liquid ammonia enters the evaporator from a set of spray headers installed at the top and mid-plane of the tube bundle. The evaporator is designed to be easily modified to simulate a system four times as large. This flexibility permits a range of vapor velocities to be investigated. Pool boiling has also been considered as an optional design concept to allow further variations of the system. This operational flexibility provides a significant variation in OTEC system performance to be investigated with a single heat exchanger system.

The work was performed by a team headed by TRW. TRW was responsible for project management and overall system design and integration. Mechanical design and fabrication of the heat exchangers was performed by Southwestern Engineering Company (SEC). Additional support was provided by C F Braun in condenser design and drawing review. The evaporator thermal design was based on experiments conducted by Union Carbide Corporation, Linde Division, on single tube heat transfer and vapor velocity effects, and on UCC/Linde's design tradeoff studies. UCC/Linde also provided the surface enhancement which was applied to half the evaporator tubes. Finally, review of the static and dynamic analyses of the heat exchangers was performed by J. H. Wiggins Company.

The work in this contract was performed in four phases: Conceptual and Preliminary Design, Detail Design, Fabrication, and Installation and Test Support. This report documents the design which took place during the third and fourth phases. This is largely confined to more detailed design of the changes proposed in Engineering Change Proposals 31372.001,

-.002, -.003, and -.004 (References 7, 8, 9, and 10) and design changes resulting from solution of fabrication difficulties. Previous design work is documented in the Preliminary Design Report (Reference 1) and the Detailed Design Report (Reference 2).

The project effort was conducted by TRW under Department of Energy (DOE) contract number DE-AC03-77ET-20444.

2. SYSTEM DESIGN

2.1 SYSTEM DESCRIPTION

The 1 MWe Heat Exchangers test loop uses warm seawater to evaporate liquid ammonia which then passes through a throttle valve to simulate a turbine before being condensed with cold seawater and recycled. The test loop is installed in a below-decks compartment of a T-2 tanker and will be operated at sea.

The evaporator and condenser are the major hardware items and their design and construction was TRW's responsibility under Contract Number DE-AC03-77ET-20444. TRW was also responsible for associated hardware required or recommended for operation, including an evaporator drain tank, a phase separator, the evaporator reflux loop, and instrumentation of both heat exchangers. This responsibility extended from subsystem design and specification to procurement of the hardware.

The support systems for the heat exchangers were provided under a separate DOE contract with Global Marine Development, Inc. (GMDI). These included the ship-platform, cold water pipe, throttle valve, seawater pumps, condensate return pump, tube cleaning system (Amertap), data acquisition system, and all storage, handling and safety systems.

2.2 SYSTEM MODIFICATIONS

A number of modifications to the system made subsequent to the Detailed Design Phase had an impact on the Final Design. All were a result of DOE direction. The following should be noted (documentation where applicable follows in Appendix A).

Amertap Cleaning System. The detailed design assumed that the cleaning system was M.A.N. brushes. Subsequent direction required compatibility with an Amertap cleaning system. The effects of the change are described in Reference 9.

Turbine Simulator. The effects of a turbine will be simulated using a throttle valve. The DOE dropped plans for a desuperheater and with it the need to recycle an additional 9000 lb/hr of liquid ammonia through the condenser. A TRW proposal to provide a turbine as rejected.

Design Loading Conditions. The design load for vertical and lateral accelerations was increased from 0.1g to 0.5g, with consequent changes to the mechanical design of the units and their support structures. A discussion is given in Reference 9.

Recommended Equipment. The scope of the TRW contract was broadened to include hardware recommended for operation of the reflux loop, but previously designated GFE. This equipment includes flowmeters, drain and vent valves, pump motors, valve controls, and relief valves. This change is discussed in Reference 8 and the items are specified in References 4 and 5.

Optional Equipment. A number of options proposed by TRW (Reference 7) were included in the scope of work. These options are evaporator mid-plane feed, vapor shrouds, pool boiling, enhanced tubes, phase separator, condensate drain measurements and instrumented tubes. The design of these options is documented in succeeding sections of this report.

3. EVAPORATOR AND ASSOCIATED SYSTEMS DESIGN

3.1 EVAPORATOR SYSTEM DESCRIPTION

The evaporator specifications are given in Reference 3. Associated with the evaporator are the reflux loop to provide variation in liquid loading, a drain tank, and a phase separator to remove entrained liquid from the exit vapor. As part of the plan for operational flexibility, the following options have been included in the evaporator: mid-plane feed, vapor shrouds, pool boiling and enhanced tubes. The basic design of the evaporator system is documented in References 1 and 2 with the exception of these options and such changes as were directed by the DOE or required by practical difficulties of fabrication. Details of the changes are given in the following sections. All evaporator design documentation is included in Appendix B.

3.2 REFLUX LOOP

The reflux loop is shown schematically in TRW drawing number 6850-91-05. The equipment for the loop is specified in Reference 5. Due to additional direction all components of the loop were procured under this contract rather than by GMDI, although some alterations were made to accommodate GMDI design choices. The following changes were made:

Drain Line. The drain line size was increased from 18 inches to 20 inches and the filter changed from 16 inches to 20 inches to reduce the friction losses. Extra NPSH margin was needed because GMDI was unable to provide the requested static head to the reflux pumps. In this case, the next largest size was selected by GMDI.

Pump Motors and Controllers. A DC motor and DC variable speed power supply were finally selected for the reflux pumps. Only the motor was procured by TRW. An AC motor was originally recommended by TRW because the explosion proof requirement cannot be met by DC. GMDI preferred DC power and was concerned about the size and weight of the AC power supply so they designed a vented enclosure for the pump motors. This obviated the explosion proof requirement.

3.3 DRAIN TANK

The internal baffle arrangement for the drain tank was criticized at the Detailed Design Review as being inadequate to break up wave motion. These structures were redesigned to the present configuration as shown in SEC drawing EM-97068. Eight axial baffles were added at the tips of the eight 45° angled baffles.

3.4 PHASE SEPARATOR AND DRAIN

An external phase separator (formerly called a demister) was approved for the reasons discussed in Reference 7. The unit is a pressure vessel containing both mesh and hook-and-vane separators, as specified in Reference 6. The liquid is returned to the drain tank via a drain line containing a meter and dump valve (see TRW drawing number 6850-91-38).

3.5 FLEXIBLE OPERATION

The evaporator includes several design features that permit operation in more than one mode: vapor shrouds, enhanced tubes, mid-plane feed and pool-boiling. The rationale for these features is given in the proposal, Reference 7. Final design of these features is summarized below.

Vapor Shrouds. A system of removable neoprene-gasketed metal plates is used for control of the ammonia vapor flow path. They can be used, in an altered configuration, to control liquid level in the pool boiling design discussed below. The shroud layouts are shown in TRW drawings 6850-91-20, -21, and -22.

Enhanced Tubes. The lower half of the evaporator tube bundle was tubed with 3151 enhanced surface tubes, of a type known as "High Flux" (Union Carbide, Linde Division). The ammonia side heat transfer coefficient is improved from 880 to 4400 $\text{ft}^2\text{-hr-}^{\circ}\text{F/Btu}$ so that the expected output of the mixed bundle is increased from the baseline of 256,000 lb/hr to over 300,000 lb/hr at the same operating conditions. This additional vapor volume can be accommodated by the original equipment, including the condenser, phase separator, and relief valves. Pool boiling performance is expected to improve even more dramatically: h_a increases from 540 to 5500 $\text{ft}^2\text{-hr-}^{\circ}\text{F/Btu}$. As an additional consequence of the use of two types of tubes in the evaporator, a splitter plate was installed in the outlet waterbox, to enable measurement

of two separate outlet temperatures. The splitter plate design and function are discussed in Reference 8. A tie-in to pump P-2 was recommended for the mid-plane feed system so that higher reflux rates could be supplied to the enhanced tubes for separate testing.

Mid-Plane Feed. A separate feed system was designed to provide make-up liquid to the center of the bundle and allow more control over reflux ratio. The equipment and layout is similar to that selected for separate testing of enhanced tubes.

Pool-Boiling. A schematic of the pool boiling feed and drainage system is shown in TRW drawing number 6850-91-17. Pool boiling performance is expected to be about 282,000 lb/hr of ammonia evaporate. The shrouds and a differential pressure type level sensor are used for liquid level control. Operation of the system is described in Reference 11.

3.6 WATERBOX DESIGN

A problem peculiar to the evaporator arose during final assembly of the waterboxes to the shell. Due to the great size of the units, the waterboxes deformed sufficiently under their own weight to prevent their being hung and the flanges bolted up to provide a watertight seal. This problem was resolved by welding the waterboxes to the skirts.

4. CONDENSER DESIGN

4.1 CONDENSER DESCRIPTION

The condenser specifications are given in Reference 3. Associated with the drain tank are five drainpots, one for each drain, in place of a drain tank. The condensate recycle for desuperheat has been eliminated from the system. Condensate is returned to the evaporator by a pump provided by GMDI. The basic design is documented in References 1 and 2. Drainpot design is documented in Appendix C.

4.2 CONDENSER DRAINPOTS

The drainpots are designed to permit measurement of condensate flow rate for each of five zones along the condenser length. The rationale for this option is discussed in Reference 7.

Each drainpot contains a level element and drains into a line containing a turbine flowmeter. All drain lines empty into a common header which feeds the condensate return pump. The level glass originally proposed for each drainpot was dropped because of a U. S. Coast Guard prohibition.

5. INSTRUMENTATION

5.1 INSTRUMENTATION DESCRIPTION

The 1 MWe heat exchangers contain a variety of instruments to obtain data, primarily on heat transfer performance, but also to evaluate certain operating parameters. The instrumentation includes resistance temperature detectors (RTD's), pressure sensors, differential pressure sensors, and level sensors. Contractual direction increased the scope to provide for procurement of reflux loop and condenser drains flowmeters. A system for monitoring flow rate and temperature drop in individual heat exchanger tubes, called the instrumented tube loops, has been added. Additional instrumentation associated with the pool boiling option, phase separator drain, and condenser drainpots is discussed as part of the design of those systems elsewhere in this report. All equipment for instrumentation is specified in the Instrumentation Specification, Reference 4. Design documentation for the following additions are in Appendix D.

5.2 PRESSURE AND DIFFERENTIAL PRESSURE INSTALLATIONS

Each pressure and differential pressure sensor requires a system of vents, drains, and isolation valving to ensure correct operation. These systems have also been designed for differential pressure-type level elements. The recommended installations are shown in TRW drawings 6850-91-30, -31, -32, -33, -34, -35, -36, -37, -45 and -46.

5.3 INSTRUMENTED TUBE LOOPS

The instrumented tube loops measure flow rate and temperature drop for 20 representative tubes in each exchanger so that local heat transfer performance can be evaluated. The same variables are measured for four tubes that are partially blocked to prevent entrance of Amertap balls and thus remain uncleaned. A third group of six tubes operate in a closed loop of deionized water that is heated or chilled to return it to inlet temperature and recycled. These tubes function as a control group since they will experience no fouling. The rationale and use of the system is discussed in the original proposal, Reference 7. Final design of the system necessitated changes to the proposed design, due to the added requirement

for Amertap compatibility. This design is discussed in the proposal for inclusion of Amertap, Reference 9. Other documents relating to the instrumented tube loops design are TRW drawings showing flowmeter and RTD installation, 6850-91-39, -40, -41, -42, -43, and -44.

6. MATERIALS AND PROCESSES

6.1 MATERIALS AND PROCESSES SCOPE

This section of the Final Design Report is concerned with materials selection for corrosion/erosion control and with fabrication processes. Final design included solutions to the following problems: titanium tube-to-tubesheet welding, cleanliness control, waterbox and skirt coating selection, and shroud seal selection. The design is documented in Appendix E.

6.2 TITANIUM WELDING PROGRAM

A complete procedure was developed for making the titanium tube-to-tubesheet welds. This procedure was qualified by pull-out strength, air-soap leak, and dye penetrant tests. Also, during fabrication it became necessary to provide additional procedures for dealing with problems that arose. A special procedure was developed for welding a series of joints in which the tubesheet was accidentally chamfered, and detailed instructions for maintaining cleanliness were issued. Cleanliness is highly important to producing acceptable titanium welds.

6.3 WATERBOX AND SKIRT COATING

The waterbox, skirt, and the frames for Amertap netting were all coated with a neoprene coating. Due to difficulties in uniform application, the coating required some repair. This procedure was performed after shipment and is accordingly discussed in the proposal for shipyard repairs, Reference 10.

6.4 SHROUD SEALS

A neoprene gasket bonded to the shroud was selected for sealing the space between the shroud and the framework to which the shrouds are bolted. This eliminates the need for installation of a separate gasket and speeds handling during shroud configuration changes.

7. REFERENCES

1. TRW document, "1 MWe Heat Exchangers for Ocean Thermal Energy Conversion (OTEC), Preliminary Design Report," July 1978.
2. TRW document, "1 MWe Heat Exchangers for OTEC, Detailed Design Report," No. 80.6806-JES-282, 19 June 1980.
3. TRW specification, "1 MWe Heat Exchangers for Ocean Thermal Energy Conversion (OTEC), Part I - System Specification," SY4-3, 78.6806-JES-947, Revision B, 10 June 1980.
4. TRW specification, "1 MWe Heat Exchangers for Ocean Thermal Energy Conversion (OTEC), Part II - Instrumentation Specification, SY4-3, 78.6806-JES-947, Revision A, 12 May 1980.
5. TRW specification, "1 MWe Heat Exchangers for Ocean Thermal Energy Conversion (OTEC), Part III - Auxiliary Equipment Specification," SY4-3, 78.6806-JES-947, Revision A, 12 May 1980.
6. TRW specification, "1 MWe Heat Exchangers for Ocean Thermal Energy Conversion (OTEC), Part IV - Phase Separator Specification," SY4-3, 79.6806-JES-297, Revision B, 10 September 1979.
7. TRW proposal, "1 MWe Heat Exchangers for Ocean Thermal Energy Conversion (OTEC), Proposal for DOE Directed Design Changes," Proposal No. 31372.001, 28 February 1979.
8. TRW proposal, "1 MWe Heat Exchangers for Ocean Thermal Energy Conversion (OTEC), Proposal for DOE Directed Design Changes," Proposal No. 31372.002, 27 August 1979.
9. TRW proposal, "1 MWe Heat Exchangers for Ocean Thermal Energy Conversion (OTEC), Proposal for DOE Directed Design Changes," Proposal No. 31372.003, 19 December 1979.
10. TRW proposal, "1 MWe Heat Exchangers for Ocean Thermal Energy Conversion (OTEC), Proposal for DOE Directed Design Changes," Proposal No. 31372.004, 25 February 1980.
11. TRW document, "1 MWe Heat Exchangers for OTEC, Operations and Maintenance Manual," 80.6806-JES-270, 20 June 1980.

APPENDIX A

SYSTEM DESIGN

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK - REDONDO BEACH - CALIFORNIA 90278

INTEROFFICE CORRESPONDENCE

8712.6.80-010

TO: J. E. Snyder
(W/3 copies
ap. A & B)

CC: R. H. Douglass
A. Kaplan
L. R. Mortaloni
S. J. Voce

DATE: 29 February 1980

SUBJECT: OTEC-1 Evaporator
Structural Adequacy

FROM: A. Barrett

BLDG. R5 MAIL STA. B320 EXT. 63610

1.0 Introduction

In response to the request by R. J. Glumace, (per letter, Reference 1) GMDI report RPT-4086-C815 (Reference 2) was reviewed. The GMDI report presented a finite element analysis of the OTEC-1 evaporator shell and support structure. Results of the GMDI analysis indicated that excessively high stresses would occur in the evaporator shell due to inertial loadings.

A review of the GMDI model and analysis results revealed that the radial restraint provided by the tube sheet in the actual structure was not included in the GMDI analysis model. Since this was felt to be crucial to an accurate analysis of the shell stress, an independent finite element analysis of the evaporator shell was performed by TRW Structures. In addition to the tube sheet, a more refined mesh, and other modeling modifications considered to be necessary for an accurate stress analysis were included in the TRW model.

A cross-validation of the GMDI and TRW analysis methods was accomplished by independent analysis of two test cases in which TRW and GMDI each analyzed identical models.

2.0 Results Summary

Inclusion in the analysis model of both the tube sheet and a more refined mesh in the region of maximum stress proved to be necessary for an accurate stress analysis. Adding the tube sheet resulted in a large reduction in deformation and stress, compared to the "both ends free" GMDI model results.

The stresses resulting from the TRW model analysis are within ASME Code* limits for both the Department of Energy (DOE) specified load factors** and

the (lower) 16 feet sea state load factors. This is based on an assumed maximum weight of 534,000 lbs., applicable to the "pool boiling" condition.

Closely-matching stress and displacement results were obtained by GMDI and TRW Structures for the two different models analyzed independently. This provides cross-validation of the analysis methods used.

Approved

A. Kaplan, Head
Loads & Technology Section

* Boiler and Pressure Vessel Code, Sect. VIII, Div. 2

** $\eta_{\text{Long}} = .2 \text{ G}$, $\eta_{\text{Lateral}} = .5 \text{ G}$, $\eta_{\text{Vertical}} = 1.5 \text{ G}$

TABLE OF CONTENTS

	Page No.
1.0 Introduction	1
2.0 Results Summary	1
3.0 Structural Criteria	4
3.1 Loads	
3.2 Allowable Stress	
4.0 Review of GMDI Report RPT 4086-C815	6
5.0 TRW Finite Element Analysis Model and Results	8
6.0 Evaluation of TRW Stress Analysis Results per ASME Code	11
7.0 Cross-Validation of Analysis Methods	16
Figures	19
References	25
Appendices	
A. TRW Finite Element Model Plots	
B. TRW Finite Element Analysis Results: Computer Printout	
B1. Stress Model: DOE: 0 psi (OTEC 1 data file)	
B2. Stress Model: 16' sea: 0 psi (OTEC 4 data file)	
B3. Stress Model: DOE: 190 psi (OTEC 8 Case 5 data file)	
B4. Stress Model: 16' sea: 190 psi (OTEC 8 Case 6 data file)	
B5. Validation Model 1 (OTEC 5 data file)	
B6. Validation Model 2 (OTEC 6 data file)	
B7. Validation Model 3 (OTEC 3 data file)	
B8. Validation Model 4 (OTEC 2 data file)	
C. Condenser and Phase Separator: Assessment of Structural Adequacy	

3.0 Structural Criteria

3.1 Applied Loads

The load conditions are combinations of "sea states" (inertial load factors), and "operating modes" (internal pressure and fluid weight).

Inertial load factor sets considered are as follows:

DOE Load Factors

$$\begin{aligned}n_x \text{ (Longitudinal)} &= .2 \text{ G} \\n_y \text{ (Lateral)} &= .5 \text{ G} \\n_z \text{ (Vertical)} &= 1.5 \text{ G}\end{aligned}$$

The DOE load factors exceed the GMDI computed maximum load factors for the 16' sea state.

16' Sea State

$$\begin{aligned}n_x \text{ (Longitudinal)} &= .0635 \text{ G} \\n_y \text{ (Lateral)} &= .246 \text{ G} \\n_z \text{ (Vertical)} &= 1.309 \text{ G}\end{aligned}$$

The 16' sea load factors are obtained by applying a factor of $2 \times (1.86) = 3.72$ to the calculated RMS accelerations, Reference 3, page 24. The 16' sea state is considered to be the worst-case sea state that would actually occur in combination with the highest-weight operating condition (pool-boiling mode).

"Pool boiling" is the operating mode considered in combination with the DOE and 16' sea inertial load factors. In the pool boiling mode, the tubes are full of sea water and the shell outside the tubes is filled with liquid ammonia. Combined structure plus fluid weight is 534,000 lb. A 118 psi gage internal over-pressure occurs during normal operation in pool boiling mode. This is bounded by the two pressures, 190 psi, (relief valve pressure) and 0 psi, applied in the structural analysis. The 0 psi condition is included since the pressure stress may have a relieving effect on the inertial load stress at some locations.

A "flooded weight" of 640,000 lb. is identified in Reference 2, page 24. The 640,000 lb. weight could only occur due to a rupture of one or more sea-water tubes while the shell was simultaneously de-pressurized. This is an abnormal condition for which the stress limits described in subsection 3.2 do not apply. By implication the minimum factor of safety against catastrophic collapse for the flooded condition is

$$* 3 \times \frac{534,000}{640,000} = 2.5, \text{ if the limits of page 5 are met.}$$

* See Footnote Page 5

3.2 Allowable Stresses

The ASME Boiler and Pressure Vessel Code, Section VIII, div. 2, Appendix 4 is the applicable source for allowable stresses (Reference 4). This is referred to in subsequent paragraphs as the "Code". The specific limits applicable to the evaluation of the stresses in the shell at the highest-stressed location, the edge of the support, are either:

Paragraph 4-133 $P_L + P_b \leq 1.5 S_m$

where:

$P_L + P_b$ = the maximum magnitude of the local primary membrane plus primary bending stress intensity associated with equilibrating the worst-case design or operating applied mechanical load set.

S_m = the tabulated design stress intensity value at applicable temperature. S_m is set at 1/3 ultimate stress or 2/3 yield stress whichever is lower.

or: Paragraph 4-136.3 *Applied load. $\leq$ 2/3 "lower bound collapse load" (Limit Analysis)

If either the limit of paragraph 4-133 or the limit of paragraph 4-136.3 is satisfied, the Code requirement is met. Note that the limit of paragraph 4-133 is more restrictive, but simpler to apply than the limit of paragraph 4-136.3.

The evaporator shell and doubler materials are SA 516 GR 70 and SA 515 GR 70 respectively. The operating temperature range is between 40°F and 80°F for both the shell and doubler, thus:

$$S_m = 23.3 \text{ ksi}$$

$$F_{TY} = 38.0 \text{ ksi}$$

$$F_{TU} = 70.0 \text{ ksi}$$

Reference Code Table ACS-1, page 52 and 53.

* This limit assures a factor of safety of 3 against catastrophic collapse. The "Lower-Bound Collapse Load" as defined in the Code is approximately half the catastrophic collapse load.

4.0 Review of GMDI Report - RPT 4086-C815 (Reference 2)

GMDI used a 3-D shell finite element analysis with the NASTRAN computer program to calculate the stresses in the evaporator, shell and support structure. A 13' length of cylindrical shell was included with the support structure in the analysis model. Figure 1, page 4 illustrates the correspondence between the model and the design configuration.

Inertial loads were applied to the finite element model at points a & b, Reference Figure 1. Derivation of the 5 vector (3 forces and 2 moments) load set applicable to each end of the cylindrical segment was accomplished using the GENSAP program with a beam model of the entire evaporator. The beam model was considered by TRW Structures to be a valid representation of the evaporator shell and supports for the purpose of computing axial loads, shears and moments in the evaporator shell cross-sections. Therefore the loads applied to the shell segments were considered by TRW Structures to be realistic for each of the inertial load conditions investigated.

However, although the loads applied to the finite element model were realistically calculated, there were some important aspects of the finite element model of the shell which caused serious inaccuracies in the calculated shell stress and deformation. In order of importance these are:

1. The radial restraint exerted by the tube sheet was not included in the analysis model.
2. The shell mesh was relatively coarse in the critical region at the edge of the support.
3. The radial distance between the doubler and the shell was not included.

(1) Neglecting the tube sheet resulted in modeling the cylindrical segments as being completely free radially at both ends. As can be seen in Figures 3 through 5, pages 21-23, this resulted in very large out-of-round distortions over the entire length of the cylindrical shell segment, due to the lateral and vertical applied loadings. The high circumferential bending stresses obtained in the GMDI analysis are associated with the large deformations, particularly the large curvature change evident in Figure 3. The large distortions shown in Figure 4 are physically impossible since the location coincides with the weld of the shell to the tube sheet. The shell would actually be constrained to remain circular at this location.

(2) The first shell element adjacent to the support extended over a 12° arc circumferentially. The mid point of this element is 8" away from the edge of the support. Since the constant strain elements used in the analysis predict the average stress over the region of the element, and since the stress is increasing rapidly with increasing proximity to the support, the calculated stress is lower than the maximum stress which would actually occur for the assumed boundary constraints and applied loads.

(3) Omitting the radial distance between the shell and the doubler from the model results in omitting to calculate the membrane component of stress which occurs in the shell and doubler, at the region of maximum stress, due to the imposed bending curvature of the shell and doubler.

5.0 TRW Finite Element Analysis Model and Results

The finite element analysis model used by TRW Structures to calculate stress in the OTEC-1 evaporator shell is illustrated in Figure 6, page 24. A detailed "map" of the model defining the relative location of nodes, members, and elements, is given in Appendix A. Coordinates of the nodes and other numerical data associated with the model are given in the computer printouts in Appendix B.

Key features of the model are described in the following:

1. The length and location of the model corresponds to the GMDI model. This permits using the same loads.
2. Shell elements, 15" in the radial direction and 3" thick are used to represent the tube sheet. Stiff spokes are used to apply load to the end welded to the tube sheet.
3. Flexible spokes are used to apply load to the radially unconstrained end. This is similar to the boundary conditions at the corresponding end on the GMDI model. The spokes are so flexible that the relative displacement between the midpoint of the spokes and the shell is over 100 times greater than the relative radial displacements from any point on the circumference of the free end of the shell to another. This assures "zero radial constraint" to a high degree of accuracy.
4. The radial distance between the doubler and the shell is included in the model. The shear stiffness of the weld connection is represented in the doubler-shell connection.
5. The nodal points on the doubler coinciding with the welded attachment to the support structure are constrained against displacement in directions coinciding with the plane of the plate members of the support structure. (Inspection of the GMDI results indicated that the displacements of the shell at the attachment to the support were negligible).
6. Inertial loads are applied to the shell at the two ends of the cylinder, at the intersection of the midpoint of the spokes. The loads applied are identical to those applied in the GMDI analysis.
7. A refined mesh is used in the regions of maximum stress (at the edge of the support). The shell and doubler elements adjacent to the saddle support extend 3.5° circumferentially. The midpoint of those elements is located 2.4" from the edge of the support. This is only 20% larger than the combined shell+doubler thickness (2.0 in.) and is therefore close enough to the support to accurately predict the maximum bending stress.
8. The conical segment of the shell immediately adjacent to the tube sheet is modeled. (In the GMDI model, this portion of the shell was modeled as cylindrical.)

Four load cases were run. These included the DOE load factors and the 16' sea load factors, each with 0 psi internal pressure and with 190 psi internal pressure.

Stress analysis results for the above described model and loads are tabulated in Table 1, page 10. For the 16' sea loads, all stresses are below the ASME Code (simplified) stress limit, $1.5 S_m$. For the DOE loads, the stress in one element only, (the doubler on the +X side, adjacent to the support, at the edge nearest the tube sheet) exceeds the $1.5 S_m$ allowable stress. However, the less restrictive alternate Code limits based on a Limit Analysis evaluation procedure are met for the DOE load factors. See section 6.0 of this report. Note that the zero pressure condition is more critical than the maximum pressure condition. This is because the internal pressure has a relieving effect on the critical stresses, which are always in a compressive sense in the circumferential direction.

The TRWSAP computer program was used to perform the above described finite element analysis. A complete description of the features of this program is given in Reference 6.

Table 1

Stress Analysis Results

(Using the model as documented in Sect. 5.0 and Figure 6)

LOAD FACTORS		PRESSURE		MAX. LATERAL * DEFLECTION		MAX. SHELL STRESS,** LB/IN ²		MAX. DOUBLER STRESS,** LB/IN ²		REF. DATA FILE APP. B
DOE	16'SEA	0 PSI	190 PSI	VALUE	NODE NO.	VALUE	ELE. NO.	VALUE	ELE. NO.	
X		X		-.25	321	23,400	526	43,900	540	OTEC 1
X			X	-.29	321	17,600	523	36,600	540	OTEC 8 CASE 5
	X	X		-.15	321	15,900	526	30,000	540	OTEC 4
	X		X	-.19	321	16,800	523	22,900	540	OTEC 8 CASE 6

* At any location between the "tube sheet end" of the cylinder and the support. Includes lateral translation, bending deformation, and uniform radial expansion.

** "Effective" stress, Von Mises.

PREPARED A. BARRETT2/15/80

REPORT NO.

PAGE 11

CHECKED _____

MODEL STEC-1 EVAPORATOR6.0 EVALUATION OF STRESS PER A.S.M.E. BOILER & PRESSURE
VESSEL CODE SECT VIII DIV. 2 (REF (4) IN LIST OF REFS.)

THE SHELL & DOUBLER STRESS AT THE SUPPORT LOCATION IS A LOCALIZED MEMBRANE PLUS BENDING STRESS ASSOCIATED WITH THE EQUILIBRATION OF AN APPLIED MECHANICAL (INERTIAL) LOADING. THEREFORE IT IS CLASSIFIED AS A $P_L + P_b$ STRESS. THE FOLLOWING ALTERNATE LIMITS APPLY:

- EITHER $P_L + P_b \leq 1.5 S_m$ - - REF. (4) APPENDIX 4, PAR. 4-136
- OR IT MUST BE SHOWN THAT THE APPLIED LOAD IS EQUAL TO OR LESS THAN $\frac{2}{3}$ THE "LOWER BOUND COLLAPSE LOAD" BASED ON LIMIT ANALYSIS PRINCIPLES. REF. (4) APPENDIX 4, PAR. 4-136.3

16' SEA STATE, "POOL BOILING" OPERATING CONDITION
(WT. = 534,000 LB.)

$$P_L + P_b = 30,200 \text{ LB/IN}^2 \quad \text{REF. PG. 10}$$

$$1.5 S_m = 1.5(23.3) = 35,000 \text{ LB/IN}^2 \quad \text{REF. (4) PG. 53}$$

SA 516 @ 120°F

$$P_L + P_b = 30,200 < 1.5 S_m = 35,000$$

$\therefore$ COND. 4-136 IS MET FOR THE 16' SEA STATE

D.O.E. LOAD FACTORS, "POOL BOILING" OPERATING CONDITION
(WT. = 534,000 LB.)

$$P_L + P_b = 43,900 \text{ LB/IN}^2 \quad \text{REF. PG. 10}$$

$$P_L + P_b > 1.5 S_m \therefore \text{COND. 4-136 IS NOT MET FOR THE D.O.E. LOAD FACTORS @ WT. = 534,000}$$

THE STRESSES DUE TO THE D.O.E. LOAD FACTORS ARE EVALUATED PER THE LESS RESTRICTIVE ALTERNATE CRITERIA OF PAR. 4-136.3 (LIMIT ANALYSIS)

SEE FOLLOWING PAGE. A-12

PREPARED A. BARRETT 2/12/80 REPORT NO.

PAGE 12

CHECKED _____

MODEL _____

6.0 (CONT.)EVALUATION FOR D.O.E. LOAD FACTORS PER PAR. 4-136.3

SECT A-A, AS ILLUSTRATED ON PAGE 13 IS SELECTED AS A BASIS FOR COMPUTING A LOWER BOUND TO THE COLLAPSE LOAD. (SECT A-A INCLUDES THE LOCATION OF MAX. STRESS.) THE APPLIED STRESSES ON SECT. A-A ARE RESOLVED INTO AXIAL LOAD & MOMENT RESULTANTS. THE "FULLY PLASTIC" MAGNITUDE IS CALCULATED FOR EACH RESULTANT. IT IS ASSUMED THAT EACH RESULTANT WOULD INCREASE PROPORTIONALLY, WITH INCREASING APPLIED EXTERNAL LOAD. THIS IS A CONSERVATIVE ASSUMPTION, SINCE IT IGNORES THE REDISTRIBUTION OF LOAD TO OTHER, LOWER-STRESSED REGIONS, WHICH WOULD OCCUR FOR LOADS CAUSING YIELD IN THE HIGHEST-STRESSED REGION. IN CALCULATING THE RATIO OF APPLIED LOAD TO THE COLLAPSE LOAD, A LINEAR INTERACTION IS ASSUMED AMONG THE "RESULTANT RATIOS" (APPLIED RESULTANT/FULLY PLASTIC RESULTANT). THIS IS CONSERVATIVE. (REF. ⑦ PG. 172) THEREFORE, THE PROCEDURE USED RESULTS IN A VERY CONSERVATIVE CALCULATION OF THE RATIO OF THE APPLIED LOAD TO THE "LOWER BOUND COLLAPSE LOAD". THE TRUE COLLAPSE LOAD WOULD BE HIGHER THAN THE COLLAPSE LOAD IMPLIED BY THIS ANALYSIS.

CALCULATION OF RATIO OF APPLIED LOAD TO THE LOWER-BOUND COLLAPSE LOAD. (REF PGS. 14 & 15)

$$\frac{P}{P_o} + \frac{M_x}{M_{x_o}} + \frac{M_z}{M_{z_o}} = \frac{.2189(10)^6}{2.1(10)^6} + \frac{.4428(10)^6}{1.163(10)^6} + \frac{1.707(10)^6}{13.728(10)^6}$$

$$= .104 + .381 + .122 = .608$$

COND. 4-136.3 IS SATISFIED FOR $\frac{P}{P_o} + \frac{M_x}{M_{x_o}} + \frac{M_z}{M_{z_o}} \leq \frac{2}{3}$

$.608 < \frac{2}{3} \therefore$ COND 4-136.3 IS SATISFIED

NOTE THAT THE "LOWER BOUND COLLAPSE LOAD" IS DEFINED IN THE CODE AS ACTUALLY A LOWER BOUND TO THE LOAD CAUSING FINITE PLASTIC DEFORMATION. THE LOAD REQUIRED TO CAUSE CATASTROPHIC COLLAPSE WOULD BE TWICE AS HIGH, BASED ON THE RATIO OF ULTIMATE TO YIELD STRESS. THEREFORE A FACTOR OF SAFETY OF 3 AGAINST CATASTROPHIC FAILURE IS MAINTAINED.

TRW.

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK • REDONDO BEACH, CALIFORNIA

PREPARED A. BARRETT2/15/80

REPORT NO.

PAGE 13

CHECKED _____

MODEL OTEC-1 EVAPORATOR6.0 (CONT.)STRESS DISTRIBUTION IN SHELL AND DOUBLER ELEMENTS
AT THE LOCATION OF MAXIMUM STRESS LOAD CONDITION

{ D.O.E. LOAD FACTORS
WEIGHT = 534,000 LB.

SYMBOLS :

INDICATES MEMBRANE STRESS

INDICATES BENDING STRESS,
(RIGHT HAND VECTOR)

ELEMENT NO. XXX (SEE APPENDIX A)

STRESS MAGNITUDES ARE IN UNITS LB/IN²

PREPARED A. BARRETT 2/15/80

REPORT NO.

PAGE 14

CHECKED _____

MODEL OTEC-1 EVAPORATOR

6.0 (CONT.)

SECT A-A REF. PG 13

LOAD CONDITION : D.O.E. LOAD FACTORS
WT. = 534,000 LB. (PUL BUILDING WND.)

CALCULATION OF RESULTANT APPLIED MOMENTS & AXIAL
LOAD BASED ON THE STRESS DISTRIBUTION SHOWN
ON PAGE 13

$$P_{\text{RESULTANT}} = 8.5(1.25)(5200 + 5460 - 1860 - 1860) \\ + 8.5(.75)(-15,200 - 33,700) = -218,900 \text{ LB.} \\ \text{(COMPRESSIVE)}$$

LOCATION OF NEUTRAL AXIS (MEASURED FROM SAME REFERENCE)
AS Z₀, PG. 15

$$\bar{Z} = \frac{\sum AZ}{\sum A} = \frac{2(17)(1.25)^2}{55.25} + \frac{17(.75)(1.25 + \frac{.75}{2})}{55.25} = .856$$

$$M_{x \text{ RESULTANT}} = \left(\bar{Z} - \frac{1.25}{2}\right)(8.5)(1.25)(5200 + 5460 - 1860 - 1860) \\ + \left(1.25 + \frac{.75}{2} - \bar{Z}\right)(8.5)(.75)(15,200 + 33,700) \\ + \frac{(8.5)(1.25)^2}{6}(18,000 + 20,400 + 23,000 + 20,000) \\ + \frac{8.5(.75)^2}{6}(12,200 + 13,700) = \\ = 17,000 + 225,000 + 180,200 + 20,600 = 442,800 \text{ IN. LB.}$$

REF. SKETCH
PG. 13

$$M_{z \text{ RESULTANT}} = 8.5(1.25)\left(8.5 + \frac{8.5}{2}\right)(5200 + 1860) \\ + 8.5(1.25)\left(\frac{8.5}{2}\right)(5460 + 1860) \\ + 8.5(.75)\left(\frac{8.5}{2}\right)(-15200 + 33,700) \\ = 956,400 + 330,500 + 420,000 = 1,706,900 \text{ IN. LB.}$$

REF. SKETCH
PG. 13

PREPARED A. BARRETT

3/15/80

REPORT NO.

PAGE 15

CHECKED _____

MODEL STEC-1 EVAPORATOR

6.0 (CONT.)

SECT A-A

REF. PG. 13

CALCULATION OF YIELD MOMENT ABOUT X0 AXIS

LOCATION OF X0 AXIS :

$$\begin{aligned} 2(17) Z_0 F_y &= 2(17)(1.25 - Z_0) F_y + 17(.75) F_y \\ 4(17) Z_0 &= [2(1.25) + .75] 17 \\ Z_0 &= .8125 \end{aligned}$$

VIEW B-B

$$\begin{aligned} M_{0x} &= 2(17) \frac{Z_0^2}{2} F_y + 2(17) \frac{(1.25 - Z_0)^2}{2} F_y + 17(.75) F_y (1.25 - Z_0 + \frac{.75}{2}) \\ &= 30.61 F_y = 30.61(38,000) = 1.163(10)^6 \text{ IN. LB} \end{aligned}$$

CALCULATION OF YIELD MOMENT ABOUT Z AXIS

(IGNORING THE NOTCH)

$$\begin{aligned} M_{0z} &= 1.5 M_{Ez} = 1.5 \frac{(1.25)[(2)(17)]^2}{6} F_y = 361.25 F_y \\ &= 13.728(10)^6 \text{ IN. LB} \end{aligned}$$

CALCULATION OF YIELD AXIAL LOAD

$$\begin{aligned} P_0 &= A F_y = [2(17)(1.25) + 17(.75)] F_y = 55.25 F_y \\ &= 2.10(10)^6 \text{ LB.} \end{aligned}$$

SEE REF. (7) FOR DERIVATION OF "FULLY PLASTIC" MOMENT.

7.0 Cross Validation of Analysis Methods

In addition to the model described in Section 6.0, four other models were analyzed by TRW Structures. The purpose of these additional models was to provide cross-validation of analysis methods with GMDI and to test the effect of modeling refinements. The first cross-validation model duplicated the model used by GMDI for report RPT-4086-C815 (Reference 2). This model was created by making the following modifications to the model defined on Figure 6: a) change conical elements to cylindrical b) remove tube sheet plate elements c) substitute flexible spokes for stiff spokes at the tube sheet end of the cylinder d) use a plate element extending 12° circumferentially (instead of 3.5° as in the Figure 6 model) to match the GMDI mesh in this region e) restrain shell nodes against θ_y and θ_z rotation* (Reference coordinate system Figure 6) to be consistent with the GMDI model f) delete the shear connection between the doubler and the shell for elements away from the edge of the support to eliminate the effect of the radial "standoff". As can be seen from Table 2, row 1 (page 24) the results of this model closely matched the results of the GMDI model per RPT 4086-C815.

The second cross-validation model was the same as the first, except that the plate elements representing the tube sheet were added. This case was suggested by GMDI as an additional cross-validation case. GMDI then modified their own model to match, and ran the same loads as input by TRW. As can be seen from row 2 of Table 2, the TRW and GMDI results again matched very closely. As well as providing further assurance of the validity of analysis methods, this also illustrates the structural importance of the tube sheet.

A third validation model was run to test the effect of mesh refinement. This was similar to the first cross-validation model, (and the GMDI model per RPT 4086-C815) except that the mesh was refined in the region adjacent to the support, per the model of Figure 6. Results of this model are presented in row 3 of Table 2. This shows the importance of the mesh refinement in the region of maximum stress. It indicates that the GMDI model per RPT 4086-C815 underpredicted the maximum stress by a factor of 2, for the constraint assumptions made.

* In the GMDI model the rotational degree of freedom of shell nodes, other than the rotation about an axis parallel to the ϕ of revolution of the cylinder, were restrained. This was done to reduce the size of the model.

The fourth validation model was the same as the third, except that the radial "standoff" between the doubler and the shell was included. The results, row 4 of Table 2 show an increase in stress in the doubler, and a corresponding decrease in stress in the shell, compared to the results when the radial standoff is not included. This shows that including the radial standoff is important in predicting the maximum local stress in the shell and doubler.

Table 2

Validation Models: Results

(Ref. Sect. 7.0 for a description of the models & discussion of results)

APPLIED LOADS: DOE Load factors with 0 psi in all 4 cases

I. D. NO.	MESH** REFINEMENT		TUBE SHEET		DOUBLER STANDOFF		MAX. LATERAL **** DEFLECTION, IN.		MAX. SHELL STRESS LB/IN ²		MAX. DOUBLER STRESS,*** LB/IN ²		REF. TRW DATA FILE APP. B
	2 1/2"	8"	YES	NO	YES	NO	TRW	GMDI*	TRW	GMDI*	TRW	GMDI*	
1		X		X		X	-5.08	-4.60	67,300	74,500	61,800	53,500	OTEC 5
2		X	X			X	- .33	- .34	12,700	15,300	12,900	9,700	OTEC 6
3	X			X		X	-5.90	---	120,800	---	124,900	---	OTEC 3
4	X			X	X		-4.49	---	79,400	---	163,000	---	OTEC 2

* Reference 2

** The circumferential distance from the highest stressed region (the edge of the saddle support) to the centroid of the adjacent element is used as a measure of the mesh refinement. (see pg. 7, item 2 & pg. 8, item 7.)

*** The TRW & GMDI models are not identical with respect to doubler modeling. In cases 1-3, the doubler is not connected tangentially to the shell in the TRW model, whereas it is in the GMDI model.

**** At any location between the "tube-sheet end" and the support

PREPARED A. BARRETT

4/18/80

REPORT NO.

PAGE 19

CHECKED _____

MODEL QTEC-1 EVAPORATOR

FIG. 1

ILLUSTRATION OF EVAPORATOR CONFIGURATION & FINITE ELEMENT MODEL

EVAPORATOR SKETCH

DRY WEIGHT OF EVAPORATOR = 252,000 LB. REF. (2) Pg. 9

G.M.D.I. FINITE ELEMENT MODEL SKETCH

PREPARED A. BARRETT

1/18/80

REPORT NO.

PAGE 20

CHECKED _____

MODEL OTEC-1 EVAPORATOR

FIG 2

SKETCH OF SUPPORT, DOUBLER, & SHELL SEGMENT

WELD OF DOUBLER TO VERTICAL PLATES OF SUPPORT IS DOUBLE FILLET, 3/8" BOTH SIDES OF VERTICALS, OR, SINGLE 3/4" FILLET.

REF: ⑤ DWG. BM-97062 REV. 8 "FIXED SUPPORT ASSY"
DM-77740 "SUPPORT MOD."

TRW.

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK • REDONDO BEACH, CALIFORNIA

PREPARED A. BARRETT 1/16/80

REPORT NO.

PAGE 21

CHECKED _____

MODEL OTEC -1 EVAPORATOR

CROSS-SECTION THRU EVAPORATOR SHELL
AT (AFT) EDGE OF FWD. SUPPORT

FIG. 3

PLOT OF CALCULATED DEFLECTED SHAPE OF THE
EVAPORATOR SHELL AT THE SUPPORT LOCATION

VIEW C-C REF. FIG. 1 PG. 19

GMDI ANALYSIS RESULTS
PER REF. (2)

REF: (2) APPENDIX B (NASTRAN OUTPUT)
SUBCASE 1 DISPLACEMENTS FOR NODES 3031-3060

SUBCASE 1 = OPERATIONAL FULL BULKING, TANK & AMMONIA LOAD
 $g_x = .2$, $g_y = .5$, $g_z = 1.5$ A-22

PREPARED A. BARRETT

1-24-80

REPORT NO.

PAGE 22

CHECKED _____

MODEL OTEC-1 EVAPORATOR

FIG 4

GMDI ANALYSIS RESULTS PER REF. 2

PLOT OF CALCULATED DEFLECTED SHAPE OF EVAPORATOR SHELL

AT THE LOCATION OF THE WELD TO THE TUBE SHEET

VIEW D-D , FIG 1 , PG. 19

NOTE: THE OUT-OF-ROUND DEFORMATION SHOWN IS
PHYSICALLY IMPOSSIBLE, SINCE THE SHELL WOULD
BE UNSTRAINED TO REMAIN CIRCULAR AT THIS LOCATION
BY THE TUBE SHEET.

NODE NOS. 4121 - 4150

LOAD SUBCASE 1

REF. (2) AP. 8

A-23

PREPARED A. BARRETT

REPORT NO.

PAGE 23

CHECKED _____

MODEL OTEC-1 EVAPORATOR

SECT A-A

**CROSS-SECTION THRU SHELL
LOCATED 2.6' FWD. OF SUPPORT &**

FIG. 5

GMDI ANALYSIS RESULTS PER REF. 2

PLOT OF CALCULATED DEFLECTED SHAPE OF EVAPORATOR SHELL
AT A LOCATION BETWEEN THE SUPPORT AND THE END

VIEW E-E FIG. 1 PG. 19

REF. 2 AP. B NODE NOS. 4031-4060,
LOAD SUBCASE 1

FIG 6

TRW STRESS MODEL

(TRW/SP)

References

1. Letter 79-ETEC-DRF-3968, From R. J. Glumace, Rockwell International Energy Technology Engineering Center, (E.T.E.C.), to J. E. Snyder, TRW, Dtd 29 November 1979, "Review of GMDI Report RPT-4086-C815."
2. Global Marine Development Inc. (GMDI) Document No. RPT-4086-C815, "Technical Report on OTEC-1 Compartment Evaporator Tank & Modified Support Structure Finite Element Analysis".
3. GMDI RPT-4086-C726, "Technical Report on Ship Motions Analysis", Rev. 1, Dtd 10 August 1979.
4. American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code, Sect VIII Div. 2, (July 1977 Version).
5. OTEC Evaporator Engineering Drawings by Southwestern Engineering, Los Angeles.
6. TRW Document MEL-75-C-96, "TRW Structural Analysis Program (TRWSAP)" Dtd 1 July 1975, (Reissued September 1979.)
7. "Plastic Analysis of Structures", Philip G. Hodge Jr., McGraw-Hill, 1959.

INTEROFFICE CORRESPONDENCE

5515.2.79-271

TO: J. E. Snyder

M. P. Bianchi
L. D. Berman
CC: R. H. Douglass
L. J. Meyers
M. H. Seidman
A. M. Sprouse

DATE: 19 June 1979

SUBJECT: Amertap Ball Guiding System - 1 MWe OTEC

FROM: L. A. Rosales

BLDG.
01MAIL STA.
2220EXT.
52630

References 1): IOC 5515.2.79-218, "Visit to Southwestern Engineering on Foam Waterbox Closure, 9 May 1979", M. P. Bianchi to L. A. Rosales, dated 14 May 1979.

The Materials Engineering Department has been requested to evaluate materials and methods for guiding the Amertap cleaning balls to and from the heat exchanger tube fields. A concern had been voiced that the balls would be "trapped" in eddy currents within the waterboxes causing an effective loss in the number of cleaning balls and, possibly, a non-uniform ball distribution. If this is a real problem, two solutions present themselves for study; viz., a) adding more balls until an equilibrium condition is set up (the numbers trapped are replaced) and b) building a guiding system within the waterboxes. The first solution is very simple and should be evaluated. This memo concerns approaches to the second solution.

Of the two heat exchangers, the evaporator presents a more complex problem since the tube field is rectangular. Therefore, the water flow containing the balls must change from a circular cross-section at the inlet flange of the waterbox to a rectangular cross-section at the tubesheet. This profile reverses on the outlet side of the evaporator. The condenser tube field is circular except for a "cut-off" of a chord on the upper side. This will simplify the guiding system somewhat. Both heat exchangers contain multiple penetrations which must be accommodated. The number of penetrations is particularly high on the outlet sides of the heat exchangers. Two types of penetrations are present; viz., temperature sensor leads and their supports which are mounted to the waterboxes, and instrumented tube hoses and their supports which are mounted to the tubesheet skirts.

Approaches to the guide design include the use of flow vanes to direct the stream onto the tube field and to minimize rotational flow components, the filling of the outer portion of the waterboxes with a solid material to form the desired flow passage, and the implanting of a guiding screen or wall within the waterbox to contain the balls within the desired flow envelope.

The use of flow vanes was considered as a straight-forward simple approach to the problem. Conically shaped cylinders would be welded into the waterboxes to direct the flow onto the tubefield. However, this approach does not guarantee that balls would not find their way into the "dead" spots and become trapped. Therefore, this approach was discontinued.

Filling the dead areas of the waterboxes with a solid material, preferably foam, is a viable approach. A polyurethane foam of approximately 2 to 4 LB/FT³ density is the leading candidate material.

The method of foaming is to build wooden forms with proper supports, assemble the forms to the waterboxes (either prior to or after waterbox assembly), inject the foam/catalyst, room temperature cure, remove the forms, and spray-up or hand-lay a fiberglass reinforced plastic coating over the foam surfaces. This coating is required to prevent flaking or erosion of the foam and to provide a strong surface for supporting workers and scaffolding. A description of the equipment and manufacturing steps is given in Reference 1. Two problems are encountered with this type of guiding system. First, the forms must be constructed to clear the instrumented tube hose supports and the thermistor supports. This greatly complicates the form design and installation/removal. Secondly, the instrumented tube hose connections must be made after foaming. This requires that the foam be removed around the hose fittings to allow the clamp to be attached and secured. As a result, a large amount of dead space around the hose fittings will be created. Some means of either rebuilding the foam around these fittings and fiberglassing the surface or of accounting for the effect of these dead spaces will have to be developed. The foaming would be done after the completion of the heat exchanger hydrotest so that the operation will cause an extension of the schedule to accommodate the form fabrication, the foaming steps, and form removal. Foaming may be done at either SECO or at the Portland, Oregon shipyard.

A simpler approach is to construct guiding screens withing the waterboxes. Screen support rings or frames would be fabricated in parallel to the heat exchanger fabrication and installed by bolts just prior to hydrotesting. Screens will then be attached to the supports by tying and this operation can be performed either at SECO or the shipyard. A sketch of the supports are shown in Figures 1 through 3 for an

evaporator and condenser inlet, respectively.

A circular ring is attached to the waterbox just downstream of the inlet by means of four or more lugs. The ring may be made of either bar or rod stock. The ring material can be steel with a corrosion resistant coating. For the evaporator, two rectangular frames are mounted in the tubesheet skirt by lugs. The frame closest to the tubesheet must be segmented so that it can be installed past the instrumented tube hose supports. These frames would be installed prior to the instrumented tube hoses. At this time, the skirt screen is installed (either before or after the waterbox attachment) by tying to the periphery of the frames. Slits are made into the screen material to fit around the penetrating hoses and hose supports. After RTD installation and waterbox attachment, screen segments are tied between the inlet support ring and the support frame nearest the skirt flange. Again, the screen must be fitted around the RTD supports and leads. The outlet side of the evaporator is of the same design except that more skirt and waterbox penetrations must be accommodated. In addition, a flow splitter plate which is mounted to the outlet skirt penetrates partially into the waterbox, see Figure 2. Therefore, the rectangular frames must be split at this plane and attached to the splitter plate.

The frames in the condensor can be somewhat simplified, Figure 3. Two circular frames are mounted in the waterbox. The larger of the two rings has a "cutoff chord" to match the tubefield on the top. A simple rectangular frame is mounted on a horizontal plane in the shell skirt. No ring is required in the skirt near the tubesheet since the tubefield extends to very near the skirt wall.

Two types of screen material were investigated. The first is a woven polyester or polyamide screen material and the second is standard nylon fishing netting. The woven screen is stiffer but has the disadvantage that the individual strands are not secure at the crossing points so that shifting can occur. If this happens, local areas may have openings greater than the ball diameter and some escape and trapping may occur. Additionally, the cut edges of the woven screen must be heat sealed or secured in some way to prevent unravelling. The nylon netting appears to be a better choice. Anchovie netting with openings of approximately .413 inch square is readily available and easily cut and tied to conform to difficult patterns. Each cross over is knotted so that the size of the openings is quite stable. Twenty five foot wide lengths of netting can be

Page 4

Amertap Ball Guiding System - 1 MWe OTEC
5515.2.79-271

obtained for \$4.00/linear foot (quote from U. S. Sports Netting, Long Beach, Calif.) which is very inexpensive. The netting can be installed at anytime after the frames are bolted to their mounting lugs.

Of the three methods discussed above (flow vanes, filling with a solid material, or use of a guiding screen) the guiding screen appears to be the best approach to assure proper Amertap ball flow at a minimum impact to the fabrication procedures and schedule. In order to implement this approach, immediate design activity must be started at SECO to design the frames and, especially, the mounting lugs. These mounting lugs must be installed prior to the waterbox and skirt neoprene coating operations.

LAR:dj

Figure 1. SCREEN SUPPORTS - EVAPORATOR
(INLET SIDE)

A-32

Figure 2. SCREEN SUPPORTS - EVAPORATOR SKIRT
(OUTLET SIDE)

Figure 3. SCREEN SUPPORTS - CONDENSER
(INLET SIDE SHOWN)

APPENDIX B

EVAPORATOR SYSTEM DESIGN

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK - REDONDO BEACH - CALIFORNIA 90278

INTEROFFICE CORRESPONDENCE

79.6852.4.5-21

TO: J. E. Snyder

CC: M. H. Seidman

DATE: 8 March 1979

SUBJECT: Transmittal of Suction Head
Requirements for 1MWe Evaporator
Reflux Pumps

MS
FROM: A. M. Sprouse
BLDG. 81 MAIL STA. 1578 EXT. 52130

Attached are calculations of required static head for the suction lines of the 1MWe evaporator reflux pumps, including the previously optional pump P-3 (for mid-plane feed). Note that the most severe requirement is for pump P-2 which needs at least 13.1 ft from bottom of drain tank to pump centerline. Pumps P-1 and P-3 require 11.9 and 10.0 feet respectively. Also note that these calculations do not include the effects of platform acceleration.

Approved:

P. T. Fukunaga
P. T. Fukunaga

AMS:mn

Attachment

PREPARED A.M. SPROUSE

REPORT NO.

PAGE 1

CHECKED MHS

REFLUX PUMP SUCTION LINE
JAN 17 1979

MODEL 111We HX's

I OBJECTIVE: DETERMINE REQUIRED STATIC HEAD FROM BOTTOM OF DRAIN TANK TO PUMP CENTERLINE

II APPROACH: COMPARE LINE LOSSES TO PUMP NPSH_R, ASSUME T @ PUMP = T_{set} (i.e. HEATING NEGLIGIBLE)

III SYSTEM STATEMENT
PUMP P-1

$$\text{STATIC HEAD} = 7' 9 \frac{1}{8}'' + Z = 7.76' + Z = H_s$$

FITTING LOSSES:

- 1- 18" INTERNAL UPSET INLET
- 1- 18" LR 90° ELL, W
- 2- 18" x 16" ECC. RED. T EXP. W
- 2- 18" WELD TEE
- 1- 18" x 12" CONC. RED. W
- 1- 12" LR 90° ELL, W

K	W, lb/hr (MAX)
1.0 ^b	1,280,000
0.1 ^b	"
0.019, 0.013 ^c	"
0.45 ^b	"
0.302 ^a	1,024,000
0.13 ^b	"

B-3

^a CRANE, "FLOW OF FLUIDS", PUB. 410, 1977 ed., p. A-26

^b VALUES FROM GOULD'S PUMP MANUAL, 1976 ed., SxN 16C, p. 340ff
VALUES ARE FOR FLANGED FITTINGS AS WELDED NOT GIVEN.

PREPARED A.M. SPROUSE

REPORT NO.

PAGE 2

CHECKED MHS

REFLUX PUMP SUCTION LINE

MODEL 111W₂ HX'S

JAN 19 1979

ELEMENT LOSSES

16" FILTER (F-1)

$$\Delta p = 0.25 - 0.35 \text{ psid}$$

12" BUTTERFLY (V-5)

$$C_v = 5000$$

PIPE LOSSES

6" - 18" SCH 40 C.S. WELDED @ 1,280,000 lb/hr

3 1/2' - 18" SCH 40 C.S. WELDED @ 1,024,000 lb/hr

2' - 12" SCH 40 C.S. WELDED @ 1,024,000 lb/hr

PUMP MANUFACTURER'S RECOMMENDED $NPSH_R = 8'$ (PUMP P-1)

IV. CALCULATIONS:

LINE FLOWS

$$18" @ 1,280,000 \text{ lb/hr} = 9.357 \text{ ft}^3/\text{sec} \quad (\rho = 38 \text{ lb/ft}^3)$$

$$18" \text{ PIPE, } A = 1.553 \text{ ft}^2, D = 1.4063 \text{ ft.}$$

$$V = 6.03 \text{ ft/sec}$$

$$Re = \frac{\rho V D}{\mu} = 3.22 \times 10^6 \quad (\mu = 0.36)$$

$$12" @ 1,024,000 \text{ lb/hr} = 7.485 \text{ ft}^3/\text{sec}$$

$$V = 4.82 \text{ ft/sec}$$

$$Re = 2.57 \times 10^6$$

$$12" @ 1,024,000 \text{ lb/hr} = 7.485 \text{ ft}^3/\text{sec}$$

$$12" \text{ PIPE, } A = 0.777 \text{ ft}^2, D = 0.9965 \text{ ft}$$

$$V = 9.63 \text{ ft/sec}$$

$$Re = 3.64 \times 10^6$$

HEAD LOSS, PIPING, H_p

$$H_p = \frac{4f V^2 L}{2gD} \quad \text{WHERE } f = \text{FRICTION FACTOR}$$

FOR 18" PIPE, $\epsilon/D \approx 0.0001$, $Re = 3.22 \times 10^6$, $4f \approx 0.012$

MOODY CHART, e.g. A.G. HANSEN, FLUID MECHANICS,
WILEY & SONS, 1967, p. 420, FIG. 11.2

PREPARED A.M. SPROUSE
CHECKED MHS
MODEL 1MWe OTEC HX's

REPORT NO.

PAGE 3

REFLUX PUMP SUCTION LINE

JAN 19 1979

$$H_p = \frac{(0.012)(6.03)^2(1.5)}{2 \cdot 32.2 \cdot 1.4063} = 0.0024 \text{ ft}$$

FOR 18" PIPE, $\epsilon/D \approx 0.0001$, $Re = 2.57 \times 10^6$, $4f \approx 0.0125^c$

$$H_p = \frac{(0.0125)(4.82)^2(3.5)}{2 \cdot 32.2 \cdot 1.4063} = 0.011 \text{ ft}$$

FOR 12" PIPE, $\epsilon/D \approx 0.00015$, $Re = 3.64 \times 10^6$, $4f \approx 0.013^c$

$$H_p = \frac{(0.013)(9.63)^2(Z)}{2 \cdot 32.2 \cdot 1.9965} = 0.019 Z \text{ ft.}$$

$$\underline{H_p = 0.0134 + 0.019 Z}$$

HEAD LOSS, FITTINGS, H_e

$$H_e = Z(K_i V_i^2 / 2g)$$

$$H_e = \frac{(6.03)^2}{64.4} [1. + .1 + .019 + .013 + 2(.45)] +$$

$$\frac{(4.82)^2}{64.4} [.302] + \frac{(9.63)^2}{64.4} [.13] =$$

$$\underline{H_e = 1.44 \text{ ft}}$$

HEAD LOSS, ELEMENTS, H_v , H_f

$$C_v = Q \sqrt{\text{Sp. Gr.} / \Delta p_v}^d, \quad Q \equiv \text{Flow, GPM}$$

$$Q = \frac{1,024,000}{500 \cdot .609} = 3363 \text{ GPM} \quad \text{FOR V-5}$$

$$\Delta p_v = \left(\frac{Q}{C_v} \right)^2 (\text{Sp. Gr.}) = \left(\frac{3363}{5000} \right)^2 (.609)$$

$$\Delta p_v = 0.276 \text{ psid}$$

$$\Delta p_f = 0.35 \text{ MAX.}$$

$$H = \frac{2.31 \Delta p}{\text{Sp. Gr.}}$$

^d CONTROL VALVE HAN

B-5

PREPARED A. M. SPROUSE

REPORT NO.

PAGE 4

CHECKED MHS

REFLUX PUMP SUCTION LINE

MODEL 1 MWe HX's

JAN 19 1979

$$H_f + H_v = \frac{2.31 (.35 + .276)}{.609}$$

$$\underline{H_f + H_v = 2.37 \text{ ft}}$$

V. REQUIRED ELEVATION ABOVE P-1

$$H_s - H_d - H_p - (H_f + H_v) \geq \text{NPSH}_R$$

$$7.76 + Z \geq 8 + 1.44 + 2.37 + .0134 + .019 Z$$

$$Z(1 - .019) \geq 4.06$$

$$Z \geq 4.14$$

$$H_s \geq 7.76 + 4.14$$

$$\underline{\underline{\geq 11.9 \text{ ft}}}$$

PREPARED A.M. SPROUSE

REPORT NO.

PAGE 5

CHECKED MHS

REFLUX PUMP SUCTION LINE

MODEL 1MW₂ HX'S

JAN 17 1979

VI

SYSTEM STATEMENT

PUMP P-2

↑
33"
:
:
:
13 1/2"
15"
4 5/8"
3 5/8"
4 1/8"
Z
15"

$$H_s = 87'' - 17\frac{1}{8}'' + Z = 7.406' + Z$$

18" SYSTEM SAME AS GIVEN P. 1 & P. 2 EXCEPT LENGTH OF 18" PIPE = 2.0 ft

FITTING LOSSES

- 1-18" INTERNAL UPSET INLET
- 1-18" LR 90° ELL, W
- 2-18" x 16" ECC. RED. & EXP, W.
- 2-18" WELD TEE
- 1-18" x 10" CONC. RED., W
- 1-10" LR 90° ELL, W
- 1-10" x 8" ECC. RED., W

K
1.0^b
0.1^b
0.019, 0.013^a
0.45^a
1.086^b
0.14^a
0.105^b

W, lb/hr
1,280,000
"
"
1,280,000/640,000
512,000
"
"

ELEMENT LOSSES

- 16" FILTER
- 10" BUTTERFLY (V-6)

$$\Delta p = 0.25 - 0.35 \text{ psid}$$

$$C_v = 2100$$

PIPE LOSSES

- 6"-18" SCH 40 C.S. WELDED, 1,280,000 lb/hr
- 2'-18" SCH 40 C.S. WELDED, 640,000 lb/hr
- Z-10" SCH 40 C.S. WELDED, 512,000 lb/hr

PUMP MANUFACTURER'S RECOMMENDED NPSH_R = 9' (PUMP P-2)

PREPARED A.M. SPROUSE

REPORT NO.

 PAGE 6

 CHECKED MHS
REFLUX PUMP SUCTION LINE

 MODEL 1MW, HX's
JAN 19 1979
VII. CALCULATIONS:
LINE FLOWS

18" @ 1,280,000 lb/hr, SEE p. 2

 18" @ 640,000 lb/hr = 4.678 ft³/sec

 $V = 3.01 \text{ ft/sec}$
 $Re = 1.61 \times 10^6$

 18" @ 512,000 lb/hr = 3.74 ft³/sec

 $V = 2.41 \text{ ft/sec}$

 NO PIPE RUNS, $\therefore Re$ NOT COMPUTED

 10" @ 512,000 lb/hr = 3.74 ft³/sec

 FOR 10" PIPE, $A = 0.5475 \text{ ft}^2$, $D = 0.835$
 $V = 6.836 \text{ ft/sec}$
 $Re = 2.17 \times 10^6$

NO 8" PIPE RUNS OR FITTINGS

HEAD LOSS, PIPING, H_p

FOR 18" PIPE @ 1,280,000 lb/hr, 6' LONG, SEE p. 2

 $H_p = 0.0025 \text{ ft}$

FOR 18" PIPE @ 640,000 lb/hr, 2' LONG

 $\epsilon/D \approx 0.0001$, $Re = 1.61 \times 10^6$, $4f \approx 0.013$
 $H_p = \frac{(0.013)(3.01)^2(2)}{2 \cdot 32.2 \cdot 1.4063} = 0.0026 \text{ ft.}$

 FOR 10" PIPE @ 512,000 lb/hr, Z ' LONG

 $\epsilon/D \approx 0.00018$, $Re = 2.17 \times 10^6$, $4f \approx 0.0135$
 $H_p = \frac{(0.0135)(6.836)^2 Z}{2 \cdot 32.2 \cdot 0.835} = 0.0117 Z \text{ ft.}$
 $H_p = 0.0025 + 0.0026 + 0.0117 Z$
 $= \underline{\underline{0.0051 + 0.0117 Z}}$

PREPARED A.N. SPROUSE

REPORT NO.

PAGE 7

CHECKED MHS

MODEL 1 MWc HX's

REFLUX PUMP SUCTION LINE

JAN 19 1979

HEAD LOSS, FITTINGS, H_e

$$H_e = \frac{(6.03)^2}{64.4} \cdot (1.582) + \frac{(3.01)^2}{64.4} \cdot (.45) + \frac{(2.41)^2}{64.4} \cdot (1.086) + \frac{(6.836)^2}{64.4} \cdot (.14 + .105) =$$

$$H_e = 1.23 \text{ ft}$$

HEAD LOSS, ELEMENTS, $H_v + H_f$

$$\Delta p_v = \left(\frac{Q}{C_v} \right)^2 (\text{Sp. Gr.})$$

$$C_v = 2100, Q = \frac{512,000}{500 \cdot .609} = 1681 \text{ FOR V-6}$$

$$\Delta p_v = \left(\frac{1681}{2100} \right)^2 (.609)$$

$$\Delta p_v = 0.39 \text{ psid}$$

$$\Delta p_f = 0.35 \text{ MAX.}$$

$$H_v + H_f = \frac{2.31 (.39 + .35)}{.609} = 2.81 \text{ ft}$$

VIII. REQUIRED ELEVATION ABOVE P-2

$$H_s - H_e - H_p - (H_f + H_v) \geq \text{NPSH}_R$$

$$7.406 + Z \geq 9 + 1.23 + .0051 + .0117Z + 2.81$$

$$Z (1 - .0117) \geq 5.639$$

$$Z \geq 5.71 \text{ ft}$$

$$H_s \geq 7.406 + 5.71$$

$$\underline{\underline{\geq 13.1 \text{ ft}}}$$

PREPARED A. M. SPROUCE AMS
CHECKED AMS
MODEL 1 MW. HX'S

REPORT NO.

PAGE 8

REFLUX PUMP SUCTION LINE
FEB 20 1979

IX. SYSTEM STATEMENT

PUMP P-3

$$\text{STATIC HEAD} = H_s = 88 \frac{5}{8}'' + Z = 7.385' + Z$$

FITTING LOSSES:

	<u>K</u>	<u>W_{max}, lb/hr</u>
1-18" INTERNAL UPSET INLET	1.0 ^b	1,280,000
1-18" LR 90° ELL, W	0.1 ^b	"
2-18" x 16" ECC. RED. & EXP., W	0.019, 0.013 ^a	"
2-18" WELD TEE, BRANCH x RUN	0.45 ^b	1,280,000 / 128,000
1-18" WELD TEE, RUN	0.07 ^b	640,000
1-18" x 12" CONC. RED., W	0.302 ^a	128,000
1-12" x 6" CONC. RED., W	3.186 ^a	"
1-6" LR 90° ELL, W	0.17 ^b	"

ELEMENT LOSSES:

16" FILTER (F-1) $\Delta p = 0.25 - 0.35$ psid
6" BUTTERFLY (V-7) $C_v = 850$

PIPE LOSSES:

6"-18" SCH 40 C.S. WELDED @ 1,280,000 lb/hr
2'-18" SCH 40 C.S. WELDED @ 640,000 lb/hr
6'-18" SCH 40 C.S. WELDED @ 128,000 lb/hr
2'-6" SCH 40 C.S. WELDED @ 128,000 lb/hr

PUMP MANUFACTURER'S RECOMMENDED NPSH_r = 7' (PUMP P-3)

PREPARED A.M. SPROUSE

REPORT NO.

PAGE 9

CHECKED MHS

REFLUX PUMP SUCTION LINE

MODEL 1 MW_e HX'S

FEB 20 1970

X. CALCULATIONS :

LINE FLOWS

18" @ 1,280,000 lb/hr, SEE p.2

18" @ 640,000 lb/hr, SEE p.6

18" @ 128,000 lb/hr = 0.936 ft³/sec

V = 0.602 ft/sec

$$Re = \frac{\rho V D}{\mu} = 3.22 \times 10^5$$

6" @ 128,000 lb/hr = 0.936 ft³/sec

6" PIPE, A = 0.2006 ft², D = 0.5054 ft

V = 4.666 ft/sec

$$Re = \frac{\rho V D}{\mu} = 8.96 \times 10^5$$

HEAD LOSS, PIPING, H_p

FOR 18" PIPE @ 1,280,000 lb/hr, 6" LONG, SEE p.2

$$H_p = 0.0024 \text{ ft}$$

FOR 18" PIPE @ 640,000 lb/hr, 2' LONG, SEE p.6

$$H_p = 0.0026 \text{ ft}$$

FOR 18" PIPE @ 128,000 lb/hr, 6' LONG

$$\epsilon/D \approx 0.0001 \quad Re = 3.22 \times 10^5, \quad H_f = 0.0155$$

$$H_p = \frac{(0.0155)(0.602)^2 6}{2 \cdot 32.2 \cdot 1.4063} = 0.00037 \text{ ft}$$

FOR 6" PIPE @ 128,000 lb/hr, 2' LONG

$$\epsilon/D \approx 0.0003 \quad Re = 8.96 \times 10^5, \quad H_f = 0.016$$

$$H_p = \frac{(0.016)(4.67)^2 (2)}{2 \cdot 32.2 \cdot 0.505} = 0.0107 \cdot Z \text{ ft}$$

$$H_p = 0.0024 + 0.0026 + 0.00037 + 0.0107 \cdot Z$$

$$= \underline{\underline{0.00537 + 0.0107 Z}}$$

PREPARED A. M. SPROUSE

REPORT NO.

 PAGE 10

 CHECKED MHS

 MODEL 1 MWe HX's
REFLUX PUMP SUCTION LINE
FEB 20 1979
HEAD LOSS, FITTINGS, H_e

$$H_e = \frac{1}{64.4} \left[(16.03)^2 (1.582) + (3.01)^2 (.07) + (.602)^2 (.45 + .302) + (1.20)^2 (3.186) + (4.67)^2 (.17) \right]$$

$$H_e = \underline{1.036 \text{ ft}}$$

HEAD LOSS, ELEMENTS, $H_v + H_f$

$$\Delta P_v = (Q/C_v)^2 (S_p \cdot G_r)$$

$$C_v = 850, Q = \frac{128,000}{500 \cdot .609} = 420. \quad \text{FOR V-7}$$

$$\Delta P_v = \left(\frac{420}{850} \right)^2 (.609) = 0.149 \text{ psid}$$

$$\Delta P_f = 0.35 \text{ psid, max}$$

$$H_v + H_f = \frac{2.31 (.149 + .35)}{.609} = \underline{1.89 \text{ ft}}$$

XI. REQUIRED ELEVATION ABOVE P-3

$$H_s - H_e - H_p - (H_f + H_v) \geq \text{NPSH}_R$$

$$(7.385' + Z) \geq 7 + 1.036 + .00537 + .0107 \cdot Z' + 1.89'$$

$$Z(1 - .0107) \geq 2.546$$

$$Z \geq 2.57 \text{ ft}$$

$$H_s \geq 7.385 + 2.57 \text{ ft}$$

$$\underline{H_s \geq 10. \text{ ft.}}$$

XII. CONCLUSION : HEIGHT OF DRAIN TANK IS FIXED

 BY THE MOST STRINGENT REQUIREMENT: PUMP P-2, WHICH NEEDS 13.1 ft FROM BOTTOM OF DRAIN TANK TO PUMP Φ .

PREPARED A. M. SPROUSE

CHECKED MHS

MODEL 1MW₂ HX's

REPORT NO.

PAGE

REFLUX PUMP SUCTION LINE

JAN 18 1979

APPENDIX A CIRCULATION OF K FOR REDUCERS AND EXPANDERS^a

$$K_r = \frac{0.8 \sin \frac{\theta}{2} (1 - \beta^2)}{\beta^4} \quad \theta = 2 \arctan \frac{D_2 - D_1}{2L}, \quad \beta = \frac{D_1}{D_2}$$

$$K_e = \frac{2.6 \sin \frac{\theta}{2} (1 - \beta^2)^2}{\beta^4} \quad \beta^2 = \frac{A_1}{A_2}$$

18":	D = 17.25	A = 233	18" x 16"	L = 15"
16":	D = 15.25	A = 182.65	18" x 12"	L = 15"
12":	D = 12.00	A = 113.10	18" x 10"	L = 15"
10":	D = 10.02	A = 78.85	10" x 8"	L = 7'
8":	D = 7.98	A = 50.03	12" x 6"	L = 8"
6":	D = 6.065	A = 28.89		

$$18" \times 16": K_r = \frac{0.8 \sin \frac{133}{2} (1 - 0.784)}{0.784^2} = 0.019$$

$$16" \times 18": \quad K_d = \frac{2.6 \text{ Air} \cdot \frac{1.33}{2} (1 - 0.784)^2}{0.784^2} = 0.013$$

$$18 \times 12": \quad K_r = \frac{0.8 \sin \frac{.346}{2} (1 - .485)}{.485^2} = 0.362$$

$$18'' \times 10'' : K_r = \frac{0.8 \sin \frac{.473}{2} (1 - .338)}{.338} = 1.086$$

$$10 \times 8": \quad K_r = \frac{0.8 \sin \frac{289}{2} (1 - .634)}{.634} = 0.105$$

12' x 6" : $K_r = \frac{0.8 \sin \frac{710}{2} (1 - .255)}{(.255)^2} = 3.186$ FEB 20 1979

FEB 20 1979

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK - REDONDO BEACH - CALIFORNIA 90278

INTEROFFICE CORRESPONDENCE

79.6852.4.4-36

TO: J. E. Snyder

CC: K. P. Cramer
A. M. Sprouse
M. J. Watson

DATE: 15 May 1979

SUBJECT: Phase Separator Design
AdequacyFROM: M. H. Seidman *MHS*
BLDG. 81 MAIL STA. 1578 EXT. 52130

This document discusses the design adequacy and limitations of the phase separator being built by Peerless Manufacturing Company for the LMWe Heat Exchanger for OTEC Project. It is provided in fulfillment of Deliverable 4 for PWA 3773-21 (Task 1-2, a).

Peerless has provided (Reference 1) curves of separation efficiency and pressure drop as a function of percent of nominal flow, Figures 1 and 2. Nominal flow is defined in the figures as 267,000 lb/hr. The separation efficiency is shown as 100% out to a value of 120% of nominal flow. This separation efficiency of 100% means design efficiency, Reference 2, which will result in a predicted outlet vapor quality of "approximately 99.0 to 99.5%," Reference 3. It does not mean that the unit will remove 100% of the liquid but was drawn at that level "because of the scale of the figure," Reference 2.

Reference 1 states the following: "Please note that the graph of Separation Efficiency vs. Percent Nominal Flow is an approximation. It demonstrates that at some point beyond rated flow, usually about 120% of rated flow, the separation efficiency begins to deteriorate and as the flow increases, the deterioration becomes more rapid. This curve is an estimation based on air-water tests of mist extractors in various installation configurations."

Note also the word of caution on Figure 1. The approximation applies to when and by how much the efficiency falls off with increasing flow. It could go to perhaps 130% before falling off, Reference 2.

The separation efficiency is shown in Figure 1 as a function of ammonia flow. This curve is valid for values of incoming ammonia quality greater than 60 to 80%, Reference 2. The vane separators can handle 3 gpm per square foot of frontal area of the vane boxes. From Reference 4 this frontal area is at least 22 square feet. (It could be as high as 28 square feet; this will be defined when detailed drawings are received.) Thus the vane separators can handle at least 66 gpm without any degradation in efficiency. However, the capacity of the phase separator is much larger because it has a quieting plate and venetian blinds (primary) prior to the vane separator (secondary), Reference 4. This primary separator "will knock down large quantities of liquid prior to the mist extractor" (i.e., secondary separator), Reference 2.

The current design of the OTEC system indicates a liquid flow from the phase separator of 10,670 lb/hr, Reference 5, or 35 gpm. Thus the phase separator should adequately handle any flow quality reasonably expected from the 1MWe OTEC system without degrading its efficiency.

The nominal flow of 267,000 lb/hr, Reference 5, to the phase separator was based on smooth tubes in the evaporator. The current design is for the lower half of the evaporator tube bundle to utilize tubes whose outside surface is enhanced by a proprietary UCC/Linde process. In addition, the evaporator is also designed to operate in a pool boiling mode. The anticipated vapor flow rates leaving the evaporator (i.e., inlet to phase separator) are 111% and 106% of nominal design flow for the standard spray bundle mode and for the pool boiling mode respectively, Reference 6. These are within the operating range for maximum separation efficiency.

In Figure 2 the pressure drop through the phase separator is shown versus the flow rate. At 115% of nominal flow the pressure drop is 0.5 psi. A change in inlet quality will not significantly affect the pressure drop. There would be some increase in pressure drop due to a decrease in quality. This is caused by filling the inlet nozzle with more liquid. There would be no effect at the outlet nozzle. Peerless does not have any quantitative numbers for this effect, Reference 2.

The curve presented in Figure 2 can be used to estimate the pressure drop through the phase separator at any flow rate up to 120% of nominal. Extrapolation of the curve to flow rates up to 140% of nominal should present no problem.

Conclusions

Review of the limited data and information provided by Peerless, References 1 and 7, indicates that the phase separator will meet the design requirement of providing greater than 99.5% quality at design operating conditions. At the ammonia flow rates expected during operation with the enhanced tubes and at operation in the pool boiling mode there will be no degradation in separation efficiency. Thus outlet ammonia quality will not be affected. Similarly at ammonia flow rates less than nominal, the outlet ammonia quality is not altered.

Assuming that 50 percent of the entering liquid ammonia is removed in the primary (venetian blind) separator, then the phase separator will operate at design efficiency for ammonia inlet qualities at least as low as 85 percent. Recall as stated earlier that Reference 1 indicated that phase separator performance will not be degraded down to an inlet quality of 60 to 80 percent. In any case, the current design should be adequate for any anticipated system operating point.

For any anticipated operating point, the pressure drop will be less than 0.5 psi. As inlet ammonia quality decreases, there will be some small, but undefined, increase in pressure drop. This change should still keep the pressure drop within the bounds of 0.5 psi or less.

Approved: _____

P. T. Fukunaga

Figure 1

B-16
SEPARATION EFFICIENCY (%)

Figure 2

PRESSURE DROP (IN. H₂O)

PERCENT NOMINAL FLOW

REFERENCES

1. Letter from D. Van Calhoun, Peerless to A. L. Yarden, SEC, April 5, 1979.
2. Telephone conversation between Van Calhoun, Peerless Manufacturing and M. H. Seidman, TRW, April 25, 1979.
3. Peerless Manufacturing Company Quotation, Proposal No. M6-1856, from Sherrill Stone to A. L. Yarden, SEC, January 26, 1979.
4. Peerless Manufacturing Company Drawing Number 6B-36092, Revision 1, February 22, 1979.
5. "Process Flow Diagram", TRW Drawing Number 6850-37-01, Revision 3, 23 February, 1979.
6. Private conversation between A. M. Sprouse and M. H. Seidman, May 7, 1979 (To be published in paper presented at sixth OTEC conference).
7. Sherman, R.M., "Determination of the Performance Characteristics of a Standard Eight Inch Wide Peerless Vane Type Liquid-Gas Separator", Peerless Test Report No. E-1547, 26 August, 1974

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK - REDONDO BEACH - CALIFORNIA 90278

INTEROFFICE CORRESPONDENCE

79.6852.4.4-70

TO: J. E. Snyder

CC: A. M. Sprouse

DATE: 21 November 1979

SUBJECT: Phase Separator Drain Line

FROM: M. H. Seidman *MHS*

BLDG. 81 MAIL STA. 1578 EXT. 52130

The attached IOC presents the analysis of the subject item. It also includes the specifications for the equipment required for operation of the subject system.

A second attachment presents calculations for selection of the control valve in the drain line.

The level sensor in the phase separator drain has been changed from a displacer type to a Δ -P type. The manufacturer does not recommend displacer type level controls for a marine environment.

Approved:

P. T. Fukunaga

MHS:cm

Attachments

TRW

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK - REDONDO BEACH - CALIFORNIA 90278

INTEROFFICE CORRESPONDENCE

79.6852.3.11-36

TO: M. H. Seidman

CC: P. R. Edris
P. L. Trubey

DATE: 16 November 1979

SUBJECT: OTEC-1; 1 MWe. Revision Design
of Phase Separator Liquid Return
System.

FROM: P. G. Otero *P. G. Otero*
BLDG. 81 MAIL STA. 1578 EXT. 52130

Reference: TRW IOC 79.6852.3.11-33, from P. G. Otero; Subject:
1 MWe Design of Phase Separator Liquid Return System;
dated: 28 September 1979.

Phase Separator Liquid Return design is completed. Design is based upon the evaporator operating at 120% of nominal flow and producing vapor of 96% quality. Total liquid return to the evaporator drain tank is 42 G.P.M.

Attached design package includes:

- Analysis justifying line and component sizing
- "B" size P&ID with design restraints noted
- Procurement specifications for all equipment items

Reference document No. 79.6852.3.11-33 is obsolete, and to be replaced by present document.

PO:df

cc: A. Nagler *A.N.*
P. Otero
M. Seidman

AVOID VERBAL ORDERS

TO: P. R. Edris Bldg. 81 Mail Sta. 1578 Date 9/24/79
FROM: R. A. Wise *RAM* Bldg. 81 Mail Sta. 1569 Ext. 63374
SUBJECT: 1 MW Phase Separator Liquid Return Line Flow Rate

The flow rate analysis that was performed on the phase separator liquid return line shows that the maximum liquid ammonia flow rate is 69.3 GPM with the flow control valves wide open. The required flow rate is 42 GPM which can be obtained by partially closing the control valve. The analysis is attached. The analysis assumed 40 feet of 4" line and 4' of 1-1/2" line with a head of 10 feet. The configuration is shown in Figure 1.

If the head is reduced to 9', the flow rate will be 65.8 GPM.

The C_v of the control valve partially closed to obtain the 42 GPM is 14.1 GPM/psi.

FIGURE 1

NOTES:

1. LCV-1 TO BE CLOSE COUPLED TO EVAPORATOR DRAIN TANK CONNECT NOZZLE
2. FE-4 REQUIRES 2'0" UPSTREAM AND 2'0" DOWNSTREAM STRAIGHT RUN
3. LINE 171-A-1 1/2 TO BE RESTRICTED TO 5'-0" MAX LENGTH
4. LIC-2 TO BE PROVIDED BY SIC

PREPARED R.A. WISE 9/24/79 REPORT NO.

PAGE 1

CHECKED [Signature] 11/19/79

PROJECT

1. MW PHASE SEPARATOR
DRAIN CIRCUIT-MAIN LOOP

I.) PRESSURE DROP EQUATION

$$\Delta P_L = \frac{K_L V_L^2}{2g} \quad V_L = \frac{Q_L}{A_L} = \frac{Q}{A_L} \quad Q \text{ IS THE SAME FOR COMPONENTS}$$

$$\Delta P_L = \frac{K_L \left(\frac{Q}{A_L}\right)^2}{2g} = \frac{K_L Q^2}{A_L^2 \cdot 2g} = K_L Q^2 \quad K_L = \frac{K_L}{2g A_L^2}$$

$$\Delta P_T = \sum K_L Q^2 = Q^2 \sum K_L \quad K_T = \sum K_L = \sum \frac{K_L}{2g A_L^2}$$

$$\underline{\underline{\Delta P_T = K_T Q^2}}$$

II.) POT TO 4" LINE LOSS FACTOR

$$\frac{A_p}{A_L} = \frac{\pi D_p^2}{4} = \left(\frac{D_p}{D_L}\right)^2 \quad D_p = 16" \quad D_L = 3.826"$$

$$\frac{A_p}{A_L} = \left(\frac{3.826}{16}\right)^2 = .0572 \quad \text{FROM FIGURE 2 } K_L' = .475$$

$$A_L = \frac{\pi D_L^2}{4} = \frac{\pi (3.826)^2}{4} \quad A_L = .07984 \text{ FT}^2$$

$$K_L = \frac{.475}{2 \times 32.174 \times (.07984)^2} \quad K_L = 1.15804$$

III.) FRICTION LOSS FROM POT TO VALVE

ASSUME $Q = 42 \text{ GPM} = .09358 \text{ FT}^3/\text{sec}$

$$A = .07984 \text{ FT}^2 \quad V = \frac{Q}{A} = \frac{.09358}{.07984} \quad V = 1.172 \frac{\text{FT}}{\text{sec}}$$

$$\rho = 38.0 \frac{\text{#}}{\text{FT}^3} \quad D = 3.826" \quad \gamma = .359 \frac{\text{#}}{\text{FT-HR}}$$

FIGURE 2
LOSS FACTORS FOR SUDDEN CONTRACTIONS

PREPARED R.A. WISE 9/24/79 REPORT NO. _____
 CHECKED ms 11/19/79 _____
 PROJECT _____

PAGE 3

$$N_{RE} = \frac{VDP}{\mu} = \frac{3600 \times 1.172 \times \frac{3.826}{12} \times 38}{.359} = 1.42 \times 10^5$$

FROM FIGURE 3 $\frac{\epsilon}{D} = .000015$ $L = 40'$

FROM FIGURE 4 $f = .0170$ $D_2 = 3.826''$

$$K_2 = \frac{fL}{D_2} \cdot \frac{A_2}{2gA_2^2} \quad A_2 = .07984 \text{ FT}^2$$

$$K_2 = \frac{.0170 \times \frac{40}{\frac{3.826}{12}}}{64.348 (.07984)^2}$$

$$K_2 = 5.20$$

IV.) BUTTERFLY VALVE

FROM POSI-SEAL CATALOG FOR 4" BUTTERFLY VALVE $C_v = 404 \text{ GPM/PSI}$

$$K_3 = \frac{889.76 D_3^4}{C_v^2 \times 2g A_3^2} \quad D_3 = 4'' \quad A_3 = .07984 \text{ FT}^2$$

$$K_3 = \frac{889.76 (4)^4}{(404)^2 \times 64.348 (.07984)^2}$$

$$K_3 = 3.4024$$

V.) 90° BEND

$$\frac{R'}{D} = 1.5 \text{ (MINIMUM PRESSURE DROP)}$$

FIGURE 3.

RELATIVE ROUGHNESS FOR DUCT MATERIAL

FIGURE 4
FRICTION FACTORS

PREPARED R.A. WISE

REPORT NO.

PAGE 6

CHECKED MLG 11/19/79

PROJECT

FROM FIGURE 5 $K'_B = .32$

FOR LENGTH OF BEND ASSUME $1/4$ OF A CIRCLE OF RADIUS R'

$$\frac{R'}{D} = 1.5 \quad D = 3.826 \quad R' = 1.5 \times 3.826 = 5.739"$$

$$C = 2\pi R' \quad L = \frac{C}{4} = \frac{2\pi R'}{4} = \frac{\pi R'}{2} = \frac{\pi \times 5.739}{2} \quad L = 9"$$

FROM SECTION III $f = .0170 \quad A_4 = .07984 F_1^2$

$$K_4 = \frac{K'_B + \frac{fL}{D}}{2g A_{10}^2} = \frac{.32 + \frac{.017 \times 9}{12}}{\frac{3.84/12}{64.348 (.07984)^2}} \quad K_4 = .8776$$

VI.) 90° BEND

FROM SECTION II

$$K_5 = .8776$$

VII.) 4" TO 2" REDUCER

$$\tan \frac{\theta}{2} = \frac{1}{1 \frac{5}{16}} \quad \theta = 74.6^\circ$$

FROM FIGURE 6

$$K_6' = .135$$

$$K_6 = .5779$$

FIGURE 5
LOSS FACTORS FOR 90° ROUND BENDS

FIGURE 6
LOSS FACTOR FOR CONVERGING NOZZLE

PREPARED R.A. WISE

9/13/79

REPORT NO.

PAGE 9

CHECKED AW

11/19/79

PROJECT

$$K_6 = \frac{K_5'}{2g A_5^2} \quad D_6 = 1.939 \quad A_6 = .020506 \text{ FT}^2$$

$$K_6 = \frac{.135}{64.348 (.020506)^2}$$

$$\underline{K_6 = 4.9892}$$

VIII) 2" TO 1 1/2" REDUCER

$$\tan \frac{\theta}{2} = \frac{.25}{1.375} \quad \theta = 20.6^\circ$$

$$\text{FROM FIGURE 6} \quad K_7' = .02$$

$$D_7 = 1.5" \quad A_7 = .012272 \text{ FT}^2$$

$$K_7 = \frac{.020}{64.348 (.012272)^2}$$

$$\underline{K_7 = 2.0638}$$

IX.) FRICTION LOSS FROM REDUCER TO C.V. AND C.V. TO 90° BEND

$$Q = 42 \text{ GPM} = .09358 \text{ FT}^3/\text{sec} \quad A_8 = .012272$$

$$D_8 = 1 1/2" \quad V = \frac{Q}{A_8} = \frac{.09358}{.012272} \quad V_8 = 7.626 \text{ FT/sec}$$

$$\rho = 38 \text{ #/FT}^3 \quad \mu = .359 \text{ #/FT-sec}$$

$$N_{RE} = \frac{V_8 D_8 \rho}{\mu} = \frac{3600 \times 7.626 \times \frac{1.5}{12} \times 38}{.359} = 3.63 \times 10^5$$

$$\text{FROM FIGURE 3} \quad \frac{\epsilon}{D} = .00004 \quad L = 4'$$

$$\text{FROM FIGURE 4} \quad f = .0138$$

$$K_8 = \frac{f L_8}{2g A_8^2} = \frac{.0138 \times 4}{\frac{1.5}{12}} \div 64.348 (.012272)^2$$

$$\underline{K_8 = 45.5695}$$

PREPARED R. A. WISE 12/13/79 REPORT NO. _____
 CHECKED AKG 11/19/79 _____
 PROJECT _____

PAGE 10

XI) CONTROL VALVE

FROM POST-SEAL CATALOG FOR 1 1/2" BUTTERFLY VALVE $C_v = 38$ GPM/PSI

$D_g = 1.5"$ $A_g = .012272 \text{ FT}^2$

$K_g = \frac{889.76 (1.5)^4}{(38)^2 \times 64.348 (.012272)^2}$ $K_g = 321.8851$

XII) 1 1/2" TO 2" DIFFUSER

FROM SECTION VIII $\theta = 20.60^\circ$

$\frac{A_1}{A_2} = \frac{(1.5)^2}{(1.939)^2} = .5984$

$A_{10} = .012272 \text{ FT}^2$ FROM FIGURE 7 $K'_{10} = .08$

$K_{10} = \frac{.08}{64.348 (.012272)^2}$ $K_{10} = 8.2551$

XIII) 2" TO 4" DIFFUSER

FROM SECTION VI $\theta = 74.6^\circ$

$\frac{A_1}{A_2} = \left(\frac{2}{4}\right)^2 = .25$

FROM FIGURE 7 $K'_{11} = .58$

$A_{11} = .020506 \text{ FT}^2$

$K_{11} = \frac{.58}{64.348 (.020506)^2}$ $K_{11} = 21.4354$

FIGURE 7
LOSS FACTORS FOR ROUND DIFFUSERS

PREPARED R. A. WISE 9/13/79
CHECKED AMG 11/19/79
PROJECT _____

REPORT NO. _____

PAGE 12

XIII.) 90° BEND

FROM SECTION V $K_{12} = 1.8776$

XIV.) EXPANSION LOSS TO TANK

$\frac{A_1}{A_2} \approx 0$ FROM FIGURE 8 $K_{13}' = 1.0$ $A_{13} = 1.07984 \text{ FT}^2$

$$K_{13} = \frac{1}{64.348 (1.07984)^2} \quad \underline{\underline{K_{13} = 2.4379}}$$

XV.) TOTAL LOSS FACTOR

$$K_T = \sum_{i=1}^{13} K_i = 1.15804 + 5.20 + 3.4024 + .8776 + .8776 + 4.9892 + 2.0638 + 45.5695 + 321.8881 + 8.2551 + 21.4354 + .8776 + 2.4379$$

$$\underline{\underline{K_T = 419.03224}}$$

XVI.) FLOW RATE

$$\Delta P_T = h = 10' \quad h = K_T Q^2$$

$$Q = \sqrt{\frac{h}{K_T}} = \sqrt{\frac{10}{419.032}} \quad \underline{\underline{Q = .1544 \frac{\text{FT}^3}{\text{SEC}} = 69.33 \text{ GPM}}}$$

XVII.) C_v OF CONTROL VALVE WITH 42 GPM

$K_q = 321,8881$ WITH VALVE WIDE OPEN

SUBTRACT K_q FROM K_T B-34

FIGURE 7
LOSS FACTORS FOR SUDDEN EXPANSIONS

PREPARED R. A. WISSE

REPORT NO.

PAGE 14

CHECKED AMG

11/19/78

PROJECT

$$K_T' = K_T - K_g = 419.0322 - 321.8881 \quad K_T' = 97.1541$$

LET K_{14} = LOSS FACTOR OF $1\frac{1}{2}$ " VALVE
CLOSED SO THAT 42 GPM WILL PASS THRU SYSTEM,

$$Q = 42 \text{ GPM} = .0935828 \frac{\text{FT}^3}{\text{SEC}}$$

$$h = 10'$$

$$Q = \sqrt{\frac{h}{K_T' + K_{14}}}$$

$$K_T' + K_{14} = \frac{h}{Q^2}$$

$$K_{14} = \frac{h}{Q^2} - K_T' = \frac{10}{(.0935828)^2} - 97.1541$$

$$K_{14} = 1044.6982$$

$$C_v = \sqrt{\frac{889.76 D_g^2}{K_{14} \times 2g A_g^2}} = \sqrt{\frac{889.76 (1.5)^2}{1044.6982 \times 64.348 (.012272)^2}}$$

$$C_v = 14.06 \frac{\text{GPM}}{\text{PSI}}$$

NOTES:

1. LCV-1 TO BE CLOSE COUPLED TO EVAPORATOR DRAIN TANK CONNECT NOZZLE
2. FE-4 REQUIRES 2'0" UPSTREAM AND 2'0" DOWNSTREAM STRAIGHT RUN
3. LINE 171-A-14 TO BE RESTRICTED TO 8'-0" MAX LENGTH
4. LIC-2 TO BE PROVIDED BY SIC

ITEM	DESCRIPTION	QUANTITY	SIZE & CONN
LE/LT-2	DIFFERENTIAL PRESSURE TRANSMITTER ROSEMOUNT MODEL 1151DP, 4-20 MA OUTPUT SIGNAL	1	1/2 INCH RPT
LCV-1	FLOW CONTROL VALVE: FISHER GLOBE VALVE MODEL 657-ED	1	1 1/2"-300# FLANGELESS
FE-4	FLOW METER: CONTOLOTRON ULTRASONIC, 4-20 MA ANALOG OUTPUT, TYPE 241E	1	1 1/2" SCHEDULE 80 C.S. PIPE
V-14	HAND SHUT-OFF VALVE: FISHER MODEL NO. 1075 - 8500	1	4"-300# FLANGELESS
LY-2	ELECTRO PNEUMATIC TRANSDUCER FISHER MODEL NO. 546, 4-20 MA INPUT SIGNAL, 3-15 PSIG OUTPUT SIGNAL	1	

B-37

1	11-2-79	REVISED INSTRUMENTATION AND MINIMUM HEIGHT REQUIREMENT	LEV	5
LYO	DATE	REVISION	BY	APP
DESIGNED BY: L. VILLALFANDO		1 MWb HEAT EXCHANGER FOR OTEC		
CHECKED BY: A. J. G.		PHASE SEPARATOR DRAIN PIPING AND INSTRUMENTATION DIAGRAM		
APPROVALS		FACILITIES, MECHANICAL AND POWER INTEGRATION LABORATORY		
PROJECT OFFICE		6850-91-38		
RELEASED BY: [Signature]		77 02 21 1		

LEVEL DIFFERENTIAL PRESSURE TRANSMITTERS

MANUFACTURER: Rosemount

MODEL NO.: 1151DP or Equal

ITEM NUMBERS: LE/LT-3,1, LE/LT-3,2, LE/LT-3,3, LE/LT-3,4, LE/LT-3,5, LE/LT-2

DIFFERENTIAL UNIT:

LEVEL TRANSMITTER

SUPPRESSION AND ELEVATION: Integral

DIFFERENTIAL UNIT: Diaphragm

BODY MATERIAL: Carbon/Steel

WETTED PARTS: 316 S.S.

BODY RATING: 2000 psig

DIFFERENTIAL CONNECTION: 1/2 inch NPT

MOUNTING: Yoke

SCALE RANGE: 0-100 Linear

OUTPUT SIGNAL WILL INCREASE WITH INCREASE IN DIFFERENTIAL PRESSURE

TRANSMITTER

TYPE: Electronic

OUTPUT: 4-20 ma

POWER SUPPLY: 24 VDC

LOAD IMPEDANCE: 600 OHMS

ACCURACY: $\pm 0.2\%$ of SpanLINERARITY: $\pm 0.1\%$ of SpanHYSTERESIS: $\pm 0.05\%$ of Span

ELECTRICAL CONNECTION: 1/2 inch NPT Electrical Conduit Hub

ELECTRICAL CLASSIFICATION: Class 1, Division 2, Group D

COPPER, COPPER BEARING MATERIAL AND VITON ARE NOT PERMITTED ON ANY ITEMS

WETTED WITH AMMONIA.

SERVICE: Condenser drain pots, item Numbers; LE/LT-3,1, LE/LT-3,2, LE/LT-3,3
LE/LT-3,4, LE/LT-3.5.

Phase separator, Item Number LE/LT-2

SIZE	FSCM NO.	Level Differential Pressure Transmitters	REV.
A	11982		
SCALE		SHEET	1

ITEM NO.	PROCESS FLUID	LEVEL RANGE (IN.)	LEVEL RANGE IN. OF WATER	SPECIFIC GRAVITY PROCESS FLUID	NOMINAL TEMP °F	NOMINAL PRESSURE psig
LE/LT-3,1	Liquid Ammonia	14	0-8.8	0.627	48	73.7
LE/LT-3,2	Liquid Ammonia	14	0-8.8	0.627	48	73.7
LE/LT-3,3	Liquid Ammonia	14	0-8.8	0.627	48	73.7
LE/LT-3,4	Liquid Ammonia	14	0-8.8	0.627	48	73.7
LE/LT-3,5	Liquid Ammonia	14	0-8.8	0.627	48	73.7
LE/TL-2	Liquid Ammonia	14	0-8.5	0.607	72	118.7

SIZE A	FSCM NO. 11982	Level Differential Pressure Transmitters	REV.
SCALE		SHEET 2	

SONIC FLOWMETER

ITEM NO.: FE/FT-4

MANUFACTURER: Clampitron, or Badgermeter Model No. 150 or equal

LINE NO.: 171-A-1-1/2

FLUID: Liquid Ammonia

FLOW RANGE: 0-60 GPM

SCALE RANGE: 0-60 GPM

NOMINAL FLOW: 42 GPM

NOMINAL VELOCITY: 7.64 FT/ SEC

NOMINAL TEMPERATURE: 72°F

NOMINAL PRESSURE: 120.85 psig

SPECIFIC GRAVITY: 0.607

PIPE SIZE & SCHEDULE: 1-1/2 in. Sch. 80

INSIDE PIPE DIAMETER: 1.500 in. Nominal

PIPE MATERIAL: ASTM A-106, Grade B

TRANSDUCER: 240 series, externally mounted with integral mounting clamps.

The electrical enclosure shall be Class I, Division 2, Group D.

Transducer shall be furnished with 30 feet transducer cable.

TRANSMITTER:

TRANSMITTER: Model No. 241E.

OUTPUT SIGNAL: 4-20 ma. at 500 ohms (direct acting)

POWER SUPPLY: 115 VAC, 60 Hz

ACCURACY: $\pm 1\%$ over specified rangeLINEARITY: $\pm 1\%$ REPEATABILITY: $\pm 1/2\%$

ELECTRICAL ENCLOSURE: Class I, Division 2, Group D

1 MWe Heat Exchanger for OTEC

SIZE	FSCM NO.	Phase Separator Drain Line	REV.
A	11982	SONIC FLOWMETER FE/FT-4	
SCALE		SHEET	16

ITEM NO.: LCV-1

MANUFACTURER: Fisher Model No. 657-ED or equal

LINE NO.: 171-A-1-1/2

FLUID: Liquid Ammonia

NOMINAL FLOW: 42 GPM

INLET PRESSURE: 120.85 psig

NOMINAL PRESSURE DROP: 1 psi

NOMINAL TEMPERATURE: 72°F

SPECIFIC GRAVITY: 0.607

VALVE BODY SIZE: 1-1/2 in.

PORT SIZE: 1-7/8 in.

TYPE OF BODY: Globe

BODY MATERIAL: Carbon Steel

END CONNECTION: 300 #R.F.

PACKING: Teflon Asbestos

GUIDING: Cage

VALVE CHARACTERISTIC: Linear

TRIM MATERIAL: 316 s.s.

CV FOR NOMINAL FLOW: 32.7

FURNISHED CV: 39.2

ACTUATOR TYPE: Diaphragm

FAIL POSITION: Closed

INPUT SIGNAL: 3-15 psig

△P FOR ACTUATOR SIZING: 180 psi

POSITIONER REQUIRED: No

ITEM NO.: LY-2

Control valve shall be furnished with an electro-pneumatic transducer Fisher Model No. 546, direct acting, 4-20 MA. input, 3-15 psig output, 20 psig air supply pressure, electrical enclosure, Class 1, Division 2, Group D.

Transducer and filter pressure regulator Fisher Model No. 67FR shall be assembled on valve prior to delivery. Copper, copperbearing materials and Viton are not permitted in all items wetted with ammonia.

1 MWe Heat Exchanger for OTEC

SIZE	FSCM NO.	Phase Separator Drain Line CONTROL VALVE SPEC. LCV-1	REV.
A	11982		
SCALE		SHEET	18

HAND VALVE

ITEM NO.: V-14

MANUFACTURER: Fisher Model No. 1075-8500 or equal

LINE NO.: 171-A-4

FLUID: Liquid Ammonia

NOMINAL PRESSURE: 119.6 psig

NOMINAL TEMPERATURE: 72°F

SPECIFIC GRAVITY: 0.607

VALVE BODY SIZE: 4 in.

TYPE OF BODY: Butterfly

BODY MATERIAL: Carbon Steel

END CONNECTION: 300 LB. Flangeless

TRIM MATERIAL: 316 s.s.

SEAL MATERIAL: TFE

ACTUATOR TYPE: Manual

1 MWe Heat Exchanger for OTEC

SIZE	FSCM NO.	Phase Separator Drain Line	REV.
A	11982	HAND VALVE V-14	
SCALE		SHEET 19	

PREPARED P. O. + w

11/12/79

REPORT NO.

PAGE 5

CHECKED MHS

4/15/79

MODEL

CONTROL VALVE CALCULATION

REF. A.V.O. FROM: R.A. WISE TO P.R. EDRIS, 9/24/79, IMN PHASE SEPARATOR LIQUID RETURN LINE FLOW RATE

PRESSURE DROP AVAILABLE FOR CONTROL VALVE

$$K_T = 419.03 \quad K' \text{ AT VALVE} = 321.88$$

$$K = K_T - K' = 419.03 - 321.88 = 97.15$$

$$H = K Q^2 \quad Q = 42 \text{ GPM} = \frac{42 \text{ GAL}}{\text{MIN}} \times \frac{\text{MIN}}{60 \text{ SEC}} \times \frac{\text{FT}^3}{7.48 \text{ GAL}} =$$

$$H = 97.15 \times .0935^2 \quad Q = .0935 \text{ FT}^3/\text{SEC}$$

$$H = 0.8508 \text{ FT. (HEAD LOSS THRU THE SYSTEM WITHOUT C.V.)}$$

$$\text{TOTAL } H = 9 \text{ FT}$$

$$H_{\text{FOR C.V.}} = 9 - 0.8508 = 8.15 \text{ FT}$$

$$P = \frac{H \times C}{144} = \frac{8.15 \times 38}{144} = 2.15 \text{ LB/IN}^2$$

CONTROL VALVE CALCULATION (SEE FLASHING CALCULATION)

$$T = 72^\circ \text{F}$$

$$P_1 = 120.85 \text{ PSIG}$$

$$P_2 = 119.85 \text{ PSIG}$$

$$SG = \frac{37.9}{62.39} = .607$$

$$\Delta P = 1 \text{ PSI}$$

$$CV = Q \sqrt{\frac{SG}{\Delta P}} = 42 \left(\frac{.607}{1} \right)^{1/2} = 32.74$$

$$CV = 32.74$$

VALVE: FISHER MODEL ED, 1 1/2" BODY. LINEAR.

PREPARED

Pablo Otero

11/12/79

REPORT NO.

PAGE 16

CHECKED

MHS

11/15/79

MODEL

FLASHING CALCULATION

REF. FISHER CATALOG NO. 10 PAGE 2-11

$$\Delta P_{allow} = K_M (P_1 - Y_C P_V)$$

ΔP_{allow} = MAXIMUM ALLOWABLE DIFFERENTIAL
PRESSURE FOR SIZING PURPOSES, PSI

$$P_1 = \text{BODY INLET PRESSURE, PSIA} = 135.55$$

$$P_V = \text{VAPOR PRESSURE OF LIQUID AT BODY
INLET TEMPERATURE, PSIA} = 133.4$$

$$K_M = \text{VALVE RECOVERY COEFFICIENT} = .68 \text{ FOR ED, 1 1/2 IN. VALVE}$$

Y_C = CRITICAL RATIO PRESSURE

VAPOR PRESSURE - PSIA

CRITICAL PRESSURE - PSIA

$$\text{VAPOR PRESSURE AT } 72^\circ\text{F} = 133.4 \text{ PSIA}$$

$$\text{CRITICAL PRESSURE OF NH}_3 = 1634 \text{ PSIA}$$

$$\frac{\text{VAPOR PRESSURE}}{\text{CRITICAL PRESSURE}} = \frac{133.4}{1634 \text{ PSIA}} = 0.08$$

SEE FIGURE 2 CRITICAL PRESSURE RATIOS OF LIQUIDS OTHER THAN WATER

$$Y_C = 0.870$$

$$\Delta P_{allow} = 0.68 (135.55 - 0.87 (133.4))$$

$$\Delta P_{allow} = 13.25 \text{ PSI}$$

$$\Delta P = \text{TOTAL HEAD AVAILABLE FOR CONTROL VALVE} = 2.15 \text{ LB/IN}^2$$

IF ΔP IS LESS THAN ΔP_{allow} , THEN THE ΔP SHOULD BE USED IN THE
LIQUID SIZING EQUATION.

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK - REDONDO BEACH - CALIFORNIA 90278

INTEROFFICE CORRESPONDENCE

79.6853.1-030

TO: J.E. Snyder

CC: A.H. Hausrath
M.H. Seidman

DATE: 8 October 1979

SUBJECT: External Pressure Tests for 1 MeW
OTEC Heat Exchanger Tubes

FROM: T.R. Baumann
BLDG. 81
MAIL STA. 1513

TAB
EXT. 62635

INTRODUCTION

This memorandum summarizes the results of structural test of the 1 MeW Heat Exchanger Tubes conducted on 27 April 1979 and previously reported in Reference 1.

SUMMARY

Four round titanium tube specimens for the OTEC 1 MeW Heat Exchanger were tested to determine their ability to withstand external pressure. The results of these tests are shown in Figure 1. All specimens achieved pressures above the theoretical limits prior to collapse. This is consistent with data obtained in previous tests on round titanium tubes (References 2 through 5).

TUBE TESTS

Tube Tests were conducted on four titanium tube specimens which are shown in Figures 2a through 2d. The tubes were approximately 48 inches in length and one inch in diameter. Tubes A-1 and A-3 were externally coated, over the entire surface, with Lindy enhanced tubing coating. Tubes A-2 and A-4 each had a small section which was not coated. Aluminum sleeves were cemented to each tube end to provide a smooth surface for a pressure seal (O-Ring). Details of the test fixture and test procedure can be found in References 2 through 4. All tubes achieved pressures which were much higher than ASME design requirements and were above the curves derived from previous tests on steel tube sections (adjusted to account for difference in material). All tests were also above the theoretical limits for perfectly round tubes. Previous tests by TRW on round titanium tubes have also shown this characteristic. Aluminum tubes tested by TRW have failed below the theoretical round tube limit.

TRB:p

REFERENCES

- 1) TRW IOC 79.6853.3-013, "Quick Look Test Report for 1 Mw Heat Exchanger Tube Structural Tests (Test Series A)
- 2) TRW IOC 78.6853.3-18, "Test Report for Toshiba Area Enhanced Titanium Tubes Under External Pressure," 6 December 1978, T.R. Baumann, TRW Systems Group
- 3) TRW IOC 78.6853.3-015, "OTEC Aluminum Tube Structural Tests", 27 Nov 1978, T.R. Baumann, TRW Systems Group
- 4) TRW IOC 78.6853.3-008, "Test Report for OTEC Heat Exchanger Tube Tests", 1 June 1978, T.R. Baumann, TRW Systems Group
- 5) TRW IOC 78.6853.3-007, "OTEC Heat Exchanger Tube Structural Trade-Off and Analysis," 30 May 1978, T.R. Baumann, TRW Systems Group

FIGURE 1
1 MeW HEAT EXCHANGER TUBE TESTS

LEGEND

- ROUND TUBE THEORY
- - - STEEL TUBE TESTS
- - - OUT OF ROUND (1 PERCENT)
- ASME CODE TEST DATA

NOTE: CURVES ARE FOR 1 INCH DIAMETER (O.D.) TUBES

Specimen Number	Collapse Pressure
Specimen A-1 (QC-803)	680 PSI
Specimen A-2 (QC-804)	770 PSI
Specimen A-3 (QC-805)	970 PSI
Specimen A-4 (QC-806)	820 PSI
Average Value	810 PSI

FIGURE 2a

TEST SPECIMENS (BEFORE TESTING)

FIGURE 2b

TEST SPECIMENS (AFTER TESTING)

FIGURE 2c
TEST SPECIMENS (TOP ENDS)

FIGURE 2d
TEST SPECIMENS (BOTTOM ENDS)

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK • REDONDO BEACH • CALIFORNIA 90278

INTEROFFICE CORRESPONDENCE

79.6852.4.5-47

TO: J. E. Snyder III

CC: M. H. Seidman

DATE: 19 July 1979

SUBJECT: Mid-Plane Feed of Enhanced
Tubes - 1MWe Evaporator

FROM: A. M. Sprouse
BLDG. 81 MAIL STA. 1578 EXT. 52130

Attached are calculations of velocities and pressure losses in the mid-plane spray headers at varying flow rates. These calculations indicate that the enhanced tube half-bundle can be fed at reflux rates of up to 2:1 using the mid-plane spray headers. Flow distribution is acceptable for all rates, but at 2:1 the flow velocity is at or near the operable limit. Higher reflux rates can be achieved by using the upper spray systems.

Approved:

P. T. Fukunaga

AMS:cm

PREPARED A.M. SPROUSE

REPORT NO.

PAGE 1

CHECKED MH-S.

7/19/79

FEED ENHANCED TUBES IN
SPRAY MODE

MODEL 1 MW HX's for OTEC

OBJECTIVE: FIND SUITABLE FEED RATES AND ESTIMATE PUMPING POWER REQUIRED FOR MID-PLANE FEED OF ENHANCED TUBES

APPROACH: CALCULATE LINE LOSSES FOR SEVERAL REFLUX RATES. COMPARE WITH AVAILABLE PUMP POWER.

SYSTEM DESCRIPTION:

MAIN HEADERS: 2 @ 10.5' LONG, 4" X-STD, ID = 3.826"

SPRAY TUBES: 28 @ 9.5' LONG, 1 1/2" STD, ID = 1.61"

NOZZLES: 210 - 3/8" HH20W, ID = 0.35"

TOTAL FLOW: 171,100 lb/hr LIQ NH₃ @ 72°F @ NO REFLUX

CALCULATIONS:

I. AT 1:1 REFLUX, TOTAL FLOW = 342,200 lb/hr

4" HEADER, $\dot{m} = 171,100 \text{ lb/hr} = 1.25 \text{ ft}^3/\text{sec}$

$$V = 1.25 / 0.080 = 15.63 \text{ fps}$$

$$Re = \frac{38 \cdot 15.63 \cdot 3.826 \cdot 3600}{12 \cdot 0.355} = 1,920,350$$

$$\epsilon \sim 0.01$$

$$\epsilon/D = 0.031$$

$$4f = 0.057$$

$$H = 0.057 \left(\frac{15.63}{64.4} \right)^2 \left(\frac{10.5 \cdot 12}{3.826} \right) = 7.11 \text{ ft} = 1.88 \text{ psi}$$

1 1/2" HEADER, $\dot{m} = 12,221 \text{ lb/hr} = 0.089 \text{ ft}^3/\text{sec}$

$$V = 0.089 / 0.014 = 6.38 \text{ fps}$$

$$Re = \frac{38 \cdot 6.38 \cdot 1.61 \cdot 3600}{12 \cdot 0.355} = 329,855$$

$$\epsilon \sim 0.01$$

$$\epsilon/D = 0.074$$

$$4f = 0.085$$

$$H = 0.085 \left(\frac{6.38}{64.4} \right)^2 \left(\frac{9.5 \cdot 12}{1.61} \right) = 3.80 \text{ ft} = 1.01 \text{ psi}$$

MAXIMUM TOTAL HEADER LOSS = 1.88 + 1.01 = 2.89 psi

PREPARED A.M. SPROUSE

REPORT NO.

PAGE 2

CHECKED MHS

7/19/79

MODEL 1 MW HX's for OTEC

FEED ENHANCED TUBES IN
SPRAY MODE

$$\begin{aligned}\frac{3}{8}" \text{ NOZZLE} : \dot{m} &= 1629.5 \text{ lb/hr} = 0.0119 \text{ ft}^3/\text{sec} \\ V &= 0.0119 / 0.00067 = 17.8 \text{ fps} \\ H &= (1.0) \frac{(17.8)^2}{64.4} = 4.91 \text{ ft} = 1.29 \text{ psi}\end{aligned}$$

$$\text{TOTAL LOSSES} : 1.09 + 2.89 = 4.18 \text{ psi}$$

$$\begin{aligned}\text{NOZZLE LOSS, CAPACITY IS } 4.74 \text{ GPM NH}_3 @ \Delta p &= 40 \text{ psid} \\ \dot{m} &= 1629.5 \text{ lb/hr} = 5.35 \text{ GPM} \\ \Delta p &= 40 (5.35 / 4.74)^2 \\ &= 50.96 \text{ psid}\end{aligned}$$

FLOW DISTRIBUTION:

$$\begin{aligned}\Delta p_{\text{inlet}} &\leq 0.1 \Delta p_{\text{nozzle}} \\ 4.2 &\leq 0.1 (50.96) = 5.1 \text{ psi} \quad \text{OK}\end{aligned}$$

$$KE_{\text{inlet}} \leq 0.1 \Delta p_{\text{nozzle}} = 5.1 \text{ psi}$$

$$KE_{\text{inlet}} = \frac{1.1 (6.38)^2}{64.4} = 0.695 \text{ ft} = 0.18 \text{ psi} \leq 5.1 \quad \text{OK}$$

DISTRIBUTION APPEARS TO BE O.K.

II. @ NO REFLUX, FLOW IS 171,100 lb/hr

$$\begin{aligned}4" \text{ HEADER, } \dot{m} &= 85,550 \text{ lb/hr} = 0.625 \text{ ft}^3/\text{sec} \\ V &= 0.625 / 0.08 = 7.81 \text{ fps}\end{aligned}$$

$$Re = \frac{38 \cdot 7.81 \cdot 3.826 \cdot 3600}{12 \cdot 0.355} = 959,561$$

$$\epsilon \approx 0.01$$

$$\epsilon/D = 0.031$$

$$4f = 0.057$$

$$H = 0.057 \frac{(7.81)^2}{64.4} \left(\frac{10.5 \cdot 12}{3.826} \right) = 1.77 \text{ ft} = 0.46 \text{ psi}$$

$$\begin{aligned}1\frac{1}{2}" \text{ HEADER, } \dot{m} &= 6111 \text{ lb/hr} = 0.045 \text{ ft}^3/\text{sec} \\ V &= 0.045 / 0.014 = 3.19 \text{ fps}\end{aligned}$$

$$Re = \frac{38 \cdot 3.19 \cdot 1.61 \cdot 3600}{12 \cdot 0.355} = 164,927$$

MAY 16 1979

PREPARED A.M. SPROUSE mg
CHECKED MHS 7/19/79
MODEL 1 MW_e HV's for OTEC

REPORT NO.

PAGE 3

FEED ENTRANCED TUBES IN
SPRAY MODE

$$\epsilon \approx 0.01, \epsilon/D = 0.074, 4f = 0.085$$

$$H = 0.085 \frac{(3.19)^2}{64.4} \left(\frac{9.5 \cdot 12}{1.61} \right) = 0.96 \text{ ft} = 0.251 \text{ psi}$$

$$\text{MAX. HEADER LOSS} = 0.46 + 0.25 = 0.71 \text{ psi}$$

$$\frac{3}{8}'' \text{ NOZZLE, } m = 814.7 \text{ lb/hr} = 0.00596 \text{ ft}^3/\text{sec}$$

$$V = 0.00596 / 0.00067 = 8.9 \text{ fps}$$

$$H = 1.0 \frac{(8.9)^2}{64.4} = 1.23 \text{ ft} = 0.32 \text{ psi}$$

$$\text{TOTAL INLET LOSSES: } 0.71 + 0.32 = 1.03 \text{ psi}$$

$$\text{NOZZLE LOSS, } m = 814.7 \text{ lb/hr} = 2.675 \text{ GPM}$$

$$\Delta p = 10 \left(\frac{2.675}{2.56} \right)^2 = 10.9 \text{ psid}$$

FLOW DISTRIBUTION

$$\Delta p_{\text{inlet}} \stackrel{?}{\leq} 0.1 \Delta p_{\text{nozzle}}$$

$$1.03 \leq 0.1 (10.9) = 1.09 \quad \text{OK}$$

$$KE_{\text{inlet}} = \frac{1.1 (3.19)^2}{64.4} = 0.174 \text{ ft} = 0.046 \text{ psi}$$

$$KE_{\text{inlet}} \stackrel{?}{\leq} 0.1 \Delta p_{\text{nozzle}}$$

$$0.046 \leq 1.09 \quad \text{OK}$$

FLOW WILL BE ACCEPTABLE

III AT 2:1 REFLUX, TOTAL FLOW = 513,300 lb/hr

$$4'' \text{ HEADER, } m = 256,650 \text{ lb/hr} = 1.876 \text{ ft}^3/\text{sec}$$

$$V = 1.876 / 0.08 = 23.45 \text{ fps}$$

Wow!

$$Re = \frac{38 \cdot 23.45 \cdot 3.826 \cdot 3600}{12 \cdot 0.355} = 2,881,288$$

$$\epsilon \approx 0.01, \epsilon/D = 0.031, 4f = 0.057$$

$$H = 0.057 \frac{(23.45)^2}{64.4} \left(\frac{10.5 \cdot 12}{3.826} \right) = 16.0 \text{ ft} = 4.23 \text{ psi}$$

$$1\frac{1}{2}'' \text{ HEADER, } m = 18,332 \text{ lb/hr} = 0.134 \text{ ft}^3/\text{sec}$$

$$V = 0.134 / 0.014 = 9.57 \text{ fps}$$

$$Re = \frac{38 \cdot 9.57 \cdot 1.61 \cdot 3600}{12 \cdot 0.355} = 494,782$$

MAY 17 1979

PREPARED A.M. SPEOUSE

REPORT NO.

PAGE 4

CHECKED MHS

7/19/79

FEED ENHANCED TUBES IN
SPRAY MODE

MODEL 1 MWL HK's for OTEC

$$\epsilon \approx 0.01, \epsilon/D = 0.074, 4f = 0.085$$

$$H = 0.085 \frac{(9.57)^2}{64.4} \left(\frac{9.5 \cdot 12}{1.61} \right) = 8.56 \text{ ft} = 2.26 \text{ psi}$$

$$\text{MAXIMUM HEADER LOSS} = 4.23 + 2.26 = 6.49 \text{ psi}$$

$$\frac{3}{8}'' \text{ NOZZLE; } \dot{m} = 2444.3 \text{ lb/hr} = 0.0179 \text{ ft}^3/\text{sec}$$

$$V = 0.0179 / 0.00067 = 26.67 \text{ fps}$$

$$H = 1.0 \frac{(26.67)^2}{64.4} = 11.04 \text{ ft} = 2.91 \text{ psi}$$

$$\text{TOTAL INLET LOSSES: } 6.49 + 2.91 = 9.4 \text{ psi}$$

$$\text{NOZZLE LOSS, } \dot{m} = 2444.3 \text{ lb/hr} = 8.027 \text{ GPM}$$

$$\Delta p = 80 (8.027/6.4)^2 = 125.8 \text{ psi}$$

FLOW DISTRIBUTION

$$\Delta p_{\text{inlet}} \leq 0.1 \Delta p_{\text{nozzle}}$$

$$9.4 \leq 0.1 (125.8) = 12.6 \quad \text{OK}$$

$$\text{K.E. inlet} = 1.1 \frac{(9.57)^2}{64.4} = 1.564 \text{ ft} = 0.412 \text{ psi}$$

$$\text{KE inlet} \leq 0.1 \Delta p_{\text{nozzle}}$$

$$0.412 \leq 12.6 \quad \text{OK}$$

PROBABLY OK: FLOW DISTRIBUTION IS ACCEPTABLE, BUT
THE FLUID VELOCITY MAY BE TOO HIGH FOR PRACTICAL
OPERATION

SUMMARY:

CASE	Δp_{inlet} HEADERS	Δp_{inlet} NOZZLES	Δp NOZZLES	TOTAL Δp	$\dot{m}_a$, lb/hr	V_{max} , fps
NO REFLUX	.71	.32	10.9	11.93	171,100	7.81
1:1	2.89	1.29	50.96	55.14	342,200	15.63
2:1	6.49	2.91	125.8	135.2	513,300	23.45

PREPARED A. M. SPROUSE

REPORT NO.

PAGE 1

CHECKED MHS

7/14/79

MID-PLANE FEED PRESSURE LOSSES

PROJECT 1MW_e HX's

1. PROBLEM: 3 CONFIGURATIONS HAVE BEEN PROPOSED FOR ATTACHMENT OF NOZZLES TO THE MID-PLANE FEED SPRAY TUBES.

SCHEME 1 - NOZZLE WELDED IN PLACE. GIVES BEST COVERAGE (6.6") BUT NOZZLE NOT REMOVABLE. SLIGHT DISCONTINUITY ON INSIDE OF PIPE

SCHEME 2 - NOZZLE SCREWED INTO FITTING WELDED INTO TUBE. COVERAGE SLIGHTLY LESS THAN IN SCHEME 1 (6.25"); NOZZLE REMOVABLE, BUT MAJOR DISRUPTION OF FLOW INSIDE OF PIPE

SCHEME 3 - NOZZLE SCREWED INTO FITTING WELDED TO OUTSIDE OF PIPE. SMOOTH TRANSITION FOR FLOW AND NOZZLE REMOVABLE, BUT COVERAGE ONLY 5.4"

REMOVABLE NOZZLE WITH GREATEST COVERAGE PREFERRED. THIS IS SCHEME 2. IS FLOW DISRUPTION TOLERABLE?

2. APPROACH: ESTIMATE PRESSURE LOSS IN A SCHEME 2 SYSTEM AND COMPARE WITH DROP ACROSS NOZZLE.

3. SYSTEM DESCRIPTION:

MAIN HEADERS: 2 - 10.5' LONG, 4" X-STG., ID = 3.826"

SPRAY TUBES: 28 - 9.5' LONG, 1 1/2" STD., ID = 1.61"

NOZZLES: 210 - 3/8" HH20W, ID ≈ 0.35"

TOTAL FLOW: 128,000 lb/hr H₂O. NH₃ @ 72°F

4. MAXIMUM HEADER Δp

$$\text{IN 4" HEADER, } \dot{m} = \frac{128,000}{2} = 64,000 \frac{\text{lb}}{\text{hr}} = 0.468 \frac{\text{ft}^3}{\text{sec}}$$

$$A_{4"} = \frac{\pi (3.826)^2}{4 \cdot 144} = 0.080 \text{ ft}^2$$

PREPARED A. N. SPROUSE

REPORT NO.

PAGE 2

 CHECKED MHS

7/25/79

 PROJECT 1 NIW6 HX'S

MID-PLANE FEED PRESSURE LOSSES

$$V = 0.468 / 0.080 = 5.85 \text{ ft/sec}$$

$$Re = \frac{\rho V d}{\mu} = \frac{38 \cdot 5.85 \cdot 3.826 \cdot 3600}{12 \cdot 0.355} = 719,000$$

$$E = 0.01' * \text{AVERAGE VALUE FOR RIVETED PIPE (ASSUMES RIVETED PIPE HAS SIMILAR FLOW DISTURBANCES)}$$

$$E/D = 0.031$$

$$4f^* = 0.057$$

$$H = 4f \frac{V^2}{2g} \left(\frac{L}{D} \right) = 0.057 \left(\frac{5.86^2}{64.4} \right) \left(\frac{10.5 \cdot 12}{3.826} \right)$$

$$H = 1.00 \text{ ft} = 0.264 \text{ psi}$$

$$\text{IN } 1\frac{1}{2}'' \text{ HEADER, } \dot{m} = \frac{128,000}{28} = 4571 \frac{\text{lb}}{\text{hr}} = 0.033 \text{ ft}^3/\text{sec}$$

$$A_{1\frac{1}{2}} = \frac{\pi (1.61)^2}{4 \cdot 144} = 0.014 \text{ ft}^2$$

$$V = 0.033 / 0.014 = 2.36 \text{ ft/sec}$$

$$Re = \frac{\rho V d}{\mu} = \frac{38 \cdot 2.36 \cdot 1.61 \cdot 3600}{12 \cdot 0.355} = 122,000$$

$$E/D = (0.01)(12) / 1.61 = 0.074$$

$$4f = 0.085$$

$$H = 4f \frac{V^2}{2g} \left(\frac{L}{D} \right) = 0.085 \left(\frac{2.36^2}{64.4} \right) \left(\frac{9.5 \cdot 12}{1.61} \right)$$

$$H = 0.52 \text{ ft} = 0.137 \text{ psi}$$

* HANSEN, FLUID MECHANICS, WILEY, 1967 p. 420ff

PREPARED A.M. SPROUSE

REPORT NO.

PAGE 2A

CHECKED

PROJECT 1MW₂ HK's

MID PLANE FEED PRESSURE LOSSES

LOSS IN TRANSITION FROM 4" TO 1½" HEADER

APPROXIMATE AS SUDDEN CONTRACTION COMBINED
WITH 1½" 45° ELL.

$$D_1/D_2 = 3.826 / 1.61 = 2.38$$

$$K = 0.4 \quad \text{REDUCTION}$$

$$K = 0.2 \quad \text{45° ELL}$$

$$H = \frac{(.4 + .2) (2.36 \text{ fps})^2}{2 \cdot 32.2} = 0.052 \text{ ft} = 0.014 \text{ psi}$$

$$\begin{aligned} \text{MAXIMUM TOTAL HEADER LOSS} &= 0.26 + 0.14 + 0.01 \\ &= 0.41 \text{ psi} \end{aligned}$$

PREPARED A.M. SPROUSE

REPORT NO.

PAGE 3

 CHECKED MHS

 PROJECT 1 MW_e HX's

MID-PLANE FEED PRESSURE LOSSES

5. RELATIVE NOZZLE LOSSES:

 GOULDS PUMP MANUAL (3rd Ed., p. 424) GIVES
RESISTANCE COEFFICIENTS FOR VARIOUS TYPES OF INLETS
AS FOLLOWS:

 THESE CONFIGURATIONS CORRESPOND ROUGHLY TO THE
NOZZLE ATTACHMENT SCHEMES AS INDICATED.

$$\text{AT NOZZLE INLET, } \dot{m} = \frac{128,000}{210} = 609.5 \frac{\text{lb}}{\text{hr}} = 0.0045 \frac{\text{ft}^3}{\text{sec}}$$

$$A_{\text{inlet}} = \frac{\pi (.35)^2}{4 \cdot 144} = 0.00067 \text{ ft}^2$$

$$V = \frac{.0045}{.00067} = 6.72 \text{ ft/sec}$$

$$H = \frac{K V^2}{2g} = K \left(\frac{6.72^2}{64.4} \right) = 0.7 K$$

$$\text{SCHEME 1: } 0.35 \text{ ft} = 0.09 \text{ psi}$$

$$\text{SCHEME 2: } 0.7 \text{ ft} = 0.18 \text{ psi}$$

$$\text{SCHEME 3: } 0.035 \text{ ft} = 0.009 \text{ psi}$$

6. TOTAL, HEADER + INLET LOSSES, SCHEME 2

$$\Delta p = 0.71 + 0.18 = \underline{\underline{0.59 \text{ psi}}}$$

PREPARED A.M. SPROUSE

REPORT NO.

PAGE 4

CHECKED MHS

7/29/79

MID-PLANE FEED PRESSURE LOSSES

PROJECT 1MW₆ HK'S

7. COMPARISON OF (HEADER + INLET LOSSES) WITH DROP ACROSS NOZZLE ORIFICE

EACH NOZZLE HAS A CAPACITY OF 1.5 GPM H₂O AT $\Delta p = 5$ psid *

CAPACITY CORRECTION FACTOR = $\frac{1}{\sqrt{\text{Sp.Gr.}}} = 1.28$ NH₃
 $\therefore$ NOZZLE CAPACITY = 1.92 GPM NH₃ @ 5 psid

THROUGHPUT, $\dot{m} = 609.5$ lb/hr = 2.0 GPM

$$\text{GPM} \propto \Delta p^{\frac{1}{2}}$$

$$\Delta p = 5 \left(\frac{2.0}{1.92} \right)^2 = \underline{\underline{5.42 \text{ psid}}}$$

FOR GOOD FLOW DISTRIBUTION,

$$(K.E.)_{\text{inlet}} \leq 0.1 \Delta p \quad \text{AND}$$

$$(\Delta p)_{\text{inlet}} \leq 0.1 \Delta p$$

FOR THE SECOND CRITERION: $0.59 \leq 0.1 (5.4)$ CLOSE

FOR THE FIRST CRITERION: $K.E. = \frac{\alpha V_a^2}{2g}$

$$\alpha \equiv \text{AVG. VELOCITY CORRXN} = 1.1, \text{ TURBULENT FLOW}$$

$$K.E. = \frac{1.1 (2.36)^2 (0.61)}{2 \cdot 32.2 \cdot 2.31} = 0.025 \leq 0.54 \quad \underline{\underline{OK}}$$

8. $\therefore$ AT MAXIMUM LOSS, SCHEME 2 WILL PROBABLY STILL HAVE A GOOD FLOW DISTRIBUTION.

* SPRAYING SYSTEMS CO. INDUSTRIAL CATALOG 27, p. 18
PERRY, CHEMICAL ENGINEERS' HANDBOOK (4th Ed.)
p. 5-45

The information disclosed here is proprietary rights of Southwestern Engineering. Neither this document nor the information disclosed herein is reproduced or transferred to other documents or used or disclosed to others for manufacturing purposes, or for any other purposes, except as specifically authorized in writing by Southwestern Engineering.

SCHEME 3

SCHEME 2

SCHEME 1

JOB NO.
6043

PROPOSED ARRG T FOR
MID- PLANE SPRAY NOZZLES

DRWG
SIZE
REV
A 1018

DATE
3.15.79
BY
ALL

CHK
APP

B-60

PREPARED A. M. SPROUSE

REPORT NO.

PAGE 1

CHECKED

ENHANCED SURFACE PERFORMANCE

MODEL

1 MW HX's for OTEC

OBJECTIVE: CALCULATE EVAPORATIVE MASS RATE FROM HIGH FLUX BUNDLE.

BACKGROUND: DATA WAS TAKEN BY LINDE ON BOTH SMOOTH AND HIGH FLUX TUBES IN BOTH THIN-FILM AND POOL-BOILING EVAPORATION. THIS DATA IS PLOTTED, SEE FIGURE 1. IN ADDITION, SYSTEM PERFORMANCE BASED ON FIXED PARAMETERS SUCH AS AREA, FOULING FACTOR, INLET TEMPERATURES, ETC., HAS BEEN PLOTTED AS A FUNCTION OF h_a AND Q/A , FIGURE 2. INTERSECTION OF THE DATA (HERE PLOTTED AS h_a VS Q/A , WHERE $h_a = Q/A/\Delta T$) GIVES THE OPERATING POINTS.

OPERATING POINTS:

		h_a	Q/A	U	$\dot{m}_a$
HIGH FLUX	POOL-BOILING	5450	2994	556	173,536
"	" THIN FILM	4430	2950	544	171,096
SMOOTH	"	880	2230	364	129,337
"	POOL-BOILING	540	1865	289	108,097

WHERE $U = (0.001614 + h_a)^{-1}$ AND

$$\dot{m}_a = \frac{Q/A \cdot A}{533.72}, \quad A = 30,935 \text{ HIGH FLUX } \frac{1}{2} \text{ BUNDLE}$$

$$= 30,955 \text{ SMOOTH } \frac{1}{2} \text{ BUNDLE}$$

PERFORMANCE:

POOL-BOILING : $\dot{m}_a = 173,536 + 108,097 = 281,633 \text{ } \frac{1}{2} \text{ hr}$

SPRAYED BUNDLE : $\dot{m}_a = 171,096 + 129,337 = 300,433 \text{ } \frac{1}{2} \text{ hr}$

RELIEF REQMTS:

USGS CRITERIA ARE INSENSITIVE TO VAPOR RATE
USING CRITERIA IN S.O.W. : 10% OF CYCLE FLOW

$$A = \frac{\dot{m}_a (460 + T)^{\frac{1}{2}}}{CK (P)(M)^{\frac{1}{2}}} = \frac{30,400 (460 + 103.5)^{\frac{1}{2}}}{.340 (209 + 14.7)(17)^{\frac{1}{2}}}$$

$$A = 2.30 \text{ SQ. IN.}$$

ORIFICE ON ORDER IS SIZE $L \leftrightarrow 2.85 \text{ SQ. IN.}$

SINGLE TUBE PERFORMANCE DATA

MAY 02 1979

- (1) HIGH FLUX FILM FOLLOWS
- (2) HIGH FLUX THIN FILM
- (3) SMOOTH FILM THIN FILM
- (4) SMOOTH THIN FILM BOILING

APR 27 1979

PREPARED A.M. SPROUSE

REPORT NO.

PAGE 1

CHECKED

MODEL 1MWe HK's FOR OTEC

EVAPORATOR OPERATION
POOL BOILER & ENHANCED TUBES

OBJECTIVE: CONSIDER OPERATION IN VARIOUS MODES

1. SPRAYED BUNDLE
 - a. FULL BUNDLE
 - b. ENHANCED ONLY
 - c. SMOOTH ONLY
2. POOL BOILER
 - a. FULL BUNDLE
 - b. ENHANCED ONLY
 - c. SMOOTH ONLY

FACTORS FOR CONSIDERATION ARE

1. FEED RATE
2. CONDENSER ADJUSTMENT
3. SEPARATION REQMT
4. RELIEVING RATE

DATA: BASED ON CALCULATED HEAT TRANSFER PERFORMANCE (SPROUSE CALCS 5/14/79), THE HALF-BUNDLES WILL PRODUCE AN EVAPORATIVE VOLUME AS FOLLOWS

	POOL-BOILING	SPRAYED BUNDLE
HIGH-FLUX	173,536	171,096
SMOOTH	108,097	129,337
TOTAL	281,633 lb/hr	300,433 lb/hr

1. FEED RATE

SPRAYED BUNDLE: @ NO REFLUX, VOLUMES SUPPLIED WILL HAVE TO BE EQUAL TO THOSE QUOTED ABOVE. HOWEVER, FOR GOOD DISTRIBUTION RATIOS OF AT LEAST 1:1 WILL BE REQUIRED. ANALYSIS HAS SHOWN (SPROUSE CALCS, 5/16/79) THAT IT IS POSSIBLE TO USE THE MID-PLANE FEED SYSTEM TO SUPPLY RATIOS OF 1:1 TO THE HIGH-FLUX TUBES OPERATING ALONE. EITHER PUMP 1 OR 2 WOULD HAVE TO BE PIPED UP FOR THIS PURPOSE. THE UPPER SPRAY SYSTEM HAS BEEN DESIGNED EXPRESSLY TO SUPPLY THE SMOOTH TUBES AT REFLUX RATIOS OF UP TO 9

THE TOTAL BUNDLE CAN BE SUPPLIED AT HIGHER REFLUX RATIOS BECAUSE THE REFLUX FALLING FROM THE TOP SUPPLEMENTS THE MID-PLANE FEED SYSTEM.

@ 2:1 REFLUX —

TOP FEED 387,900 lb/hr
REFLUXED FROM TOP 258,600 lb/hr
MID-PLANE FEED 254,700 lb/hr
TOTAL TO ENHANCED TUBES 513,300 lb/hr
TOTAL FEED 642,600 lb/hr

PREPARED A. M. SPROUSE

REPORT NO.

PAGE 2

CHECKED

 MODEL 1 MW_e HX's FOR OTEC
EVAPORATOR OPERATION
POOL-BOILER & ENHANCED TUBES

@ 4:1 REFLUX —

TOP FEED 646,500 lb/hr
 REFLUXED FROM TOP 517,200 lb/hr
 MID-PLANE FEED 338,300 lb/hr
 TOTAL TO ENHANCED TUBES 855,500 lb/hr
 TOTAL FEED 984,800 lb/hr

POOL BOILER: REQUIRED FEED RATES WILL BE AS GIVEN IN DATA ABOVE. IN ADDITION, THERE WILL BE LOSSES TO DRAIN SYSTEM BASED ON THE SPILLOVER CONTROL OF LIQUID. SINCE SOME SLOSHING WILL OCCUR, SPILLOVER LOSSES OUGHT TO BE AT LEAST 5% OF THE TOTAL SUPPLIED; THUS FEED MUST BE AT LEAST

HIGH FLUX ONLY : 182,700 lb/hr
 SMOOTH ONLY : 113,800 lb/hr
 TOTAL BUNDLE : 296,500 lb/hr

IN FEEDING THE POOL BOILER, THE MAJOR HEAD REQUIREMENT IS THE STATIC HEAD OF LIQUID ABOVE THE PUMP. THIS HEIGHT WILL BE ABOUT 10.5' IN THE VESSEL DURING FULL OPERATION AND ~ 4.5' DURING HALF OPERATION. THE LIQUID LEVEL IN THE DRAIN TANK IS ~ 3.5' BELOW THE EVAPORATOR. PIPING AND VALVE LOSSES BETWEEN 0.35 AND 2.5 psi ARE EXPECTED OVER THE FLOW RANGE. THIS MEANS THE TOTAL HEAD REQUIREMENT IS BETWEEN ~ 2.4 - 6.2 psi (9.3 - 23.5 ft)

2. CONDENSER ADJUSTMENT

THE OPERATING CONDITIONS IN THE CONDENSER HAVE BEEN CALCULATED FOR A RANGE OF POSSIBLE CONDENSING RATES (SEE ATTACHED GRAPH). THE OPERATING TEMPERATURES ARE AS FOLLOWS, IN °F:

	POOL-BOILING	SPRAYED BUNDLE
HI-FLUX	45.2	45.2
SMOOTH	43.3	43.9
TOTAL	48.4	49.0

THESE NUMBERS ARE BASED ON SIMULTANEOUS SOLUTION OF THE EQNS

$$T_{we} = T_{ac} - \frac{(T_{ac} - T_{wi})}{\exp(UA/m_w C_{pw})}$$

$$m_a = \frac{m_w C_{pw} (T_{we} - T_{wi})}{.975 \Delta H_{fg} |_{T_{ac}}}$$

B-65

MAY 18 1979

PREPARED A.M. SPROUSE

REPORT NO.

PAGE 3

CHECKED

MODEL 1 MW₂ HK's for OTEC

EVAPORATOR OPERATION
POOL-BOILER & ENHANCED TUBES

3. SEPARATION RQMT

CURVES FOR Φ -SEPARATOR PERFORMANCE ARE ATTACHED. PERFORMANCE DETERIORATES BEYOND 120% OF NOMINAL FLOW (267,000 LB/HR, 2 Φ FLOW). ASSUMING LIQUID CARRYOVER IS DIRECTLY PROPORTIONAL TO EVAPORATIVE RATE, THE MAXIMUM 2 Φ FLOW IS 267 (300.4/256) = 313,300 LB/HR, OR 117% OF NOMINAL.

⇒ THEREFORE, Φ SEPARATOR CAPACITY IS ADEQUATE
PRESSURE DROP @ 117% IS 14.4 "H₂O = 0.52 psi

⇒ THEREFORE, Δp IS TOLERABLE

4. RELIEVING RATE

CONTRACT REQUIREMENT IS FOR 10% OF CYCLE FLOW. THE MAXIMUM RELIEF RATE WOULD THEN BE 30,043 LB/HR.

$$A = \frac{W}{PCK \sqrt{M/T}} \times = \frac{30,043}{(223.7)(340)(17/563.5)^{1/2}} = \underline{2.274 \text{ SQ. IN.}}$$

COMPARE SELECTED VALVE W/ SIZE L ORIFICE, 2.853 SQ. IN.

⇒ RELIEF VALVE MEETS CONTRACT RQMTS FOR CAPACITY EVEN UNDER MAXIMUM FLOW CONDITIONS

THE U.S. COAST GUARD REGULATIONS (SUB-CHAPTER F, 54.15) ARE BASED ON VESSEL CAPACITY, NOT EVAPORATIVE RATE, SO THE INCREASE IN MASS FLOW RATE HAS NO EFFECT ON COMPLIANCE WITH USCG REGS.

⇒ POOL-BOILER & ENHANCED BUNDLE MEET USCG REG REQUIREMENTS FOR RELIEF.

CONDENSER PERFORMANCE

MAR 21 1978

Figure 1

88-B
SEPARATION EFFICIENCY (%)

Figure 2

PRESSURE DROP (IN. H₂O) $\times 0.036 = \text{psi}$

PREPARED A.M. SPROUSE

REPORT NO.

PAGE 1

CHECKED _____

MODEL _____

HIGH FLUX TUBE POOL BOILING PERFORMANCE SOLUTION

OBJECTIVE: FIND AMMONIA VAPORIZATION RATE FOR
 POOL BOILING CASES

(1) SMOOTH TUBE BUNDLE

(2) HIGH FLUX TUBE BUNDLE

EQUATIONS:

$$Q = UA \Delta T_{em} = \dot{m}_w C_{pw} (T_{wi} - T_{wo}) = \dot{m}_a C_{pa} (T_{ae} - T_{ac}) + \dot{m}_a \Delta H_{fg} |_{T_{ae}}$$

$$\Delta T_{em} = \frac{T_{wi} - T_{wo}}{\ln \left[\frac{T_{wi} - T_{ae}}{T_{wo} - T_{ae}} \right]}$$

$$\frac{1}{U} = \frac{1}{h_{sw}} \left(\frac{A_o}{A_i} \right) + \frac{1}{h_w} \left(\frac{A_o}{A_{av}} \right) + \frac{1}{h_f} \left(\frac{A_o}{A_i} \right) + \frac{1}{h_a}$$

$$\begin{aligned} A &= 63,540 \text{ ft}^2 \\ \dot{m}_w &= 42,211,000 \text{ lb/hr} \\ C_{pw} &= 0.9564 \text{ BTU/lb-}^\circ\text{F} \\ T_{wi} &= 80^\circ\text{F} \\ C_{pa} &= 1.12 \text{ BTU/lb-}^\circ\text{F} \\ T_{ae} &= 72^\circ\text{F} \\ T_{ac} &= 48^\circ\text{F} \end{aligned}$$

$$\begin{aligned} \Delta H_{fg} |_{T_{ae}} &= 506.6 \text{ BTU/lb} \\ Y_{hsw} &= 0.000827 \text{ ft}^2\text{-hr-}^\circ\text{F/BTU} \\ A_o/A_i &= 1.059 \\ Y_{hw} &= 0.000203 \text{ ft}^2\text{-hr-}^\circ\text{F/BTU} \\ A_o/A_{av} &= 1.029 \\ Y_{hf} &= 0.0005 \text{ ft}^2\text{-lb-}^\circ\text{F/BTU} \end{aligned}$$

Q , T_{wo} , $\dot{m}_a$, AND h_a ARE UNDETERMINED.

SOLUTION TECHNIQUE CONSISTS OF SELECTING A T_{wo} AND
 CALCULATING A SET OF SOLUTIONS FOR Q , h_a , AND $\dot{m}_a$.
 PLOTTING THESE SOLUTIONS AGAINST THE LINDE DATA FOR
 SINGLE TUBE POOL-BOILING WILL GIVE AN INTERSECTION
 AT WHICH THE SYSTEM WILL OPERATE.

SOLVING:

$$\frac{Q}{A} = \frac{\dot{m}_w C_{pw} \Delta T}{A} = 635.36 (80 - T_{wo}) \quad \text{Eqn 1}$$

$$U = \frac{\dot{m}_w C_{pw} \Delta T}{A \Delta T} \ln \left[\frac{T_{wi} - T_{ae}}{T_{wo} - T_{ae}} \right] = \left[\sum a_i \frac{1}{h_i} \right]^{-1}$$

$$U = 635.36 \ln \left[\frac{8}{T_{wo} - 72} \right] = \frac{1}{(0.001614 + Y_{ha})} \quad \text{Eqn 2}$$

$$\dot{m}_a = \frac{Q}{C_{pa}(T_{ae} - T_{ac}) + \Delta H_{fg} |_{T_{ae}}} = \frac{Q}{533.48} \quad \text{Eqn 3}$$

JUL 10 1978

PREPARED A.M. SPROUSE

REPORT NO.

PAGE 2

CHECKED _____

MODEL _____

**HIGH FLUX POOL BOILING
PERFORMANCE SOLUTION**

SYSTEM PERFORMANCE : SOLVING EQN'S 1 & 2, PLOT FIG. 1

<u>T_{wa}</u>	<u>h_a</u>	<u>Q/A</u>
75.2	9643	3050
75.5	3450	2859
76	1523	2541
77	576	1906
78	259	1271

10,725	3131
4159	2135
1673	2077
607	1957
227	1175

LINDE SMOOTH TUBE DATA : PLOT, FIG 1

<u>Q/A</u>	<u>ΔT</u>	<u>h_a</u>
1500	3.2	469
2000	3.5	571
3000	3.9	769
4000	4.25	941
5000	4.5	1111

LINDE HIGH FLUX TUBE DATA : (SINGLE TUBE)

<u>Q/A</u>	<u>ΔT</u>	<u>h_a</u>
2500	.465	5376
3000	.555	5405
4000	.72	5556
5000	.88	5682
6000	1.05	5714

SOLUTIONS : USING FIG 2 TO FIND m_a FOR Q/A AT INTERSECTIONS

	<u>Q/A</u>	<u>h_a</u>	<u>m_a, lb/hr</u>
SMOOTH TUBE	1850	540	215,000
HIGH FLUX TUBE	2970	5400	343,000

NOTE: THIS IS AN APPROXIMATE SOLUTION. NO ATTEMPT HAS BEEN MADE TO DEAL WITH SUCH PROBLEMS AS :

- VARIATION IN ΔT ALONG TUBE
- VARIATION IN h_a WITH STATIC HEAD
- HEAT LOSS THROUGH POOL SPILLOVER

PREPARED A.M. SPROUSE

REPORT NO.

PAGE 3

CHECKED _____

MODEL _____

SPRAYED BUNDLE PERFORMANCE

FOR COMPARISON, LINDE DATA ON THIN-FILM EVAPORATION (USED TO APPROXIMATE SPRAYED BUNDLE PERFORMANCE), IS LISTED BELOW:

HIGH FLUX SURFACE:

<u>Q/A</u>	<u>ΔT</u>	<u>h_a</u>
2000	0.52	3846
3000	0.67	4478
4000	0.79	5063
5000	0.90	5556

SMOOTH TUBE: $Q/A = 856 (\Delta T)^{1.0354}$

<u>Q/A</u>	<u>ΔT</u>	<u>h_a</u>
2000	2.27	881
3000	3.36	894
4000	4.43	902
5000	5.50	909

PLOTTING THESE ON FIG. 1 ALSO AND USING FIG 2 TO FIND $\dot{m}_a$, GIVES THESE SOLUTIONS

SOLUTIONS:

	<u>Q/A</u>	<u>h_a</u>	<u>$\dot{m}_a$, lb/hr</u>
HIGH FLUX TUBE	2920	4400	510,000
SMOOTH TUBE	2200	880	256,000

i.e. USING A BUNDLE OF HIGH FLUX TUBES COULD
DOUBLE OUTPUT

FIG 2: NH_3 MASS FLOW AS A FUNCTION OF HEAT FLUX

of
BTU
ha

FIG. 1: SOLUTION OF POOL-BOILING PERFORMANCE

INTEROFFICE CORRESPONDENCE

79.6852.3.4-37

TO: A. M. Sprouse

CC: P. R. Edris

G. M. Kikm

DATE: 18 June 1979

SUBJECT: 1MWe Heat Exchangers Design
Recommendations for Flooded
Bundle Concept.FROM: E. J. DiCamillo
BLDG. 81 MAIL STA. 1578 EXT. 52130

Reference: (1) IOC Number 79.6852.4.5-33, A. M. Sprouse to
P. R. Edris, Subject "Design Requirements of
Pool Boiler Piping, 1 MWe HX's Phase III, Tasks
2.1C and 2.1F", dated 6 June 1979.

In reference to the two tasks cited in the reference memo, transmitted herewith are the procurement specifications and a marked copy of the revisions to be made to the P&ID. In addition, recommendations pertaining to the SECO proposed design for overflow level control and a review of the recirculation capabilities of the existing reflux pumps are included.

RECOMMENDATIONS

Instrumentation (See Figure 2) - It is recommended that two liquid level sensors be installed, one at each end of the evaporator. The information obtained from these sensors will be useful in determining whether one end of the tube bundle is uncovered or what period of oscillation and depths of exposure the tube bundle may experience under different sea states. Temperature, pressure and flow measurement can be accomplished using the baseline instruments.

Since the level sensors must be at opposite ends of the evaporator, additional instrumentation ports are required near the bottom and close to the tube sheet, one at each end. The top side connections may be made by installing tee's on the baseline pressure ports.

Piping and Valves - As shown on the P&ID mark-up, one 6 inch check valve and one 2 inch hand valve are required in the ammonia feed line to the evaporator. The purpose of the check valve is to prevent reverse flow and consequent flooding of the drain tank whenever the pump stops. The shut-off valve is to permit by-passing around the check valve and control of the drain back rate when emptying the evaporator.

The piping will be a continuation of the existing 6 inch carbon steel pipe of the same specification as baseline. It is recommended that a 6 inch flanged tee be installed in the baseline piping at a location convenient to the lower side of the evaporator. This tee should be blind flanged on the flooded bundle outlet. During the flooded bundle testing, the tee outlet to the spray header would be sealed with an inline blind flange (spectacle blind).

Level Control - The design shown on SEC drawing number EM-97407 was reviewed. The scheme is basically sound, but the minor modifications shown in Figure 1 will permit full or half tube bundle testing without a shut down and modification to the tube bundle shrouds and external overflow piping. The addition of two valves at the overflow outlet for the lower bundle test and the incorporation of overflow slots at the lower bundle overflow level will permit testing of the full bundle with the valves closed and the lower half bundle with the valves open. This also eliminates the need for the 6 inch valves shown at the drain tank inlet.

The concept of four 6 inch overflow lines to the upper overflow level is a good one as it will provide adequate venting between the top of the evaporator and the top of the drain tank. This is another reason for leaving the 6 inch lines connected to the higher overflow level, even when testing the lower bundle.

This design makes maximum use of baseline systems including the OTEC-1 condensate return system and requires a minimum of new equipment.

Approved: _____

P. T. Fukunaga
P. T. Fukunaga

ED:df

Attachments

Appendix A
Appendix B
Appendix C

Specification: _____

VALVE
SPECIFICATION SHEET

Equipment List
Item No.: _____

Function: By-Pass Return Valve (Butterfly Type)

Manufacturer: _____

Model and Part No.: _____

Size and Type of End Connections: 2" - 300# ANSI R.F. (Wafer Type)

Maximum Design Pressure (psig): 265
(based on a safety factor of 4 to 1)

System Operating Pressure (psig): 115 (Nominal)

Flow Coefficient: Cv = 55 MIN.
(if known)

Materials:

Body: Carbon Steel

Trim: 316 Stainless Steel

Seat: Teflon

Service: Liquid Anhydrous Ammonia

Type of Operation: Manual

Extras (as applicable):

Cylinder Operating Pressure: N/A

Limit Switches: N/A

Actuating Solenoid Valve Item Number: N/A

Position of Cylinder in Fail Position: N/A

Set Pressure: N/A

Miscellaneous Requirements: Raised face must be concentrically machined. Ball
or gate valves are not permitted. Leak rate shall meet ANSI B16.104-1976,
Class VI. Construction materials must be suitable for use in a sea air
environment or have an approved protective coating. Copper, copper bearing
materials and viton are not permitted on any items wetted with ammonia.

SIZE A	FSCM NO. 11982	REV.
SCALE None	B-77	SHEET

Specification: _____

VALVE
SPECIFICATION SHEET

Equipment List
Item No.: _____

Function: Check Valve

Manufacturer: Daniel Industries or equal

Model and Part No.: Model 1603-AC or equal (Contoured disc)

Size and Type of End Connections: 6" - 300# ANSI R.F. (Wafer type)

Maximum Design Pressure (psig): 265
(based on a safety factor of 4 to 1)

System Operating Pressure (psig): 115

Flow Coefficient: Cv = 1125 MIN.
(if known)

Materials:

Body: Carbon Steel

Trim: 316 Stainless Steel

Seat: Teflon Seal

Service: Liquid Anhydrous Ammonia

Type of Operation: Spring

Extras (as applicable):

Cylinder Operating Pressure: N/A

Limit Switches: N/A

Actuating Solenoid Valve Item Number: N/A

Position of Cylinder in Fail Position: N/A

Set Pressure: N/A

Miscellaneous Requirements: Raised flange must be concentrically machined. Leak
rate shall meet ANSI B16.104-1976, Class VI. Construction materials must
be suitable for use in a sea air environment or have an approved protective
coating. Copper, copper bearing materials and viton are not permitted on
any items wetted with ammonia.

SIZE A	FSCM NO. 11982	REV.
SCALE None	SHEET	

SPECIFICATION SHEET

LEVEL DIFFERENTIAL PRESSURE TRANSMITTER

(2 Required)

Sybron/Taylor Model 3323TD41264-06 or equal
with 316 SST element construction.

Differential Unit

Level Transmitter

Diaphragm Type with remote seals and 35 feet capillary lines

Cast Aluminum Housing

316 SST Diaphragm

Max. Element Working Pressure: 1500 psig

Differential Connection: 1/2 Inch butt weld schedule 40 pipe

Scale: Linear

Measurement Increase Output Increase

Level Range 0 - 156 in., Ammonia, SG .609

Instrument Range 0 - 105 in., H₂O

Electronic Transmitter

Accuracy: $\pm 0.5\%$

Output: 4 - 20 MA

Electrical Connection: 1/2 Inch NPT Electrical Conduit Hub

Power Supply: 24 VDC

Load Impedance: 860 Ohms, Max.

Electrical Classification: Cl. 1, Div. 1, Gr. D

Copper, copper bearing materials and viton are not permitted on any items
wetted with ammonia.

Level Range 0 - 156 In.

Other Sources: Foxboro, Rosemount, Honeywell

SIZE	FSCM NO.			REV.
A	11982	LE/LT - 50 & 51		
SCALE			SHEET	

TRW

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK • REDONDO BEACH, CALIFORNIA

PREPARED SPW

REPORT NO.

PAGE

CHECKED _____

FLOODED BUNDLE

MODEL 1MWc HX15

OVERFLOW LEVEL CONTROL

FIG. 1

B-80

PREPARED

9/15

REPORT NO.

PAGE

CHECKED

MODEL

1 MWc HX'S

FLOODED BUNDLE

LIQUID LEVEL INSTRUMENT REQ'TS

FIG 2

1. PROVIDE LEVEL SENSORS AT BOTH ENDS OF EVAPORATOR (2 REQ'D)
2. SENSOR & TRANSMITTER CAN BE COMBINED INTO A SINGLE DEVICE SUCH AS HONEYWELL'S MODEL DST-4115-0011-11-02-5.
3. PROVIDE SHUT-OFF VALVES ON SENSING LINES AT CONNECTIONS TO EVAPORATOR.
4. LEVEL MEASUREMENT REQ'TS ARE BETWEEN 0 & 156 INCHES (13 FT). OPERATING LEVELS ARE AT APPROX 78 INCHES AND 138 INCHES.
5. TRANSMITTER OUTPUT SHALL BE 4-20 MA FOR THE MEASUREMENT RANGE.

CRITERIA FOR NEW P&ID
FOR POOL BOILING
(See Attached Sketch)

1. Prepare a new P&ID (per A. Sprouse).
2. Use 6850.91-05 for basis but show primarily pool boiling piping and new components, check valve, bypass shut-off valve and instrumentation.
3. Show also reflux loop with pump and instrumentation used for pool boiling.
4. See Figure 1, and Figure 2, and attached sketch.

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK • REDONDO BEACH, CALIFORNIA

PREPARED _____
CHECKED _____
PROJECT _____

REPORT NO.

PAGE

--

APPENDIX-A

CALCULATIONS

PREPARED EJ DICHAVALLO 6/11/79
 CHECKED R.M.K. 6/26/79
 PROJECT 1 MW. HX'S

REPORT NO.

PAGE 1

FLOODED BUNDLE
 (POOL BOILING)

DETERMINE PUMPING RATE FOR EVAPORATOR
 FEED DURING POOL BOILING.

REFERENCE DATA:

MEMO: SPROUSE TO EDRIS, 79.6852.45-33,
 6 JUN 79

HEAT TRANSFER PERFORMANCE MUST BE
 TESTED FOR EACH TUBE BUNDLE & BOTH
 TOGETHER

UPPER BANK TUBE BUNDLE

EVAP. RATE

3153 SMOOTH TUBES - 108,000 lb/HR, Q_1

LOWER BANK TUBE BUNDLE

3151 HI-FLUX TUBES - 174,000 lb/HR, Q_1
 282,000 lb/HR, Q_3

MINIMUM SPILLOVER & LEAKAGE

5% OF FEED RATE

FEED RATES

$$\frac{Q \times 1.05 \frac{\text{lb}}{\text{HR}}}{60 \frac{\text{MIN}}{\text{HR}} \times 5.1 \frac{\text{lb}}{\text{GAL}}} = Q \times 0.00343$$

$$108,000 \times 0.00343 = 370 \text{ GPM}$$

$$174,000 \times 0.00343 = 597 \text{ GPM}$$

$$967 \text{ GPM}$$

PREPARED

hjs

6/12

REPORT NO.

PAGE 2

CHECKED

DMK

6/26/79

PROJECT

1 MWc HX'S

FLOODED BUNDLE
(POOL BOILING)

ASSUMING EACH OPERATIONAL TEST TO BE AT STEADY STATE THE EVAPORATOR FEED RATE REQ'TS. ARE 370, 597, & 967 GPM FOR THE UPPER BUNDLE, LOWER BUNDLE & FULL BUNDLE RESPECTIVELY. THE CONDENSATE RETURN SYSTEM (P-106) MUST BE CAPABLE OF RETURNING 5% LESS THAN THESE VALUES, OR:

$$\frac{370}{1.05} = 352 \text{ GPM (UPPER BUNDLE)}$$

$$\frac{597}{1.05} = 568 \text{ GPM (LOWER BUNDLE)}$$

$$\approx 920 \text{ GPM (BOTH)}$$

THE ABOVE ARE COMPUTED VALUES & IT IS REASONABLE TO ASSUME THAT ACTUAL VALUES COULD BE SIGNIFICANTLY DIFFERENT SINCE THE PURPOSE OF TESTING IS TO MEASURE THE PERFORMANCE.

THERE ARE FOUR "EXISTING" POTENTIAL PUMPS WHICH COULD BE CANDIDATES FOR FEEDING THE AMMONIA TO THE EVAPORATOR THEY ARE THE THREE REFLUX PUMPS, P-1, P-2, & P-3 PROVIDED WITH THE EVAPORATOR AND THE CONDENSATE RETURN PUMP PROVIDED BY THE SIC.

FACTORS BEARING ON THE DECISION ARE:

- 1) THE CONDENSATE FEED PUMP MUST

PREPARED

ELH

6/12

REPORT NO.

PAGE 3

CHECKED

90MK

6/26/79

PROJECT *1M111e HX*

FLOODED BUNDLE
(POOL BOILING)

OPERATE WHILE TESTING.

- 2) THE OVERFLOW AND LEAKAGE MUST BE MADE UP IN THE EVAPORATOR
- 3) NONE OF THE REFLUX PUMPS CAN ACCOMMODATE THE LOW FLOW RATE REQ'D FOR MAKE-UP ALONE
- 4) A RANGE OF FLOW RATES MUST BE ACCOMMODATED
- 5) AN OVERFLOW SCHEME SIMILAR TO THE ONE SHOWN ON SECO DWG EM-97407 WILL BE PROVIDED FOR OVERFLOW CONTROL AND RETURN TO THE DRAIN TANK.

THE FOLLOWING ARE THREE POSSIBLE WAYS TO ACCOMPLISH THIS

- 1) CONNECT RETURN CONDENSATE (P-106) LINE TO POOL BOILER INLET ON EVAPORATOR. ADD A NEW PUMP TO RETURN OVERFLOW AND LEAKAGE TO THE EVAPORATOR (17 TO 50 GPM)
- 2) SAME AS 1) EXCEPT USE ONE OF REFLUX PUMPS AND CIRCULATE A LARGER OVERFLOW QUANTITY
- 3) LET CONDENSATE RETURN TO EVAPORATOR DRAIN TANK AS IT NORMALLY DOES. MODIFY LINE 105-D-6 FOR CONNECTION TO THE POOL BOILER INLET ON THE EVAP. THIS WILL UTILIZE THE MID RANGE

PREPARED

JHS

6/1/79

REPORT NO.

PAGE 4

CHECKED

DMK

6/26/79

PROJECT

3) (CONT'D)

REFLUX PUMP; P-2 WITH A VARIABLE OUTPUT OF 415-1660 GPM TO FEED THE EVAPORATOR. THIS PUMP ALONG WITH THE FLOW CONTROL VALVE IN THE LINE WILL PERMIT FLOW ADJUSTMENT TO THE DESIRED RATE. ALSO THIS LINE IS EQUIPPED WITH FLOW MEASURING DEVICE WHICH IS NECESSARY TO PERMIT SATISFACTORY TEST OPERATION AND PERFORMANCE MEASUREMENT.

THE ECONOMICAL CHOICE IS CLEARLY METHOD 3) AS IT AVOIDS THE DUPLICATION OF EQUIPMENT AND OR PIPING.

PREPARED

J/D

6/14/79

REPORT NO.

PAGE 5

CHECKED

DMK

6/26/79

FLOODED BUNDLE

PROJECT 1 MWe HX'S

COMPUTE C_v FOR 2" BY-PASS VALVE

LIMIT ΔP TO 2 PSI @ 100 GPM

$$C_v = GPM \sqrt{\frac{S.G.}{\Delta P}}, \quad S.G. = .62$$

$$GPM = 100$$

$$\Delta P = 2$$

$$C_v = 100 \sqrt{\frac{.62}{2}}$$

$$C_v = 55 \text{ MIN.}$$

TRW.DEFENSE AND SPACE SYSTEMS GROUP
ONE SPACE PARK • REDONDO BEACH, CALIFORNIA

PREPARED

6/15/79

REPORT NO.

PAGE 6

CHECKED

DML

6/26/79

PROJECT LMW HX'S

FLOODED BUNDLE

DETERMINE MAX NH_3 HOLDUP REQ'D FOR
POOL BOILING

= FULL EVAP SHELL VOL (BET TUBE SHEETS)

- UPPER ULLAGE CORD SEGMENT (z' DEPTH)

- 6304 TUBES (1" OD)

L = LENGTH BETWEEN TUBE SHEETS, 39 FT

D = INSIDE SHELL DIA ≈ 13 FT

$$\text{VOL} = \frac{\pi}{4} D^2 L = .785 \times 169 \times 39 = 5174 \text{ Ft}^3$$

(38,700 GAL)

$$Q = R^2 L \left[\left(\frac{\pi \theta^\circ}{180} \right) - \sin \theta \cos \theta \right] *$$

(CBI, BULL H, TABLES & FORMULAS

B-90

PREPARED

FLS

REPORT NO.

PAGE 7

CHECKED

6MIL

6/28/79

FLOODED BUNDLE

PROJECT

1.2.1.1.2 HX

WHERE

$Q = \text{VOL, Ft}^3 \text{ (FOR CORD SEGMENT)}$

$R = \text{RADIUS CYL, Ft} = 6.42'$

$L = \text{LENGTH, CYL.}$

$d = \text{RADIAL DISTANCE, CORD TO SHELL}$

SEE DIA PREV PAGE

$$\theta = \text{SEE DIA, } \cos \theta = \frac{R-d}{R}$$

$$\theta = \arccos \frac{6.42 - 2}{6.42} = (.688) \quad 46.5^\circ$$

$$Q = 6.42^2 \times 39 \left[\frac{3.14 \times 46.5}{180} - .785 \times .688 \right]$$

$$Q = 502 \text{ Ft}^3$$

TUBE VOL.

$$6304 \times .785 \times \left(\frac{1}{12}\right)^2 \times 39 = 1340 \text{ Ft}^3$$

NH₃ REQ'D IN EVAPORATOR

$$= 5174 - 502 - 1340 = 3332 \text{ Ft}^3 = 24,923 \text{ gal.}$$

FOR TOTAL REQ'D IN LOOP

ASSUME DRAIN TANK 1/2 FULL

" 152 Ft³ IN PIPING (1100 GAL)

PREPARED

ELJ

REPORT NO.

PAGE 8

CHECKED

DMK

6/26/79

FLOODED RUINER

PROJECT

1.1100 LX

DRAIN TANK VOL (54 X 32' L)

$$\frac{1}{2} \left(\frac{\pi}{4} \times 5^2 \times 32 \right) = 314 \text{ Ft}^3 = 2350 \text{ GAL}$$

TOTAL NH₃ REQ'D IN LOOP FOR
POOL BOILING

EVAPORATOR	24,923 GAL
DRAIN TANK (1/2)	2,350
PIPING	1,100
	<hr/> 28,373

PLUS ON-BOARD INVENTORY	6,000
	<hr/> 34,373

35,000 GAL NH₃

REQ'D FOR POOL BOILING

TRW.

ONE SPACE PARK • REDONDO BEACH, CALIFORNIA

PREPARED Gerald Kikin
CHECKED _____
PROJECT _____

REPORT NO. _____

PAGE _____

APPENDIX-B

RECOMMENDATIONS FOR AUGMENTED CIRCULATION
AND
POOL BOILING PUMP ANALYSIS

APPENDIX B

Page B-1

1 MW HEAT EXCHANGER POOL BOILING

RECOMMENDATIONS FOR AUGMENTED CIRCULATION DURING POOL BOILING EXPERIMENTATION.

Reference: McAdams, 3rd Edition, Page 378.

The reference shows that the heat flux rate in boiling water at one atmosphere is significantly affected by pool boiling recirculation.

Similar data does not appear to be available for ammonia. However, it would be prudent to provide for some capability for flow rates higher than that required based upon full vaporization of the nominal ammonia flows of:

Upper half of evaporator	352 gpm
Lower half of evaporator	568 gpm
Full evaporator	920 gpm

Figure 1 shows the head-flow characteristics for variable speed pump 1 and 2 super imposed on the system head flow requirements for the pool boiling pump and piping mode.

Intersection of the pump curve and the system curve gives a head-flow operating point. It appears that either P1 or P2 may be used to provide recirculation flows during the pool boiling mode of operation. The desired flow is achieved by pump speed control and operation of the appropriate flow control valve.

It should be pointed out that Figure 1 only shows the capability for pumps P1 and P2 to deliver various recirculation flows. The return flow characteristic of the liquid ammonia is shown in Figure 2.

Figure 2 shows the liquid NH_3 head required to return the liquid NH_3 to the sump by a gravity head as a function of flow through each of the four 6 inch return lines.

Table 1 enumerates the recirculation ratios attainable using pump 2 for either a stable, level, ocean platform with liquid ammonia overflow through four 6 inch diameter pipes or a rocking platform with liquid ammonia overflow through two 6 inch diameter pipes.

TABLE 1: Pool Boiling Recirculation Ratios

	Stable Platform	Rocking Platform
Full Bundle	2.8	2.8
Lower Bundle	5.1	4.9

Recirculation ratio is defined as the non-evaporated liquid NH_3 flow rate returned from the top of the pool boiling surface to the sump from which it flows to the pump suction.

$$\text{Recirculation ratio} = \frac{w_3}{w_2}$$

It should be noted that any sub-cooling of liquid NH_3 in the sump will in effect provide additional driving head for the recirculation drain lines due to the fact that the pressure in the evaporator is that of saturated NH_3 and the pressure in the sump is lower due to lower saturation pressure of the slightly cooler liquid ammonia in the sump. For example, only 1°F of sub-cooling results in the equivalent of 4.9 feet of addition head.

FIGURE B1

FIG. B2 POOL BOILING LIQUID NH₃ OVERFLOW LINE PRESSURE DROP VS FLOW

LIQUID NH₃ FLOW PER RETURN LINE* 6PM

* THERE ARE FOUR RETURN LINES

B-97

J. K. K.
6/26/79

TRW.

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK • REDONDO BEACH, CALIFORNIA

PREPARED Gerald Kikin

REPORT NO. _____

PAGE 85

CHECKED _____

PROJECT _____

APPENDIX-B

BACKUP ANALYSIS AND CALCULATIONS

PREPARED

J. K. Pen

6/15/77

REPORT NO.

PAGE 86

CHECKED

J. K. Pen

4/26/74

PROJECT

1 MW H.E.

*1 MW H.E. POOL BOILING
- RECIRC. CAPABILITY*

Problem: With existing 1 MW H.E. reflux pyp, what pool boiling recirculation rate can be achieved.

Approach - With Pool Boiling piping and using pyps P1 and/or P2, we need pool boiling evaporation rates of 375-9786 PM. Above and beyond this net flow out of the evaporator, what recirculation flow can be achieved with present pyps and piping.

Need to determine system head flow curve for pool boiling piping. This is plotted against pump head curve which can be varied ~~either~~ with variable speed drive. System head curve is modified with butterfly valve FCV 2.

Superposition of system + pump head flow curve gives operating points.

Start with Pump P2

6" Lin. from pump discharge to 8" inlet nozzle through side of tank

PREPARED D. Kiken

6/15/79

REPORT NO.

PAGE B7

CHECKED 79

6/26

PROJECT

1 MW Pool Boiling
- RECIRC CAP.

1-90° elbow, 1 45° elbow

expansion from 6" pipe to 8" nozzle

pipe elevation 4' 9"

elev. at mid sup. 18' 9"

Butterfly
valve.

FCV-2

$K=2.9 \quad \theta=10^\circ$

+ Check
value.

$$\begin{array}{r} 18'9'' \\ - 4'9'' \\ \hline 14'0'' \end{array} + \frac{39mm}{43mm} \times 10 =$$

$$14 + 9.07 \approx 23.1 \text{ ft.}$$

+ 4.4 Total of bundle.

pump discharge to mid pt. of EVAP.

27.5 ft. to
Top of Bundle.

long radius elbow
45° elbow.

check valve

downward swing
conventional swing

sudden expansion 6" - 8" resistance coeff $K=0.1 \theta$

SPS #26
{ none

K based on vel in small pipe.

$$h_e = K \frac{V^2}{2g}$$

PREPARED G. Keken

6/15/79

REPORT NO.

PAGE B 8

CHECKED E/S

6/26

1 MW Pool Boiling

PROJECT

6" Straight Pipe: $L \approx 34'$

Mid pt. Evap to Mid pt sup. = 10'

Mid pt sup to Top Pool Boil shroud. ~ 15'

Mid pt sup to Top of pool Boiling shroud. = $14\frac{1}{2}'$

$$\text{Friction loss: } h_e = f \frac{L}{D} \frac{V^2}{2g}$$

$$Re = \frac{D V \rho}{\mu}$$

$$\mu_{\text{H}_2\text{O}} = 0.11 \text{ cp.}$$

$$\rho = 38 \text{ lb/ft}^3$$

$$f \approx \frac{.021}{(Re)^{0.2}}$$

long radius elbow 90°
" " 45°

Check valve.

Butterfly valve

6" Pipe, $L = 34'$

Sudden expansion.

NO
(4)
(11)

$$K = .75$$

$$K = 0.2$$

$$K = 2.0$$

$$K = 0.29 \quad (G=100)$$

$$K = 1.00$$

$$K_{\text{total}} = 5.29$$

$K_{\text{Tot.}}$

1.80

.20

2.00

0.29

1.00

PREPARED

J. Kukun

6/19/79

REPORT NO.

PAGE 89

CHECKED

JIS

6/26

PROJECT

IMWHE POOL BOILING
- RECIRC CAPACITY

6" Pipe

$$V = \frac{GPM / 7.48}{(60 \cdot (785) (\frac{6}{12})^2)} = 0.0114 \text{ GPM}$$

$$\Delta P_{friction} = f \frac{L}{D} \frac{V^2}{2g} = \left(\frac{.021}{Re^{0.2}} \right) \frac{L}{D} \frac{(0.0114)^2 \text{ GPM}^2}{64.4}$$

$$Re = \frac{DV\rho}{\mu} = \frac{(\frac{6}{12})(0.0114)(38) \text{ GPM}}{(.12) / 1480} = 2686 \text{ GPM}$$

$$(Re)^{0.2} = (2686 \text{ GPM})^{0.2} = 4.85 (\text{GPM})^{0.2}$$

$$\Delta P_{friction, ft} = \frac{.021}{4.85 (\text{GPM})^{0.2}} \frac{L}{D} \frac{(0.0114)^2 (\text{GPM})^2}{64.4}$$

① $\Delta P_{friction} = \frac{.00013 \times L \times \text{GPM}^{1.8}}{8.7 \times 10^{-9}} \quad L=34 \quad \frac{2.97 \times 10^{-7} \text{ GPM}^{1.8}}{2.97} *$

② Fittings $\Delta P = K \frac{V^2}{2g} = (5.29) \left[\frac{(0.0114) \text{ GPM}}{64.4} \right] = 1.07 \times 10^{-5} (\text{GPM})^2 *$

③ Static head: Fuel flood: 15'
1/2 flood: 10'

* GPM = flow per each of 4 return lines.

TRW.DEFENSE AND SPACE SYSTEMS GROUP
ONE SPACE PARK REDONDO BEACH, CALIFORNIA

PREPARED

J. K. Kizer

6/20/77

REPORT NO.

PAGE B10

CHECKED

J. K. Kizer

6/26

PROJECT

1 MW IZE. Pool Boiling
- Resrc. Capacity -

Flow, GPM [#]	Friction, ft <small>= 2.97×10^{-3} GPM^{1.8}</small>	Fittings, ft <small>= 1.07×10^{-5} GPM²</small>	Static, ft			
			Full Flood	1/2 Flood	Full Flood	1/2 Flood
100	.0012	0.107	15	10	15.1	10.1
200	.0041	0.428	15	10	15.4	10.4
400	.0143	1.712	15	10	16.7	10.7
600	.0297	3.852	15	10	18.9	13.9
1000	.0746	10.7	15	10	25.8	20.8
1500	.155	24.1	15	10	39.3	34.3
2000	.26	42.8	15	10	58.1	53.1
2500	.39	66.9	15	10	82.3	77.3
3000	.54	96.3	15	10	111.84	106.8
4000	.90	171.2	15	10	187.1	182.1

PREPARED

J. K. Kuc

REPORT NO.

PAGE B11

CHECKED

PROJECT

1MW POOL BOILING
RECIRC. CAP.

ΔP SUMP TO PUMP: FRICTION & FITTINGS

Contraction

Sump to 18"

K = 0.50

18" elbow

K = 0.45

Cont. 18" → 16"

K = .05

K = 1.03 ←

Exp 16" → Filter

Contrail. Filter → 16"

Exp. 16" → 18"

K = .05

tee

K = 1.5

90° elbow

K = 0.15

1.8" x 12 contraction

K = .25

Filter

Filter, Total

K = 1.03

4.28

L Piping ~ 20'

Say expansion into filter is from 16" pipe to plane mid way between 16" entrance and 24" mid. plane of filter.

$$\text{area ratio} = \frac{(.785)(16^2)}{\frac{20}{12} \times \frac{28}{12}} = .36$$

$$\text{eq. dia. ratio} = 0.6$$

Crane, pg A-26

K_{exp} = 0.40

K_{cont.} = 0.28

PREPARED

G. Kibin

6/19/79

REPORT NO.

PAGE B12

CHECKED

8/1/79

PROJECT

1 MW HE Pool Boiler

Reirc. Capability

Filter Screen

$$\text{say } \frac{A_{\text{screen}}}{A_{\text{filter area}}} = .75$$

$$\frac{V_0, A_0}{A_f}$$

$$C_d \sim 0.4$$

fig 4 page 2 Sect. 465.5

D. E. H.T. + Fluid Flow Handbook

$$\Delta P, \text{ ft} = \frac{1}{(C_d)^2} \left(\frac{A_0}{A_f} - 1 \right) \frac{V_0^2}{2g}$$

$$= \frac{1}{(0.4)^2} \left(\frac{1}{.75} - 1 \right) \frac{V_0^2}{2g} = 2.0 \frac{V_0^2}{2g}$$

V_0 is velocity in filter tank.
need to relate to pipe velocity.

$$\Delta P, \text{ ft (related to 16" pipe)} = \frac{2.0 \left(\frac{20 \times 24}{144} \right) V_D^2}{(0.785 \left(\frac{16}{12} \right)^2) 2g} = 2.0 \left[\frac{(0.785 \left(\frac{16}{12} \right)^2) V_D^2}{\frac{20 \times 24}{144}} \right] \frac{1}{2g}$$

$$\Delta P, \text{ ft} = 0.35 \frac{V^2}{2g} \quad \text{filter screen}$$

entrance

0.40

exit

0.20

K Total

B-105

1.03

PREPARED D. K. Kien

6/19/79

REPORT NO.

PAGE B13

CHECKED J. S.

6/26

144 WFE. POOL BOILER

PROJECT

Revic. Capability

$$\text{Pipe friction} = f \frac{L}{D} \frac{V^2}{2g}$$

$$18" \text{ pipe } V = \frac{\text{GPM} / 7.48}{60 \left(\frac{785}{12} \right)^2} = 0.00126 \text{ GPM}$$

$$\Delta P_{\text{friction ft.}} = \left(\frac{.021}{Re^{.2}} \right) \frac{L}{D} \frac{(\text{.00126 GPM})^2}{64.4}$$

$$= \frac{(.021)}{\left(\frac{D V \rho}{\mu} \right)^{.2}} \frac{L}{D} \frac{(1.59)(10^{-6})(\text{GPM})^2}{64.4}$$

$$= \frac{(.021)}{\left(\frac{V \rho}{\mu} \right)^{.2}} \frac{L}{D^{1.2}} \frac{1.59 \cdot 10^{-6} \text{ GPM}^2}{64.4}$$

$$= \frac{.021}{\left(\frac{\rho}{\mu} \right)^{.2}} \frac{L}{D^{1.2}} \frac{V^{1.8}}{64.4}$$

$$= \frac{.021}{\left(\frac{38}{.12/1488} \right)^{.2}} \frac{L}{\left(\frac{18}{12} \right)^{1.2}} \frac{(\text{.00126})^{1.8} (\text{GPM})^{1.8}}{64.4}$$

$$= \frac{(.021)}{13.63} \frac{L}{1.63} \frac{(6.03) 10^{-6} (\text{GPM})^{1.8}}{64.4}$$

Pump Section

$$\Delta P_{\text{friction}} = 8.83 \times 10^{-11} \text{ GPM}^{1.8} L$$

GPM = Total flow

PREPARED

J. K. K.

6/19/79

REPORT NO.

PAGE 814

CHECKED

7/10

6/26

PROJECT

14WHE. Pool Boil.
Flex. Capability.

Fittings.

$$\Delta P = K \frac{V^2}{2g} = 4.28 \frac{(0.00126 Q)^2}{64.4} = 1.06 \times 10^{-7} Q^2$$

For example at $Q = 2400$ gpm

$$\Delta P_{\text{fittings}} = 1.06 \times 10^{-7} (2400)^2 = 0.611 \text{ ft.}$$

$$\Delta P_{\text{fric.}} = 8.83 \times 10^{-11} (2400)^{1.8} (20) = 0.0021 \text{ ft.}$$

Suction line
sump to pump inlet.

ΔP_{total}

$$= 0.613 \text{ ft.}$$

$$\Delta P_{\text{sump to pump}} = 1.06 \times 10^{-7} (Q)^2 + 1.77 \times 10^{-9} (Q)^{1.8} \text{ ft.}$$

$Q = \text{gpm.}$

PREPARED D. Robin

6/21/74

REPORT NO.

PAGE B15

CHECKED [Signature]

6/26

1 MW 4002 BOILER

PROJECT

RECIRC. FLOW - PUMP CAP. VS RPM

Pump Head & Flow vs Speed

Flow	370	GPM	upper bundle
	597	GPM	lower bundle
	967	GPM	Both upper + lower

$$\frac{Q_1}{Q_2} = \frac{S_1}{S_2} = \frac{\sqrt{H_1}}{\sqrt{H_2}}$$

Pump Affinity Law
For Fixed diameter
Impeller

10x21A pump # 1

rated speed, 1770 rpm

Q	H ₁₇₇₀	800 RPM	
		Q	H
0	213	0	99.6
1000	207	684	96.8
2000	195	1368	91.2
3000	180	2051	84.2
4000	150	2735	70.1

Pump NO 2

6x21A

rated speed 1770 rpm

Q	1770 RPM	550 RPM		1000 RPM		850 RPM	
	H	Q	H	Q	H	Q	H
0	507	0	49	0	162	0	117
500	502	155	48.5	282	160	240	116
1000	493	311	47.6	565	157	480	114
1500	472	447	45.6	847	151	720	109
2000	425	621	41.0	1130	136	960	98
2500	362	777	34.9	1412	116	1201	83.5
3000	267	847	24.3			1441	61.6

TRW.

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK • REDONDO BEACH, CALIFORNIA

PREPARED Gerald Kikin

REPORT NO. _____

PAGE _____

CHECKED _____

PROJECT _____

APPENDIX-C

POOL BOILING LIQUID NH_3 OVERFLOW
LINE ANALYSIS

PREPARED

G. K. K.

6/19/74

REPORT NO.

PAGE *C1*

CHECKED

J. S.

6/20

PROJECT

1MW H.E. Pool Boiling
NH₃ FLOW RETURN LINES

Problem Determine flow capacity of pool boiling overflow lines return lig. NH₃ to EVAP. SUMP.

Lig. Pressure drop

1. Entrance $K = .23$
2. Elbow, 90 $K = 0.415$
3. Contractions 10" - 6" $K = \left[1 - \left(\frac{d_1}{d_2}\right)^2\right]^2 = \left[1 - \left(\frac{6}{10}\right)^2\right]^2 = [1 - .36]^2 = .55$
4. 6" Pipe Friction
5. 45° elbow $K = 0.2$
6. 6" Gate Valve $K = 0.17$
7. Expansion $K = 1.0$

$$\Sigma K = 2.45$$

PREPARED D. Kiki 6/19/79 REPORT NO.

PAGE 02

CHECKED S/S 6/24

PROJECT _____

1 MW HE, Pool Boiling
NH₃ FLOW RETURN LINES

Pipe friction

A 400 gpm, $\mu \sim 0.12$ cp, $D = 6"$

$f = 0.017$ Lane 3-9

$$\Delta P, \text{ ft.} = f \frac{L}{D} \frac{V^2}{2g}$$

$$V = \frac{Q / 7.48}{(1.05)(\frac{6}{12})^2 60} = 0.114 Q$$

$$Q = GPM$$

$$\text{friction drop } \Delta P_{\text{ft.}} = 4.54 \times 10^{-7} Q^{1.8} = \left\{ \begin{array}{l} L = 266 \text{ ft} \\ D = 6" \end{array} \right\}$$

fittings $K = 2.45$

$$\Delta P_{\text{ft.}} = K \frac{V^2}{2g} = 2.45 (2.02 \times 10^{-6} Q^2) = 4.95 \times 10^{-6} Q^2$$

$$\Delta P_{\text{total}} = \frac{4.95 \times 10^{-6} Q^2 + 4.54 \times 10^{-7} Q^{1.8}}{\text{fittings } L = 266 \text{ ft.}}$$

Full
Floored
Bundle

$$\Delta P_{\text{total}} = (10^{-6}) [4.95 Q^2 + .454 Q^{1.8}]$$

1/2 Floored Bundle

$$\Delta P_{\text{Fittings}} = 4.95 \times 10^{-6} Q^2$$

$$\Delta P_{\text{friction}} = 3.67 \times 10^{-7} Q^{1.8}$$

$$\left\{ \begin{array}{l} L = 21' \\ D = 6" \end{array} \right\}$$

$$\Delta P_{\text{Total}} = 10^{-6} [4.95 Q^2 + 0.367 Q^{1.8}]$$

ΔP 's are plotted as shown on Fig B2

FIG. B2 POOL BOILING LIQUID NH₃ OVERFLOW LINE PRESSURE DROP VS FLOW

* THERE ARE FOUR RETURN LINES

B-112

J. Kiker
6/26/79

PREPARED

D. K. Kim

6/25/77

REPORT NO.

PAGE *C4*

CHECKED

[Signature]

6/26

PROJECT

IMW H.E. POOL BOILING
OVERFLOW RETURN LINE FLOW CAPACITY

For Full flooded bundle, gravity head available $\approx 15'$
From fig B2, we see the overflow line capability at
this head is ≈ 1700 GPM.

For a level, non-rocking ocean platform, the
maximum overflow would then be 6800 GPM.
However for a sea state that causes rocking of
the ocean platform, it is expected that at any
one time, liquid ammonia overflow would be
flowing through only two return lines.
Under this rocking situation the flow would
then be 3400 GPM.

The various overflow capability is summarized
as follows:

Table C-1

PLATFORM MOTION POOL BOILING SETUP	TOTAL OVERFLOW CAPABILITY, GPM	
	NON-ROCKING (4 RETURN LINES)	ROCKING (2 RETURN LINES)
FULL FLOODED BUNDLE	6800	3400
1/2 FLOODED BUNDLE	5600	2800

The available head for the 1/2 flooded bundle
 $\approx 10'$.

PREPARED H. Kikui 6/25/79
CHECKED [Signature] 6/26
PROJECT _____

REPORT NO.

PAGE C5

1 MW H.E. POOL BOILING
OVERFLOW RETURN FLOW & PUMP CAPABILITY

Correlation of PUMP CAPACITY AND (2) overflow Return pipe capacity.

Problem: What is maximum pool boiling recirculation flow for full flooded and $\frac{1}{2}$ flooded bundles.

From Figure B1, Maximum flow capability of Pump P2 is ≈ 3500 GPM
Maximum capability of A is ≈ 3700 GPM

Full Flooded Bundle

Therefore maximum pump flow capability using either P1 or P2 is $\approx 3500 - 3700$ GPM.

$$\text{Recirculation rate (Pump P2)} \approx \frac{3500 - 920}{920} = 2.8$$

The 3500 GPM pump flow is limiting compared to the 6800 GPM drain overflow capability of four 6" drain lines.

However the 2 drain line maximum flow is ≈ 3400 which is about the same as the 3500 GPM pump flow capability.

$\frac{1}{2}$ flooded bundle: From Table C1

4 lines { Flow limit due to Pump P2 ≈ 3500 GPM
4 overflow line flow limit 5600 GPM
Therefore Pump limits this case.

2 lines { Pump P2 flow ≈ 3500 GPM
2 overflow line limit ≈ 2800 GPM
Therefore 2 overflow lines limit this case

PREPARED

J. K. K. K.

6/26/79

REPORT NO.

PAGE C6

CHECKED

J. K. K. K.

6/26

17. W. H. E. Pool Boiling
Recirculation Ratios.

PROJECT

Lower Bundle Recirculation Ratio -

Pump limit
4 lines

$$= \frac{3500 - 570}{570} = 5.1$$

2 lines
limit

$$\frac{2800}{570} = 4.9$$

RESULTS

RECIRCULATION RATIOS

	PLATFORM MOTION	
	NON-ROCKING	ROCKING
FULL BUNDLE	2.8	2.8
LOWER BUNDLE	5.1	4.9

$$\text{Recirculation Ratio} = \frac{W_3}{W_2}$$

APPENDIX C
CONDENSER DESIGN

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK • REDONDO BEACH • CALIFORNIA 90278

INTEROFFICE CORRESPONDENCE

79.6852.3.10-13

TO: M. Seidman

CC: C. Schumann
P. Edris
K. Cramer

DATE: 24 May 1979

SUBJECT: Condenser Drain Pots Evaluation per PWA DE-AC03-
77ET-20444 Item 6 of Deliverables JN 3773-21

FROM: P. Trubey *P29*
BLDG. 81 MAIL STA. 1578 EXT. 52130

Reference: IOC 79.6852.3.10-11, Dated 4 May 1979, Same Subject

Attached herewith is the completed and checked version of the evaluation previously delivered, unchecked, with the above referenced IOC. There have been minor modification and clarifications incorporated, although the basic results have not changed appreciably.

PT:df

Attachment

PREPARED P.L. Trubey

5-4-79

REPORT NO.

PAGE 1

CHECKED C.T. SCHUMANN

5-24-79

PROJECT OTEC - 1 MWE

(4122 places)

CONDENSER DRAIN PDT EVALUATION

I PURPOSE:

This analysis is performed to evaluate the operability of the drain pots, and associated level, located on the Condenser Test Article. Five drain pots are provided and the change in liquid level within the drain pots reflects system dynamic fluctuations, and is herein addressed. Ship motions are considered to be as identified in the proposal using 16 foot significant wave height.

II DESIGN CRITERIA:

1. Nominal flow rate per condenser zone is 163.24 gpm.
2. Pitch angle in 16 ft. significant wave height seas is 1.56° , maximum
3. Roll angle (16 ft seas) = 2.12° , quartering seas, and natural roll period is 16 seconds.

PREPARED P.L. Trubey 5-4-79

REPORT NO.

PAGE 2

CHECKED _____

CONDENSER DRAIN POT EVALUATION

PROJECT OTEC - 1 MWeIII APPLICABLE CODES, STANDARDS, REFERENCES:

1. OTEC-1 Early Ocean Test Platform - GMDI proposal No. 030178 dated 3 April 1978
2. Crane Technical Paper No. 410, Flow of Fluids
3. Advanced Engineering Mathematics, 3rd edition E. Kreystig, 1972.
4. Introduction to Fluid Mechanics - John and Haberman dated 1971
5. Marks' standard Handbook for Mechanical Engineers, 7th edition - Baummeister, 1967
6. Drawings - Drain Pot Details, OTEC Condenser DM-97162, Rev. 3

EM-96989, Rev. 2 OTEC Condenser, Mechanical outline Drawing

EM-96993, Rev. 3 OTEC Condenser, Shell Baffles

PREPARED P. L. Trubey 5-4-79
CHECKED _____
PROJECT _____

REPORT NO. _____

PAGE 3

CONDENSER DRAIN POT EVALUATION

IV ASSUMPTIONS:

1. Sea response spectra is sinusoidal waveform (pitch rate)
2. Yaw motion has no effect on fluid flow characteristics.
3. maximum operating sea state is 16 foot significant wave height.
4. Fluid viscous properties not considered, although Newtonian properties are assumed to hold.
5. Flow into drain hole is approximated by considering it as a dam with flow over the edge.
6. Physical configuration allows the fluid to go through sinusoidal fluctuations at twice frequency of actual ship pitch rate.

PREPARED P.L. Trubey

5-4-79

REPORT NO.

PAGE 4

CHECKED _____

PROJECT OTEC-1 MW_e

CONDENSER DRAIN POT EVALUATION

II CALCULATIONS:

Flow rate to be used = 180 GPM / drain pot zone.

$$\text{Nominal flow} = 163.24 = \left(\frac{816.2 \text{ GPM}}{5 \text{ zones}} \right)$$

$$\text{excess} = \frac{180 - 163.24}{163.24} \times 100\% = 10.3\% \text{ excess}$$

The following configuration shall be used to evaluate the condenser zone:

Assume 8" ID for this evaluation of drain pot inlet size.

Drain pot inlet:

Open top flow area:

$$7.981" \text{ ID} \Rightarrow A = \pi D^2 / 4 = 50.03 \text{ in}^2$$

bottom tube drain holes (inlet nozzle to pot volume)
4 x 1/2 x 4" dia (approx)

PREPARED P.L. Trubey

5-4-79

REPORT NO.

PAGE 5

CHECKED _____

PROJECT OPEC-2 MWE

CONDENSER DRAIN POT EVALUATION

drains flow area:

$$A_d = 2 \times \frac{\pi(4')^2}{4} = 25.13 \text{ in}^2$$

⇒ The governing liquid flow area is 25.13 in² (bottom drain).

Available drain pot flow area available is

$$ID_1 = 23.25 \text{ in}$$

$$ID_2 = 7.981 \text{ in}$$

$$OD_2 = 8.625 \text{ in}$$

$$\begin{aligned} A_{Av} &= \frac{\pi(23.25)^2}{4} - \frac{\pi(8.625^2 - 7.981^2)}{4} \\ &= 416.2 \text{ in}^2 \\ &= 2.89 \text{ ft}^2 \end{aligned}$$

$$\text{Volume per foot} = 2.89 \text{ ft}^2 \times 1 \text{ ft} \times 7.48 \text{ gal/ft}^3 = 21.6209 \text{ gal.}$$

Note 1)

For this analysis, only one zone of the condenser will be considered and it will be assumed to be typical of all 5 zones within the condenser.

2) The angle inscribed by the 8" dia. hole is only about 8.5°, but this exceeds 2 times the roll rate, and roll need not be considered)

PREPARED P.L. Trubey 5-2-79

REPORT NO.

PAGE 6

CHECKED _____

PROJECT OTEC - 1MWe

CONDENSER DRAIN POT EVALUATION

CALCULATIONS, Cont.

I. Determine hold-up volume

1.

θ = angle of intersection

r = radius of base

$$\begin{aligned} \text{Volume} &= H \left(\frac{2}{3} a^3 - cB \right) / (r-c) \\ &= H \left[a \left(r^2 - \frac{1}{3} a^2 \right) - r^2 c \text{ rad } u \right] / (r-c) \end{aligned}$$

B = area of base

Given ψ = pitch angle = 1.56° (max., 16' significant wave)

for $H = 4' - 13/4'' = 4.145833 \text{ ft.}$, find h

$$\begin{aligned} h &= H \tan 1.56^\circ = 4.145833 (.0272337) \\ &= 0.1129071 \text{ ft.} \end{aligned}$$

PREPARED P. L. Trubey 5-2-79
 CHECKED _____
 PROJECT OTEC - 1 MWe

REPORT NO.

PAGE 7

CONDENSER DRAIN POT EVALUATION

$\theta = \text{inscribed angle}$

$$\begin{aligned} c &= r - h \\ &= 4.50' - .1129070707 \\ &= 4.387092929 \text{ ft.} \end{aligned}$$

To find angle θ , note that $\cos \theta = \frac{c}{r} = \frac{4.3870929}{4.50} = .9749095398$

$$\theta = 12.8618726^\circ, 2\theta = 25.72374521^\circ$$

$$B = \text{Area of base (segment)} = \frac{1}{2} r^2 (\text{rad } A - \sin A) \quad A = 2\theta$$

rad A = radian measure of angle A

$$A = 25.72374521^\circ$$

$$\text{Rad } A = .4489640498$$

$$\sin A = .4340324825$$

$$\begin{aligned} B &= \frac{1}{2} (4.5^2) (.4489640498 - .4340324825) \\ &= .1511821189 \text{ ft}^2 \end{aligned}$$

Volume of Ungula: $V = H \left(\frac{2}{3} a^2 - cB \right) / (r - c)$

$$H = 4.145833$$

$$a = 4.5 \sin 12.86^\circ = 1.001706359$$

$$c = 4.387092929$$

$$B = .1511821189$$

$$r = 4.50 \text{ ft}$$

PREPARED P.L. Trubey

5-2-79

REPORT NO.

PAGE 8

CHECKED _____

PROJECT OTEC - 1 MWe

CONDENSER DRAIN PDT EVALUATION

$$V = 4.145833 \left[\frac{2}{3} (1.001706359)^3 - 4.387092929 (1.1511821189) \right] / (4.5 - 4.387092929)$$

$$= 4.145833 [.6700852113 - .6632500048] / (.112907071)$$

$$= 0.2509818617 \text{ ft}^3$$

$$\text{Volume (gal.)} = .2509818617 (7.481) = 1.877595307$$

2. However, to accomplish flow of 180 gpm into the 8" diameter hole, the head must be increased to accomplish this flow rate.

$$\left(\frac{Q}{b} \right)_{\max}^2 = \left(\frac{8}{27} \right) g h_o^3 \quad \text{(From Fluid Mechanics by John & Haberman)}$$

also, for max flow condition $h = \frac{2}{3} h_o$
(for flow height at entry to hole)

where Q = Flow in CFS
 b = width of flow area (assume πD)
 g = 32.174 ft/sec²
 h_o = height of fluid in hold-up above hole height.

$$\text{let } b = \pi D = \pi (8") / 2 = 2.094395102 \text{ ft.}$$

$$Q = 180 \text{ GPM } (7.481 \times 60) = .401015907 \text{ CFS}$$

$$h_o^3 = \left(\frac{27}{8} \right) \left(\frac{Q}{b} \right)_{\max}^2 \times \frac{1}{g} = \frac{27}{8 \times 32.174} \left(\frac{.401015907}{2.094395102} \right)^2 = .0038456937$$

$$h_o = (.0038456937)^{\frac{1}{3}} = 0.1566720638 = 1.880064765 \text{ inches}$$

$$h = \frac{2}{3} (1.880065) = 1.2533765 \text{ in.}$$

PREPARED P.L. Trubey 5-2-79

REPORT NO.

PAGE 9

CHECKED _____

CONDENSER DRAIN POT EVALUATION

 PROJECT OTEC - 1 Mwe

To evaluate new hold-up volume, evaluate average height of fluid around opening of drain hole

$$\psi = 1.56^\circ$$

$$\begin{aligned}\Delta \text{elev} &= 8^\circ \tan 1.56^\circ \\ &= 0.2178709304 \text{ inches}\end{aligned}$$

⇒ For average depth,

$$\text{use } h = 1.25337651 + (.5)(.2178709304) = 1.362311975 \text{ in.}$$

$$\begin{aligned}\therefore h_o &= \frac{3}{2} (1.362311975) = 2.043467963 \text{ inches} \\ &= .1702889969 \text{ ft.}\end{aligned}$$

3. Using this value, re-calculate volume of hold-up with flow constant at 180 GPM into drain hole.

$$\text{new values: } h = .1129070707 + .1702889969 = .2831960676 \text{ ft.}$$

$$C = r - h = 4.5 - .2831960676 = 4.216803932 \text{ ft}$$

$$\theta = \cos^{-1} \frac{4.216803932}{4.500} = 20.43522048^\circ$$

$$A = 2\theta = 40.87044096^\circ$$

$$B = \text{area of base} = \frac{1}{2} r^2 (\text{rad } A - \sin A)$$

$$\text{rad } A = .7133237615, \sin A = .6543507798$$

$$\begin{aligned}B &= \frac{1}{2} (4.5)^2 (.7133237615 - .6543507798) \\ &= .5971014391 \text{ ft}^2\end{aligned}$$

PREPARED P. L. Trubey 5-2-79
 CHECKED _____
 PROJECT OTEC - IMWe

REPORT NO.

PAGE 10

CONDENSER DRAIN POT EVALUATION

Re-calculate value of H :

$$\tan \psi = .2831960676$$

$$H = .2831960676 / \tan \psi = 10.3986729 \text{ ft.}$$

Since, however, the compartment is only 8.292' long, the additional volume must be subtracted from the hold-up volume.

⇒ calculate total volume and subtract excess due to baffle plate interference.

$$\text{Total Volume of Ungula} = V = H(4/3a^3 - cB) / (r - c)$$

$$H = 10.3986729 \text{ ft}$$

$$a = 4.5 \sin \theta = 1.571166635$$

$$c = 4.216803932 \text{ ft}$$

$$B = 0.5971014391 \text{ ft}^2$$

$$r = 4.50 \text{ ft.}$$

$$V = H[2/3(a^3) - cB] / (r - c)$$

$$V = 10.3986729 [0.0678245238] / 0.283196068$$

$$= 2.490447846 \text{ ft}^3 = 18.63104034 \text{ gal. Total volume which includes excess;}$$

PREPARED P.L. Trubey 5-2-79

REPORT NO.

PAGE 11

CHECKED _____

CONDENSER DRAIN POT EVALUATION

PROJECT OTEC- 1 MW

4. Calculate excess storage quantity:

$$H = 10.3986729 - 8.291666667 = 2.107006233 \text{ ft}$$

$$h = H \tan 1.56^\circ$$

$$= 0.057381926 \text{ ft}$$

$$C = 4.5 - .057381926 = 4.442618074 \text{ ft}$$

$$r = 4.5 \text{ ft}$$

$$\theta = \cos^{-1} \frac{4.442618074}{4.5} = 9.159705496^\circ$$

$$A = 2\theta = 18.31941099^\circ$$

$$\begin{aligned} B = \text{area of base} &= \frac{1}{2} r^2 (\text{rad } A - \sin A) \\ &= \frac{1}{2} (4.5)^2 (.3197340388 - .3143140897) \\ &= 0.0548769846 \text{ ft}^2 \end{aligned}$$

$$a = 4.5 \sin 9.159705496 = .7163411542 \text{ ft.}$$

Volume of excess (lingula shape)

$$V = H \left(\frac{2}{3} a^3 - CB \right) / (r - C)$$

$$V = \frac{2.107006233 \left[\frac{2}{3} (.7163411542)^3 - (4.442618074)(.0548769846) \right]}{(4.5 - 4.442618074)}$$

$$= 0.046275833 \text{ ft}^3$$

$$= 0.346189507 \text{ gal.}$$

5. Actual hold-up quantity (minimum, excluding viscous forces in draining fluid)

$$V = 18.63104034 - 0.346189507 = 18.28485083 \text{ gal.}$$

PREPARED P.L. Trubey 5-2-79
CHECKED _____
PROJECT OTEC-1 MWe

REPORT NO.

PAGE 12

CONDENSER DRAIN POT EVALUATION

6. As the ship pitches to the opposite direction, it passes through the mid-point or level plane, calculate the height of fluid, flow rate, and change of fluid level resulting from this pitching motion.

For Volume = 18.28485083 gal, Length = 8'-3 1/2", the segment area is as follows:

$$B = \frac{18.28485083 \text{ gal}}{7.481 \text{ gal} \times 8.292 \text{ ft}} = 0.2947745141 \text{ ft}^2$$

Find h corresponding to $B = 0.2947745141 \text{ ft}^2$

$$B = \frac{1}{2} (r^2) (\text{rad } A - \sin A)$$

$$r = 4.5 \text{ ft}$$

$$(\text{rad } A - \sin A) = \frac{2 \times 0.2947745141}{4.5^2} = 0.0291135322$$

$$\text{Find } A; \quad \text{rad } A = \sin A + 0.0291135322$$

$$A = 32.197686^\circ \quad (\text{by iteration})$$

$$\text{since } A = 2\theta, \quad \theta = \frac{A}{2} = 16.098843^\circ$$

$$c = r \cos \theta$$

$$\theta = 16.098843^\circ$$

$$r = 4.5'$$

$$c = 4.323531393 \text{ ft}$$

$$h = r - c = .176468607 \text{ ft}$$

$$h = 2.117623289 \text{ in.}$$

PREPARED P.L. Trubey 5-3-79

REPORT NO.

PAGE 13

CHECKED _____

CONDENSER DRAIN POT EVALUATION

PROJECT OTEC - 1 MWe

7. Using $h = 2.15$ inches (h_0 for flow over dam), calculate the maximum flow rate resulting from this height change.

as before, let $b = Md = 2.094395102$ ft.

$$h_0 = L = .1764686074 \text{ ft}$$

Flow over dam (model)

$$\left(\frac{Q}{b}\right)_{\max}^2 = \left(\frac{8}{27}\right) g h_0^3$$

where:

$$g = 32.174 \text{ ft/sec}^2$$

$$h_0 = 0.1764686074 \text{ ft.}$$

$$b = 2.094395102$$

$$Q = \text{flow (CFS)}$$

$$\left(\frac{Q}{2.094395102}\right)^2 = \frac{8 \times 32.174 \times (.1764686074)^3}{27}$$

$$= 0.0523882215$$

$$\Rightarrow Q = 2.094395102 (0.0523882215)^{1/2}$$

$$= .4793750664 \text{ CFS}$$

$$= 215.1722923 \text{ GPM}$$

$$\Delta \text{flow} = 215.1722923 - 180$$

$$= 35.1722923 \text{ GPM}$$

For graphic purposes, use 40 gpm variation

8. Evaluate height change in drain pot as a function of flow variation

PREPARED P.L. Trubey

5-3-79

REPORT NO.

PAGE 14

CHECKED _____

PROJECT OTEC - 1 MWe

CONDENSER DRAIN POT EVALUATION

note: On beginning pitch motion, flow drops instantaneously then stabilizes at maximum pitch angle. Upon returning from pitch angle the flow increases until mid point is reached whereupon the flow passes through the nominal rating to low flow again as pitch direction is reversed.

∴ Assume the sinusoidal flow variation as shown above, and evaluate the change in drain pot level as a result of this motion.

$\dot{w}_1$ = average flow rate = 180 gpm (into pot)

$\Delta \dot{w}$ = change in flow rate

ω = oscillation period

t = time (seconds) for oscillation

f = frequency = $1/\omega t$, cycles per second.

PREPARED P.L. Trubey

5-3-79

REPORT NO.

PAGE 15

CHECKED _____

PROJECT OTEC- I MWe

CONDENSER DRAIN POT EVALUATION

$\dot{w}_2$ = flow rate out of pot, 180 gpm, const.

dh = change in level inside drain pot

A = cross sectional area of drain pot available for fluid hold-up.

$$\dot{w}_1 = 180 + 40 \sin \omega t$$

where ω = oscillation period
($\frac{1}{2}$ of pitch oscillation period)

$\Delta \dot{w} = \dot{w}_1 - \dot{w}_2$ = change of flow in pot due to fluctuations.

$$\frac{d\dot{w}}{dt} = (180 + 40 \sin \omega t) - 180 = 40 \sin \omega t \text{ (gpm)}$$

To effect the measurability in change of fluid height within the drain pot, the cross sectional area and fluid accumulation (or loss) rate must be related to height changes.

$$\frac{w \text{ (gal/min)}}{7.48 \text{ gal/ft}^3 \times A \text{ (ft}^2\text{)}} = h \text{ (ft)} = \frac{w}{7.48 A} \frac{\text{ft}}{\text{min}}$$

$$A = 2.89 \text{ ft}^2$$

$$\therefore (7.48 A) \frac{dh}{dt} = \frac{d\dot{w}}{dt}$$

$$\frac{dh}{dt} = \frac{d\dot{w}}{dt} \left(\frac{1}{7.48 A} \right)$$

$$\text{Let } \left(\frac{1}{7.48 A} \right) = K_1 \text{ (const.)}$$

$$\frac{1}{K_1} = 7.48 (2.89)$$

$$= 21.62 \text{ gal/ft}$$

PREPARED P. L. Trubey 5-3-79

REPORT NO.

PAGE 16

CHECKED _____

CONDENSER DRAIN POT EVALUATION

PROJECT OTEC-1 MWe

Evaluating flow per second:

$$V(\text{volume}) = \frac{180}{60}t + \frac{40t}{60} \sin \omega t$$

$$dV = K_1 dt + K_2 \sin \omega t dt$$

$$\int \sin ax dx = -\frac{1}{a} \cos ax + C$$

$$V = K_1 t - \frac{K_2}{\omega} \cos \omega t + C_1$$

$$C_1 = 0$$

$$\Delta V = \text{Volume change} = V_{in} - V_{out}$$

$$= (K_1 t - \frac{K_2}{\omega} \cos \omega t) - (K_1 t)$$

$$= -\frac{K_2}{\omega} \cos \omega t$$

$$\omega = \text{period} = \frac{2\pi}{t}$$

$t = \text{time for one full cycle}$
(1/2 pitch rate)

evaluate K_2

$$K_2 = 40 \frac{\text{gal}}{\text{min}} \frac{\text{min}}{60 \text{ sec}} = 0.667 \text{ gal/sec.}$$

From page 5, the in flux rate can fill up at 21.62009 gal/ft

$$\Rightarrow K_2 = \text{height variable} = \frac{0.66667 \text{ ft}}{21.62009 \text{ sec}}$$

$$= 0.0308355176 \text{ ft/sec}$$

$$\approx 0.031 \text{ ft/sec.}$$

$$\Rightarrow \Delta h = K_1 \Delta V = -\frac{0.031}{\omega} \sin \omega t$$

PREPARED P. L. Trubey 5-3-79

REPORT NO.

PAGE 17

CHECKED _____

PROJECT OTEC - 1 mwe

CONDENSER DRAIN POT EVALUATION

$$\Delta h (\text{in feet}) = -\frac{0.031}{\omega} \cos \omega t$$

⇒ since $\omega = \frac{1}{2}$ pitch period of ship
 $\omega = 2\pi / t$, maximum (or minimum) h
occurs at $0, \pi$

6 CPM

ship pitch period T_1 (seconds)	ω $2 \times 2\pi / T_1$	time t (sec)	Δh above havg. ft (inches)
10 sec.	1.2566	0	-0.0247 (0.3)
	1.2566	1.25	0.0 (0.0)
	1.2566	2.5	+0.0247 (0.3)
	1.2566	3.75	0.0 (0.0)
	1.2566	5	-0.0247 (0.3)
16 sec.	0.7854	0	-0.0395 (0.47)
	0.7854	2	0.0 (0.0)
	0.7854	4	+0.0395 (0.47)
	0.7854	6	0.0 (0.0)
30 sec.	0.4188	0	-0.0740 (-.89)
	0.4188	3.75	0.0 (0.0)
	0.4188	7.5	+0.0740 (-.89)
	0.4188	11.25	0.0 (0.0)
60 sec.	0.2094	0	-0.1480 (-1.78)
	0.2094	25	0.0 (0.0)
	0.2094	15	+0.1480 (-1.78)
	0.2094	22.5	0.0 (0.0)

Note: This method of solution will not hold if
the ships pitch rate gets exceedingly
long, as Δh can approach infinity,

PREPARED P.L. Trubey

5-4-79

REPORT NO.

PAGE 18

CHECKED _____

PROJECT OTEC - 2MWe

CONDENSER DRAIN POT EVALUATION.

Note, cont.

although the actual flow stabilizes
as the pitch changes to a lower rate.

The results of this evaluation are that
the overall change in fluid height is negligible
compared to the assumed fluctuation rate, and
the greatest concern is the condenser hold-up
quantity.

PREPARED P.L. Trubey

5-4-79

REPORT NO.

PAGE 19

CHECKED _____

PROJECT OTEC - 1 MW_e

CONDENSER DRAIN POT EVALUATION

For interest, evaluate the vapor vent area available in the drain hole as a function of increased flow. Assume as before that the flow behaves as if flowing over a dam, and that the flow rate is constant.

$$Q_{cfs} = Q_{gpm} / (7.481 \text{ gal/ft}^3 \times 60 \text{ sec/min})$$

$$b = \text{constant} = 2.094395 \text{ ft (from page 8)}$$

$$h_o^3 = \left(\frac{27}{8 \times 32.174} \right) \left(\frac{Q_{cfs}}{b} \right)^2$$

$$h = {}^{2/3} h_o \quad (h \text{ is reduced height at edge of hole})$$

$$A_1 = bh \quad (A_1 \text{ is area utilized by equiv. flow})$$

$$A_o = \frac{\pi D^2}{4}, \quad D = \text{diameter of hole (ft)}, \quad D = \left(\frac{2}{3}\right) \text{ ft}$$

$$\Delta A = A_o - A_1 \text{ (ft}^2\text{)}, \quad 144 \times \Delta A = \text{flow area for vapor (in}^2\text{)}$$

$$\Delta A = \left[\frac{\pi \left(\frac{2}{3}\right)^2}{4} - \frac{2 \times b}{3} \left[\left(\frac{27}{8 \times 32.174} \right) \left(\frac{Q_{gpm}}{b \times 7.481 \times 60} \right)^2 \right]^{1/3} \right] \times 144 = \Delta A \text{ (in}^2\text{)}$$

Q(gpm)	ΔA (in ²)	Q(gpm)	ΔA (in ²)
150	22.3696	270	8.9874
160	21.1432	280	7.9744
170	19.9421	290	6.9734
180	18.7643	300	5.9838
190	17.6082	310	5.0052
200	16.4721	320	4.0370
210	15.3549	330	3.0788
220	14.2552	340	2.1303
230	13.1721	350	1.1911
240	12.1046	360	0.2607
250	11.0518	370	(-) 0.6610
260	10.0130		

PREPARED P.L. Tubey 5-4-79
 CHECKED _____
 PROJECT OTEC - 1 MWe

REPORT NO.

PAGE 20

CONDENSER DRAIN POT EVALUATION

Plot of Flow (GPM) vs Available Vent area (ΔA)

Note: This evaluation does not consider viscous forces, surface tension, and real fluid behavior in down-flow drain.

PREPARED P.L. Trubey 5-4-79

REPORT NO.

PAGE 21

CHECKED _____

CONDENSER DRAIN POT EVALUATION

PROJECT CTEC-1 MWe

VI SUMMARY OF RESULTS:

Review of results :

The results of this analysis indicates that at nominal flow rates the drain pot inlet size (8") is satisfactory and will yield a hold-up volume within the condenser of about 18.7 gallons, maximum. This volume is required to provide sufficient driving impetus to flow 180 gpm through the drain pot inlet, with the ship pitched 1.56° (16 ft. significant sea). As the ship pitches slowly the rate increases although the hold-up does not significantly change. The resulting level change within the drain pot itself is very small (eg., at pitch rate of 30 seconds period, the level fluctuates less than one inch). This assumes inviscid flow, essentially massless effects on fluid accelerations, and a predictably sinusoidal pitch waveform.

As a quick exercise, a simple calculation was performed to determine the hold-up height and flow capability if the flow-rate into the condenser drain pot was increased to 360 gpm. The results indicate that sufficient flow area is not available to accomodate the greatly increased flow as insufficient area remains to allow the fluid to be self-venting. Vapor hold-up and blocking of the drain pot would effect greatly fluctuating levels within the drain pot. The analysis does not address adequately the self-venting aspects which would certainly become a consideration as the flow area reduces. As the flow quantity exceeds the

PREPARED P. I. Tubey

5-9-78

REPORT NO.

PAGE 22

CHECKED _____

PROJECT CTEC-1MWe

CONDENSER DRAIN POT EVALUATION

venting capability, the viscous effects of the liquid phase-vapor phase interface will cause an increase in the hold-up quantity, resulting in surging or "bubbling", and the previously indicated drain pot level fluctuations. For these reasons, flow rates greater than 30% or so above nominal would have to be addressed and evaluated more closely to determine acceptability.

APPENDIX D

INSTRUMENTATION DESIGN

INTEROFFICE CORRESPONDENCE

80.6852.4.5-015

TO: J. E. Snyder

CC:

DATE: 28 May 1980

SUBJECT: Calculated Flow Rates
Through IMWe Instrumented
Tubes

FROM: A. M. Sprouse

BLDG. 81 MAIL STA. 1578 EXT. 52130

An emended version of Bob Wise's flow rate calculation for the instrumented tubes is attached. Specifically an error in converting units of psi to feet of head was corrected; pages 14, 20, and 30 (sections C-5, C-13, C-16, D, H, and I) are affected. A comparison of the old and new flow rates is shown in tabular form below. No redesign of the loops is necessary to accommodate the reduction in flow rate.

	Evaporator Tube Flow Rate, GPM	Condenser Tube Flow Rate, GPM
Average Tube	13.10	12.33
Instrumented Tube, Previous Calculation	13.58	12.74
Instrumented Tube, Corrected Calculation	13.14	12.32

Approved:

P. T. Fukunaga

AMS:cm

Attachment

PREPARED

R.A. WISE 5/30/79

REPORT NO.

PAGE 1 OF 31

CHECKED

JUL 30 1979

1 MW HEAT EXCHANGERS

MODEL

I. PURPOSE

THE PURPOSE OF THIS ANALYSIS IS TO DETERMINE THE FLOW RATE THRU THE INSTRUMENTED LOOPS OF THE 1 MW HEAT EXCHANGER.

II. APPROACH AND ASSUMPTIONS

1. THE EVAPORATOR CONFIGURATION IS SHOWN IN FIGURE 1 AND THE CONDENSER CONFIGURATION IS SHOWN IN FIGURE 2. FIGURE 3 GIVES THE COMPONENTS FOR BOTH.
2. USE 1.25 HOSE
3. RADIUS RATIO FOR BENDS = 2.0 (R/D)
4. FLOW FOR EVAPORATOR TUBE = 13.08 GPM AND CONDENSER TUBE = 12.32 GPM
5. TOTAL FLOW RATE FOR EVAPORATOR IS 82,378.36 GPM AND THE TOTAL FLOW RATE FOR THE CONDENSER IS 68107 GPM
6. THE NUMBER OF TUBES IN THE EVAPORATOR IS 6304 AND THE NUMBER OF TUBES IN THE CONDENSER IS 5526

FIGURE 1

FIGURE 2

FIGURE 3

PREPARED R. A. WISE

REPORT NO.

PAGE 5

CHECKED AMJ JUL 30 1979

MODEL

1 MW HEAT EXCHANGER -
EVAPORATORIII. CALCULATIONS2. PRESSURE DROP IN MAIN PATH - EVAPORATOR1. CONTRACTION FROM WATER BOX TO TUBES, STATION 1
STATION NO. REFERS TO FIGURE 1

TUBE ID = .944" NO. OF TUBES = 6304

AREA OF TUBES = $A_T = \frac{\pi (.944)^2}{4 \times 144} \times 6304 = 30.64 \text{ FT}^2$

WATER BOX ID = 12' AREA OF WATER BOX = $A_W = 113.1 \text{ FT}^2$

$\frac{A_T}{A_W} = \frac{30.64}{113.1} = .271$ FROM FIGURE 8 $K = .37$

$Q = 13.08 \text{ GPM} = 0.029144 \frac{\text{FT}^3}{\text{SEC}}$ AREA OF TUBE = A

$A = .0048604 \text{ FT}^2$ $V = \frac{Q}{A} = \frac{.029144}{.0048604} = 6.0 \frac{\text{FT}}{\text{SEC}}$

$\Delta P = \frac{KV^2}{2g} = \frac{.37(6)^2}{2 \times 32.174}$

$\Delta P = 0.2070 \text{ FT}$

2. FRICTION LOSS IN TUBE, STATION 1 TO 2

$L = 38.2'$ $D = .944"$ $P = 64 \text{ PSF} \cdot 3$ $\eta = 3.63 \frac{\text{FT} \cdot \text{HR}}{\text{FT} \cdot \text{HR}}$

$V = 6 \text{ FT/SEC}$ $NRE = \frac{3600 V D P}{\eta} = \frac{3600 \times 6 \times .944 \times 64}{3.63} = 3.0 \times 10^4$

FROM FIGURE 6 $\frac{e}{D} = .00006$ FROM FIGURE 7 $f = .023$

$\Delta P = f \frac{L}{D} \frac{V^2}{2g} = .023 \times \frac{38.2(6)^2}{.944 \times 12} = 6.2484 \text{ FT}$

3. EXPANSION FROM TUBE TO WATER BOX, STATION 2

FROM SECTION 1 $\frac{A_T}{A_W} = .271$ FROM FIGURE 4

$K = .54$ $V = 6 \frac{\text{FT}}{\text{SEC}}$

$\Delta P = \frac{.54(6)^2}{64.348}$

$\Delta P = .3021 \text{ FT}$

FIGURE 4
LOSS FACTORS FOR SUDDEN EXPANSIONS

FIGURE 5
LOSS FACTOR FOR CONVERGING NOZZLE

PREPARED R. A. WISE

REPORT NO.

PAGE 8

CHECKED *[Signature]* JUL 30 1973

MODEL

1 MW HEAT EXCHANGER - EVAPORATOR

4. LOSS IN WATER BOX, STATION 2 TO 3

$$V = 1.62 \frac{\text{FT}}{\text{sec}} \quad L = 7.4 \text{ FT} \quad D = 12' \quad P = 64 \quad \mu = 3.63$$

$$N_{RE} = \frac{VD\rho}{\mu} = \frac{3600 \times 1.62 \times 12 \times 64}{3.63} = 1.234 \times 10^6$$

$$\frac{\epsilon}{D} = .00015 \text{ FROM FIGURE 6} \quad f = .014 \text{ FROM FIGURE 7}$$

$$\Delta P = f \frac{L}{D} \frac{V^2}{2g} = .014 \times \frac{7.4}{12} \frac{(1.62)^2}{64.348} \quad \Delta P = 0.0004 \text{ FT}$$

5. CONTRACTION LOSS, STATION 3

$$ID = 6' \quad A = \pi (6')^2 = 28.27 \text{ FT}^2 \quad Q = 183.6 \frac{\text{FT}^3}{\text{sec}}$$

$$V = \frac{Q}{A} = \frac{183.6}{28.27} \quad V = 6.50 \frac{\text{FT}}{\text{sec}}$$

$$\frac{A_3}{A_2} = \frac{6^2}{12^2} = .25 \text{ FROM FIGURE 8} \quad K = .37$$

$$\Delta P = K \frac{V^2}{2g} = .37 \frac{(6.5)^2}{64.348} \quad \Delta P = 0.2429 \text{ FT}$$

6. FRICTION LOSS, STATION 3 TO 4

$$V = 6.5 \frac{\text{FT}}{\text{sec}} \quad N_{RE} = \frac{3600 \times 6.5 \times 6 \times 64}{3.63} = 2.48 \times 10^6$$

$$L = 1.25' \quad D = 6' \quad \frac{\epsilon}{D} = .00025 \quad f = .015$$

$$\Delta P = f \frac{L}{D} \frac{V^2}{2g} = .015 \times \frac{1.25}{6} \frac{(6.5)^2}{64.348} \quad \Delta P = 0.0021 \text{ FT}$$

7. EXPANSION JOINT, STATION 4 TO 5

$$L = 2' \quad ID_5 = 6' \quad ID_4 = 6'1" = 6.0833' \quad \frac{A_5}{A_4} = .973$$

$$K_E = .01 \quad K_L = .01 \quad K_T = .02 \quad V = 6.5 \frac{\text{FT}}{\text{sec}}$$

USE 1 JOINT PER INCH SO THERE IS 24 CONTRACTIONS & EXPANSIONS. $K = .02 \times 24 = .48$

$$\Delta P = K \frac{V^2}{2g} = .48 \frac{(6.5)^2}{64.348} \quad \Delta P = 0.3152 \text{ FT}$$

FIGURE 6

RELATIVE ROUGHNESS FOR DUCT MATERIAL

D-12

FIGURE 7
FRICTION FACTORS

FIGURE 8:
LOSS FACTORS FOR SUDDEN CONTRACTIONS

PREPARED R. A. WISE

REPORT NO.

PAGE 12

CHECKED AMJ JUL 30 1979

MODEL

1 MW HEAT EXCHANGER -
EVAPORATOR & INST. LOOP

8. FRICTION LOSS, STATION 5 TO 6

$$D=6' \quad L=2' \quad f=.015 \quad V=6.5 \frac{FT}{SEC}$$

$$\Delta P = .015 \frac{2}{6} \frac{(6.5)^2}{64.348} \quad \Delta P = 0.0033 FT$$

9. CONTRACTION LOSS, STATION 6 TO 7

$$L=3.2' \quad D_1=6' \quad D_2=5' \quad \tan \frac{\theta}{2} = \frac{5}{3.2} \quad \theta = 17.8^\circ$$

$$K=.016 \quad V=9.35 FT/SEC$$

$$\Delta P = .016 \frac{(9.35)^2}{64.348} \quad \Delta P = 0.0217 FT$$

10. TOTAL PRESSURE DROP IN MAIN LOOP

$$h_L = .2070 + 6.2484 + .3021 + .0004 + .2429 + .0021 + .3152 + .0033 + .0217$$

$$h_L = 7.3431 FT$$

b. PRESSURE DROP FOR INSTRUMENTED LOOP - EVAPORATOR

FROM FIGURE 1 AND 3 THE PRESSURE DROP FOR

THE INSTRUMENTED LOOP IS $P_{T1} - P_{T14} = K_T Q^2$

K = TOTAL LOSS FACTOR FOR INSTRUMENTED LOOP AND
Q = FLOW RATE THRU INSTRUMENTED LOOP,

P_{T14} = TOTAL PRESSURE AT INST. LOOP OUTLET = $P_{S14} + \frac{V_I^2}{2g}$
 V_I = VELOCITY OF SEAWATER IN INST. LOOP.

$$P_{S14} = P_{S7} \quad P_{T14} = P_{S7} + \frac{V_I^2}{2g} \quad P_{T1} - P_{S7} - \frac{V_I^2}{2g} = K Q^2$$

THE PRESSURE DROP FOR THE MAIN LOOP IS

$$P_{T1} - P_{T7} = h_L \quad P_{T7} = P_{S7} + \frac{V_M^2}{2g} \quad V_M = \text{VELOCITY IN MAIN LOOP}$$

$$P_{T1} - P_{S7} - \frac{V_M^2}{2g} = h_L \quad P_{S7} = P_{T1} - \frac{V_M^2}{2g} - h_L$$

$$P_{T1} - P_{T1} + \frac{V_M^2}{2g} + h_L - \frac{V_I^2}{2g} = K Q^2 \quad K Q^2 + \frac{V_I^2}{2g} = \frac{V_M^2}{2g} + h_L$$

$$V_I = \frac{Q}{A_I} \quad K Q^2 + \frac{Q^2}{2g A_I^2} = \frac{V_M^2}{2g} + h_L$$

$$V_M = 9.35 \quad h_L = 7.3431 FT$$

$$\frac{V_M^2}{2g} + h_L = \frac{(9.35)^2}{64.348} + 7.3431 = 8.7017 FT$$

$$Q = \sqrt{\frac{\frac{V_M^2}{2g} + h_L}{K_T + \frac{1}{2g A_I^2}}}$$

PREPARED R.A. WISE

REPORT NO.

PAGE 13

CHECKED

JUL 30 1979

MODEL

1 MW HEAT EXCHANGER

C. LOSS FACTORS FOR INSTRUMENTED LOOP

USE THE FOLLOWING EQUATIONS TO DETERMINE PRESS. DROP.

$$\Delta P = \frac{K' V^2}{2g} = K Q^2 \quad K = \frac{K'}{2g A^2}$$

$$\Delta P = f \frac{L}{D} \frac{V^2}{2g} = K Q^2 \quad K = \frac{f L D}{2g A^2}$$

1. CONTRACTION LOSS FROM WATER BOX TO TUBES, STATION 1

STATION 1 REFERS TO FIGURE 1

FROM SECTION 1 OF 2 $K' = .37$ $A = .0048604 \text{ FT}^2$

$$K = \frac{.37}{64.348 (.0048604)^2} \quad \underline{\underline{K = 243.40}}$$

2. FRICTION LOSS IN TUBE, STATION 1 TO 2FROM SECTION 2 OF 2 $L = 38.2 \text{ FT}$ $D = .944 \text{ IN}$ $f = .023$ $A = .004604 \text{ FT}^2$

$$K = \frac{f L D}{2g A^2} = \frac{.023 \times 38.2}{\frac{.944}{12}} \quad \underline{\underline{K = 7347.19}}$$

3. CONTRACTION LOSS AT STATION 1' DUE TO INSERTINSERT ID = .88" TUBE ID = .944 $A_I = .004224 \text{ FT}^2$ $\frac{A_I}{A_T} = .869$ FROM FIGURE 8 $K' = .035$

$$K = \frac{.035}{(.004224)^2 \times 64.348} \quad \underline{\underline{K = 30.49}}$$

STATION NO REFERS TO FIGURE 3

$$\text{TOTAL PRESSURE DROP THRU LOOP} = P_{T1} - P_{T14} + (P_{T14'} - P_{T7}) = K_T Q^2 +$$

$$(P_{T14'} - P_{T7})$$

$$= P_{T1} - P_{T7} = K_T Q^2 + P_{T14} - P_{T7} \quad P_{T14'} = P_{S14} + \frac{V_{I2}^2}{2g}; \quad P_{T7} = P_{S7} + \frac{V_{I2}^2}{2g}$$

$$P_{T1} - P_{T7} = K_T Q^2 + P_{S14} + \frac{V_{I2}^2}{2g} - P_{S7} + \frac{V_{I2}^2}{2g}$$

$$P_{S14} = P_{S7}$$

$$P_{T1} - P_{T7} = K_T Q^2 + P_{S7} + \frac{V_{I2}^2}{2g} - P_{S7} - \frac{V_{I2}^2}{2g}$$

$$P_{T1} - P_{T7} = K_T Q^2 + \frac{V_{I2}^2}{2g} - \frac{V_{I2}^2}{2g} = h_L$$

PREPARED R. A. WISE 5/30/79

REPORT NO.

PAGE 14

CHECKED

MAY 27 1980

MODEL

1 MW HEAT EXCHANGER

~~3. LOSS FACTORS FOR INSTRUMENTATION LOOP~~~~USE THE FOLLOWING EQUATIONS TO DETERMINE PRESSURE DROP:~~

~~$$\Delta P = \frac{K' V^2}{2g} = \frac{K' Q^2}{A^2 2g} = K Q^2$$~~

~~$$K = \frac{K'}{A^2 2g}$$~~

~~$$\Delta P = f \frac{L}{D} \frac{V^2}{2g}$$~~

~~$$K = \frac{fL}{D A^2 2g}$$~~

4. EXPANSION LOSS AT STATION 1'

$$\frac{A_1}{A_2} = \left(\frac{0.944}{1.25} \right)^2 = 0.57 \quad K' = 0.18$$

$$A = \frac{\pi (0.944)^2}{4 \times 144} = 0.00486 \text{ FT}^2$$

$$K = \frac{0.18}{(0.00486)^2 \times 64.348}$$

$$K = 118.43$$

5. FRICTION LOSS, STATION 1' TO 3'

FROM FIGURE 3

$$L = 8.6'$$

FROM TABLE 1

$$Q = 156 \text{ GPM}$$

$$\Delta P = 2.45 \text{ PSI}$$

$$= 0.0334 \text{ FT}^3/\text{sec}$$

$$100 \text{ FT.}$$

$$\Delta P = \frac{5.54}{100} \times 8.6 = K (0.0334)^2$$

$$K = 426.85$$

5.54 $\frac{\text{ft}}{100 \text{ ft}}$

6. BEND LOSS AT STATION 2'

FROM FIGURE 9

$$\text{FOR } \frac{R}{D} = 2$$

$$K' = 0.24$$

$$A = 0.008522$$

$$K = \frac{K'}{A^2 2g} = \frac{0.24}{(0.008522)^2 \times 64.348}$$

$$K = 51.35$$

FIGURE 9
LOSS FACTORS FOR 90° ROUND BENDS

PRESSURE LOSS IN HOSE

TABLE I

Page 16

WATER FLOW PRESSURE LOSS (PSI Per 100 Feet of Hose)

FLOW OF WATER IN U.S. GAL. PER MIN.	FLOW OF WATER IN CU. FEET PER SEC.	ACTUAL INTERNAL DIAMETER, INCHES								
		1/4	3/8	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2
0.5	.001	0.4								
1.5	.003	3.02	1.01	0.42						
2.5	.005	7.75	2.58	1.08						
5	.011	27.8	9.27	3.86	0.95	0.32	0.13			
10	.022	99.5	33.2	13.8	3.38	1.14	0.47	0.12		
15	.033		71.0	29.6	7.25	2.45	1.01	0.25	0.08	
20	.044		121.0	50.3	12.4	4.15	1.71	0.42	0.14	
25	.055			76.5	18.7	6.34	2.60	0.64	0.22	
30	.066			108.0	26.5	8.96	3.68	0.90	0.30	0.13
35	.077			142.0	34.8	11.8	4.83	1.18	0.40	0.17
40	.088				44.7	15.1	6.20	1.52	0.51	0.21
45	.099				55.0	18.6	7.65	1.87	0.63	0.26
50	.110				67.5	22.8	9.35	2.28	0.78	0.32
60	.132				94.3	31.8	13.1	3.19	1.08	0.45
70	.154				126.0	42.5	17.5	4.25	1.44	0.60
80	.176					54.6	22.5	5.48	1.86	0.77
90	.198					67.5	27.8	6.80	2.30	0.95
100	.223					81.5	33.5	8.19	2.78	1.15
125	.278					124.0	50.6	12.4	4.20	1.73
150	.334						72.1	17.6	6.97	2.46
175	.390						94.5	23.1	7.83	3.23
200	.446						122.0	29.6	10.1	4.15
225	.501							36.8	12.5	5.15
250	.557							44.6	15.2	6.28
275	.613							53.3	18.1	7.45
300	.688							62.5	21.2	8.75
325	.724							72.5	24.6	10.2
350	.780							83.2	28.2	11.7
375	.836							94.5	32.1	13.3
400	.891							107.0	36.2	14.9
450	1.00								44.9	18.6
500	1.11								54.5	22.5
600	1.34								76.5	31.6
700	1.56								102.0	42.1
800	1.78								131.0	53.9
900	2.00									66.8
1000	2.23									81.4
1100	2.45									97.0
1200	2.67									114.0
1300	2.90									132.0
1400	3.12									
1500	3.34									
1600	3.56									
1800	4.01									
2000	4.45									

NOTE: The pressure loss experienced by a liquid flowing through a hose depends on the rate of flow, the viscosity of the liquid, the hose I.D., the smoothness of the tube, and the hose length. This chart shows the relationship between rate of flow, I.D., and pressure loss for water at 68° F with a vis-

cosity of one centipoise. The pressure loss is directly proportional to the length of the hose, therefore, the data shown can be easily extended by use of proportions, e.g., the pressure drop for 50 feet of hose length is half that for 100 feet.

PREPARED R.A. WISE 5/31/79

REPORT NO.

PAGE 17

CHECKED

MODEL

1 MW HEAT EXCHANGER

7. FRICTION LOSS, STATION 3' TO 4'

$$L = 2' \quad D = 1.25" \quad \phi = 15 \text{ GPM} = 0.0334 \text{ FT}^3/\text{sec}$$

$$\text{FROM TABLE 2} \quad \Delta P = 4.45 \frac{\text{FT}}{100 \text{ FT}}$$

$$\Delta P = \frac{4.45}{100} \times 1 = K (0.0334)^2 \quad \underline{K = 39.89} \quad (f = 0.0194)$$

8. LOSS IN 1 1/4" BALL VALVE, STATION 4' TO 5'

$$K' = .30 \quad A = 0.008522 \text{ FT}^2 \quad \text{FOR } 1.25" \text{ VALVE}$$

FROM HILLS-MCCANNA

$$K = \frac{.3}{(0.008522)^2 \times 64.4} \quad \underline{K = 64.19}$$

9. LOSS IN TEE, STATION 5' TO 6'

$$\text{FROM FIGURE 9 FOR } \frac{R'}{D} = .5 \quad K' = .88$$

$$K = \frac{.88}{(0.008522)^2 \times 64.4} \quad \underline{K = 188.31}$$

10. FRICTION LOSS, STATION 6' TO 7'

THIS IS THE SAME AS TWICE ITEM 7

$$\underline{K = 79.78}$$

11. LOSS IN ORIFICE, STATION 7' TO 8'

USE AN AREA RATIO OF 0.64

$$\text{FROM FIGURE 10} \quad K' = 0.70 \quad C = .765$$

(THIS WILL GIVE A VENA CONTRACTA LOSS OF ABOUT 12")

$$\text{ORIFICE SIZE} = 1.0" \quad A = .005454 \text{ FT}^2$$

$$K = \frac{.70}{(.005454)^2 \times 64.4} \quad \underline{K = 365.69}$$

TABLE 2
Friction loss in plastic pipe schedule 80

Page 18

Velocity measured in ft./sec., Loss in feet of water head per 100 ft. of pipe.

GALLS PER MIN.		1/2"		3/4"		1"		1 1/4"		1 1/2"		2"		2 1/2"		3"		3 1/2"		4"	
Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss
2	2.74	6.72	1.08	1.51																	
4	5.48	24.2	2.97	5.45	1.79	1.94	1.00	.99	.75	.577											
6	8.23	51.2	4.45	11.5	2.68	3.94	1.50	.82	1.08	.575	.85	.307									
8	11.0	86.9	5.94	19.6	3.57	5.69	2.00	1.39	1.45	.84	.87	.385	.41	.877							
10	13.7	132.0	7.42	29.6	4.46	8.60	2.50	2.10	1.82	.96	1.09	.376	.76	.315	.485	.879					
12			8.91	41.5	5.34	12.0	3.00	2.94	2.18	1.55	1.30	.987	.91	.361	.572	.855					
15			11.1	52.7	6.7	27.4	3.7	2.45	2.77	2.4	1.85	.985	1.14	.343	.727	.685	.94	.855			
18			13.4	87.9	8.05	25.5	4.50	6.75	3.27	2.82	1.96	.818	1.36	.340	.873	.316	.45	.856			
20			14.8	107	9.02	30.9	5.00	7.57	3.95	3.47	2.17	.996	1.51	.414	.97	.340	.72	.868	.96	.857	
25					11.2	54.8	6.25	11.4	4.55	5.25	2.71	1.51	1.9	.825	1.21	.312	.90	.303	.495	.855	
30	.53	.825			13.4	65.3	7.50	16.0	5.45	7.58	3.26	2.11	2.27	.874	1.44	.297	1.08	.345	.84	.877	
35	.62	.874			15.6	86.9	8.75	21.5	6.58	9.78	3.80	2.81	2.45	1.16	1.70	.396	1.26	.392	.975	.303	
40	.71	.845			17.9	111	10.0	27.5	7.26	12.5	4.35	3.99	3.05	1.49	1.94	.507	1.44	.346	1.12	.332	
45	.795	.854																			
50	.88	.865	.42	.827	11.2	55.9	8.26	15.6	4.89	4.46	3.41	1.86	2.18	.429	1.63	.306	1.25	.344			
55	.975	.878	.476	.832	12.5	61.5	9.08	18.9	5.45	5.41	3.79	2.25	2.42	.766	1.80	.372	1.40	.399			
60	1.06	.891	.74	.839	13.7	69.2	10.00	22.0	5.98	6.44	4.16	2.66	2.67	.912	1.99	.445	1.55	.357			
65	1.15	.906	.80	.844	15.0	57.8	10.9	26.5	6.52	7.61	4.54	3.16	2.92	1.07	2.17	.522	1.67	.379			
70	1.23	.921	.86	.851	16.1	67.0	11.8	30.7	7.06	8.84	4.92	3.66	3.14	1.25	2.35	.604	1.81	.325			
75	1.33	1.38	.925	.857	17.5	77.1	12.7	35.5	7.61	10.1	5.30	4.20	3.99	1.45	2.55	.691	1.95	.371			
80	1.41	1.155	.98	.865	18.8	87.4	13.6	40.1	8.15	11.5	5.66	4.79	3.64	1.62	2.70	.787	2.08	.421			
85	1.50	1.174	1.04	.872	20.0	98.2	14.5	45.2	8.69	12.9	6.05	5.36	3.88	1.83	2.89	.886	2.25	.477			
90	1.59	1.193	1.11	.880	21.2	110	15.4	50.5	9.03	14.5	6.45	6.02	4.10	2.04	3.05	.962	2.34	.531			
95	1.67	1.213	1.20	.889	22.5	122	16.5	55.9	9.78	16.1	6.81	6.55	4.35	2.27	3.25	1.10	2.51	.582			
100	1.76	1.24	1.23	.898					10.9	18.6	7.57	8.13	4.85	2.76	3.67	1.34	2.79	.719			
110	1.95	1.29	1.36	.917	18.2	68.2			12.0	23.4	8.35	9.68	5.35	3.29	3.97	1.60	3.07	.855			
120	2.11	1.329	1.48	.937	20.0	81.5			13.0	27.4	9.08	11.4	5.80	3.87	4.35	1.88	3.35	1.00			
130	2.3	1.381	1.60	.959	21.8	95.4			14.1	31.8	9.84	13.2	6.30	4.46	4.69	2.18	3.65	1.16			
140	2.47	1.437	1.72	.982	23.4	111			15.2	36.5	10.6	15.1	6.80	5.12	5.05	2.50	3.91	1.35			
150	2.65	1.496	1.85	.997	25.4	127			16.5	41.5	11.5	17.2	7.27	5.87	5.41	2.84	4.19	1.52			
160	2.82	1.559	1.97	1.02					17.4	46.7	12.1	19.4	7.75	6.58	5.78	3.20	4.47	1.71			
170	3.0	1.626	2.08	1.04	17.4	52.2			18.5	52.2	12.9	21.7	8.20	7.37	6.14	3.58	4.75	1.91			
180	3.16	1.696	2.22	1.06	20.0	61.5			19.6	58.5	13.6	24.2	8.60	8.18	6.50	3.97	5.02	2.12			
190	3.36	1.769	2.34	1.08	20.6	64.4			20.6	64.4	14.4	26.6	9.20	9.05	6.85	4.39	5.30	2.35			
200	3.52	1.846	2.46	1.11	21.7	70.5			21.7	70.5	15.1	29.5	9.70	9.96	7.22	4.84	5.58	2.58			
220	3.68	1.91	2.71	1.15	23.9	84.1			23.9	84.1	16.7	34.9	10.6	11.9	7.94	5.78	6.14	3.08			
240	4.23	1.18	2.96	1.24	26.1	98.7			26.1	98.7	18.2	41.0	11.6	15.9	8.66	6.77	6.70	3.62			
260	4.58	1.37	3.20	1.31	28.3	115			28.3	115	19.7	47.5	12.6	18.2	9.38	7.85	7.24	4.19			
280	4.94	1.57	3.45	1.38					21.2	54.5	13.5	18.6	10.1	9.02	7.82	7.82	4.79				
300	5.29	1.79	3.69	1.47					22.7	62.0	14.4	21.1	10.8	10.2	8.38	8.38	5.45				
320	5.64	2.01	3.94	1.54					24.2	69.9	15.5	25.7	11.5	11.5	8.94	8.94	6.16				
340	5.99	2.24	4.19	1.60					25.8	78.2	16.5	28.6	12.5	12.9	9.50	6.91					
360	6.35	2.51	4.45	1.65					27.2	86.9	17.4	29.5	13.0	14.3	10.0	7.66					
380	6.70	2.77	4.68	1.71					28.8	96.1	18.6	32.6	13.7	15.8	10.6	8.46					
400	7.05	3.05	4.93	1.77					30.5	106	19.4	35.9	14.4	17.4	11.2	9.31					
450	7.95	3.79	5.54	1.98							21.8	44.6	16.2	21.6	12.5	11.6					
500	8.87	4.61	6.16	2.21							23.2	54.1	18.1	26.5	14.0	14.1					
550	9.70	5.50	6.77	2.49							24.5	64.9	19.9	31.4	15.5	16.8					
600	10.6	6.44	7.39	2.69																	
650	11.5	7.47	8.00	3.12																	
700	12.5	8.60	8.65	3.58																	
750	13.2	9.77	9.24	4.07																	
800	14.1	11.0	9.85	4.58																	
850	15.0	12.5	10.5	5.12																	
900	15.9	13.7	11.1	5.69																	
950	16.7	15.1	11.7	6.29																	
1000	17.6	16.6	12.3	6.91																	
1100	19.4	19.8	13.5	8.27																	
1200	21.1	23.3	14.8	9.75																	
1300				11.3																	
1400																					
1500																					
1600																					
1700																					
1800																					
1900																					
2000																					
2200																					
2400																					
2600																					
2800																					
3000																					
3200																					
3400																					
3600																					
3800																					
4000																					
4200																					
4400																					
4600																					
4800																					
5000																					
5200																					
5400																					
5600																					
5800																					
6000																					
6200																					
6400																					

*Data shown is calculated from Williams and Hazen formula $H = \frac{2.025}{C^{1.852}} \frac{V^{1.852}}{D^{4.87}} L$ using C=150. For water at 60°F. Where H = head loss, V = fluid velocity ft./sec., D = diameter of pipe, ft.
C = coefficient representing roughness of pipe interior surface.

FIGURE 10
LOSS FACTOR FOR FLAT PLATE ORIFICES

PREPARED R. A. WISE 5/31/79
 CHECKED ML MAY 27 1980
 MODEL _____

REPORT NO.

PAGE 20

1 MW HEAT EXCHANGER

12. FRICTION LOSS, STATION 8' TO 9'

THIS IS THE SAME AS ITEM 10.

$K = 79.78$

13. FRICTION LOSS, STATION 9' TO 11'

$L = 15'$ FROM FIGURE 3

FROM TABLE 1 FOR $Q = 15 \text{ GPM} = .0334 \frac{\text{FT}^3}{\text{SEC}}$ $\Delta P = 2.45 \frac{\text{PSI}}{100 \text{ FT}}$ 5.54 ft

$\Delta P = K Q^2$ $5.54 \times \frac{15}{100} = K (.0334)^2$ $K = 744.51$

14. FRICTION LOSS, STATION 11' TO 14'

THIS IS THE SAME AS ITEM 4 $K = 39.89$

15. BEND LOSS AT STATION 10'

THIS IS THE SAME AS ITEM 6 $K = 51.35$

16. LOSS IN 1 1/4" BALL VALVE, STATION 11' TO 12'

THIS IS THE SAME AS ITEM 8

$K = 64.19$

16. TOTAL LOSS FACTOR

$K_T = 243.4 + 7347.19 + 30.49 + 118.43 + 426.85 + 51.35 +$
 $39.89 + 64.19 + 188.31 + 79.78 + 365.69 + 79.78 +$
 $744.51 + 39.89 + 51.35 + 64.19$

$K_T = 9935.21$

d. FLOW RATE THRU TUBE-EVAPORATOR

FROM SECTION b - $\frac{V_m^2}{2g} + h_L = 8.7017 \text{ FT}$ $D_I = 1.25"$

$A_I = .008522 \text{ FT}^2$ $\frac{1}{2g A_I^2} = 214$ $K_T + \frac{1}{2g A_I^2} = 10,149.29$

$Q = \sqrt{\frac{V_m^2 + h_L}{K_T + \frac{1}{2g A_I^2}}} = \sqrt{\frac{8.7017}{10,149.29}} = .02928 \frac{\text{FT}^3}{\text{SEC}}$ $Q = 13.14 \text{ GPM}$

PREPARED R.A. WISE 6/18/79

REPORT NO.

PAGE 21

CHECKED [Signature] JUL 30 1979

MODEL

OTEC-1MW - MAXIMUM
CLOSING ANGLE OF BALL VALVE

E. MAXIMUM CLOSING ANGLE OF BALL VALVES

USE A 1 1/4" BALL VALVE

HEIGHT & LENGTH OF HOLE = 1.278"

MINIMUM HEIGHT TO PASS AMCTAP BALL = .944"

$$L = .639 \tan \theta \quad h = 1.278 - L$$

$$\frac{.944}{h} = \cot \theta \quad .944 = (1.278 - .639 \tan \theta) \cot \theta$$

$$1.278 \cot \theta - .639 \sin \theta = .944 \quad \underline{\underline{\theta = 22.1^\circ}}$$

$$L = .2595$$

$$h = 1.0185$$

$$\cot \frac{\theta}{2} = \frac{.305}{.639}$$

$$\theta = 123^\circ$$

$$A_s = \text{AREA OF SHADOWED AREA} = \frac{\pi r^2 \theta}{360} - \frac{r^2 \sin \theta}{2}$$

$$A_s = \frac{\pi (.639)^2 \times 123}{360} - \frac{(.639)^2 \sin 123}{2}$$

$$A_s = 0.267 \text{ IN}^2$$

PREPARED R. A. WISE

REPORT NO.

PAGE 22

CHECKED AMJ JUL 30 1979

MODEL

$$A_T = \text{AREA OF TOTAL CIRCLE} = \pi r^2$$

$$A_T = \pi (.639)^2 \quad A_T = 1.283 \text{ IN}^2$$

$$\text{AREA OF FLOW} = A_T - A_S = A_F = 1.283 - .267$$

$$A_F = 1.016 \text{ IN}^2 \quad \text{PROTECTED AREA} = 1.016 \cos 22.1 = A_P = .9414$$

$$A_L = \text{AREA OF LINE} = \pi \left(\frac{1.25}{4} \right)^2 = 1.2272$$

$$\frac{A_P}{A_L} = \frac{.9414}{1.2272} = .767$$

$$\frac{A_F}{A_T} = \frac{1.016}{1.283} = .792$$

K_C = CONTRACTION LOSS FROM FIGURE 1

K_E = EXPANSION LOSS FROM FIGURE 2

$$K_C = .08 \quad K_{E1} = .04 \quad K_{C2} = .06 \quad K_{E2} = .05$$

FROM FIGURE 3 AND FIGURE 4 FOR

$$R'/D = .5 \quad \text{AND} \quad \theta = 22.1^\circ$$

$$K_1 = .88 \times .4 = 0.352$$

$$K_0 = .88 \times .4 = 0.352$$

$$K_T = K_C + K_1 + K_{C2} + K_{E1} + K_0 + K_{E2} = .08 + .352 + .04 + .06 + .352 + .05$$

$$\underline{K_T = .938}$$

THIS IS THE INCREASE IN
LOSS FACTOR THRU THE VALVE
FOR THE MAXIMUM ALLOWABLE
CLOSING ANGLE OF 22.1° . ADD
THIS TO THE FULLY OPEN LOSS
FACTOR OF 0.30

$$K = \frac{.938}{(.008522)^2 64.348}$$

$$\underline{K = 200.72}$$

FIGURE 1
LOSS FACTORS FOR SUDDEN CONTRACTIONS

FIGURE 2
LOSS FACTORS FOR SUDDEN EXPANSIONS

FIGURE 3
LOSS FACTORS FOR 90° ROUND BENDS

FRACTION OF 90° BEND - K

1.4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0

0 20 40 60 80 100 120 140 160 180

BEND ANGLE - θ

FIGURE 4

CORRECTION FACTORS FOR ROUND
BENDS OTHER THAN 90°

PREPARED R. A. WISE

REPORT NO.

PAGE 27

CHECKED

ALY

JUL 30 1979

MODEL

FLOW RATE WITH MAXIMUM
CLOSING ANGLE ON BALL VALVE -
EVAPORATOR

F. FLOW RATE WITH BALL VALVES CLOSED TO MAXIMUM-EVAP.

$$K_T = 9282.23 \quad (\text{Page 20 ITEM 16})$$

SINCE THERE ARE 2 BALL VALVES ADD
 $2 \times 200.72 = 401.44$ TO K_T

$$K_T' = 9282.23 + 401.44 \quad \underline{K_T' = 9683.67}$$

$$K_T' + \frac{1}{2gA_f^2} = 9603.87 + \frac{1}{(0.008012)^2 \times 64.348} = 9897.65$$

$$\frac{V_m^2}{2g} + h_L = 8.7017$$

$$Q = \sqrt{\frac{8.7017}{9897.65}} = .02965 \frac{\text{FT}^3}{\text{SEC}} \quad \underline{Q = 13.31 \text{ GPM}}$$

PREPARED R.A. WISE 7-30-79 REPORT NO.

PAGE 28

CHECKED _____

MODEL _____

1 MW HEAT EXCHANGERS -
CONDENSERG. PRESSURE DROP IN MAIN PATH - CONDENSER1. CONTRACTION FROM WATER BOX TO TUBES, STATION 1
STATION NO. REFERS TO FIGURE 2

TUBE ID = .944" NO. OF TUBES = 5526

$$\text{AREA OF TUBES} = A_T = \frac{\pi (.944)^2}{4 \times 144} \times 5526 = 26.86 \text{ FT}^2$$

WATER BOX ID = 9' AREA OF WATER BOX = $A_W = 63.62 \text{ FT}^2$

$$\frac{A_T}{A_W} = \frac{26.86}{63.62} = .422 \text{ FROM FIGURE 8 } K = .29$$

 $Q = 12.32 \text{ GPM} = .02745 \frac{\text{FT}^3}{\text{SEC}}$ $A = \text{AREA OF TUBE}$

$$A = \frac{\pi (.944)^2}{4 \times 144} = .0048604 \text{ FT}^2 \quad V = \frac{Q}{A} = \frac{.02745}{.0048604} = 5.65 \frac{\text{FT}}{\text{SEC}}$$

$$\Delta P = \frac{KV^2}{2g} = \frac{.29(5.65)^2}{64.348} \quad \underline{\underline{\Delta P = 0.1439 \text{ FT}}}$$

2. FRICTION LOSS IN TUBE, STATION 1 TO 2 $L = 41.4 \text{ FT}$ $D = .944"$ $E = 64 \frac{\text{#}}{\text{FT}}$ $\gamma = 3.63 \frac{\text{#}}{\text{FT}^3 \text{ - H}_2\text{O}}$

$$V = 5.65 \frac{\text{FT}}{\text{SEC}} \quad \text{NRE} = \frac{3600 \times 5.65 \times .944 \times 64}{12 \times 3.63} = 2.82 \times 10^4$$

FROM FIGURE 6 $\frac{E}{D} = .00006$ FROM FIGURE 7 $f = .024$

$$\Delta P = f \frac{L}{D} \frac{V^2}{2g} = .024 \times \frac{41.4}{.944} \frac{(5.65)^2}{64.348} \quad \underline{\underline{\Delta P = 6.2659 \text{ FT}}}$$

3. EXPANSION FROM TUBE TO WATER BOX, STATION 2FROM SECTION 1 $\frac{A_T}{A_W} = .422$ FROM FIGURE 4 $K = .35$ $V = 5.65 \text{ FT/SEC}$

$$\Delta P = .35 \frac{(5.65)^2}{64.348} \quad \underline{\underline{\Delta P = .1736 \text{ FT}}}$$

PREPARED R. A. WISE

REPORT NO.

PAGE 29

CHECKED _____

MODEL _____

1 MW HEAT EXCHANGER -
CONDENSER

4. LOSS IN WATER BOX, STATION 2 TO 3

$$D = 9' \quad A = 63.62 \text{ FT}^2 \quad Q = 68080.32 \text{ GPM} = 151.69 \frac{\text{FT}^3}{\text{SEC}}$$

$$V = \frac{Q}{A} = 2.384 \frac{\text{FT}}{\text{SEC}} \quad N_{RE} = \frac{3600 \times 2.384 \times 9 \times 64}{3.63} = 1.36 \times 10^6$$

$$\frac{E}{D} = .000017 \text{ FROM FIGURE 6} \quad f = .012 \text{ FROM FIGURE 7}$$

$$L = 4' \quad \Delta P = f \frac{L}{D} \frac{V^2}{2g} = .012 \times \frac{4}{9} \frac{(2.384)^2}{64.348} \quad \Delta P = 0.0005 \text{ FT.}$$

5. CONTRACTION LOSS, STATION 3

$$D_1 = 9' \quad D_2 = 6' \quad \frac{A_2}{A_1} = .444 \text{ FROM FIGURE 8} \quad K = .275$$

$$A_2 = 28.274 \text{ FT}^2 \quad Q = 151.69 \frac{\text{FT}^3}{\text{SEC}} \quad V = \frac{Q}{A} = 5.365 \frac{\text{FT}}{\text{SEC}}$$

$$\Delta P = K \frac{V^2}{2g} = .275 \frac{(5.365)^2}{64.348} \quad \Delta P = 0.1230 \text{ FT.}$$

6. FRICTION LOSS STATION 3 TO 5

$$D = 6' \quad L = 2.12' \quad V = 5.365 \frac{\text{FT}}{\text{SEC}} \text{ FROM SECTION 5}$$

$$\frac{E}{D} = .000025 \text{ FROM FIGURE 6}$$

$$N_{RE} = \frac{3600 \times 5.365 \times 6 \times 64}{3.63} = 2.04 \times 10^6 \text{ FROM FIGURE 7} \quad f = .0115$$

$$\Delta P = f \frac{L}{D} \frac{V^2}{2g} = .0115 \times \frac{2.12}{6} \frac{(5.365)^2}{64.348} \quad \Delta P = 0.0018 \text{ FT.}$$

7. CONTRACTION LOSS, STATION 5 TO 6

$$L = 4.8' \quad D_1 = 6' \quad D_2 = 5' \quad \tan \theta = \frac{5}{4.8} \quad \theta = 11.9^\circ$$

$$\text{FROM FIGURE 5} \quad K = .006 \quad Q = 151.69 \frac{\text{FT}^3}{\text{SEC}} \quad A = 19.63 \text{ FT}^2$$

$$V = \frac{Q}{A} = \frac{151.69}{19.63} \quad V = 7.726 \frac{\text{FT}}{\text{SEC}}$$

$$\Delta P = K \frac{V^2}{2g} = .006 \frac{(7.726)^2}{64.348} \quad \Delta P = 0.0056 \text{ FT.}$$

PREPARED R. A. WISE

REPORT NO.

PAGE 30

CHECKED

MAY 27 1980

1 MW HEAT EXCHANGER -
CONDENSER

MODEL

8. FRICTION LOSS, STATION 6 TO 7

$$D = 5' \quad V = 7.726 \frac{FT}{SEC} \text{ FROM SECTION 7} \quad L = 5.2'$$

$$\frac{\Sigma}{D} = .00003 \text{ FROM FIGURE 6}$$

$$N_{RE} = \frac{3600 \times 7.726 \times 5 \times 64}{3.63} = 2.45 \times 10^6$$

$$\text{FROM FIGURE 7} \quad f = .011$$

$$\Delta P = f \frac{L}{D} \frac{V^2}{2g} = .011 \times \frac{5.2}{5} \frac{(7.726)^2}{64.348} \quad \underline{\underline{\Delta P = .0106 FT}}$$

9. TOTAL PRESSURE DROP IN MAIN LOOP

$$h_L = .1439 + 6.2659 + .1736 + .0005 + .1230 + .0018 + .0056 + .0106$$

$$\underline{\underline{h_L = 6.7249 FT.}}$$

H. TOTAL LOSS FACTOR

$$\text{FROM SECTION K-16} \quad \underline{\underline{K_T = 9282.23}}$$

I. FLOW THRU TUBE-CONDENSER

$$V_M = 7.726 \frac{FT}{SEC} \quad \frac{V_M^2}{2g} + h_L = \frac{(7.726)^2}{64.348} + 6.7249$$

$$\frac{V_M^2}{2g} + h_L = 7.6525 FT$$

$$K_T + \frac{1}{2gA_z^2} = 9935.21 + \frac{1}{64.348(0.008522)^2} = 10,149.29$$

$$Q = \sqrt{\frac{\frac{V_M^2}{2g} + h_L}{K_T + \frac{1}{2gA_z^2}}} = \sqrt{\frac{7.6525}{10,149.29}} = .02746 \frac{FT}{SEC}$$

$$\underline{\underline{Q = 12.32 GPM}}$$

PREPARED R. A. WISE

REPORT NO.

PAGE 31

CHECKED _____

MODEL _____

1 MW HEAT EXCHANGER -
CONDENSER

J. MAXIMUM CLOSING ANGLE OF BALL VALVES

FROM SECTION E K = 401.44

K. FLOW RATE WITH BALL VALVES CLOSED TO MAXIMUM-COND.

$$K_T' = 9282.23 + 401.44 = 9683.67$$

$$K_T' + \frac{1}{2gA_f^2} = 9897.65 \quad \frac{V_m^2}{2g} + h_L = 7.6525$$

$$Q = \sqrt{\frac{7.6525}{9897.65}} = .02781 \frac{FT^3}{sec} \quad \underline{\underline{Q = 12.48 GPM}}$$

PREPARED A.M. SPROUSE

REPORT NO. 377323

PAGE 1

CHECKED _____

MODEL LMWb HX's

INSTRUMENTED LOOPS ORIFICE
PLATE ACCURACY

1. PURPOSE: Determine if drop across meter run is great enough to allow desired accuracy.
2. DATA: Orifice size: 0.94"
Line size: 1.38" I.D. (1 1/4" Sch 40)
Flow rate: 13.1 GPM
Desired accuracy: ± 0.05 GPM
Accuracy of transmitter: 0.2% of full span
Span of transmitter: 0-5 or 0-30" H₂O

3. DROP ACROSS METER:

$$q = CA \sqrt{2gh_L} \quad , \quad C = f(R_e, \beta)$$

$$h_L = \left(\frac{q}{CA} \right)^2 \frac{1}{2g}$$

$$R_e = \frac{\rho V d}{\mu} = \frac{63.78 \cdot 2.81 \cdot 3600}{2.304 \cdot 12} = 23,336$$

$$V = \frac{13.1 \times 2.228 \times 10^{-3}}{(1.38/12)^2 (\pi/4)} = 2.81 \text{ ft/sec}$$

$$\beta = 0.68$$

FROM A-19, CRANE MANUAL:

$$C = 0.70$$

$$h_L = \left(\frac{2.81}{0.7} \right)^2 \frac{1}{64.4}$$

$$h_L = 0.25 \text{ ft} = 0.11 \text{ psi}$$

RECOVERY @ PIPE TAPS WILL BE ABOUT 46%
(Chemical Engineering 2/3/75, p. 75)

$$\therefore \text{Measured } \Delta p = 0.54 \times 0.11 = 0.0598 \text{ psi}$$

PREPARED A. M. SPROUSE

REPORT NO.

PAGE 2

CHECKED _____

MODEL 1MV6 HX 5

INSTRUMENTED LOOPS ORIFICE
PLATE ACCURACY

4. Δp METER DEFLECTION

$$\text{ACCURACY} = 0.0025 \times 5 = 0.0125'' \text{ H}_2\text{O} = 0.000452 \text{ psi}$$

$$\text{SIGNAL @ 13.1 GPM} = 0.06 \text{ psi} = 1.66'' \text{ H}_2\text{O}$$

$$\Delta \text{flow of } 0.05 \text{ GPM} = 13.15 \text{ GPM}$$

$$V = \frac{13.15 \times 2.228 \times 10^{-3}}{(1.38/12)^2 (11/4)} = 2.82 \text{ ft/sec}$$

$$h_L = \left(\frac{2.82}{0.7} \right)^2 \frac{1}{64.4} = 0.252 \text{ ft} = 0.111 \text{ psi}$$

$$\text{Overall loss } 0.111 \times 0.54 = 0.0602 \text{ psi} = 1.667'' \text{ H}_2\text{O}$$

$$\therefore \text{Change in meter reading is}$$

$$1.6674 - 1.6546 = 0.0128'' \text{ H}_2\text{O}$$

5. CONCLUSION:

The meter is accurate enough to
measure changes of $\pm 0.05 \text{ GPM}$

$$'' \text{ H}_2\text{O} \times 0.03613 = \text{psi}$$

APPENDIX E

MATERIALS AND PROCESSES DESIGN

DEFENSE AND SPACE SYSTEMS GROUP
ONE SPACE PARK - REDONDO BEACH - CALIFORNIA 90278

INTEROFFICE CORRESPONDENCE

5515.2.79-153

TO: J. E. Snyder

CC: R. H. Douglass
T. C. Dvorak
J. S. Hicks
L. J. Meyers
W. I. Rogers

DATE: 2 April 1979

M. H. Seidman
A. M. Sprouse

SUBJECT: Final Report on the Tube-to-Tubesheet Weld
Development Program - 1 MWe OTEC

FROM: L. A. Rosales

BLDG.
01

MAIL STA.
2220

EXT
52630

Reference: (1) IOC 5515.2.78-336, "Status of Tube-to-Tubesheet Weld Development Program-
1 MWe", L. A. Rosales to J. E. Snyder, 11 July 1978.

INTRODUCTION

The joint TRW/SECO Weld Development Program as described in Reference 1 has been completed. Tests which have been performed since the first status report are (refer to Table I) Test No. 7, "Schedule Qualification", Test No. 8, "Weld Strength Determination", and Test No. 9, "Leak Test". In addition, weld repair and tube plugging methods were established and the documentation completed.

SUMMARY OF PREVIOUS RESULTS

At the conclusion of tests 1 through 6, the following had been established⁽¹⁾:

- a) A candidate weld schedule was established.
- b) A joint preparation and fitup procedure was established. This consisted of an abrasive oxide removal and solvent wipe of the tube end and hole, and a light pin-rolling to obtain a snug fit between tube and hole.
- c) The use of cotton plugs in the tube bore was found to be effective in reducing contamination.
- d) The use of semi-automatic hand-held TIG welders fitted with a special positioning collet was found satisfactory for accomplishing good welds.
- e) The tube joint geometry has been established. A flush tube end-to-clad surface has been selected.
- f) The minimum acceptable titanium cladding thickness required for high quality welds was established as .188 inch.

Table I TEST JOINT CONFIGURATION - PLAN

TEST NO.	WELD CONFIGURATION	TITANIUM CLADDING THICKNESS	VISUAL/ DYE PENETRANT	INSPECTION	
				MACROGRAPHIC	MICROGRAPHIC
1	Flush Tubes	.250 inch	X	X	X
2	Tube Recessed	.250 inch	X	X	X
3	Tube Protruding	.250 inch	X	X	X
4	Optimum of (1 to 3)	.125 inch	X	X	X
5	Optimum of (1 to 3)	.188 inch	X	X	X
6	Optimum of (1 to 3)	.250 inch	X	X	X
7	Optimum of (1 to 6)		X	X	X
8	Optimum of (1 to 6)		X	X	X
9	Optimum of (1 to 6)		X	X	X

Test No. 1 through 3 - Weld Schedule Development

Test No. 4 through 6 - Titanium Cladding Thickness Optimization

Test No. 7 - Schedule Qualification

Test No. 8 - Weld Strength Determination

Test No. 9 - Leak Test

The approximate number of welds anticipated are:

Test 1 through 3 - 10-15 welds each

Test 4 through 9 - 10 welds each

WELD SCHEDULE QUALIFICATION

In order to establish the reproducibility of high quality welds obtained with the candidate weld schedule, a total of 52 welds were made on a test plate having a .250 inch thickness cladding of titanium. Visual inspection of the welds showed excellent weld appearance. No evidence of contamination, such as discoloration, pin holes, or burn-back was noted. Figure 1 shows some of the welds. Dye penetrant examination showed no evidence of surface defects such as pores or cracks. The welds were sectioned, polished and etched to reveal the microstructure of the titanium. Typical micrographs are shown in Figures 2 through 4. No evidence of oxide contamination of the interface was noted although approximately one half of the examined portion of the welds exhibited small spherical porosity, (typically less than .004 inch diameter), Figure 3. The position of the pores was random although most were found in the area where fusion of the tube and cladding begins, see Figure 4. The microstructure of the weld and heat affected zone (HAZ) was normal.

Microhardness measurements were taken across the tube-to-tubesheet weld, Figure 5. No evidence of contamination at the interface was noted.

The influence of time between joint preparation and welding was investigated as part of the qualification. Seven tube joints were abrasively cleaned, solvent wiped, assembled, and set aside for 60 days. The joints were then given a solvent wipe on all exposed surfaces, the tubes lightly pin-roll expanded into the tubesheet and welded as before. The visual appearance was excellent and dye penetrant inspection revealed no surface defects. Micrographs were made and the structure was found to be similar to that found for the qualification welds. However, two of the spherical pores were larger than those noted on the qualification joints, Figure 6.

Another test plate, with two rows of tubes, was prepared to confirm weld repair techniques. Figure 7 shows two repair welds made by manual TIG welding techniques. A pin hole was repaired to simulate minor repair normally encountered. So that a burn back condition could be simulated, a tube was recessed in a hole approximately .125 inch and welded per the schedule. Almost complete lack of fusion occurred. This joint was then repaired by manual TIG welding with filler wire and is shown in the right hand photo of Figure 7. While the appearance is not of the quality as the original welds, as would be expected, no evidence of contamination (discoloration) was found and dye penetrant inspection revealed no surface defects.

The preparation for repair welds was abrasive cleaning of all exposed surfaces with a stainless steel brush followed by a solvent wipe.

In cases where tube welds are non-repairable and it is not practical to replace the entire tube, the defective hole must be plugged to prevent ammonia leakage. The standard repair procedure employed by SECO is to machine the tube-end back, insert a plug which is shaped like a deep cup and manually weld the cup to the cladding using filler metal. Sample plugs were machined from titanium bar stock (commercially pure, grade 2), see Figure 8. These plugs were manual TIG welded into twelve holes on the leak test fixture, Figure 9. Visual examination showed slight surface discoloration but no evidence of porosity, cracks, etc. Dye penetrant inspection confirmed the absence of surface defects.

In order to produce the 1 MWe Heat Exchangers at SECO for an economic cost and reasonable schedule, the practical method is to use the normal work areas for all operations. An air-conditioned enclosure consisting of a wood frame and polyethylene sheet was designed to seal around the heat exchanger shell and contain the welding equipment and operators. In order to check the performance, smaller units were designed and purchased for use at SECO on other non-titanium jobs. The enclosures proved to be extremely effective in maintaining work area cleanliness and, with some minor modifications, purchase orders have been issued for the 1 MWe Heat Exchanger enclosures. Figure 10 shows one of the smaller enclosures. Note that the work area is completely enclosed and the interior is supplied with filtered conditioned air.

The planned tube installation procedure will be to align the tubes flush with the front tubesheet, weld the tubes to the tubesheet, trim the tubes flush with the back tubesheet and weld to the back tubesheet. It is extremely important that no contaminants be introduced during the trimming operation. Our plan is to use air driven cutter, Figure 11, to perform this operation. No cutting fluids or lubricants will be used. The trimmed areas will be solvent wiped prior to welding.

QUALITY ASSURANCE

In order to maximize the weld quality, preweld samples and a combination of inspection and leak testing will be used. Separate test plates of the actual tubesheet joint configuration (cladding thickness, hole spacing, tube size and material) will be provided and test welds made prior to the startup of any production welding or after any equipment changes (electrode replacement). These preweld samples are meant to verify the equipment operability. In addition, the first few production welds at the start of a shift will be inspected prior to continuing. This procedure is designed to detect cleanliness, contamination, or fitup problems prior to welding a large number of tubes.

After welding, a visual inspection under white light will be made to detect evidence of discoloration due to oxide contamination. Normally, a light straw color may be tolerated but blue indicates that excessive contamination or poor gas coverage is occurring.

A dye penetrant inspection will follow the visual inspection to confirm the absence of surface defects. Dye penetrant inspection will be performed per SECO AS-OP-PT-2 using the low sulfur plus halogens (< one percent by weight) materials.

After tube welding has been completed, the welds shall be halogen leak tested using Freon 12 gas and a portable halogen detector. SECO AS-OP-FT-1 will be followed. The Freon 12 (dischlorodifluormethane) is compatible with titanium and will cause no stress corrosion cracking problems.

A final joint test will be the hydrostatic testing on the shell side which will be per SECO AS-383. This test will be made after the rolling operation of the tubes into the grooved tubesheets. The test pressure will be 285-psig.

Defective welds will be repaired or plugged as described above.

WELD STRENGTH

The pull-out (or push-out) strength of the tube-to-tubesheet welds were determined as a part of the weld development program. Although the weld is basically a primary seal on the seawater side and a secondary seal on the ammonia side, it will also provide redundancy for the prime structural joint which is the rolled portion. Analysis by SECO has determined that the joints must be capable of reacting an axial (pull-out) force of 401 LB in the condenser and 347 LB in the evaporator.

A test plate was fabricated with three tubes which were roll-expanded into grooved holes. No welding was employed. These tubes were tested in pull-out (push-outs) using a tensile tensile. The average pull-out force was 4640 LB, much greater than the predicted loading. The results are shown in Table II. A second test plate was fabricated using thirteen tubes. In this case, the tubes were welded to the tubesheet and no roll-expanding (other than the light pin-roll prefit) was employed. The tubes were tested, Figure 12. None of the tube welds failed. Instead, the tube ends failed by flaring at the point at which the test fixture engaged the tube. All the joints withstood a force of at least 4715 LB and the average was 5096 LB with no failure. The data are shown in Table II. The weld therefore provides redundancy for the roll-expanded portion of the tube joint. Two of the tubes tested had been repair welded as described above.

LEAK TEST

The leak tightness of the welds was tested in a special fixture containing twelve tubes, Figure 13. The tubes were welded to a mock tubesheet by the final weld schedule after the opposite end of the tubes were crimped and sealed by TIG welding. Visual and dye penetrant inspections were performed. The fixture was then pressurized to 80 psig using shop air. A soap solution was applied to the tube welds. No leaks were detected.

After completion of the tube leak test, the tube welds were ground out and the holes plugged using the titanium plugs shown in Figure 8. The plugs were welded by manual TIG welding. After visual and dye penetrant inspections, the fixture was again pressurized to 80 psig and the soap leak test repeated. No leaks were detected.

Table II RESULTS OF PUSH-OUT TESTS ON TUBE-TO-TUBESHEET JOINTS

<u>Configuration</u>	<u>Push-Out (Pull-Out) Force (LB)</u>	<u>Ave Push-Out Force (LB)</u>
Roll-Expand into Grooved T/S		
Tube 1	3830	4640
Tube 2	5040	
Tube 3	5050	
Welded*		
Tube 1	5135	5096
Tube 2	5175	
Tube 3	5140	
Tube 4	5500	
Tube 5	4950	
Tube 6	5320	
Tube 7	4780	
Tube 8	4880	
Tube 9	5195	
Tube 10	4715	
Tube 11	5325	
Tube 12	4960	
Tube 13	5375	

*All tests terminated when tube failed in compression. The weld joints did not fail.

WELD PROCEDURE

The weld procedure for welding titanium tubes to titanium clad tubesheets was documented as supplement 001 to SECO General Tube-to-Tubesheet Weld Standard AS-451. The supplement defines procedures for cleaning, welding, inspection, tube weld repairs and tube plugging. In addition, two Weld Procedure Specifications (WPS) and two Procedure Qualification Records (PQR) were prepared to define settings for tube welding (WPS No. 51-51-E-T-I and PQR T-1302) and for tube repair and plugging (WPS No. 51-51-F-TI and PQR T-1338). These documents are included as Appendix I.

The test weld records for the qualification welds (Test No. 7) the leak test welds (Test No. 9) and the tube plugging welds are included as Appendix II.

LAR:dj

Figure 1. VISUAL APPEARANCE OF QUALIFICATION WELDS. NO EVIDENCE OF CONTAMINATION NOR DEFECTIVE WELDS WAS DETECTED.

Figure 2. MICROGRAPHS OF QUALIFICATION WELDS. QUALITY APPEARS EXCELLENT WITH NO EVIDENCE OF CONTAMINATION. THE WELD MICROSTRUCTURE IS NORMAL AND THE LEAK PATH IS AT LEAST THE TUBE WALL THICKNESS. 15X and 19X.

Figure 3. SAME AS FIGURE 2. A SMALL SPHERICAL PORE ($< .004$ INCH DIAMETER) IS NOTED IN THE WELD BEAD. THE HIGHER MAGNIFICATION MICROGRAPH AT THE INTERFACE NEAR THE WELD SHOWS NO EVIDENCE OF CONTAMINATION. 19X and 100X.

Figure 4. SAME AS FIGURE 2. SMALL SPHERICAL PORES ($< .004$ INCH DIAMETER) ARE NOTED IN THE WELD AREAS.

Figure 5. MICROHARDNESS FOR WELD JOINTS. TEST CONDITION 7.

Figure 6. CROSS SECTIONS OF WELDS MADE AFTER TWO MONTH DELAY BETWEEN THE CLEANING ASSEMBLY OPERATION AND THE START OF WELDING. GENERAL QUALITY IS GOOD. ISOLATED SPHERICAL POROSITY OF UP TO .009 INCH DIAMETER WAS NOTED (SEE ARROW). 7.4X

Figure 7. HAND REPAIR WELDS (LOWER RIGHT HAND HOLE IN EACH PICTURE) OF A PIN HOLE (LEFT) AND MAJOR BURN BACK. VISUAL APPEARANCE IS GOOD WITH NO EVIDENCE OF CONTAMINATION.

Figure 8. PLUGS FOR TUBING. PLUGS ARE MACHINED FROM COMMERCIALY PURE TITANIUM ROD TO SNUGLY FIT TUBEHOLE.

Figure 9. PLUGS WELDED INTO LEAK TEST TUBESHEET. LOWER PICTURE SHOWS APPEARANCE AFTER DYE PENETRANT APPLICATION. NO DEFECTS WERE FOUND.

Figure 10. PORTABLE "CLEAN ROOM" ENCLOSING WELD AREA. FILTERED CONDITION AIR IS SUPPLIED BY THE UNIT AT THE RIGHT HAND SIDE. THE PROTECTIVE COVER SEALS AROUND THE SHELL TO GIVE FULL ENCLOSURE.

Figure 11. FLY CUTTER OF THE TYPE TO BE USED IN TRIMMING THE TITANIUM TUBES PRIOR TO WELDING. NO CUTTING FLUIDS WILL BE USED DURING THE OPERATION.

Figure 12. VIEW OF PUSH-OUT TEST SETUP. NOTE THAT TUBES HAVE "FLARED" AT TOP AT WHICH TIME THE TEST WAS TERMINATED. THE WELDS DID NOT FAIL.

Figure 13. LEAK TEST OF TUBE-TO-TUBESHEET WELDS. NO LEAKS WERE DETECTED BY SOAP SOLUTION USING 80 PSIG SHOP AIR AS A PRESSURANT.

APPENDIX I TITANIUM TUBE-TO-TUBESHEET WELD STANDARD

GENERAL TUBE TO TUBESHEET

WELD STANDARD

AS-451 SUPPLEMENT 001

- 1.0 SCOPE:
- 1.1 This supplement extends the requirements of AS-451 to cover the cleaning, handling and welding of Titanium tubes to Titanium clad tubesheet.
 - 1.2 Qualification of procedures and personnel will be in accordance with ASME SECTION III, DIV. I, CLASS 3 COMPONENTS, PARAGRAPH ND 4350.
- 2.0 CLEANING
- 2.1 TUBESHEET:
- 2.1.1 The surface oxide must be removed by sanding, light grinding or machining. It is mandatory that all traces of surface oxide be removed.
 - 2.1.1.1 Surface must be inspected and approved by the weld engineer.
 - 2.1.2 After machining and prior to staging, the tubesheets shall be immersed in Ammonium Hydroxide or rinsed in acetone to remove grease, oil and dirt.
 - 2.1.3 Subsequent to degreasing, the tubesheet shall be kept clean by completely wrapping in a weather proof paper or visqueen.
 - 2.1.4 After degreasing, the tubesheet shall only be handled with degreased handling devices and workers must wear clean white cotton gloves.
 - 2.1.5 All hardware coming in contact with tubesheets must be rinsed in clean acetone to remove all traces of organic soil.
 - 2.1.6 Once staged, the tubesheet must be fully enclosed in a clean room atmosphere. Shop dirt and dust must not be allowed to accumulate in the tube holes or the tubesheet face.
 - 2.1.7 Tube support plates shall be vacuum cleaned daily until tubes have been installed.
- 2.2 TUBES:
- 2.2.1 The tubes shall remain in the shipping containers until located in the tubing area. Container must be covered at all times except when inspecting or extracting tubes.
 - 2.2.2 The inside and outside diameter of the tube ends shall be acetone cleaned after extracting from the shipping container and prior to any mechanical cleaning.

APPENDIX I TITANIUM TUBE-TO-TUBESHEET WELD STANDARD

- 2.2.3 Oxide on the tube inside diameter shall be removed 4" back from tube end, by brushing with a stainless steel wire brush.
- 2.2.3.1 Stainless steel wire brush shall be of bottle brush design and used in conjunction with a power drill. Bottle brush shall be tapered with the large end having a diameter equal to or larger than the outside diameter of the tube.
- 2.2.5 The brush shall be cleaned in acetone after every tenth (10th) tube.
- 2.2.6 Surface oxide on the outside of the tube shall be removed 4" back from the tube end, with an abrasive flapper wheel. Flapper wheel shall be aluminum oxide resin bonded with a 240 grit. Minimum diameter of wheel shall be 3"
- 2.2.7 Tube lengths shall be free of dust immediately prior to installation. Tube ends shall be inspected by Q. A. to insure oxide removal and freedom from dust..
- 2.2.8 Tube ends shall be swabbed with acetone just prior to insertion into the tubesheet.
- 2.2.9 After insertion into the tubesheet, the tubes shall be set flush with the face of the tubesheet and pin rolled. Expander used for pin rolling shall be cleaned with acetone after every ten (10) tubes.
- 2.2.10 Tacking to secure tubes after setting shall not be permitted.

3.0 WELDING:

- 3.1 One end of the tube shall be welded and visually examined with an illuminated 3 power magnifying glass before the opposing end is set and welded.
- 3.2 Welding shall be performed with an automatic tube to tubesheet welder which shall be preprogrammed for weld speeds, amperes, slopes, and gas flow. WPS 51-51-E-TI shall be used in conjunction with appropriate data sheet for tube end welding.
- 3.3 All welding shall be performed within a covered area free from shop dust and air movement.
- 3.4 Personnel shall wear clean coveralls and clean white gloves while working in the clean room area.
- 3.5 Welding operators shall be qualified for the specific diameter tube gauge and alloy for which they will be welding.

APPENDIX I TITANIUM TUBE-TO-TUBESHEET WELD STANDARD

- 3.6 At the start of each shift and whenever the tungsten location has been adjusted, a weld on a simulated tube to tubesheet shall be made to assure proper weld configuration.
- 3.7 The tungsten shall be changed at least every fifty (50) welds to assure proper arc concentration. The tungsten shall be machine ground per figure #1.
ST-42-L
- 3.8 Once the welding operation starts, cleaning shall not be permitted. Acetone shall not be allowed in the tube weld area after final cleaning.

4.0 INSPECTION:

- 4.1 Each weld shall be visually inspected by Q. A. Visual inspection shall be performed with a 3 power illuminated magnifying glass.
- 4.2 After visual acceptance by Q. A., the unit shall be subjected to a dye-penetrant inspection.
- 4.3 After dye-penetrant acceptance, the tubes shall be subjected to an air soap test.
- 4.4 Any tubes which do not pass any of the nondestructive tests shall be noted on an NCR and must be dispositioned by the welding engineer on an individual basis.

5.0 TUBE WELD REPAIRS:

- 5.1 Tube welds, if determined by the weld engineer to be repairable, shall be repaired by the gas tungsten arc process. Filler metal addition shall be mandatory. WPS to be used in the event of a repair shall be 51-51-F-TI.
- 5.2 Removal of the weld around the entire periphery of the tube shall be accomplished by machine cutting prior to welding. Grinding shall not be permitted as a preparation for weld repair.
- 5.3 Tube repairs shall be subjected to all inspections of Par. 4.0.

6.0 TUBE PLUGGING

- 6.1 If any defective tubes or tube welds are determined unrepairable, they shall be plugged per AS-457 and WPS 51-51-F-TI.
- 6.2 Tube plug welds shall be subjected to the inspection required in Par. 4.0.

Subsidiary of
Cronus Industries, Inc.

APPENDIX I TITANIUM TUBE-TO-TUBESHEET WELD STANDARD

<p>PREP. BY <u>S.S.J.</u> APPROVED BY <u>[Signature]</u> ENGR. <u>[Signature]</u> MFG. _____ Q.A. _____</p>	<p>WELD PROCEDURE SPECIFICATION</p> <p>WPS. NO. <u>51-51-E-T I</u> REV. <u>0</u></p>	<p>DATE <u>5-18-78</u></p>																								
<p>I. APPLICABLE GWS <u>As-451, AS451 SP-001</u> PQR <u>T-1320</u> DATA SHEETS _____</p> <p>II. BASE MAT'L. TUBE SHEET OVERLAY <u>Ti-Clad</u> TUBE P-NO. <u>51</u> GROUP NO. _____ TUBE GAUGE RANGE <u>22 GA</u> TUBE O.D. RANGE <u>1"</u></p> <p>III. WELD PROCESS <u>GTAW</u> AUTOMATIC <u>X</u> MANUAL _____</p> <p>IV. ELECTRICAL CHARACTERISTICS CURRENT / POLARITY <u>DC SP</u></p> <p>V. GAS TYPE <u>ARGON</u> FLOW RATE <u>20-25 CFH</u></p> <p>VI. FILLER METAL SPEC. NO. <u>N/A</u> F-NO. _____ A-NO. _____ AWS NO. _____ TRADE NAME PREFERRED _____</p>	<p>VII. POSITION POSITION OF GROOVE <u>T, S VERT</u></p> <p>VIII. PREHEAT MIN. PREHEAT TEMP. <u>AMBIENT</u> MAX. INTERPASS TEMP. <u>300° F.</u></p> <p>IX. TECHNIQUE TRAVEL SPEED RANGE <u>8-10 IPM</u> AMPERE RANGE <u>85-90</u> PULSE _____ CONSTANT <u>X</u></p> <p>NOTES:</p> <ol style="list-style-type: none">1. Tube I. D.'s must be plugged with felt plugs during welding.2. Tubeweld head must be positioned and held secure during the entire weld cycle including pre-purge and post purge																									
<table border="1" style="width: 100%; border-collapse: collapse;"><thead><tr><th style="width: 15%;">TUBE GA.</th><th style="width: 20%;">RECESS OR PROJ.</th><th style="width: 15%;">TUBE GA.</th><th style="width: 20%;">RECESS OR PROJ.</th></tr></thead><tbody><tr><td style="text-align: center;">22</td><td style="text-align: center;">Flush</td><td></td><td></td></tr><tr><td></td><td></td><td></td><td></td></tr><tr><td></td><td></td><td></td><td></td></tr><tr><td></td><td></td><td></td><td></td></tr><tr><td></td><td></td><td></td><td></td></tr></tbody></table>			TUBE GA.	RECESS OR PROJ.	TUBE GA.	RECESS OR PROJ.	22	Flush																		
TUBE GA.	RECESS OR PROJ.	TUBE GA.	RECESS OR PROJ.																							
22	Flush

Southwestern Engineering 6111 E. Bandini Blvd., P.O. Box 54940, Los Angeles, Cal. 90054. Tel: 213/726-0641

Subsidiary of
Cronus Industries, Inc.

APPENDIX I TITANIUM TUBE-TO-TUBESHEET WELD STANDARD

		PROCEDURE QUALIFICATION RECORD		PQR NO. <u>T-1302</u> REV. <u>0</u> DATE <u>5-18-78</u>			
DATA SHEET NUMBER FOR AUTO-TUBE WELD _____							
WELD PROCEDURE SPECIFICATIONS QUALIFIED <u>51-51-E-TI</u>							
WELD PROCESS <u>GTAW</u> AUTOMATIC ☒ SEMI-AUTO ☐ MANUAL ☐							
BASE METAL S. <u>B 338</u> Grade <u>2</u> TO S. <u>B 338</u> Grade <u>2</u>							
P-NO. <u>51</u> G. _____ TO P-NO. <u>51</u> G. _____							
THICKNESS <u>22 GA</u> THICKNESS RANGE QUALIFIED <u>22GA</u>							
DIAMETER <u>1"</u>							
FILLER METAL ASME _____ AWS CLASS _____ F-NO. _____ A-NO. _____							
ASME _____ AWS CLASS _____ F-NO. _____ A-NO. _____							
FILLER METAL CHEMISTRY _____							
WIRE / ELECTRODE DIA. _____ MANUFACTURER AND TRADE NAME _____							
POSITION OF GROOVE <u>TS VERT</u>							
WELD PROGRESSION <u>360°</u>							
PREHEAT							
INITIAL PREHEAT TEMP. <u>70° F</u>							
INTERPASS TEMP. <u>200° F. Max</u>							
POST-WELD HEAT TREAT. (ACTUAL) TEMP _____							
HOLD TIME _____							
TYPE OF GAS OR GASES <u>ARGON</u>							
FLOW CFH <u>20 CFH</u>							
COMPOSITION OF GAS MIXTURE <u>100% Pure</u>							
ELECTRICAL CHARACTERISTICS							
PASS NO.	AC	DC STR	REV.	AMPS	VOLTS	TRAVEL SPD	GAS CFH
1		X		85	12	9	20

WELD JOINT DETAIL							
TECHNIQUE							
STRING OR WEAVE READ <u>Stringer</u>							
OSCILLATION _____ WIDTH _____ CYCLE/MIN _____							
MULTI PASS OR SINGLE PASS (PER SIDE) <u>Single</u>							
SINGLE OR MULTIPLE ELECTRODES <u>Single</u>							
OTHER _____							
E-27

Southwestern Engineering 6111 E Bandini Blvd. P.O. Box 54940, Los Angeles, Cal 90054. Tel: 213/726-0641

Subsidiary of
Cronus Industries, Inc.

APPENDIX I TITANIUM TUBE-TO-TUBESHEET WELD STANDARD

PREP. BY <u>SSJ</u> APPROVED BY <u>SSJ</u> ENGR. <u>SSJ</u> MFG. <u>SSJ</u> Q.A. <u>SSJ</u>		WELD PROCEDURE SPECIFICATION WPS. NO. <u>51-51-F-TI</u> REV. <u>0</u>		DATE <u>2-20-79</u>																									
I. APPLICABLE GWS <u>As-451 and supplement</u> <u>No. 001</u> PQR <u>T-1338</u> DATA SHEETS _____		VII. POSITION POSITION OF GROOVE <u>Tubesheet Vertical</u>																											
II. BASE MAT'L. TUBE SHEET CLADDING <u>Titanium</u> TUBE P-NO. <u>51</u> GROUP NO. <u>N/A</u> TUBE GAUGE RANGE <u>22 GA</u> TUBE O.D. RANGE <u>1"</u>		VIII. PREHEAT MIN. PREHEAT TEMP. <u>70° F</u> MAX. INTERPASS TEMP. <u>200° F</u>																											
III. WELD PROCESS <u>Gas Tungsten Arc</u> AUTOMATIC _____ MANUAL <u>X</u>		IX. TECHNIQUE TRAVEL SPEED RANGE <u>9.5 IPM</u> AMPERE RANGE <u>85-90</u> PULSE _____ CONSTANT <u>X</u>																											
IV. ELECTRICAL CHARACTERISTICS CURRENT / POLARITY <u>DC SP</u>		NOTES: 1. Adjacent tubes must be plugged with felt plugs during weld. 2. Auxiliary shielding must be used on the torch to prevent oxidation.																											
V. GAS TYPE <u>Argon</u> FLOW RATE <u>15-20 CFH</u>																													
VI. FILLER METAL SPEC. NO. <u>SFA. 5.16</u> F-NO. <u>51</u> A-NO. <u>N/A</u> AWS NO. <u>ER Ti-3</u> TRADE NAME PREFERRED _____ <div style="text-align: right;">E-28</div>		<table border="1"><thead><tr><th>PLUG</th><th>RECESS OR PROJ.</th><th>TUBE GA.</th><th>RECESS OR PROJ.</th></tr></thead><tbody><tr><td>AS-457</td><td>Recess 0.060"</td><td></td><td></td></tr><tr><td></td><td></td><td></td><td></td></tr><tr><td></td><td></td><td></td><td></td></tr><tr><td></td><td></td><td></td><td></td></tr><tr><td></td><td></td><td></td><td></td></tr></tbody></table>				PLUG	RECESS OR PROJ.	TUBE GA.	RECESS OR PROJ.	AS-457	Recess 0.060"																		
PLUG	RECESS OR PROJ.	TUBE GA.	RECESS OR PROJ.																										
AS-457	Recess 0.060"

Southwestern Engineering 6111 E. Bandini Blvd., P.O. Box 54940, Los Angeles, Cal. 90054, Tel: 213/726-0641

Subsidiary of
Cronus Industries, Inc.

APPENDIX I TITANIUM TUBE-TO-TUBESHEET WELD STANDARD

		PROCEDURE QUALIFICATION RECORD		PQR NO. <u>T-1338</u>		
				REV. <u>0</u>		
				DATE <u>2-20-79</u>		
DATA SHEET NUMBER FOR AUTO-TUBE WELD _____						
WELD PROCEDURE SPECIFICATIONS QUALIFIED _____						
WELD PROCESS <u>GTAW</u> AUTOMATIC ☐ SEMI-AUTO ☐ MANUAL ☒						
BASE METAL S- <u>B 338 Grade 2</u> TO S- <u>B 338 Grade 2</u>						
P-NO. <u>51</u> G- _____ TO P-NO. <u>51</u> G- _____						
THICKNESS <u>TS. 3"- Plug .080"</u> THICKNESS RANGE QUALIFIED _____						
DIAMETER _____						
FILLER METAL ASME <u>SFA. 5.16</u> AWS CLASS <u>ERTi-3</u> F-NO. <u>51</u> A-NO. <u>-</u>						
ASME _____ AWS CLASS _____ F-NO. _____ A-NO. _____						
FILLER METAL CHEMISTRY _____						
WIRE / ELECTRODE DIA. <u>1/16</u> MANUFACTURER AND TRADE NAME <u>Johnston Stainless</u>						
POSITION OF GROOVE <u>T. S. Vert</u>						
WELD PROGRESSION <u>Vert - Up</u>						
PREHEAT						
INITIAL PREHEAT TEMP. <u>70°</u>						
INTERPASS TEMP. <u>-</u>						
POST-WELD HEAT TREAT. (ACTUAL) TEMP <u>-</u>						
HOLD TIME _____						
TYPE OF GAS OR GASES <u>Argon</u>						
FLOW CFH <u>25</u>						
COMPOSITION OF GAS MIXTURE <u>100% Pure</u>						
ELECTRICAL CHARACTERISTICS						
PASS NO.	AC	DC str REV.	AMPS	VOLTS	TRAVEL SPD	GAS CFH
1		X	90	-	4	25
WELD JOINT DETAIL						
TECHNIQUE						
<u>Stringer</u>						
STRING OR WEAVE BEAD _____						
OSCILLATION _____ WIDTH _____ CYCLE/MIN _____						
MULTI PASS OR SINGLE PASS (PER SIDE) <u>Single</u>						
SINGLE OR MULTIPLE ELECTRODES <u>Single</u>						
OTHER _____

NOMINAL SIZE	PLUG - DIA		
	A	B	C
5/8"	.640"	.484"	.620"
3/4"	.765"	.609"	.745"
7/8"	.890"	.734"	.871"
1"	1.017"	.859"	.997"

REQUISITION

TYPE "TI"	
MATERIAL	
ASME SB-348	
GRADE 2	
NOMINAL SIZE	REQUIRED QUANT.
5/8	
3/4	
7/8	
1	

JOB NO.	TUBE HOLE PLUG	DRWG SIZE	REV	DATE	CHK
		AS-457		1-22-79	
	E-30			BY	APP

Southwestern Engineering Company

Subsidiary of Cronus Industries, Inc.

APPENDIX II TEST WELD RECORDS

PAGE 1 OF 1

RESEARCH & DEVELOPMENT RECORD

REV. NO. _____

DATE 1-26-79

APP. _____

SUBJECT:

O T E C W E L D I N G
R & D R E C O R D

EFFECTIVE: _____

SUPERSEDES: _____

WELD TES #T-1340

Leak Test Mockup

12 ea. 1" O. D. 22 GA. Titanium Tubes
3" THK 1/4" TI - Clad T. S.

WELD PARAMETERS FOR ALL TUBES

85 AMPS 11.5 VOLTS 9.45 IPM
20 SEC. Duration, 30 CFH 99.99% Pure Argon.
30 SEC Post Purge. 360° CW Rotation

JOINT DESIGN

Flush tube joints - pre-rolled for metal to metal contact (1/2" depth).

CLEANING

Mech/Acetone Wipe.

OPERATION OBSERVATIONS

All welds - Visually acceptable
P-T Acceptable

LEAK TEST

AIR-80 PSI - 20 min. - soap
No leaks

E-31

PREPARED BY: _____

Southwestern Engineering Company

Subsidiary of Cronus Industries, Inc.

APPENDIX II TEST WELD RECORDS

PAGE 1 OF 1

RESEARCH & DEVELOPMENT RECORD

REV. NO. _____	SUBJECT: O T E C W E L D I N G R & D R E C O R D	EFFECTIVE: _____
DATE <u>1-10-79</u>		SUPERSEDES: _____
APP. _____		

WELD TEST #T-1337A

1" O.D. 22 GA. Titanium Tubes
1 1/2" THK 1/4" TI-Clad

WELD PARAMETERS FOR ALL TUBES

85 AMPS 11.5 VOLTS 9.45 IPM
20 SEC DURATION 30 DFH 99.99% Pure Argon
30 SEC Post Purge. 360° C W Rotation

TUBE NUMBERING SEQUENCE

From top to bottom/left to right.
8 rows of tubes welded.

JOINT DESIGN

Flush tube joints - pre-rolled for metal to metal contact (1/2" depth)

CLEANING

Mechanical/acetone wipe

OPERATION NOTES

All tubes satisfactory
Qualification mock up.

E-32

PREPARED BY: _____

INTEROFFICE CORRESPONDENCE

5515.2.79-250

TO: J. E. Snyder

CC: L. D. Berman
R. H. Douglass
L. J. Meyers
M. H. Seidman
A. M. Sprouse

DATE: 7 June 1979

SUBJECT: Welding of Improperly Configured Holes on
Tubesheet - 1 MWe OTEC

FROM: L. A. Rosales

BLDG.
01

MAIL STA.
2220

EXT.
52630

- Reference 1): IOC 5515.2.78-336, "Status of Tube-to-Tubesheet Weld Development Program - 1 MWe", L. A. Rosales to J. E. Snyder, 11 July 1978.
- 2): SECO AS-451 Supplement 001, General Tube-to-Tubesheet Weld Standard, May 18, 1978.

The OTEC Heat Exchanger titanium tube-to-tubesheet weld procedure was developed for automatic welding, Reference 1. An important result of the development was a need for very good fitup between tube and tubesheet. For this reason, the holes in the tubesheet must be square; i.e., no chamfers. This requirement is documented in the SECO weld procedure, Reference 2, and its supplemental documents. In spite of precautions which were implemented by SECO, holes on all four tubesheets were found to contain full chamfers or gouges which extended for part of the hole circumferences. Hand grinding of the tubesheets has reduced the amount of chamfering but has not eliminated the problem. For this reason, a small test program was conducted to attempt to determine the limits of successful automatic welding with respect to the degree of chamfer.

A tubesheet test plate was obtained and fourteen holes purposely chamfered or gouged to simulate the observed condition. The chamfers were generally at an angle of less than 45° (see sketch on Table I) but hole No. 42 was given a shallow wide chamfer with the chamfer angle exceeding 60° . Four of the prepared holes were used for manual weld evaluation so that a total of ten welds were made on the chamfered holes with the Hobart automatic weld equipment.

The depth of the chamfers were measured and are listed in Table I. The hole numbers were assigned starting at the top left corner and counting to the right on each row.

Table I. DEPTH OF CHAMFER ON HOLE

<u>Hole No.</u>	<u>X (in.)</u>	<u>α</u>	<u>Weld</u>	<u>Results</u>
9	.041	< 45°	Man.	
12	.060	< 45°	Auto.	Burnback-Manual Repair OK
15	.058	< 45°	Auto.	Porosity
35	.040	< 45°	Auto.	OK
38	.060	< 45°	Auto.	OK
41	.041	< 45°	Auto.	OK
42	.030	> 60°	Auto.	Burnback-Manual Repair OK
44	.046	< 45°	Auto.	OK
46	.060	< 45°	Auto.	Pinhole-Auto Repair OK
48	.050	< 45°	Auto.	Burnback
50*	.045	< 45°	Auto.	Burnback
61	.040	< 45°	Man.	
63	.052	< 45°	Man.	OK
65*	.062	< 45°	Man.	Pinhole-Manual Repair OK

*Tube recessed approximately .040 inch.

The holes were cleaned according to the procedure and the tubes lightly pin-rolled into the holes. The tube end of hole No. 46 was damaged during pin-rolling but the weld was still attempted.

The results can be summarized as follows:

- a) Generally, good automatic machine welds were obtained for holes containing chamfers or gouges of depths of less than .045 inch and chamfer half angles of less than 45° .
- b) Chamfer half angles of greater than 60° caused burnback when using automatic welding.
- c) Recessing the tube from the face of the tubesheet caused defects in welds.
- d) Manual repair using filler wire was successful in all cases where a repair was attempted.
- e) One automatic machine repair was attempted on a weld containing a single pin-hole by making a second pass and this was successful.
- f) Manual welding of a chamfered hole using filler wire was successful.

It was decided not to weld holes numbers 9 and 61 at this time but to use these as part of the welder qualification (manual) test unit. In addition, no attempt was made to repair the welds on holes numbers 15, 48, and 50.

As a result of this program, the following recommendations are made:

- a) The tubesheets which have not been hand ground should be so that the severity of the chamfers and gouges should be minimized.
- b) After grinding, the chamfers and gouges should be measured and recorded so that a map of the defective holes can be made.
- c) All holes with chamfers or gouges of less than a TRW and SECO mutually agreed upon severity can be automatically welded. A starting point on this figure would be a depth X of less than .040 inch and a chamfer half angle of less than 45° . If poor results are obtained during the initial production phase, these figures can be modified at that time.

Page 4

Welding of Improperly Configured
Holes on Tubesheet - 1 MWe OTEC
5515.2.79-250

d) All holes with chamfers or gouges greater than the agreed upon limits should be manual welded using filler wire.

As an aside to this program, it was noted that several holes on one of the condenser tubesheet extend into the bevel portion of the titanium cladding. These joints should be made using manual welding with filler wire.

LAR:dj

INTEROFFICE CORRESPONDENCE

TO: J. E. Snyder

L. D. Berman
R. H. Douglass
CC: L. J. Meyers
M. H. Seidman
A. M. Sprouse

DATE: 21 May 1979

SUBJECT: Cleanliness Control of 1 MWe Heat Exchangers - OTEC FROM: L. A. Rosales

BLDG.
01MAIL STA.
2220EXT.
52630

Reference: 1) IOC 5515.2.79-153, "Final Report on the Tube-to-Tubesheet Weld Development Program - 1 MWe OTEC", L. A. Rosales to J. E. Snyder, 2 April 1979.

Cleanliness/contamination control must be strictly adhered to on the 1 MWe OTEC Heat Exchangers especially as the units are nearing the tubing and tube-to-tubesheet welding operations. Preliminary cleaning has been accomplished on most of the components and assemblies and the current effort should be aimed at maintaining the level of cleanliness achieved and preparing for the final cleaning. A discussion of the steps recommended for specific components/assemblies follows.

Waterboxes. After welding into a complete assembly, including the flange and thermistor ports, the interior will be coated. The first step in the coating process will be sandblasting. SOC-CO Plastics will sandblast both interior and exterior so that the coated waterboxes should be adequately clean upon receipt by SECO.

Evaporator Spray Headers/Piping. These components were sandblasted and have been installed into the shell. Future cleaning should include localized solvent wiping to remove small grease smudges and a final vacuum cleaning prior to the tubing operation.

Baffles/Tube Supports. These components have been sandblasted and degreased in hot ammonium hydroxide and assembled into the shells for alignment purposes. It is strongly recommended that the tubefields be protected by covering with weather proof paper or visqueen plastic. The cover should be taped to both sides of the plates. This step has not been followed to date. The covers should be kept in place until just prior to the tubing operation. At that time, they should be removed and small grease marks or other dirt removed

by local solvent wipe. The plates should be vacuumed just prior to tubing to remove dust and loose dirt from the holes.

Shells. Since the shells have been assembled outside the plant, cleanliness control is difficult. However, all efforts should be made to minimize contamination to facilitate the final cleaning operation. Large debris and major grease stains have been removed and plastic covers placed over manways and nozzles to minimize the inflow of dirt and debris. In addition, white gloves are being used by workers and foot cleaning mats placed at entry points to the shells. These mats should be used prior to a worker entering the shell, even under manways. Gloves should be changed frequently to assure cleanliness. Plywood covers for the "floors" are being used which will be removed just prior to tubing. Final cleaning steps should include localized solvent wiping and vacuum cleaning of all interior surfaces.

Tubesheets. The tubesheets were to be degreased by immersion in hot ammonium hydroxide prior to assembly to the shell. However, one condenser tubesheet and one evaporator tubesheet were installed without the degreasing step. Some attempts at cleaning these by solvent wipe and light grinding have been made but it is recommended that they be degreased by immersion after they have been cut-off the shell prior to tubing. After cleaning, the tube field should be covered with weatherproof paper or visqueen plastic as described for the baffles/supports. Just prior to tubing, the protective covers should be removed and the preweld preparation described in Reference 1, i.e., wire brushing with stainless steel brushes and acetone wipe, should be performed on the holes. Clean white gloves must be worn during this operation and Scott Soft-Cote paper towels should be used for the acetone wipe. The acetone wipe should be done after the portable weld clean-rooms have been installed (see below for description).

Tubes. The tubes are currently stored in their shipping boxes and should remain so until just prior to the tubing operation. Upon removal from the

boxes, the entire length of tube should be wiped with clean cloths to remove loose dust and dirt. The tube end should then be prepared per Reference 1. That is, the exterior of the ends should be wire brushed using stainless steel brushes and then solvent wiped with acetone. The pall-bearer type of white gloves should be used for these operations and the Scott Soft-Cote towels used for the acetone wipe. The tubes should be immediately installed to prevent collection of dust on the surface.

After tube installation, the tube ends are given a light pin roll into the tubesheet to produce a good fitup for welding. After this pin-roll, the tubesheet around the hole and the tube interior surface near the joint should be wire brushed and acetone wiped as described above. This should be accomplished in the weld clean room.

Shrouds. The shrouds will be sent to SOC-CO Plastics for application of the neoprene gaskets. Like the waterboxes, the shrouds will be sandblasted and should be clean when sent back to SECO. Any dirt or grease contamination which occurs during the fitting operations should be removed by local solvent wipe. The shroud plates should be covered when stored, preferably in boxes, to avoid contamination.

Portable Clean Room Operation. The tube-to-tubesheet joint preparation and welding will be performed in the portable clean-rooms which have been fabricated by A&M Welding. These clean rooms are made with steel frame members and heavy gage plastic walls and are collapsible to ease shipping and storage. The clean rooms seal around the end of the shell assembly. A wooden floor must be provided by SECO. Conditioned and filtered air is supplied to the clean room. Access is through a single door. A sketch of the clean room is attached.

It is strongly recommended that an entry chamber be used with each clean room. This entry area would act as a storage area for all supplies and equipment not needed for the operations performed in the clean room and as a cleaning area for tools, etc. In addition, coveralls and gloves would be put on in

the entry room prior to entering the clean room. A foot-wipe mat would be used before entering the entry chamber and another prior to entering the clean room. The only personnel allowed into the clean room would be operators for the specific operation being performed or inspectors.

A list of supplies required for the cleaning and clean-room operations is shown below. Those which should be stored in the entry chamber are marked with an asterick and those in the weld clean room by a double asterick.

Protective Clothing

SECO Standard Issue Gloves

- * White Nylon Pall Bearers Gloves
- * Hardhats

Solvents

- * Acetone
- * Dye Penetrant Solvent

Wipes

- Clean Rags
- * Scott Soft-Cote Towels

Mats

- * Foot Wipe

Tools

- ** Pin Rollers
- * Extra Rolling Pins
- ** Hobart Welding Machines
- ** Weld Guns with Collets
- * Extra Electrodes
- * Miscellaneous Hand Tools

Page 5
Cleanliness Control of 1 MWe
Heat Exchangers - OTEC
5515.2.79-231

Inspection Equipment

- * Dye Penetrant
- * Dye Penetrant Developer
- * Lights
- * Scales
- * QA Paperwork, Clipboards, Pens, etc.

LAR:dj

Figure 1. SCHEMATIC SHOWING WELD CHAMBER AND ENTRY CHAMBER

INTEROFFICE CORRESPONDENCE

5515.2.79-235

TO: J. E. Snyder

CC: S. Benesch
L. Berman
R. Douglass
L. Meyer
M. Seidman
A. Sprouse

DATE: 29 May 1979

SUBJECT: Cleaning/Cleanliness Control During Heat Exchanger
Tubing Operation - 1 MWe OTEC

FROM: L. A. Rosales

BLDG.
01MAIL STA.
2220EXT.
52630

Reference 1): IOC 5515.2.79-231, "Cleanliness Control of 1 MWe Heat Exchangers - OTEC", L. A. Rosales to J. E. Snyder, dated 21 May 1979.

The most critical heat exchanger fabrication step has long been identified as the welding of the titanium tube-to-tubesheet joint. Since titanium weldments are extremely sensitive to solid, liquid and gaseous contaminants, it is absolutely mandatory that strict cleanliness control be imposed on the steps leading up to the welding operation. It is becoming increasingly obvious that the one most critical step in this respect is the tubing operation. If the tubes are not properly cleaned prior to and during this operation, the contaminants will be trapped in the tube joints with catastrophic effect on the weldments! It is also clear that the individual steps involved in the tubing have not been addressed in sufficient detail so that cleanliness and schedule can be maintained. For this reason, an outline of a recommended procedure is included with recommendations as to the sequencing of the operations and for special equipment.

Three assembly "paths" must be subjected to close cleanliness control; viz., the tube preparation and installation, the shell/baffle-support/tubesheet assembly, and the (cut off) tubesheet/skirt assembly. These paths must be coordinated to assure compatibility of schedule as well as proper cleaning and handling. A schematic of the tubing operation is shown in Figure 1. Several substeps have been identified as potentially time-consuming and in-line with the critical path. One of these is the abrasive cleaning of the tube ends prior to installation into the shell assembly. It is recommended that this be done at least one week prior to the tubing since a result of our weld development program illustrated that this step could be performed up to two months prior to welding with no harmful effects on the weldments. The abrasive cleaning step would be performed using the three spindle air gun with abrasive flappers. This gun allows very rapid cleaning.

However, SECO only has one of these guns so that the logistics of cleaning both ends of the tubes, especially as part of the tubing operation, dictates that the recommended timing be implemented. This seems even more prudent since the current schedule calls for tubing both the evaporator and condenser simultaneously. Racks (possibly several "sawhorse" racks) must be used so that the tubes can be unpacked from their boxes, laid on the racks, cleaned, and repacked without damage. These racks would again be used for holding the tubes off the ground just prior to tubing.

Other operations which must be thought out and carefully scheduled are the tubesheet hole brushing and solvent wipes and precleaning the clean rooms. In addition, flooring and scaffolding for the clean room and the entry chamber (Reference 1) must be fabricated and be ready for installation at the proper points in time.

In order to accomplish the above steps in the very tight timeframe available, it is imperative that full-support be given by SECO Engineering, Planning, Manufacturing and Quality Assurance and the TRW Quality Assurance organizations. Due to the number of joints involved (approximately 25,000!) adequate manpower must be provided at the proper points in time to accomplish the several steps in the operations. This may impose a significant hardship on other programs within SECO and a commitment to OTEC must be made if the schedule is to be met.

LAR:dj

TUBES

- 3 spindle gun w/flappers
- use support racks

- wipe with clean rags
- insert tube guides
- use support racks

Storage of Cleanroom

Fab Floor, Scaffolds, Entry Chamber

Storage of Cleanroom

Fab Floor, Scaffolds, Entry Chamber

SHELL ASSEMBLY

- wire brush tube holes
- vacuum inside shell assembly

- acetone wipe

- acetone wipe
- cover exposed ends w/plastic sheet

- acetone wipe

- remove plastic covers from tube ends

- light pin-roll

- wire brush
- acetone wipe

CUT-OFF TUBESHEET ASSEMBLY

- degrease in NH_4OH

Figure 1. OPERATION SEQUENCE FOR TUBE INSTALLATION INTO HEAT EXCHANGER SHELL ASSEMBLY

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK - REDONDO BEACH - CALIFORNIA 90278

INTEROFFICE CORRESPONDENCE

5515.2.79-283

TO: J. E. Snyder

CC:

DATE: June 25, 1979

SUBJECT: Coordination Meeting on Steps Leading to Condenser
Tube-to-Tubesheet Welds - 1 MWe OTEC

FROM: L. A. Rosales

BLDG.
01MAIL STA.
2220EXT.
52630

Reference: SECO Weld Standard AS-451 Supplement 001 Revision No. 1, 6-11-79.

A meeting was held at SECO on 22 June 1979 to coordinate the manufacturing and cleaning steps leading up to the tube-to-tubesheet welding on the 1 MWe condenser. Present at the meeting were Dave Hodges, Jack Russell and Bill Koch of SECO and Loren Meyers and myself from TRW. Three major operations were discussed:

- Installation of the inlet tubesheet
- Tube welding on the outlet side
- Tube welding on the inlet side

The sequence of substeps agreed upon is as follows.

Installation of the Inlet Tubesheet

1) Spray the inlet side tube ends with high pressure acetone. The spraying will be performed from the top down with care to impinge on the bottom of each tube row. This represents a change in paragraph 2.2.7 of AS-451 Supplement 1 for the inlet side only. The inspection point called out in paragraph 2.2.6 is performed after spraying.

2) Hand solvent wipe the outlet side tube ends (shoe-shine method) using clean lint-free rags. This step may be performed on individual tubes just prior to their being back tubed. The inspection point in Paragraph 2.2.6 is after wiping.

3) Hang the inlet tubesheet and align by back tubing 4 or more tubes. The tubes will extend beyond the inlet tubesheet a minimum of $\frac{1}{2}$ inch when the other ends are flush with the outlet tubesheet.

4) Weld inlet tubesheet to the condenser shell.

Tube Welding on Outlet Side

- 1) Using standard pneumatically driven tools, back-tube until the tube ends protrude approximately 1/32 inch from the outlet tubesheet.
- 2) Light pin-roll the tube ends using a five-pin roller.
- 3) Wire brush the tube inside diameters.
- 4) Wire brush the face of the tubesheet.
- 5) Acetone wipe the tube joint (especially the 1/32 inch protruding tube outside diameter) using clean lint free rags.
- 6) Hand set tube ends, flush with the outlet tubesheet using a swage tool.
- 7) Insert the cotton plugs into the tubes.
- 8) Weld engineer inspection to identify joints which require manual welding. These holes will be tagged for identification.
- 9) Acetone wipe tube joints using clean lint-free rags.
- 10) Automatic TIG weld tube joints.

Tube Welding on Inlet Side

- 1) Using an internal tube cutter, trim the tube ends to within 1/16 to 1/8 inch of inlet tubesheet.
- 2) Light pin-roll the tube ends using a five-pin roller.
- 3) Using a tube trimming tool, trim the tube ends flush with the inlet tubesheet.
- 4) Wire brush the tube inside diameters.
- 5) Wire brush the tubesheet face.
- 6) Insert cotton plugs.
- 7) Weld engineer inspection to identify joints which require manual welding. These holes will be tagged for identification.
- 8) Acetone wipe joints using clean lint-free rags.
- 9) Automatic TIG weld tube joints.

Page 3
Coordination Meeting on Steps Leading
to Condenser Tube-to-Tubesheet Welds-
1 MWC OTEC
5515.2.79-283

The following items were also discussed during the meeting and agreed to:

- 1) A suitable floor covering (either linoleum or rubber) will be provided for the clean rooms and pre-entry chambers.
- 2) Floor mats will be installed as soon as they arrive at SECO. The mats will be placed at the entrances to the pre-entry chambers and the weld rooms.
- 3) White work coats will be worn for all operations (the coats are due at SECO on Monday, 25 June).
- 4) Rolls of lint-free cloth will be procured by SECO for use in the acetone wiping.
- 5) Stainless steel brushes will be used for all wire bushing.
- 6) Clean white gloves will be worn for all operations.

LAR:dj

DISTRIBUTION:

L. D. BERMAN
J. H. CHANCE
R. H. DOUGLASS
L. J. MEYERS
M. H. SEIDMAN
A. M. SPROUSE
F. C. WEAGE

SECO:

D. HODGES
W. KOCH
J. RUSSELL

DEFENSE AND SPACE SYSTEMS GROUP
ONE SPACE PARK • REDDING BEACH • CALIFORNIA 90278

INTEROFFICE CORRESPONDENCE

5515.2.79-187

TO: J. E. Snyder

CC:

DATE: 27 April 1979

SUBJECT: Visit to SOC-CO Plastics and Shroud Design
Recommendations - 1 MWe OTEC Heat Exchangers

FROM: L. A. Rosales
BLDG. MAIL STA. EXT.
01 2220 52630

As a follow-up to a quotation by SOC-CO Plastics of Cucamonga, Calif. to apply the neoprene coating to the evaporator water boxes and tubesheet/skirt areas, their plant was visited to assess their capabilities. The visit was made on 26 April 1979 by Maurice Bianchi and myself.

SOC-CO has a relatively large facility for applying a variety of coatings and liners. They coat everything from small pipe and fittings to interiors of railroad cars and large pressure vessels. They have a large grit-blasting chamber, four large ovens and three large autoclaves, the largest of which is 15 feet in diameter by 45 feet long. They are capable of coating structures either in-house at SOC-CO or at the customer's (Southwestern Engineering) facility. Mr. Jerry Kiger of SOC-CO prepared the quotation but has not seen Southwestern's fabrication area. He said that coating inside the shop area is probably unacceptable since the neoprene spray can foul and contaminate equipment such as welding machines, control panels, etc. The outside fabrication area now used for the evaporator and condenser shells should be adequate. Mr. Kiger recommends (and we agree) that the waterboxes be coated at SOC-CO since better quality control can be maintained. The tubesheet/skirt coating must be applied at SECO since it is not practical to ship the finished shell/tube assemblies. The actual spray-up and cure time to apply .060 inch of neoprene is 5 to 6 days after grit blasting. Therefore, allowing for shipping, grit blasting, coating and inspection, a total of 10 days should be scheduled for the coating operation. The waterboxes and the tubesheet/skirt coatings can be applied simultaneously since SOC-CO has adequate personnel and equipment to perform the coating at the two separate facilities.

It is the opinion of both Maurice Bianchi and myself that SOC-CO has the capability to properly apply the neoprene coating as required for the 1 MWe OTEC Heat Exchangers. In addition, they are capable of applying gaskets for the shrouds as described below.

The Materials Engineering Department has been asked to recommend materials for the shroud windows and shroud gaskets. After assessing candidate materials for the window, a tempered glass appears to be the best selection. The clear plastics will not stand up to ammonia service and crazing or cracking may result. The glass may be tempered by either heat or chemical (Chem-Cor Glass). Automobile safety glass should not be used since the plastic film used to prevent shattering will be attacked by ammonia. The glass windows can be installed in a frame with a neoprene gasket and the frame attached to the shroud with a neoprene gasket.

In order to facilitate shroud installation, it is our recommendation that neoprene gaskets be permanently attached to the shroud plate edges to seal the ammonia. Three methods have been considered, bonding mechanical fastening, and vulcanizing. Two adhesives were applied to test panels and exposed to ammonia for seven days. The adhesives were a silicone and a neoprene. Both failed by delamination. It is possible that the neoprene adhesive was not thoroughly cured which contributed to its failure. However, since this may also occur during normal shop assembly, bonding may be a high risk approach. Mechanical fastening is possible but adequate clearances for fastener heads must be allowed as well as tapped holes in the shroud plates. The best approach appears to be vulcanizing strips of soft rubber to the edges of the shroud plates. SOC-CO plastics has adequate autoclave facilities to perform this work and it is recommended that Southwestern Engineering request a quotation from them as soon as possible. The holes may be drilled or punched after the gasket application.

LAR:dj

DEFENSE AND SPACE SYSTEMS GROUP

ONE SPACE PARK • REDONDO BEACH • CALIFORNIA 90278

5515.2.79- 160

INTEROFFICE CORRESPONDENCE

TO: J.E. SNYDER
CC: L.A. ROSALES
M.H. SEIDMAN
A.M. SPROUSE
L.J. MEYERS

DATE: April 5, 1979

SUBJECT: 1 MWe FLOODED BUNDLE SHROUD SEALING

FROM: *M.P. Bianchi*
M.P. BIANCHI
BLOG. MAIL STA. EXT.
01 2230 52303

Our recommendation for sealing the shroud components is by the use of neoprene sheet stock gaskets. It is desirable from the standpoint of handling and fabrication to bond the gaskets to one of the shroud component faying surface's. We recommend GACO N-29 two component neoprene bonding agent or an RTV silicone one component adhesive/sealant such as RTV 731 or 732. The silicones must be checked for compatibility (which we are now doing) with the neoprene since inhibition can occur from certain plasticizers used in neoprene stocks.

DEFENSE AND SPACE SYSTEMS GROUP
ONE SPACE PARK • REDONDO BEACH • CALIFORNIA 90278

5515.2.79-474

INTEROFFICE CORRESPONDENCE

TO: J. E. Snyder

CC: L. D. Berman
M. P. Bianchi
M. H. Seidman
A. M. Sprouse
F. C. Weage

DATE: 17 October 1979

SUBJECT: Evaporator Skirt Coating - 1 MWe Heat Exchanger

FROM: L. A. Rosales
BLDG. 01 MAIL STA. 2220 EXT. 52630

Reference: 1) IOC 5515.2.79-463, "Implications of Procedure Changes on the Evaporator - 1 MWe OTEC Heat Exchangers", L. A. Rosales to J. E. Snyder, dated 17 October 1979.

The design of the 1 MWe Evaporator calls for the coating of the waterbox and skirt areas with a neoprene based coating (Gaco Western N-200-1 with a N-11R primer). The waterboxes have been coated but the skirt and the waterbox/skirt seal weld area are bare and must be coated on-site at Portland, Oregon. The operation may be hazardous and time consuming, see reference 1. As a result, there has been discussion concerning alternate coating materials. While an alternate material would still be hazardous to apply, the curing time would be much shorter.

The reasons for the coating are to prevent general corrosion and erosion-corrosion of the steel waterboxes and skirts and to prevent galvanic corrosion of the steel at the steel/titanium interface on the tubesheet. Neoprene was selected due to its outstanding erosion resistance and its long history of successful service in similar seawater applications. Life data of up to 32 years is available. The galvanic couple presents the most critical problem. The coating must overlap and bond to the titanium cladding in order to exclude water from the interface. This is especially important; local debonds between the titanium cladding and tubesheet are present. These debonded areas can trap water causing corrosion of the steel and, eventually, a possible leak path to the tube field. Titanium is very difficult to bond and few coating systems will form a reliable, moisture tight overlap. TRW has applied and tested the neoprene coating to titanium. A steel/titanium test joint has successfully withstood one year of exposure to seawater with no visible degradation.

It is strongly recommended that the neoprene coating be used rather than a substitute due to the lack of use data for other coating systems with titanium.