INNOVATIVE ATMOSPHERIC PRESSURE DESALINATION

Final Report

James R. Beckman
Arizona State University
Tempe, AZ

#52

Agreement Number 98-FC-81-0049
Desalination Research and Development Program

September 1999

U.S. Department of the Interior
Bureau of Reclamation
Denver Office
Technical Service Center
Environmental Resources Team
Water Treatment Engineering and Research Group

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY ***(Leave Blank)***

2. REPORT DATE
September, 1999

3. REPORT TYPE AND DATES COVERED
Final

4. TITLE AND SUBTITLE
Innovative Atmospheric Pressure Desalination
Desal R&D Program No.

5. FUNDING NUMBERS

6. AUTHOR(S)
Dr. James R. Beckman

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

Arizona State University
Department of Chemical, Bio and Materials Engineering
PO Box 876006
Tempe, Az 85287-6006

8. PERFORMING ORGANIZATION REPORT NUMBER

NA

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)
Bureau of Reclamation
Denver Federal Center
PO Box 25007
Denver CO 80225-0007

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

11. SUPPLEMENTARY NOTES

12a. DISTRIBUTION/AVAILABILITY STATEMENT

Available from the National Technical Information Service, Operations Division, 5285 Port Royal Road, Springfield, Virginia 22161

12b. DISTRIBUTION CODE

13. ABSTRACT ***(Maximum 200 words)***

A relatively new non-traditional and innovative heat efficient tower process, referred to as Dewvaporation, has been investigated and is now operational at Arizona State University. Dewvaporation technique is a specific process of humidification-dehumidification desalination, which uses air as a carrier-gas to evaporate water from saline feeds and dew form pure condensate at constant atmospheric pressure. The heat needed for evaporation is supplied by the heat released by dew fall condensation on opposite sides of a heat transfer wall. Since only a small amount of external heat is needed to establish temperature differences across the wall and since the temperature of the external heat is versatile, the external heat source can be from waste heat, from solar collectors or from fuel combustion. The unit is constructed out of thin water wettable plastics and operated at pressure drops of less than 0.1 inches of water.

14. SUBJECT TERMS--
Water Treatment, Desalting, Thermal Desalination, Dewvaporation

15. NUMBER OF PAGES
58

16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
UL	UL	UL	UL

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)

Prescribed by ANSI Std. 239-18
298-102

INNOVATIVE ATMOSPHERIC PRESSURE DESALINATION

Final Report

James R. Beckman
Arizona State University
Tempe, Arizona

Agreement Number 98-FC-81-0049
Desalination Research and Development Program

September 1999

U.S. Department of the Interior
Bureau of Reclamation
Denver Office
Technical Service Center
Environmental Resources Team
Water Treatment Engineering and Research Group

U.S. Department of the Interior
Mission Statement

The mission of the Department of the Interior is to protect and provide access to our Nation's natural and cultural heritage and honor our trust responsibilities to tribes.

Bureau of Reclamation
Mission Statement

The mission of the Bureau of Reclamation is to manage, develop, and protect water and related resources in an environmentally and economically sound manner in the interest of the American public.

Acknowledgement

This work was supported financially by the Department of the Interior, Bureau of Reclamation under Agreement No.98-FC-81-0049. The research team wishes to thank Mr. Mark Lichtwardt for his keen interest in and suggestions regarding the investigated Dewvaporation Technique.

Table of Contents

Appendices

List of Tables

List of Figures

Glossary

A	Heat transfer area (ft^2)
c_p	Heat capacity (BTU/lbmole °F)
F	Feed flow rate (lbmole/sec)
FB	Brine flow rate (lbmole/sec)
FD	Distillate flow rate (lbmole/sec)
G	Carrier gas flow rate (lbmole/sec)
ΔG	Carrier gas bleed flow rate (lbmole/sec)
$\underline{h}$	Molar enthalpy (BTU/lbmole)
h	Heat transfer coefficient (BTU/hr ft^2 °F)
HD	Hydraulic diameter (ft)
k	Thermal conductivity (BTU/hr ft °F)
L	Liquid flow rate at any position in the tower (lbmole/sec)
M	Mass transfer factor
$\dot{n}$	Molar flow rate (lbmole/sec)
P_w	Vapor pressure of water (psia)
P	Total pressure (psia)
P_f	Production density (lb/hr ft^2)
Nu	Nusselt number
Q	Energy input at the top of the tower (BTU/sec)
q	Heat flux (BTU/hr ft^2)
r	Depth of the flow media (ft)
R	Gas constant (BTU/lbmole °F)
RH	Relative humidity
S	Salinity at any position in the tower
T	Temperature (°F)
t	Thickness of heat transfer wall (ft)
U	Overall heat transfer coefficient (BTU/hr ft^2 °F)
V	Vapor loading (lbmoles of water vapor per lbmoles of carrier gas)
w	Width of the flow media (ft)
x	Mole fraction of water at any position in the tower
Z	Height of the flow media (ft)

Subscripts

B	Brine stream
D	Distillate stream
d	Dewformation side
e	Evaporation side
f	Liquid film
g	Gas
h	Top of the tower
x	Liquid phase
y	Gas phase
0	Bottom of the tower
ref	Reference point
RH	Relative humidity

LM	Logarithmic mean
Greek	
λ	Heat of vaporization of water (BTU/lbmole)
δ	Liquid film thickness (ft)
Γ	Gamma (lb/hr ft)
ρ	Density (lb/ft^3)
μ	Viscosity (lb/ft sec)
α	Temperature derivative of the heat capacity
β	Temperature derivative of vapor loading

1. EXECUTIVE SUMMARY

A relatively new non-traditional and innovative heat efficient tower process, referred to as Dewvaporation, has been investigated and is now operational at Arizona State University. All of the TASKS were successfully investigated resulting in the design and operation of a new demonstration bench-scale tower capable of 12 gallons (0.0454 m^3) per day of condensate. Two towers were built sequentially and operated for improvement of performance. Desalination of mild brackish water (800 ppm total dissolved solids (TDS)) had an average gain output ratio (GOR) of 7.8. Seawater desalination was demonstrated with a GOR of 7.5. Accounting for steam supply loss, the GOR values could be increased to 16.7 and 20. Tower number 2 is under revision to reduce heat loss at the top piping area which will further improve the observed GOR values.

Dewvaporation technique is a specific process of humidification-dehumidification desalination, which uses air as a carrier-gas to evaporate water from saline feeds and dew form pure condensate at constant atmospheric pressure. The heat needed for evaporation is supplied by the heat released by dew fall condensation on opposite sides of a heat transfer wall. Since only a small amount of external heat is needed to establish temperature differences across the wall and since the temperature of the external heat is versatile, the external heat source can be from waste heat, from solar collectors or from fuel combustion. The unit is constructed out of thin water wettable plastics and operated at pressure drops of less than 0.1 inches of water.

This bench-size database allows design for and demonstration of this technology at the 1000 gallon per day capacity. The projected capital cost for a 1000 gallon (3.79 m^3) per day unit based on laboratory data has been estimated at \$1397 which includes water heater, two pumps, one air fan (pumps and fan require 0.46 kWh per day of electricity) and a manufacturers gross margin of 30%. Totally inclusive operating cost would be \$3.35 per 1000 gallons of condensate. Lower operating costs of about \$1.50 per 1000 gallons of condensate should be realized by solar energy or atmospheric steam waste heat. The unit dimensions would be 4 feet by 4 feet by 8 feet high (1.22 m by 1.22 m by 2.44 m high).

A test unit was built to study the scaling phenomenon in the Dewvaporation process. No scaling was observed on the heat transfer walls after 2/3 of seawater and simulated seawater were evaporated at 190°F (87.78°C). Evaporation at the air/water interface instead of boiling at the wall/water interface seems to be the key to scale elimination inherent in this evaporative process.

2. BACKGROUND AND INTRODUCTION

The technology, Dewvaporation, investigated here involves the desalination of seawater and brackish water, which may find an economic niche in small plant applications.

Many technologies have been used to perform the required desalination resulting in preferred technologies based on economics (Fosselgard and Wangnick, 1989). For example, in the desalination of mild brackish (less than 1000 ppm TDS) water, Reverse Osmosis (RO) is superior to all desalination technologies. This is mainly a reflection of the fact that other technologies involve phase change (boiling) where as RO employs low-pressure pumps (less than 100 psia (6.90 bars)) to force water through semi-permeable membranes resulting in less energy consumption than that involved in a boiling process. One area where RO is ineffective in water purification is in the treatment of waters containing non-filterable suspended particulates. For example, the Colorado River contain silt in the 1 micron range which tend to foul RO membranes, thusly, increasing the maintenance and/or pretreatment costs of RO operation.

For the more TDS intense aqueous applications such as waste streams and seawater, other mechanical and thermal technologies economically compete with RO as seen by Larson *et al.* (Larson *et al.*, 1989a, 1989b). In the case of seawater desalination, the RO pump pressures increase to 1000 psia (68.95 bar) and feed waters require expensive pre-treatments in order to protect and extend the life of the membranes.

The competitive technologies to RO for seawater desalination include Mechanical Vapor Compression (MVC), Multi-Stage Flash Distillation (MSF), and Multi-Effect Distillation (MED) with and without Thermal Vapor Compression. The MVC needs shaft power to drive its compressor. The motor can be either electrically or thermally driven. For electrically driven MVC, MVC plants consume more electricity than RO units in the same seawater service. The other processes dominantly use and reuse heat as the main driver to affect temperature-driving force between boiling and condensing at staged pressures. The thermally driven plants attempt to reuse the high temperature applied heat as many times as is economically possible to minimize operating costs. This energy reuse factor economically varies from 6 to 12. As the GOR increases so does the equipment capital cost. The optimum GOR value depends on factors such as plant capacity, cost of energy, cost of materials, interest and tax rates.

2.1 Dewvaporation Philosophy

The humidification-dehumidification process involved in this paper focuses on its economic "best" niches for desalination. A potential niche fit of the Dewvaporation technology could be in the reduction of the volume of reject saline waters by an order of magnitude from RO plants such as the Yuma facility for Colorado River desalination and small seawater desalination applications.

The Dewvaporation Continuous Contacting Tower focuses on a relatively new non-traditional and innovative heat driven process using air as a carrier-gas and remaining at atmospheric pressure throughout the device. The heat source can be from low temperature solar (131°F, 55°C), waste heat, or combustible fuels (210.2°F, 99°C). Briefly, the process works for brackish desalination as viewed in Figure 1:

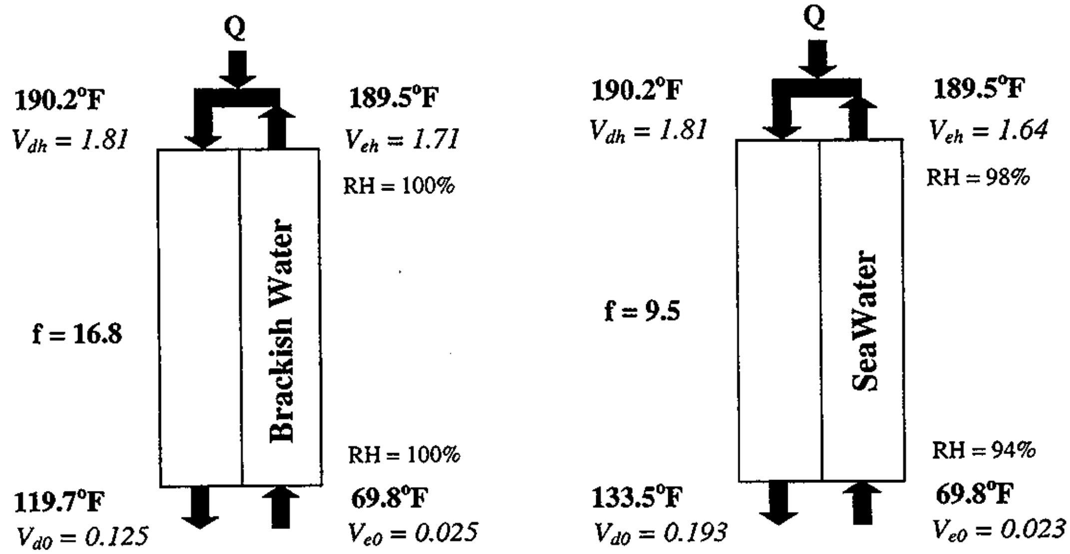


Figure 1.—Dewvaporation Continuous Contacting Tower Design Schematic

A carrier-gas such as air is brought into the bottom of the tower on the evaporation side of a heat transfer wall at a typical wet bulb temperature of 69.8°F (21°C), thereby containing about 0.025 moles of water vapor per mole of air. The wall is wetted by saline feed water, which is fed into the evaporation side at the top of the tower. As the air moves from the bottom to the top of the tower, heat is transferred into the evaporation side through the heat transfer wall allowing the air to rise in temperature and evaporate water from the wetting saline liquid which coats the heat transfer wall. Concentrated liquid leaves from the bottoms of the tower and hot saturated air leaves the tower from the top at 189.3°F (87.4°C) with a humidity of 1.71 moles of water vapor per mole of air. Heat is added to this hot air by an external heat source (in this investigation steam was used) increasing the air humidity and temperature to a V of 1.81 and 190.2°F (87.9°C) respectively. This hotter saturated air is sent back into the top of the tower on the dew formation side. The dew formation side of the tower, being slightly hotter than the evaporation side, allows the air to cool and transfer condensation heat from the dew formation side to the evaporation side. Finally, pure water condensate and saturated air leave the dew formation side of the tower at the bottom at 119.7°F (48.7°C). Total external heat needed is made up of the heat needed at the top to establish a heat transfer temperature difference and the heat needed to establish a temperature off-set between the saline feed stock and the pure water condensate. The detrimental effect of salt concentration on the energy reuse factor (or gain output ratio), f as seen in Figure 1, is given by equation 1.

$$F_{RH} = 1-(1-RH)\cdot(1+f)\cdot(1+V_{eh}) \qquad [1]$$

2.2 Predictive Model

From Figure 1 the mathematical definition of the energy reuse factor, f, is the ratio of the energy transferred through the heat transfer wall to the high temperature energy input as shown in equation 2:

$$f = \frac{V_{dh} - V_{d0}}{V_{dh} - V_{eh}} \tag{2}$$

The definition of the molar production flux, P_f, is the gas traffic times the water vapor decrease of the dew formation side of the wall divided by the wall area as shown in equation 3:

$$P_f = \frac{G}{A} \cdot (V_{dh} - V_{d0}) \tag{3}$$

Typically, the feed/condensate temperature offset is kept to 10°F (5.6°C). This can be accomplished by either an internal or external feed heat exchanger. In this analysis, the energy reuse factor, f, was 16.8. By including the heat needed for the temperature offset, the factor reduces to about 13. Actually, the product of the factor and the molar production flux, P_f, is a constant at parametric V_{eh}. The value of the constant is a function of the operating variables as shown in the following equations.

The amount of water vapor contained in the air carrier-gas is calculated by specifying the temperature, T, and calculating the vapor pressure, P_w, from equation 4 (Smith and VanNess, 1987).

$$\ln P_w = B - \frac{\lambda}{R \cdot T} \tag{4}$$

where B and λ are constants obtained by fitting a straight line to the $\ln(P_w)$ versus 1/T for the steam table. For temperature range of 32 - 212°F (0 – 100°C), B is 14 and λ/R is 5209 K (Perry, Green, and Maloney, 1984). The moles of water vapor per mole of air is then:

$$V = \frac{RH \cdot P_w}{P - RH \cdot P_w} \tag{5}$$

where the relative humidity (RH) is given as a function of salinity (S) by the following equation (Spiegler and Laird, 1980)

$$RH = 1 - 0.000538 \cdot S \tag{6}$$

The hottest temperature in the evaporating section is specified allowing the calculation of the largest value of the V_{eh} in the evaporating section of the unit. Then the change in vapor content of the carrier-gas is specified across the top of the tower by:

$$\Delta V = V_{dh} - V_{eh} \qquad [7]$$

From these specifications, the temperature difference across the heat transfer wall at any position can be described as:

$$\frac{1}{\Delta T_{LM}\big|_z} = \left[\frac{B^2 \cdot R}{\lambda}\right] \cdot \left[\frac{\left(1 + \Delta V + V_e\big|_z\right) \cdot V_e\big|_z}{\Delta V}\right] \qquad [8]$$

In this process both the film heat and mass transfer coefficients are important in establishing the overall effective heat transfer coefficient, U. For simultaneous heat and mass transfer operations involving air and water, the Lewis Number is essentially unity (McCabe, Smith, and Harriott, 1993) allowing the coefficients to be related by similitude as $k_y = h_g/c_p$. The effect of the latent energy associated with the mass transfer of water vapor can be related to the sensible heat transfer associated with the air/vapor mixture by equation 9 after Werling (Werling, 1990).

$$h_f\big|_z = h_g\big|_z \cdot \left(1 + M\big|_z\right) \qquad [9]$$

where M is expressed as:

$$M = \left(\frac{\lambda}{RT}\right)^2 \cdot \left(\frac{R}{c_p}\right) \cdot V \qquad [10]$$

Taking into account both gas film heat transfer coefficients and the thermal resistance of the heat transfer wall, then the overall effective heat transfer coefficient, U, can be expressed as:

$$\frac{1}{U\big|_z} = \frac{1}{h_{fe}\big|_z} + \frac{1}{h_{fd}\big|_z} + \frac{t}{k} \qquad [11]$$

The heat transferred through the heat transfer wall, is essentially the latent heat needed to evaporate water as:

$$q\big|_z = G \cdot \lambda \cdot \left(V_e\big|_{z+\Delta z} - V_e\big|_z\right) \qquad [12]$$

The area needed for the heat transfer wall is obtained by an energy balance (Bird, Stewart, and Lightfoot, 1960).

$$\frac{A\big|_z}{q\big|_z} = \frac{1}{U\big|_z} \times \frac{1}{\Delta T_{LM}\big|_z} \qquad [13]$$

where

$$\Delta T_{LM} = T_{yd} - T_{ye} \qquad [14]$$

Upon integrating with respect to the overall area and assuming that t/k is small compared to the gas phase resistance, equation 15 then relates the total energy reuse factor, f, and the total production flux, P_f, as follows:

$$f \cdot P_f = \left\{ \left[\frac{\lambda}{B \cdot R \cdot T} \right]^2 \cdot \left[\frac{h_g}{C_p} \right] \right\} \cdot \left(\frac{V_{eh}}{2 + V_{eh}} \right) \cdot (18) \cdot F_{RH} \qquad [15]$$

Equation 15 shows that as the temperature increases, the product of energy reuse factor and molar production flux become greater. It is also apparent that the energy reuse factor, f, and the molar production flux, P_f, are essentially related hyperbolically in a specified unit. The detrimental effect of salt concentration is also included in this expression from Equation 1.

Additionally, higher values of V_{eh}, i.e. higher temperatures, improve both f and P_f values, which is economically beneficial to the tower. However, higher temperatures are limited to the heat source temperature and the normal boiling point of water.

3. CONCLUSIONS AND RECOMMENDATIONS

From this investigation into the Dewvaporation process it can be concluded that:

- The process is capable of brackish and seawater desalination possessing a GOR of 13.1 for production density of 0.05 lb/hr ft^2 (0.24 kg/hr m^2) of heat transfer area.
- The average Operating Costs for brackish and seawater are \$3.35 per 1000 gallons (3.79 m^3) for small plants that could reduce to \$1.55 with solar energy and \$1.52 with atmospheric waste steam.
- The Capital Cost for brackish and seawater plants is \$1397 for a 1000 gallon per day plant with 6400 ft^2 (594.6 m^2) of heat transfer wall.
- Use of the horizontal serpentine airflow configuration is required for excellent air distribution needed for heat transfer.
- Enhanced heat transfer coefficients were found only with entrance region effects and not used in the final cost studies.

It is recommended that:

- Tower number 2 should be modified to reduce the heat loss from the top section piping so that higher GOR values can be demonstrated.
- A 1000 gallon per day test plant be constructed at a projected commercial cost of under \$1500. This capacity should allow the design of 100,000 gallon per day (378.5 m^3) plants leading to the million gallon per day size.
- The 1000 gallon per day unit should be mobile so that brackish water, seawater and effluent waters from reverse osmosis water treatment plants can be treated. These operations will further demonstrate the versatility and economics of the Dewvaporation process.

4. WORK PERFORMED

In summary of the work performed, Table 1 is the TIMETABLE that was proposed. Just prior to starting this investigation an original tower was constructed. However, it was not operated until January 1999. The initial tower was plagued with the preconceived notions of how to effectively distribute water and air flows. The heat transfer walls were spaced with ¼ inches (6.35 mm) dimensions for heat transfer coefficient enhancement but 80% of the area had touched and became ineffective. The condensate side also leaked so the condensate rate had to be estimated by energy balance. The tower is shown in Figure 2.

Table 1 : Research Work and Schedule

OBJECTIVE / TASK	Oct	Nov	Dec	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sept
1.0 h_g Improvement												
1.1 Build Test Unit	**											
1.2 Entrance Region	**	*										
1.3 Wall Patterns		***										
1.4 Scaling			***									
1.5 Gas Flow Oscillations				****	*							
2.0 New Wall Material												
2.1 Search	***	****	****	****	*							
2.2 Wetness Test					*							
3.0 Mathematical Model												
	**	****	****	****	*							
4.0 Tower Design												
					**	****	**					
5.0 Tower Procurement and Construction												
5.1 Procurement of Materials					**	****	****	****				
5.2 Construct							****	****				
6.0 Tower Operation												
6.1 Brackish Water									****	**		
6.2 Sea Water										**	****	
7.0 Final Report												****

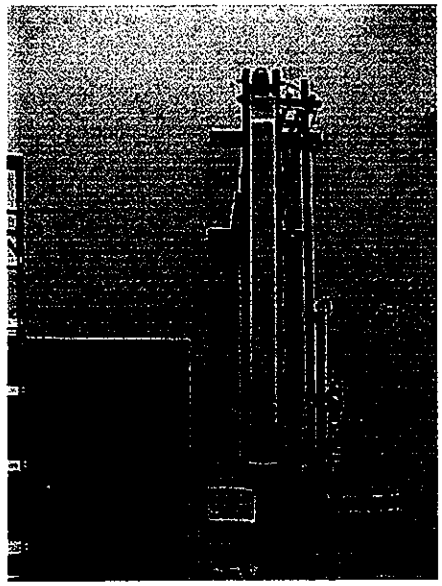


Figure 2.—Dewvaporation Continuous Contacting Tower

The original tower (Figure 2) established the need to pre-investigate the delineated TASKS of Table 1 before a successful tower could be built and operated.

TASK 1 involved the investigation into achieving the largest heat transfer coefficient possible while staying within a reasonable range of operability (i.e. temperatures, pressure drops, wall spacing) and construction ease and potential costs. Some improvements were found but only by 10% to 20% and those mainly due to laminar entrance region effects.

TASK 2 searched for new water wettable plastics that would be inexpensive and better wetting than the REXAM sheet already known. Nothing was found to be better than REXAM except polyester gauze that could wet even on polyethylene. The most interesting event was the lowering in cost of the REXAM from $0.50/ft^2 ($5.4/m^2) to about $0.10/ft^2 ($1.1/m^2) as wholesale large quantity. A different heat transfer wall material composite was researched. This composite is made up of 2 mil (0.0508 mm) polypropylene and nylon cheesecloth which costs $0.04/ft^2 (0.43/m^2).

TASK 3 was the building of a mathematical model that could be used in the designing of a new tower and in the analysis of subsequent tower data. The model was made with a VISUAL BASIC software package and is very user friendly.

TASKS 4, 5, and 6 involved the new tower design, construction and operation that would incorporate the results from all of the preliminary TASKS (1 through 3). The first tower (Tower 1) was designed with six stages that had a wall spacing of ½ inch (12.7 mm) to help insure no touching of the walls. One of these towers was built to pilot the design. But the walls did touch blocking off most of the wall area and misdirecting the airflows on each side of the wall. A new second tower (Tower 2) requiring only one stage was designed that incorporated a horizontal open-cell foam spacer. This spacer concept effectively separated ¼ inch (6.35 mm) spaced walls and insured balanced and well-directed airflow patterns. This tower was successful. Perhaps all of the problems that we encountered explain why this simplified tower technique had not been previously investigated.

The following is a detailed summary of the TASKS as listed in Table 1.

4.1 Summary of TASK 1.0 (h_g improvement)

A test unit for the determination of enhanced heat transfer coefficients was built in the ASU laboratory as shown in Figure 3. The insulated 1 foot (30.48 cm) wide by two feet (60.96 cm) high heat transfer wall unit was hinged for easy access so that the walls could be checked and re-welled during a test run. Thermocouples were used to determine wet-bulb and dry-bulb temperatures of incoming and exiting air. A hair dryer was used to pull the air stream through the test unit with the air flowrate being measured by a vain anemometer after the blower.

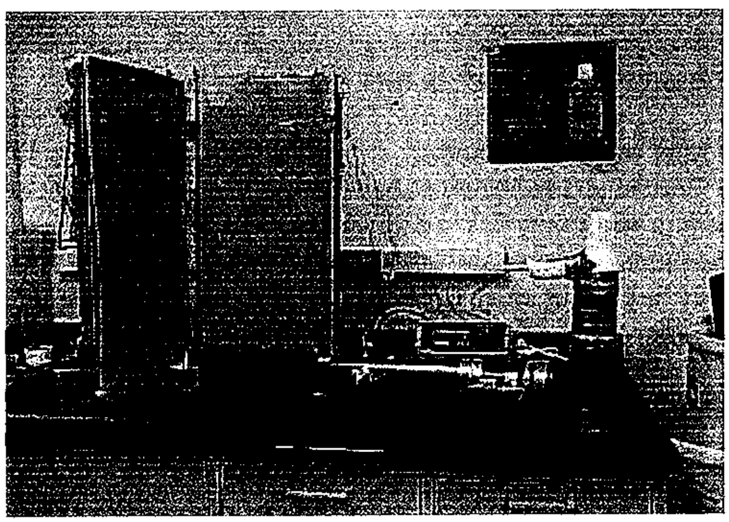


Figure 3.—Heat Transfer Coefficient (h_g) Test Equipment

Tables A1 and A2 in Appendix 1 is a summary of the heat transfer coefficients achieved by the application of the noted variables along with the resultant Reynolds Numbers and Nusselt Numbers (both data based and theoretical).

Data on Nov 12 appears to show agreement with the entrance region effect for laminar flow. That is the entrance region for laminar flow has a length of: L/D = 0.05*Re where the Prandtl Number is about 1. So for this data Re = 1300 and D = 0.5 inches (12.7 mm), therefore L = 32 inches (81.28 cm). The test device was 24 inches (60.96 cm) long allowing the improvement of the Nusselt Number over the region of flow. The test device was not made longer since the dry and wet bulb temperature difference would become error enhanced due to a temperature pinch approach. Therefore, although it's an interesting result, this kind of entrance region can not be repeated in a ten foot tall tower.

Only possible repetitive patterns can be built in the available space for a slot flow situation. The turbulent flow run showed a ratio of 0.9 which agreed with literature in that no entrance region existed and there was no enhancement in the coefficient.

Data on Nov 20 showed the results of a wavy pattern in laminar flow. There was no effective entrance region effect probably due to the flow distortion, but surprisingly, there was no enhancement effect.

Data on Nov 24 verified the laminar data of Nov 20 and showed a reduction in the heat transfer coefficient for turbulent flow at Re = 6671. This reduction may have been caused by the eight horizontal spacers used to establish the wavy sheet pattern which could have stabilized the turbulent flow at 8 positions in the device.

On Nov 25 it was visually determined that the water wettable plastic film dried out in significant regions on the plastic surface during a 10 minute run, so wet blotting paper was substituted for the plastic sheet. Drying of the surface would be reflected in a false calculated heat transfer coefficient which would be lower than that achieved (since the wetted area was actually reduced). This may help to explain the apparent lowering of the heat transfer coefficients in the previous days results for the rapidly drying surfaces at turbulent flow. Again the laminar flow runs show the entrance region enhancement effects as observed in the Nov 19 runs.

The Dec 3 runs were performed by students as a repeat of the Nov 25 runs by Dr. Beckman. The results reflected enhanced laminar flow coefficients but showed enhancement of turbulent flow coefficients with a slot spacing of 0.5 inches (12.7 mm). During those runs it was observed that the wetted sheets had become detached from the wall and may have happened for runs 3, 4, and 5. As expected, the turbulent flow run 6 showed excellent agreement with theory for a 0.25 inch (6.35 mm) slot.

The ¼ inch slot runs on Dec 4 and Dec 7 varied from laminar flow to turbulent flow conditions and showed 10% enhancement for laminar flow, but only theoretical coefficients for turbulent conditions.

On Dec 9 the slot was set to ½ inch and a first pass at air oscillatory air flow was tried. The results reflected steady state enhancement for laminar flow (about double theoretical) and a 30% enhancement for turbulent flow. Essentially, this first attempt at oscillatory behavior showed no improvement over the enhancements found at steady state conditions. More precise fitting equipment with higher frequency capability was implemented in January as scheduled to ascertain the enhancement affected by the laminar/turbulent transition zone.

The remainder of the runs were performed by Dr. Beckman from Dec 15 to Dec 18. On Dec 15, wet cheesecloth gauze was placed on the wet plotter paper to observe the enhancement effects of a rough patterned surface. Runs 1, 2 ,7 and 8 were performed with ½ inch (12.7 mm) flat wall geometry to observe the effect of Reynolds Number. High Reynolds Number of 6668 showed theoretical Nusselt Numbers but Reynolds Number of about 3500 showed an enhancement of about 30%. This result was replicated for verification. Laminar flow (run #1 on Dec 16) reflected enhanced behavior as previously observed. The effect of a wavy wall pattern was also tested on Dec 15. The wave was established by porous and non-porous horizontal spacers which produced a ½ inch amplitude wave with an 8 inch period. Runs 3, 4, 5 & 6 showed no enhancement effect for transition to turbulent Reynolds Numbers.

On Dec 16 the slot dimension was reduced to ¼ inch and flat wall geometry. Laminar and transition Reynolds Number flow produced either theoretical to detrimental effects on obtained Nusselt Numbers. Turbulent Reynolds Numbers could not be obtained due to blower pressure drop restrictions.

Dec 18 runs determined the effects of diagonally folded walls on the enhancement of Nusselt Numbers. Results showed theoretical to detrimental effects on the Nusselt Number with increasing Reynolds Number. This wall pattern reflected that which is used in flat plate heat exchanger design and as such was expected to improve the heat transfer coefficients. Although it appears that flat plate exchangers process liquids whereas this experiment involved gases, the difference in Prandtl Numbers (8 for liquids and 0.7 for air) may have an impact on the results.

From these tests the preliminary conclusions were established:

(1) ½ inch (12.7 mm) slot with laminar flow results in coefficient enhancements of 200% to 400% over theoretical for high turbulent flow.

(2) ¼ inch (6.35 mm) slot with laminar flow showed no enhancement of coefficients

(3) Higher Reynolds Numbers result in higher heat transfer coefficients for turbulent flow conditions as per theoretical

(4) Random fold, wavy and diagonal folded walls are equivalent to flat walls in enhancement effects showing 30% enhancement.

4.1.1 Task 1.4 (Scaling)

The scaling experiments were carried out on a bench scale unit that consisted of a thermal control grill mounted vertically with a sample of the plastic heat transfer sheet (with and without gauze) attached to its surface.

The unit was covered with a plastic wall allowing the air to rise in temperature, simulating the operation of the tower. Salt solutions that were tested consisted of the pacific ocean seawater and simulated seawater. Simulated seawater was prepared according to ASTM method number D1141-90 (1992) with salinity of 35 g/kg. Throughout the experiments, temperatures of the liquid film were measured along the center of the wall using thermocouples.

The test unit for scaling of simulated seawater and actual seawater was built and shown in Figure 4. Salt solution was pumped from a basin to the top of the unit and distributed unto the plastic wall by a tube with multi-punched holes. A covering allowed air to rise, heat and evaporate water from the hot solution in such a way so that the air left the top of the unit at about 190°F (87.78°C). Temperatures of the liquid falling film were monitored by three thermocouples that penetrated the plastic cover. This was the concept of a "best " simulation of the main Dewvaporation device.

Results showed that after 82% to 85% recovery of water, solids formed as crystals. The solids washed down the plastic wall and accumulated in the bottom reservoir. Some solids did get re-pumped back to the unit but did not adhere to the heat transfer surface. This phenomena is due to the fact that evaporation occurs at the liquid-air interface and not on the heat transfer wall. The solids that formed accumulated to some extent and fell to the bottom reservoir.

Seawater desalination has been planned to intake 3.5% salt solution with final rejection of 10% salt that reflects a 67% recovery. Therefore in the Dewvaporation process as proposed, solids formation should not occur.

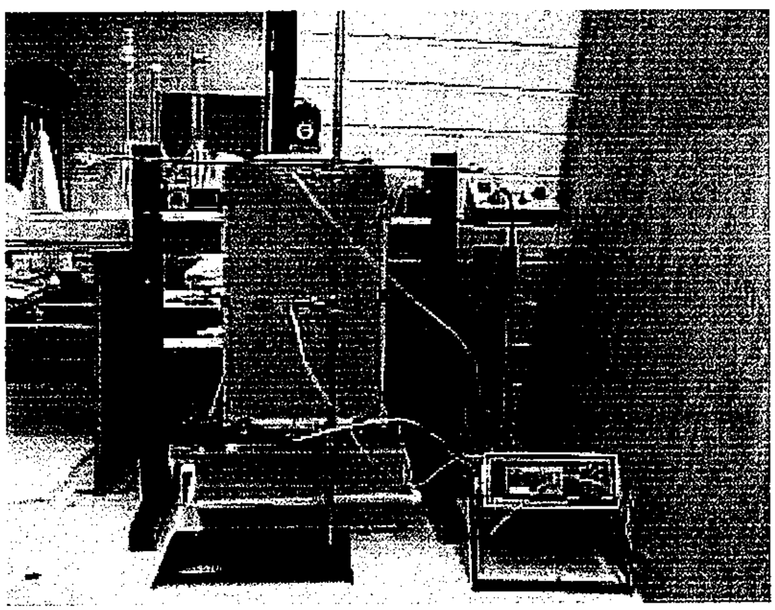


Figure 4.—Scaling Test Apparatus

4.1.2 Task 1.5 (Gas Flow Oscillation)

Appendix A shows the effect of oscillatory frequency on the film heat transfer coefficient. A butterfly valve was rotated by a variable speed motor in order to vary the gas phase flow rate frequency. The frequency was varied from 6 to 32 cycles/sec. As observed, the high and low steady state flows which bounded the oscillatory runs, had heat transfer coefficients whose average value was about 10% lower than the average coefficients calculated as a function of flow frequency. Therefore oscillatory flow did enhance the coefficients but not as significantly as originally anticipated.

4.2 Summary of TASK 2.0 (New Wall Material)

4.2.1 Task 2.1 (Search)

New plastic materials that could be used as wetable heat transfer walls were found including modified existing material, new Mylars and gauze wrap. These investigations continue through January as scheduled. The company that manufactures the currently used wettable plastic is REXAM GRAPHICS. This material was found to be better wetting after boiling with water. REXAM expressed interest in producing a better wettable film that would wet better without a boiling requirement. It appears that a surfactant was depleted from the coating after boiling which left the plastic more wettable. This material is 3 mil polyester with a wettable coating and costs $0.55/ft^2 ($5.9/m^2) retail. The cost was assumed as $0.25/ft^2 ($2.7/m^2) wholesale for the original proposal which now has reduced to $0.10/ft^2 ($1.1/m^2) large quantity wholesale.

A suitable Mylar material manufactured by DuPont's Packing Group was located. Mylar LBT was identified as a viable candidate as it was 1 mil (0.0254 mm) thick and as wettable as is produced. They sent about 2000 square feet of both one sided and two sided 'Corona electric discharged " treated for wettability. Costing of 1 mil (0.0254 mm) Mylar is $20.00/lb ($44.1/kg) or $0.13/ft^2 ($1.4/m^2) retail. In bulk, the wholesale price should be about $0.08/ft^2 ($0.86/m^2). This material was tested for wettability but was found not as wettable as the REXAM sheet. The DuPont material can not be made more wettable.

Finally an investigation was made into the possibility of a composite cloth wrap on plastic in efforts to leave "no-stone-unturned" in any approach to improve wettability. A cheesecloth wide-weave gauze was identified and tested. The wettability of this material is superior to any stand alone plastic. The gauze sells for $0.02/ft^2 ($0.22/m^2) retail from ERC Products in Maine. Materials tested were made of cotton or polyester. Both wet extremely well, but the polyester is more chemically inert. An interesting composite might be the use of DuPont Mylar with a polyester gauze covering at a material cost of about $0.10/ft^2 ($1.1/m^2). This concept would halve the cost of the heat transfer material as originally proposed. A final composite of 2 mil (0.0508 mm) polypropylene and nylon cheesecloth was selected for the base economics as it costs $0.04/ft^2 ($0.43/m^2) retail and will cost even less at large volume wholesale. This composite performs identically to the REXAM and cotton cheesecloth composite but is more chemically and bio-chemically inert.

4.2.2 Task 2.2 (Water Wettable Test)

A test for wettability was devised to establish a wettability quantitative number. The test involved the minimum liquid film flow density, gamma, where (gamma = mass of liquid flow per time per width of flow) (Bird, Stewart and Lightfoot, 1960) needed to entirely wet the heat transfer wall from top to bottom. More is OK but less flow leaves a portion of the wall dry. On the evaporative side, where this gamma has dominant importance, a dry area on the wall is a totally dead zone involving little heat transfer since M = 0 in that dry zone (h = hg*(1+ M)). The values of M vary from 90 at the top to 2 at the bottom so wettability is always important. The test rig was eight feet high and six inches wide (2.44 m high and 15. 24 cm wide) as shown in Figure 5.

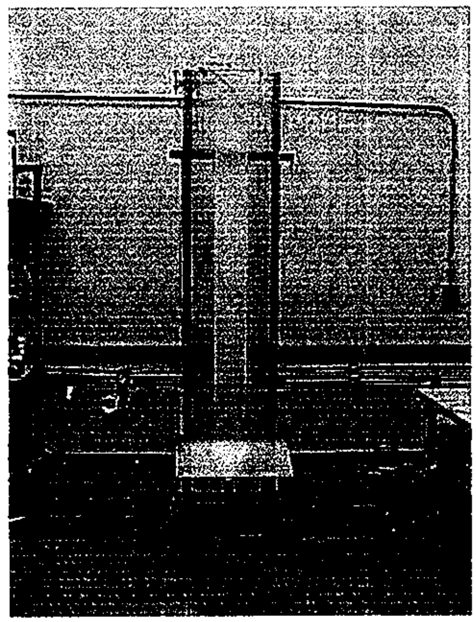


Figure 5.—Gamma Test Equipment

Materials tested along with their minimum values of gamma (100% coverage of the wall) are listed in Table 2. Clearly, gauze wrapped plastic is a superior wetting agent compared to any wettable plastic found so far. The minimum gamma can be viewed as an index that allows the width of liquid flow to be wider (thereby reducing the height of the tower as HEIGHT=HEAT TRANSFER AREA/WIDTH OF FLOW), or it can be viewed as the amount of liquid flow needed at the bottom of the tower if the liquid flow width has been set by some other constraint (thereby establishing the per cent water recovery). At any rate, smaller values of minimum gamma are preferred.

Table 2: Wettability index (MINIMUM GAMMA, lb /hr ft)

Wall Material	Min. Gamma (lb/hr ft)
Composite (Gauze covered Plastic)	4
REXAM (heat treated)	10
REXAM (untreated)	12
DuPont Mylar	68

These tests were repeated to observe the reduction in gamma for a wall wetting of 85%. Any reduction in gamma allows a shorter tower height design. Table 3 shows that for REXAM, the gamma reduced from 12 lb/hr ft (17.86 kg/hr m) at 100% coverage to 5 lb/hr ft (7.44 kg/hr m) for 85% coverage.

Table 3: Gamma for 85% Wall Coverage

Wall Material	Gamma (lb/hr ft)
Composite (Gauze covered Plastic)	1
REXAM	5

4.3 Summary of TASK 3.0 (Mathematical Model)

Theoretical model was developed based on energy and material shell balances. Model assumptions included: constant heat of vaporization, liquid film temperature is the average temperature of the gas phases, and feed temperature offset is set to be 5°F (2.78 °C) with respect to gas phase temperature of the evaporating side. Schematic of the tower is shown in Figure 6 which shows the inlet and outlet streams.

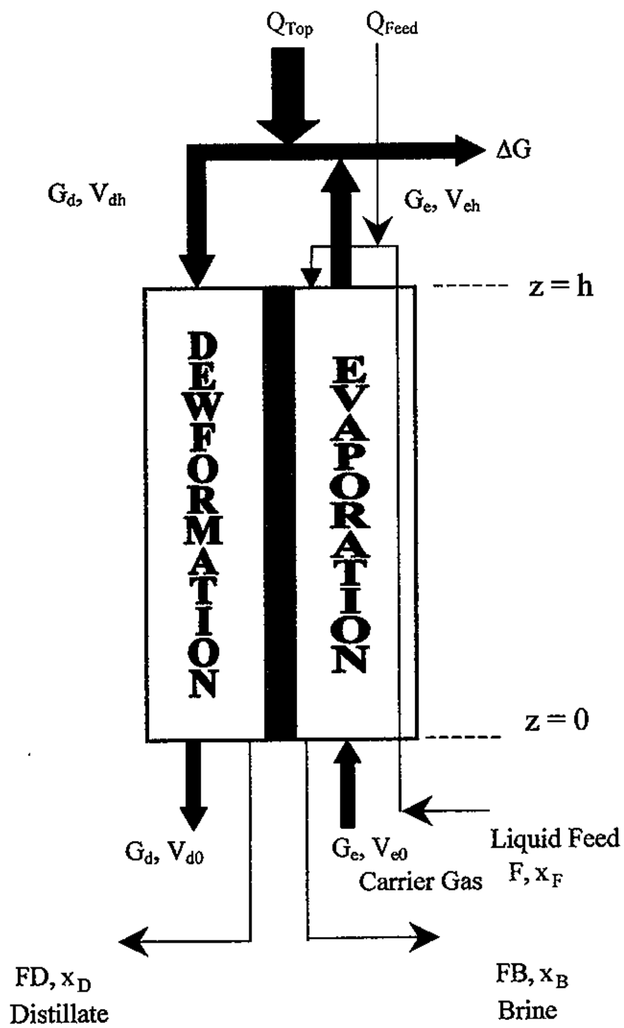


Figure 6.—Tower Model Schematic

In general the steady state energy balance is given by Equation 16:

$$\sum_{\text{all components}} \dot{n}_{in} \cdot \underline{h}_{in} - \sum_{\text{all components}} \dot{n}_{out} \cdot \underline{h}_{out} + Q = 0 \qquad [16]$$

Starting with shell balances on the evaporation side and the dewformation side, two ordinary differential equations can be obtained. These equations describe the temperature as a function of height (or area) on each side of the tower. After algebraic manipulation, the two differential equations become:

$$\frac{d}{dA}\left(T_{ye}\right) = -\left[\frac{1}{w}\right] \cdot \left[\frac{Evap2 \cdot Dew3 + Evap3 \cdot Dew1}{Evap1 \cdot Dew1 - Evap2 \cdot Dew2}\right] \qquad [17a]$$

$$\frac{d}{dA}\left(T_{yd}\right) = -\left[\frac{1}{w}\right] \cdot \left[\frac{Evap3 \cdot Dew2 + Evap1 \cdot Dew3}{Evap1 \cdot Dew1 - Evap2 \cdot Dew2}\right] \qquad [17b]$$

where

$$Evap1 = \left\{ \begin{array}{l} -c_{p_{water}} \cdot (T_{xe} - T_{ref}) \cdot G_e \cdot \beta_e - \dfrac{L_e \cdot (T_{xe} - T_{ref}) \cdot \alpha_{water}}{2} - \dfrac{L_e \cdot c_{p_{water}}}{2} \\ + G_e \cdot c_{p_{vapor}} \cdot (T_{ye} - T_{ref}) \cdot \beta_e + G_e \cdot V_e \cdot (T_{ye} - T_{ref}) \cdot \alpha_{vapor} + G_e \cdot V_e \cdot c_{p_{vapor}} \\ + G_e \cdot c_{p_{air}} + G_e \cdot \lambda \cdot \beta_e + F \cdot (T_F - T_{ref}) \cdot \alpha_{Feed} + F \cdot c_{p_{Feed}} + G_e \cdot (T_{ye} - T_{ref}) \cdot \alpha_{air} \end{array} \right\}$$

$$Evap2 = \left\{ \frac{L_e \cdot (T_{xe} - T_{ref}) \cdot \alpha_{water}}{2} + \frac{L_e \cdot c_{p_{water}}}{2} \right\}$$

$$Evap3 = \{q \cdot w\}$$

$$Dew1 = \left\{ \begin{array}{l} -c_{p_{water}} \cdot (T_{xd} - T_{ref}) \cdot G_d \cdot \beta_d + \dfrac{L_d \cdot (T_{xd} - T_{ref}) \cdot \alpha_{water}}{2} + \dfrac{L_d \cdot c_{p_{water}}}{2} \\ + G_d \cdot c_{p_{vapor}} \cdot (T_{yd} - T_{ref}) \cdot \beta_d + G_d \cdot V_d \cdot (T_{yd} - T_{ref}) \cdot \alpha_{vapor} \\ + G_d \cdot V_d \cdot c_{p_{vapor}} + G_d \cdot (T_{yd} - T_{ref}) \cdot \alpha_{air} + G_d \cdot c_{p_{air}} + G_d \cdot \lambda \cdot \beta_d \end{array} \right\}$$

$$Dew2 = \left\{ -\frac{L_d \cdot (T_{xd} - T_{ref}) \cdot \alpha_{water}}{2} - \frac{L_d \cdot c_{p_{water}}}{2} \right\}$$

$$Dew3 = \{q \cdot w\}$$

A computer code was written to numerically solve the system of equations (17a and 17b) based on the 4th order Runge-Kutta method. The model predicts the following: GOR, production density, temperature profile on the evaporation and dewformation sides, salinity as a function of height, liquid flow on each side of the tower, temperatures as a function of height, and the total area of the tower required to obtain the desired parameters.

The model was programmed into VISUAL BASIC PC software for user friendly application. An execution method and design example with display outputs (for run 806A, Appendix E) are located in Appendix B.

4.4 Summary of TASK 4.0 (Tower Design)

4.4.1 Design of Bench-Scale Tower Number 1

Taking into account all of the generated information accumulated in the first quarter of this investigation, the research team made an attempt at a plausible design of the second quarter task (Task 4.0). This design is subject to change based on continuing generated heat transfer coefficient and wettable material data. A new concern and important issue is the wettability of the heat transfer wall. The minimum gamma experiments clearly showed the needed wetting required to maintain a totally wet evaporative side wall. The value of gamma restricts the allowable width of the tower. A small value of minimum gamma allows more flexibility in the design as the actual gamma can be larger.

Table 4 shows the tower design resulting from using a 1 mil Mylar sheet with a polyester covering, GOR of 17, maximum temperature of 190°F (87.78°C), ½ inch (12.7 mm) slot spacing, maintenance of at least a minimum gamma and a range of Reynolds Number air flows using the corresponding values of the enhanced heat transfer coefficients of Table A1 and A2 (Appendix A).

Table 4: Tower Height as a function of Reynolds Number Range

Reynolds Numbers	Heat Transfer Coefficient, h_g BTU/hr ft^2 °F	Production Density, P_f lb water/hr ft^2	Height of Tower Needed ft	Wall Cost REXAM $*	Wall Cost Gauze $*
200-500	1.0 - 1.0	0.05	40	------	$700
500-1300	1.0 - 1.5	0.06	60	$1,460	$580
1300-3500	1.5 - 3.0	0.10	130	$875	$350
3500-9000	3 - 6	0.22	240	$400	$160

*cost of the heat transfer wall for a 1000 gallon/day facility

From Table 4 the conditions that gave the shortest tower or the least number of 10 feet towers connected in series was that given by the low laminar flow Reynolds Number range of 200 to 500. The laboratory bench scale unit would possibly consist of five 8 foot compartments (actually one 8 foot tower (2.44 m) with five vertical partitions). Each vertical compartment would contain, say, three evaporation and three dew formation chambers. If the chambers measured ½ inch thick by 12 inches wide and 8 feet high (12.7 mm thick by 30.48 cm wide by 2.44 m high), then the total heat transfer area would be 8 square feet/sheet x 6 sheets/chamber x 5 chambers = 240 ft^2 (22.3 m^2). The water production rate would be 240 ft^{2}x 0.05 lb/hr ft^2 = 12 lb/hr (5.44 kg/hr) condensate. The heat transfer wall would cost \$0.10/ft^2 x 240 ft^2 = \$24.00.

The entire bench scale tower would measure 12 inches by 15 inches by 8 feet (30.48 cm by 38.1 cm by 2.44 m) and require three extra pumps needed to transport inter-process evaporative liquids to the top of the next sequential tower. This formed the basis of the first design.

4.4.2 Design of Bench-Scale Tower Number 2

The design of tower number 2 was based on the conclusion that the air flows on opposite sides of the heat transfer wall were mal-distributed in tower number 1. To rectify the air flow pattern, sponge spacers were used to confine the air flow to narrower slots and to allow the slot dimension to be reduced to ¼ inch (6.35 mm) from ½ inch. This reduction in slot dimension allowed an improvement in the heat transfer coefficient as well.

4.5 Summary of TASK 5.0 (Tower Procurement and Construction)

4.5.1 Tower Number 1

The purchase and delivery of materials needed for the tower construction were obtained from January 1999 to May 1999 on schedule. The construction took place from May to June. Figure D1 through Figure D5 in Appendix D are photographs of the construction pathway for tower number one. Basically, Figures D1 and D2 show the frames of the tower. The frames are made of polycarbonate extruded dual wall. The two sets of frames have inlet and outlet air ports located at the top and bottom of the tower. The two sets are mirror images of each other: one set for evaporative air and feed waters and the other for condensate and dew formation air.

The heat transfer walls (thin REXAM sheet with gauze) were placed in between the frames as the frames were assembled one on top of the other. The frames alternated from evaporation to dew formation so that each heat transfer wall was effective. The inlet and outlet air vanes formed manifolds as shown in Figure D3 in Appendix D. These air manifolds were connected on the outside of the tower by 3 inch (7.62 cm) ABS plastic pipe fittings to facilitate external piping configurations.

The outside walls that make up the wide sides of the tower (8 ft high x 1 ft wide (2.44 m high by 30.48 cm wide)) are made of the extruded polycarbonate dual wall sheet. One of the sheets was made into the feed liquid heat exchanger by attachment of a bottom water feed inlet tube and a top exit tube. Cold feed entered the bottom and hot feed exited the top. The hot feed water was introduced to the top of the evaporation chamber.

The final "sandwich constructed" tower was pressed together (Figure C1 in Appendix C and Figure D4 in Appendix D) and further sealed with plastic tape so as to form sealed bags for pure condensate collection. The feed side waters that were unevaporated were allowed to flow to the bottom of the tower and exit to drain. The vertically mounted tower is shown in Figure 7.

The boiler as shown in Figure D5 in Appendix D was a glass flask sitting on an adjustable heater plate that was placed on top of a digital scale. The scale readings versus time gave the most accurate method of the rate of steam input to the tower.

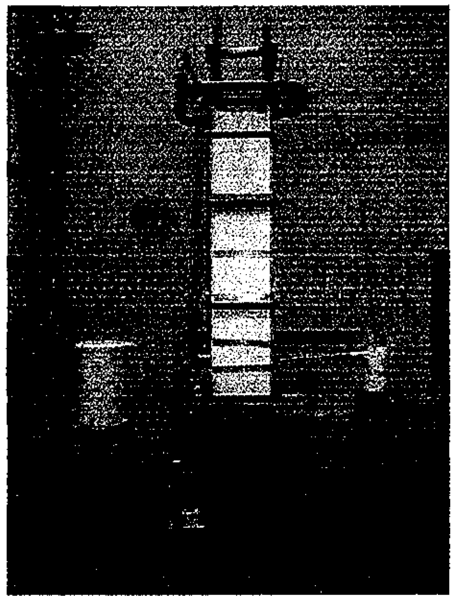


Figure 7.—Dewvaporation Tower 1 (middle) showing Boiler (Right), Feed Tank (Left) and Air Blower (left)

4.5.2 Tower Number 2

Tower number 1 was operated with brackish water for problem shake-down runs. The tower was observed to have excellent liquid wall distribution by observation through the clear plastic end walls at the top and bottom zones. But low energy reuse factors (less than 5) indicated mal-distributed air flow patterns. Inserts into the air inlet and outlet manifolds were tried but were unsuccessful in correcting the air flow patterns. A horizontal "serpentine" air flow pattern was designed and built that should achieve distributed air flow since the resultant channels are 4 inches (10.16 cm) wide not 12 inches (30.48 cm) as previously designed. These channels and air flow rates should allow Reynolds Numbers of from 150 at the tower bottom to 500 at the top. The channel spacers are constructed of thinly cut plastic sponge. Water flows down the heat transfer walls and is redistributed on the wall as it passes through the sponge spacers. Air flows to the end windows where the spacers stop and easily passes to the next level of air flow (see Figure C2 in Appendix C).

The thermocouples were arranged about the system as shown in Figure C3 in Appendix C.

Figure D6 in Appendix D is a picture of the sponge inserts that comprise the horizontal serpentine air flow channels. The spacers were held in place by three cords. Every channel had the same pattern so that air flows were countercurrent for maximum thermal driving force across the heat transfer wall. The same design as tower number one was used for tower number two. The only difference was the introduction of the sponge spacers.

4.6 Summary of TASK 6.0 (Tower Operation)

Tower number 2 was successfully operated in the desalination of brackish water and sea water (from Ventura California). The results are shown in Tables E1, E2, E3 and E4 (Appendix E).

For the brackish water runs that were made, the average GOR was 7.8 as calculated from the ratio of the condensate rate to the steam rate. The average GOR of sea water desalination runs was 7.5. The predicted detrimental effect of the slight desiccant effect of sea water was not observed. These GOR values ignore any heat loss from the steam delivery system and from the tower. There was a definite heat loss of about 200 BTU/hr (58.61 W) or 0.2 lb/hr (90.7 g/hr) of steam equivalent. If this loss occurred in the steam delivery system, then the tower received less steam which would have increased the GOR values to 16.7 for brackish water and to 20 for sea water. If the heat loss occurred in the tower, then the effective steam rate would have been reduced by 0.1 lb/hr (45.4 g/hr) since the effect of 0.2 lb/hr (90.7 g/hr) did not make it to the tower bottom. This correction would have increased the GOR values to 12.3 for brackish water and to 14 for sea water.

For all of the runs made, the tower had a background heat loss to the laboratory of about 200 BTU/hr (58.61 W) as previously stated. The energy from the steam that was introduced at the top of the tower did not appear at the tower bottom in the form of increased exhaust air temperature or exhaust humidity. The tower was well insulated with foam sheets that were 6 inches (15.24 cm) thick. The top piping area was most suspect for heat loss since the oversized ABS piping was difficult to insulate as positioned and the temperatures were high compared to all other tower temperatures. The pipes were also reversibly cemented together for ease of disassembly, for inspection, or for connections that were originally contemplated for the attachment of five more towers. The heat loss rate was comparable to 0.2 lb/hr (90.7 g/hr) of distributed steam leak. In the future the piping will be replaced with smaller sizes since only one tower is needed, and the connections will be permanently cemented to prevent any leaks. The smaller pipe will also be positioned with appropriate spacing to allow pipe insulation. All of these modifications should reduce the heat loss to less than 100 BTU/hr (29.31 W). This would result in GOR values of 12.3 and 14.

In the future, larger size capacity units should be built which would automatically reduce the heat loss problem as the ratio of production rate to externally exposed area would be reduced. Also the elimination of the top piping would help reduce the heat loss problem by designing internal steam injection plenums as was done for the economics discussion.

In summary, the GOR values for brackish and sea water were about the same having an average value of(12.3 + 14)/2 = 13.1. The production density typical of all of the runs was 0.05 lb condensate per hour per square foot of heat transfer wall (0.24 kg/hr m^2). These values were used to estimate the capital and operating costs associated with a 1000 gallon (3.79 m^3) per day plant.

5. ECONOMICS

Based on the data as presented in Tables E1, E2, E3 and E4 (Appendix E), a nominal 1000 gallon per day desalination facility has been designed and priced. The tower dimensions are 4 feet by 4 feet by 8 feet high (1.22 m by 1.22 m by 2.44 m high). The cases considered are brackish and seawater at a GOR of 13.1 and a production density 0.05 lb/hr ft^2 (0.24 kg/hr m^2). Table 5 gives a summary of the capital and operating costs for a 1000 gallon (3.79 m^3) per day unit.

The capital cost is dominated by the heat transfer wall size and material cost. The wall with spacers is 39% of the total parts list. A method of wall area reduction would be the continued investigation into improved heat transfer coefficients and by increasing the air flow rates into the turbulent Reynolds number region. Increased Reynolds numbers will invite increases in pressure drop. Higher pressure drops could require the purchase of a pressure blower instead of a fan and could require more spacers and stiffer spacers to prevent air slot collapse. These effects would increase capital cost as the wall area is reduced.

Table 5: Cost Summary of a 1000 gallon /day Desalination Plant

Capital Cost ($)		Operating Cost ($/day)*	Natural Gas	Waste Heat
Composite heat transfer wall	$250	Fuel	$2.47	$0.64
Spacers	$50	Capital Charge	$0.46	$0.46
Heater (gas controls & spray)	$115	O&M	$0.33	$0.33
Cover	$90	Electricity	$0.04	$0.04
Tubing	$50	Chemicals	$0.05	$0.05
Pumps	$60	Total	$3.35	$1.52
Fan	$50			
Feed heat exchanger sheets	$75	* Operating Cost Including Capital Charge		
Insulation	$30			
Total parts	$770			
Assembly	$208			
Total construction	$978			
Gross margin @ 30%	$419			
Total Unit Cost	1,397			

The operating costs in Table 5 are dominated by the natural gas fuel cost ($0.35/therm) which represents 74% of the total cost. Fuel costs can be reduced by less expensive fuels, higher GOR values and by the use of solar energy or waste heat. As an example, atmospheric steam which is usually generated as waste heat recovery associated in chemical plants and petroleum refineries, costs about $1/1000lb ($/453.59 kg). This steam could be used at a cost of $0.64 per day. This would reduce the operating cost from $3.38 to $1.52 per 1000 gallons (3.79 m^3) of distillate.

Other waste heat sources and solar could be essentially free. However, solar will have an additional capital charge in the operating cost table. A solar collector operating at 185°F (85°C) could be used to evaporate (not boil) water into the air stream at the top of the desalination tower. If the collector could absorb 100 BTU/hr ft^2 (315.5 W/m^2), and if the collector cost was $\$5/ft^2$ ($\$53.8/m^2$), then during daylight hours the heat from the collector could be used at an additional capital cost of \$1300. The capital charge in Table 6 would increase from \$0.46 to \$0.89 as the fuel cost reduced from \$2.47 to zero. Then the operating cost would reduce from \$3.35 to \$1.32 for the solar application.

Another consideration of the costs listed in Table 6 is the design exchange relationship between the GOR (or energy reuse factor, f) and the production density as viewed in Equation 15. If the Production density is halved then the GOR is doubled. Reflecting from Table 5, this design consideration would double the heat transfer wall area and halve the fuel cost. In summary the capital cost would increase from \$1397 to \$2057, and the operating cost would reduce from \$3.35 to \$2.35 per 1000 gallons (3.79 m^3) of distillate as shown in Table 6.

Table 6: Cost Summary of Desalination Plant Capacities

Production gallon/day	GOR	Capital Cost*	Operating Cost* Natural Gas	Operating Cost* Waste Heat
500	24	2057	2.35	1.42
1000	12	1397	3.35	1.52
2000	6	1097	5.33	2.06

* $ per 1000 gallon/day (Operating Cost Includes Capital Charge)

The case studies that could be considered are endless. Just these few examples highlight the advantages and versatility of the Dewvaporation Process.

6. REFERENCE LIST

Bird, R. B., W. E. Stewart, and E. N. Lightfoot. Transport Phenomena. John Wiley and Sons, Inc.: New York, 1960

Fosselgard, G. and K. Wangnick. "Comprehensive Study on Capital and Operational Expenditures for Different Types of Seawater Plants."*Desalination*, 76, 1989

Larson, R., *et al*. "The Carrier-Gas Process : A New Desalination and Concentration Technology", *Desalination*, 73, p. 119, 1989a

Larson, R., *et al*. The Carrier-gas Process: A New Desalination and Concentration Technology. Proceedings of the Fourth World Congress of Desalination and Water Reuse, Kuwait, Nov. 1989b

McCabe, W. L., J. C. Smith, P. Harriott. Unit Operations of Chemical Engineering, 5th ed. McGraw-Hill, Inc.: San Francisco, 1993

Perry, R. H., D. W. Green, J. O. Maloney. Perry's Chemical Engineers' Handbook. McGraw-Hill, Inc.: San Francisco, 1984

Smith, J. M., and H. C. VanNess. Introduction to Chemical Engineering Thermodynamics, 4th ed. McGraw-Hill, Inc.: San Francisco, 1987

Spiegler, K.S. and Laird, A.D. K. Principles of Desalination: Part B 2nd ed.. Academic Press: New York, 1980

Werling, P. H. Design and Experimental Operation of a Times-Two Liquid Desiccant Regeneration System. MS Thesis, Arizona State University, 1990

Appendix A

HEAT TRANSFER COEFFICIENT EXPERIMENTS

Table A1: Summary of Enhanced Heat Transfer Coefficient (h_g) - November Data

Description	Spacing	Date	Run#	Reynolds Number	h_g BTU/hr ft^2 °F	Nusselt Number Exp.	Nusselt Number Theory	Nusselt Number Ratio Exp./Theory
One random fold sheet	.5 "	12-Nov	1	1300	1.4	7.6	3	2.5
One random fold sheet	.5 "	12-Nov	2	1300	1.7	9.3	3	3.1
Both random	.5 "	12-Nov	3	1300	1.3	7.5	3	2.5
Both random	.5 "	12-Nov	4	1300	1.6	9.1	3	3
Both random	.5 "	12-Nov	5	1380	1.3	7	3	2.3
Both random	.5 "	12-Nov	6	5040	3.8	21	19	1.1
One random fold sheet	.5 "	12-Nov	7	5282	3.3	19	20	0.9
One random fold sheet	.5 "	12-Nov	8	1380	1.3	7.3	3	2.4
Wave pattern (A = .25",P=6")	.375 "	20-Nov	1	990	0.9	3.3	3	1.1
Wave pattern (A = .25",P=6")	.375 "	20-Nov	2	820	0.8	3.4	3	1.1
Wave pattern (A = .25",P=6")	.375 "	24-Nov	1	990	0.9	3.3	3	1.1
Wave pattern (A = .25",P=6")	.375 "	24-Nov	2	6671	2.6	9	24	4
Flat Plastic Sheets	.5 "	25-Nov	1	7302	3	14.1	25	0.56
Flat Plastic Sheets	.5 "	25-Nov	2	6638	10	46	22.6	2
Wetted paper sheet	.5 "	25-Nov	3	1494	1.42	6.6	3	2.2
Blotting paper	.5 "	25-Nov	4	1660	2.6	12	3	4
Blotting paper	.5 "	25-Nov	5	1245	1.7	7.8	3	2.6
Blotting paper	.5 "	3-Dec	1	1164	1.2	6.7	3	2.2

Table A2: Summary of Enhanced Heat Transfer Coefficient (h_g) - December Data

Description	Spacing	Date	Run#	Reynolds Number	h_g BTU/hr ft^2 °F	Nusselt Number Exp.	Nusselt Number Theory	Nusselt Number Ratio Exp./Theory
Blotting paper	.5 "	3-Dec	2	1164	1.9	10.3	3	3.4
Blotting paper	.5 "	3-Dec	3	Sheets touching	0	0	0	0
Blotting paper	.5 "	3-Dec	4	5042	7.6	36	19	1.9
Blotting paper	.5 "	3-Dec	5	5042	6	28.2	19	1.5
Blotting paper	.25 "	3-Dec	6	5145	7.3	21	19.3	1.1
Blotting paper	.25 "	4-Dec	1	6970	3.9	10	25	0.4
Blotting paper	.25 "	4-Dec	2	830	1	3	3	1
Blotting paper	.25 "	4-Dec	3	1162	1.2	3.3	3	1.1
Blotting paper	.25 "	4-Dec	4	1162	1.2	3.5	3	1.2
Blotting paper	.25 "	4-Dec	5	6140	9.1	25	22.4	1.1
Blotting paper	.25 "	7-Dec	1	5311	4.4	12.4	20	0.62
Blotting paper	.25 "	7-Dec	2	1162	1.7	4	3	1.3
Blotting p. Oscillating air	.5 "	9-Dec	1	1138/1220	1.7	7.8	3	262
Blotting p. Oscillating air	.5 "	9-Dec	2	3659/7969	3.3	25	20	1.2
Wet Gauze	.5 "	15-Dec	1	6668	4.1	19	23	0.83
Wet Gauze	.5 "	15-Dec	2	3497	4	18	14	1.3
Wave / alternate spacers	.5 "	15-Dec	3	3415	2.8	13	13.6	1
Wave / alternate spacers	.5 "	15-Dec	4	3415	2.8	13	13.6	1
Wave / alternate spacers	.5 "	15-Dec	5	1870	1.9	8.8	8.6	1
Wave / spacers in line	.5 "	15-Dec	6	1870	1.9	8.8	8.6	1
Wet Gauze	.5 "	15-Dec	7	6668	4.4	24.5	23.7	1
Wet Gauze	.5 "	15-Dec	8	3497	3.6	20	14	1.4
Wet Gauze	.5 "	15-Dec	9	3497	2.8	16	14	1.1
Flat Plastic Sheets	.5 "	16-Dec	1	910	1	4.6	3	1.5
Flat Plastic Sheets	.25 "	16-Dec	2	1140	1.3	3	3	1
Flat Plastic Sheets	.25 "	16-Dec	3	670	0.8	1.8	3	0.6
Flat Plastic Sheets	.25 "	16-Dec	4	1835	2.7	6.2	8.5	0.75
Flat Plastic Sheets	.25 "	16-Dec	5	2280	2.5	5.7	10	0.6
Diaganol fold	.43 "	18-Dec	1	1000	1.5	3	3	1
Diagonal fold	.43 "	18-Dec	2	2650	4.4	8.8	11	0.8
Diagonal fold	.43 "	18-Dec	3	4400	6.4	13	17	0.77

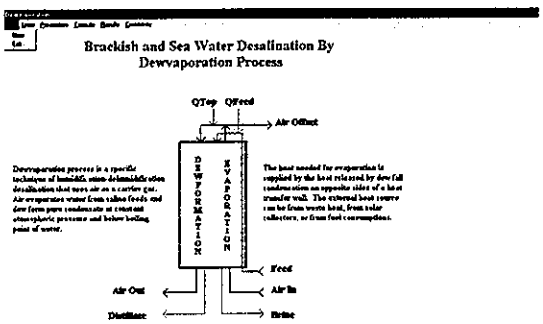
Appendix B

MATHEMATICAL MODEL – COMPUTER SIMULATION

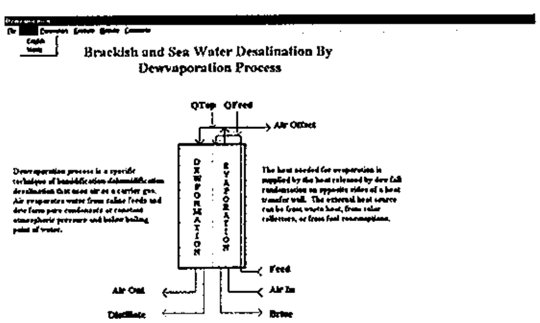
COMPUTER MODEL

The theoretical model (Differential Equations) of the DewVaporation process is solved using a 4th order Runge-Kutta (Classical method, RK4) numerical technique where the computation is performed using VISUAL BASIC programming language. The main sections of the program are File, Units, Analysis, Execute, Charts, and Information. These sections can be found in the menu bar of the main screen of the program.

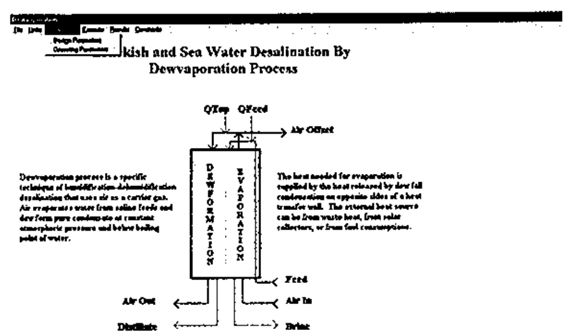
FILE: This section includes two options: New (allows the user to reset the values of the simulation), and Exit.



UNITS: This section includes two options: Metric and English. The user has the freedom to choose the units of the simulation (metric units being the default unit).



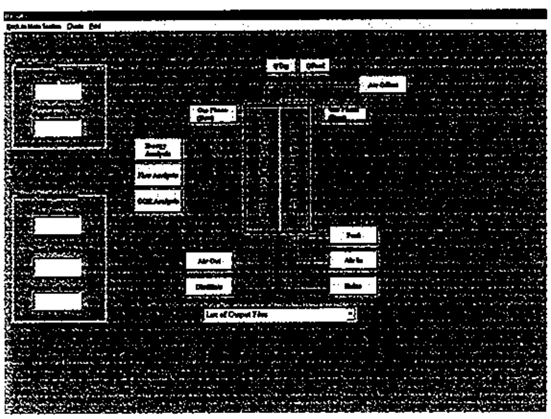
PARAMETERS: This section includes Design and Operating input parameters. The operating parameters include the gas phase temperatures on the evaporation and dewformation side at the top of the tower, the inlet air temperature, the feed temperature, feed salinity, feed temperature offset, and air offset at the top of the tower. On the other hand, the design parameters include Reynolds number for the air or Gamma (flow rate per width) for the liquid on the evaporation side at the top of the tower, run number (it is used to identify the output file numbers), recovery ratio, width of the heat transfer media (w), depth of the heat transfer media (r), thickness of the heat transfer wall, and the thermal conductivity of the heat transfer wall material. Also, the user has the choice of running the simulation with or without internal feed heat exchanger. Moreover, the user can choose to input required height of the tower, and number of heat transfer media in the tower.



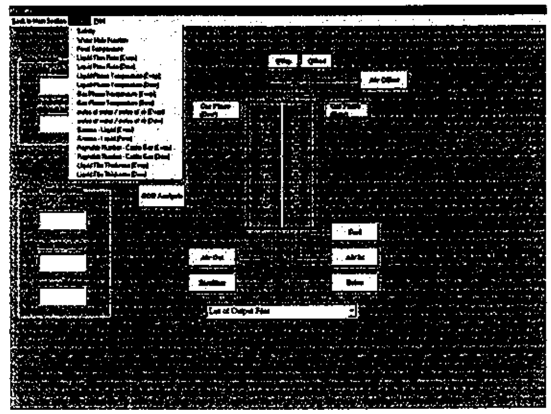
EXECUTE: is the command used to run the simulation after all the input parameters are entered. Analysis of various results can be selected after the execution of the simulation under the Results section.

RESULTS: This section contains the main results of the simulation such as GOR, total area, height of the tower, production density and number of modules needed to reduce the total height. In addition to that, specific information about the process streams is displayed separately. Process streams include Air In, Air Out, Air Offset, Feed, Brine, Distillate, Gas Phase (Dew), Gas Phase (Evap), Energy Input (Top), and Energy Offset for the feed.

The Energy, Flow, and GOR options give analysis of the overall process based on energy, flow and GOR respectively. Output files are generated each time the simulation is run and these files are identified by the "run number". These files are saved on the main directory of the computer which include Height, Area, Salinity, Water Mole Fraction (Evap), Liquid Phase Flow (Evap), Liquid Phase Flow (Dew), Liquid Phase Temperature (Evap), Liquid Phase Temperature (Dew), Feed Temperature, Gas Phase Temperature (Evap), Gas Phase Temperature (Dew), moles of water per moles of air (Evap), moles of water per moles of air (Dew), Gamma (Evap), Gamma (Dew), Reynolds Number (Evap), Reynolds Number (Dew), Liquid Film Thickness (Dew), and Liquid Film Thickness (Evap).

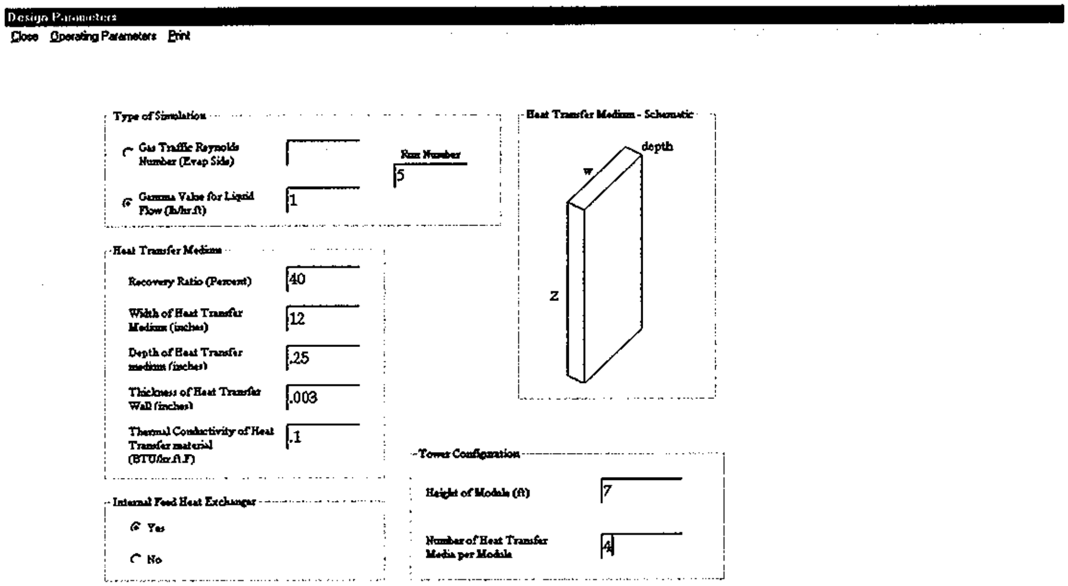


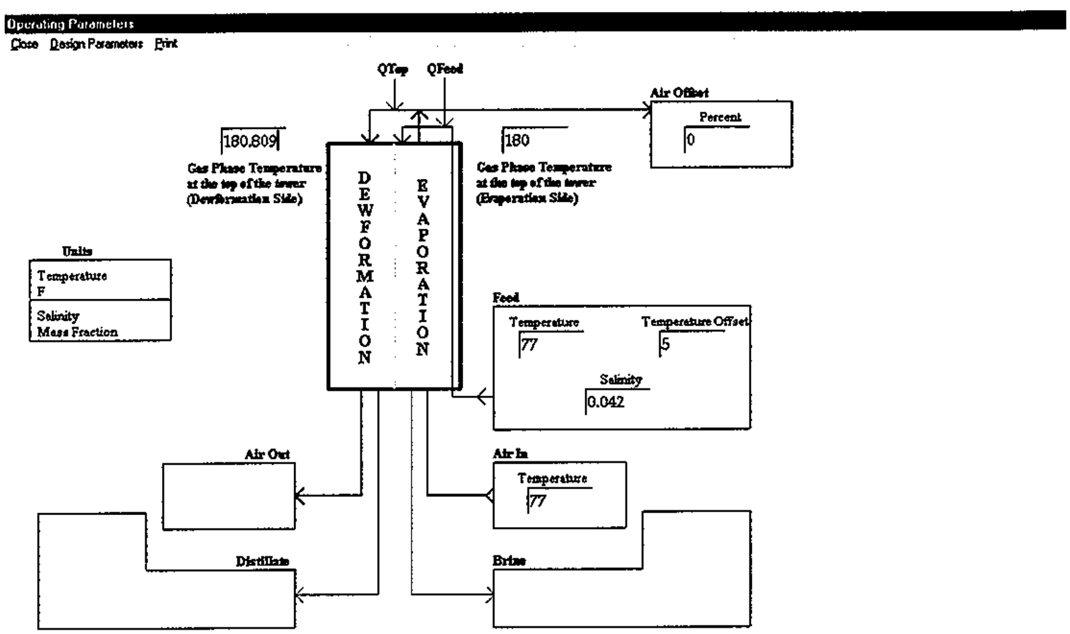
Also, the results are presented graphically which can be viewed using the Charts command under the Results section. A complete list of the charts include Salinity, Water Mole Fraction (Evap), Liquid Phase Flow (Evap), Liquid Phase Flow (Dew), Liquid Phase Temperature (Evap), Liquid Phase Temperature (Dew), Feed Temperature, Gas Phase Temperature (Evap), Gas Phase Temperature (Dew), moles of water per moles of air (Evap), moles of water per moles of air (Dew), Gamma (Evap), Gamma (Dew), Reynolds Number (Evap), Reynolds Number (Dew), Liquid Film Thickness (Dew), and Liquid Film Thickness (Evap).

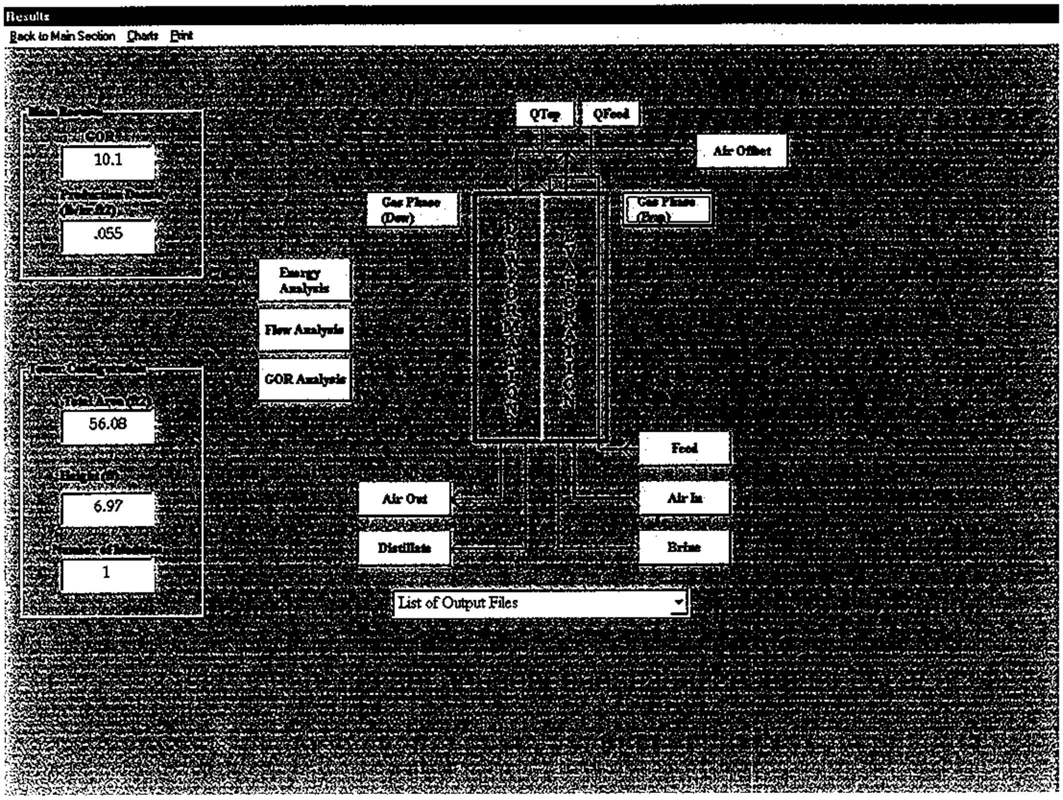


COMMENTS: This section includes the contact information.

A sample output of the simulation is given below for seawater with Salinity of 42 g/kg.






Results
Back to Main Section Charts Print
10.1
.055
56.08
6.97
1
QTop
QFeed
Air Offset
Gas Phase (Dew)
Gas Phase (Evap)
Energy Analysis
Flow Analysis
GOR Analysis
Feed
Air Out
Air In
Distillate
Brine
List of Output Files

Appendix C

DESIGN DRAWINGS OF TOWERS 1 and 2

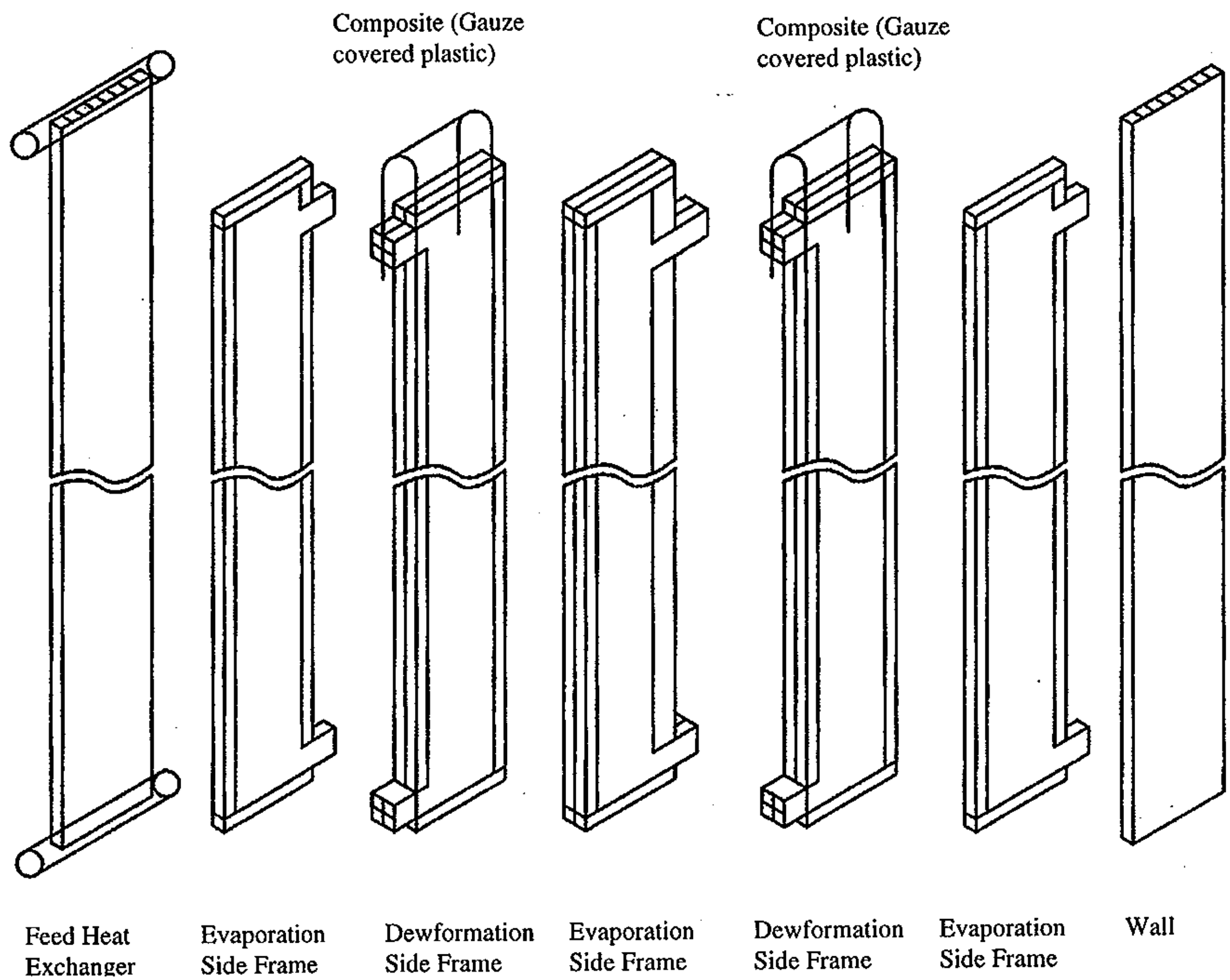


Figure C1: Tower 1 Configuration

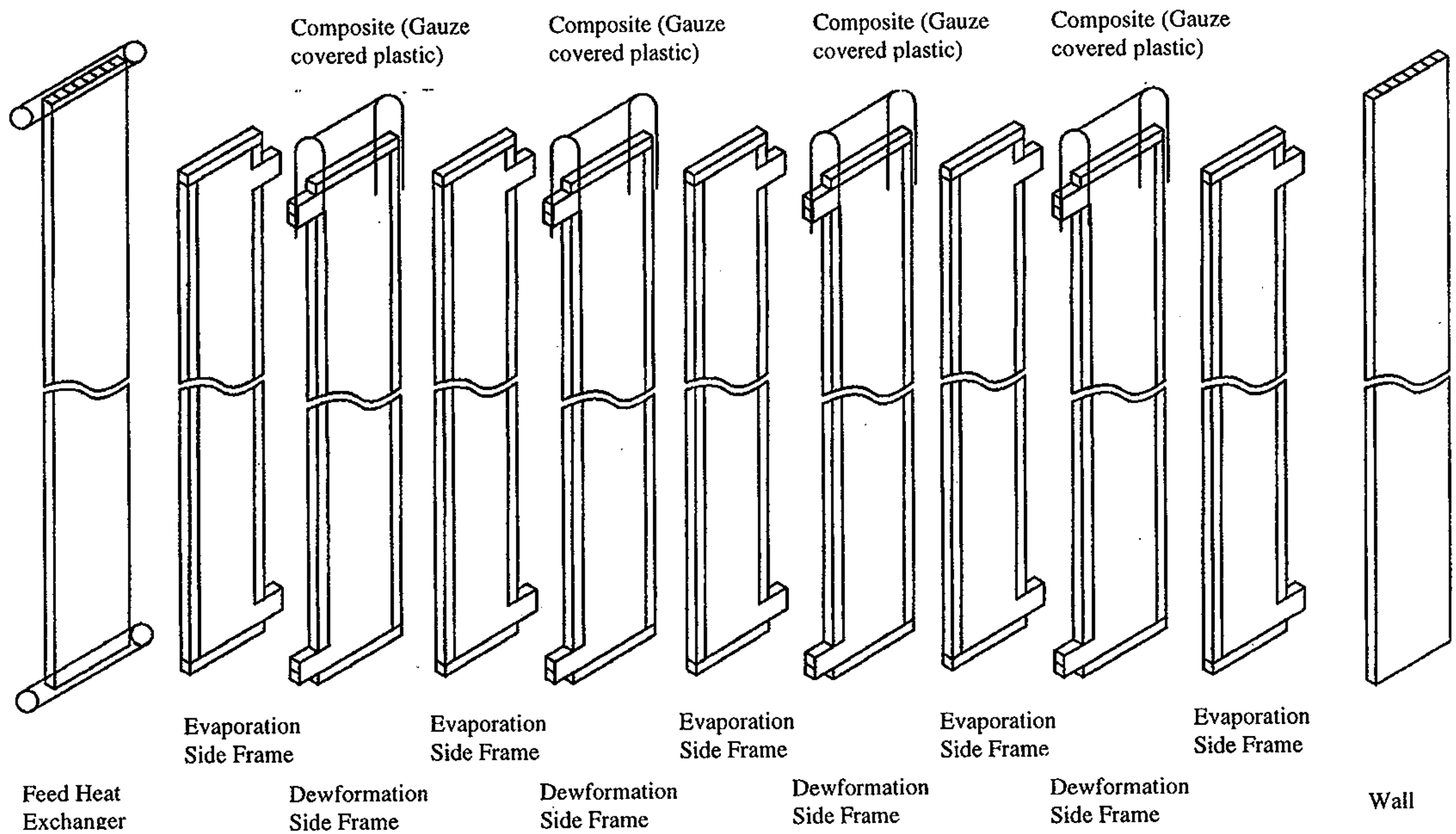
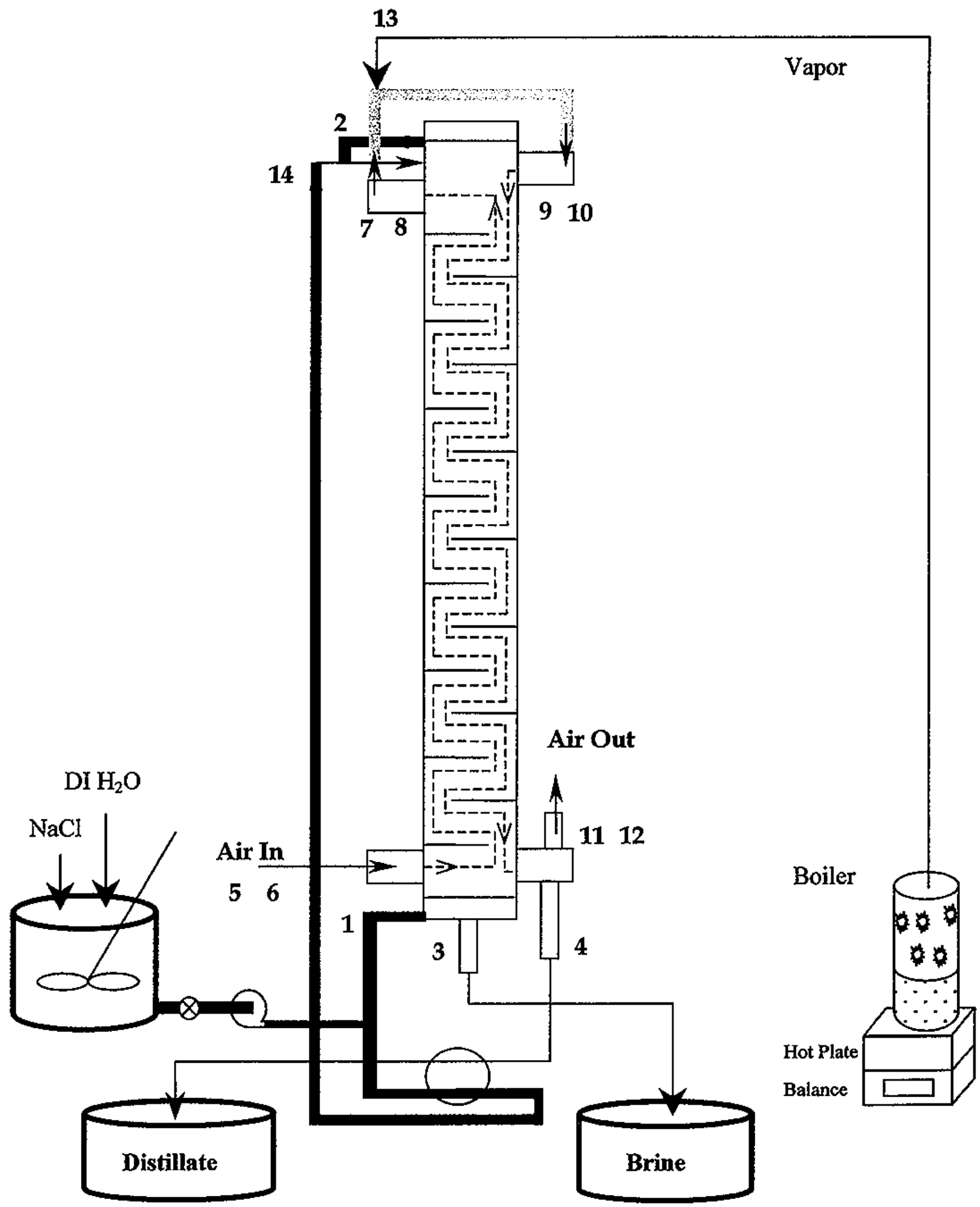


Figure C2: Tower 2 Configuration



1	*Feed*	*8*	*Water Vapor Evap Side (Wet Bulb)*
2	*Heated Feed by Air*	*9*	*Water Vapor Dew Side (Dry Bulb)*
3	*Brine*	*10*	*Water Vapor Dew Side (Wet Bulb)*
4	*Distillate*	*11*	*Saturated Air Out (Dry Bulb)*
5	*Blower Air In (Wet Bulb)*	*12*	*Saturated Air Out (Wet Bulb)*
6	*Blower Air In (Dry Bulb)*	*13*	*Heat Input to the Tower*
7	*Water Vapor Evap Side (Dry Bulb)*	*14*	*Heated Feed by Distillate*

Figure C3: Schematic of the Experimental Apparatus with Thermocouple Placement

Appendix D

ASSEMBLY PICTURES OF TOWER 1 and 2

Tower 1

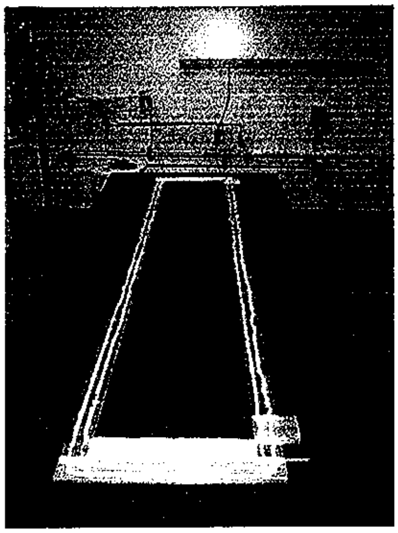


Figure D1: Evaporation Side Frame (1/4")

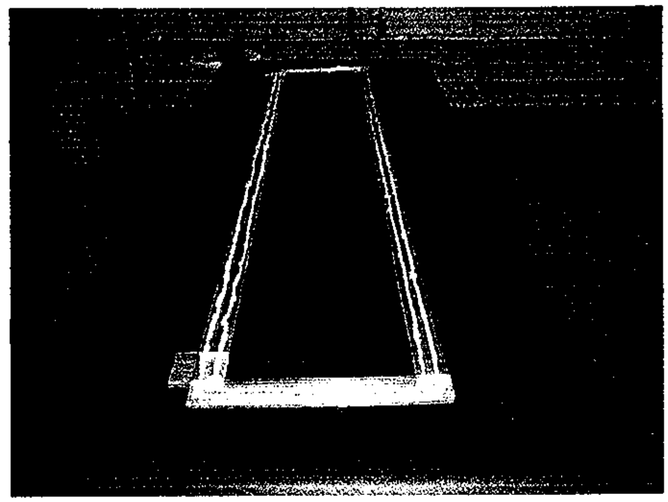


Figure D2: Dewformation Side Frame (1/2")

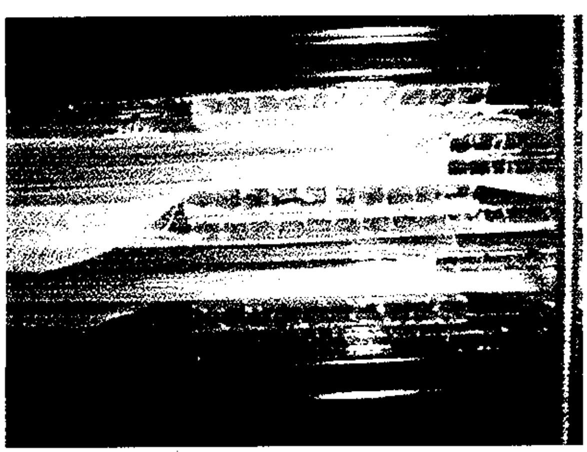


Figure D3: Air Manifold

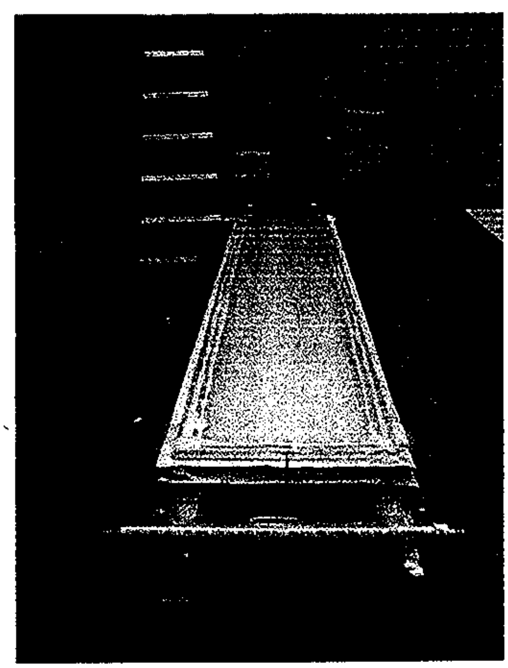


Figure D4: Tower 1 without insulation pressed and sealed on the edges

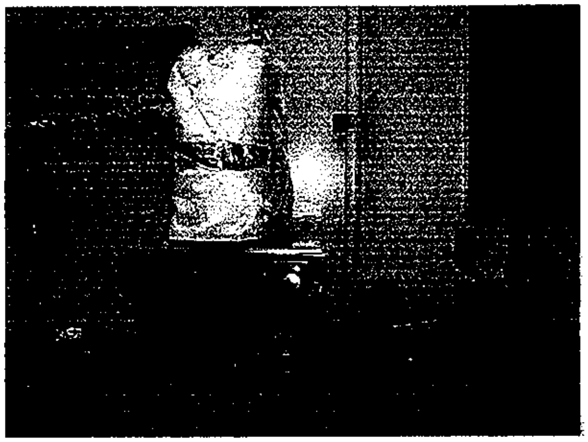


Figure D5: Boiler System showing the insulated beaker hot plate, balance and the steam line

Tower 2

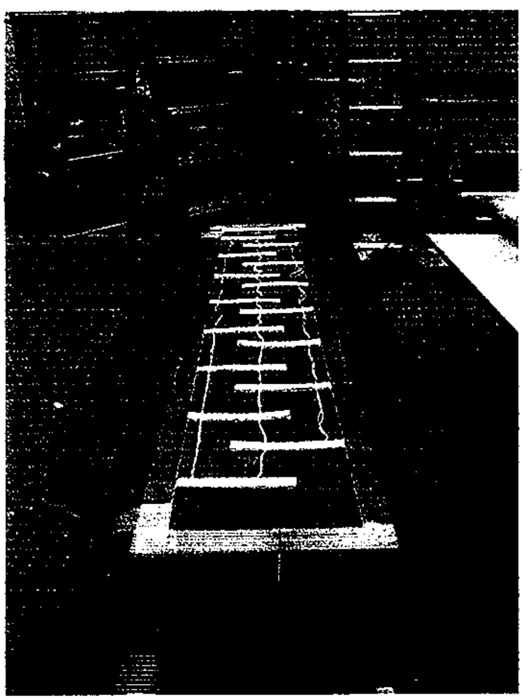


Figure D6: Sponge spacers forming the horizontal \ serpentine air flow pattern

Appendix E

TOWER OPERATION DATA

Table E1: Data Summary - Brackish water

Run Number	GOR Top of the tower	GOR Bottom of the tower	Heat Loss BTU/hr	Rate (lb/hr) Feed	Rate (lb/hr) Brine	Rate (lb/hr) Condensate	Rate (lb/hr) Steam
729A	9.29	17.74	167	6.61	3.57	3.04	0.34
805A	6.10	12.93	286	6.61	3.74	3.08	0.52
805B	7.89	19.34	177	6.61	4.41	2.22	0.29

Table E2: Data Analysis - Brackish water

Run Number	Heat Transfer Coefficient BTU/hr ft^2 °F	Air Flow Rate lbmole/hr	Top section of the tower: Evap Temp. °F	Top section of the tower: lbmole of water per lbmole of air	Top section of the tower: lbmole of water per lbmole of air	Top section of the tower: Calculated Dew Temp. °F
729A	0.48	0.14	182	1.1863	1.3212	184
805A	0.37	0.15	180	1.0514	1.2440	183
805B	0.23	0.11	181	1.0972	1.2437	183

Table E3: Data Summary - Seawater 42 g/kg

Run Number	GOR Top of the tower	GOR Bottom of the tower	Heat Loss BTU/hr	Rate (lb/hr) Feed	Rate (lb/hr) Brine	Rate (lb/hr) Condensate	Rate (lb/hr) Steam
806A	9.58	28.25	187	6.61	2.78	2.6	0.27
809A	5.42	13.34	230	6.61	4.12	1.98	0.37

Table E4: Data Analysis - Seawater 42 g/kg

Run Number	Heat Transfer Coefficient BTU/hr ft^2 °F	Air Flow Rate lbmole/hr	Top section of the tower: Evap Temp. °F	Top section of the tower: lbmole of water per lbmole of air	Top section of the tower: lbmole of water per lbmole of air	Top section of the tower: Calculated Dew Temp. °F
806A	0.26	0.13	180	1.0272	1.1426	182
809A	0.22	0.12	176	0.8339	1.0052	179

Appendix F

SI METRIC CONVERSION

Table F1: SI Metric Conversion

From	To	Multiply by
inch	cm	2.54
inch	mm	25.4
ft	m	0.3048
ft^2	m^2	0.0929
^{o}F	^{o}C	$^{o}C = (^{o}F-32)/1.8$
psia	bar	0.06895
lbmole/sec	gmole/sec	453.59
BTU/lbmole ^{o}F	J/gmole K	4.184
BTU/lbmole	J/gmole	2.326
BTU/hr ft^2 ^{o}F	W/m^2 K	5.6783
BTU/hr ft ^{o}F	W/m K	1.73073
BTU/hr	W	0.29307
BTU/hr ft^2	W/m^2	3.1546
lb	kg	0.45359
lb	g	453.59
lb/hr	g/hr	453.59
lb/hr ft	kg/hr m	1.488
lb/hr ft^2	kg/hr m^2	4.8826
lb/ft^3	g/mL	0.016
gal	m^3	0.003785