

Efficiency Improvements in a
Horizontal Humidification-Dehumidification Unit

by

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ABSTRACT

The horizontal desalination units belonging to the humidification-dehumidification family purify water using air as a carrier gas. The temperature required for separation can vary from ambient to 99 °C so waste heat, fuel combustion, or solar collectors can drive the process. A unit in which air flows horizontally affords several advantages over similar vertical “Dewvaporation” towers (as an example), including ease of construction and potentially increased efficiency. The objective was to build and test horizontal units and identify areas of potential efficiency improvements.

The desalination units consisted of: 1.) A series of aligned, corrugated, polypropylene sheets covered on the outside with absorbent, water-wettable cloth. 2.) A basin that caught saline water flowing downward from the absorbent cloth. 3.) Ten pumps to cycle the basin water back onto the cloth. 4.) An air blower on the front of the unit that drove air horizontally across the cloth, increasing the humidity of the air. 5.) A steam generator on the back of the unit producing steam that mixed with the incoming air to increase the temperature and humidity. 6) A steam box that caused the air to mix with the steam and return to flow inside the corrugations in the plastic sheets, creating a countercurrent heat exchanger as the exiting air transferred its heat to the incoming air and causing purified water to condense from the cooling, oversaturated air.

The tested unit produced distillate at a rate of 0.87 gallons per hour with 13 parts per million total dissolved solids and an energy reuse factor of 2.5. Recommendations include the implementation of a continuous longitudinal pump design, a modification of the basin to accommodate top and bottom unit center dividers, increase in insulation coverage, and optimization of air flow rate.

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LIST OF ABBREVIATIONS

Term	Abbreviation
Reverse Osmosis	RO
Meter Cubed.....	m^3
Arizona State University.....	ASU
Total Dissolved Solids	TDS
Parts Per Million	ppm
Air Vapor Content at the Inlet of the Evaporation Side.....	V_{e0}
Pound-Mole.....	lb-mol
Air Vapor Content at the Top of the Evaporation Side	V_{eh}
Air Vapor Content at the Top of the Condensation Side	V_{dh}
Air Vapor Content at the Outlet of the Condensation Side	V_{d0}
Energy Reuse Factor	f
The North And South Vertical Exit Tower.....	NASVET
Production Density.....	P_f
Carrier Gas Flow Rate	G
Area.....	A
Overall Heat Transfer Coefficient	U
Heat Transfer Coefficient of the Liquid Film on the Evaporation Side	h_{fe}
Heat Transfer Coefficient of the Liquid Film on the Condensation Side.....	h_{fd}
Thickness	t
Thermal Conductivity	k
Water Film Thickness on the Evaporation Side	δ_e

Term	Abbreviation
Water Film Thickness on the Condensation Side	δ_d
Vertical Direction.....	Z
Heat Transfer Coefficient of the Gas	h_g
Mass Transfer Factor	M
Heat of Vaporization of Water.....	λ
Gas Constant	R
Temperature	T
Heat Capacity.....	c_p
Nusselt Number	Nu
Diameter.....	D
Square foot	sqft
Millimeter	mm
Cubic feet per minute.....	cfm
Specific Enthalpy of Dry Air	h_a
Specific Enthalpy of Moist Air.....	h
Vapor Content of Moist Air.....	x
Specific Enthalpy of Water Vapor.....	h_w
Partial Pressure of Water Vapor	p_w
Atmospheric Pressure	p_a
Relative Humidity.....	RH
Heat Capacity of Dry Air, Constant Pressure	c_{pa}
Heat Capacity of Water Vapor, Constant Pressure	c_{pv}

Term	Abbreviation
Volume.....	v
Evaporation Enthalpy of Water at 0 °C	h_{we}
Ideal Gas Constant	R
Pressure	P
Moles.....	n
Heat of Brine In	H_{bi}
Temperature of Brine In.....	T_{bi}
Mass Flow Rate of Brine In.....	m_{bi}
Enthalpy of Water	\dot{H}_{wa}
Heat Capacity of Water.....	c_{pw}
Average Temperature of Basin	\bar{T}
Mass Flow Rate of Water	m_{wa}

1 INTRODUCTION

Water is one of the most important substances on Earth for sustaining life. The majority of the water on Earth is located in oceans, which have too much dissolved salt to be potable. The majority of the freshwater on Earth is in the form of glaciers. A relatively small percentage of water on earth located in rivers, lakes, and springs is accessible and can be used by humans.

Given the low availability of potable water, converting non-potable water to potable water accounts for a large industry in today's society. Desalination of water is ideal due to the vast availability of ocean water. Although desalination can be accomplished as simply as boiling water, cooling the vapor, and collecting the condensate, the energy usage of such a method would be prohibitive. A number of technologies are available to decrease the energy usage and increase efficiency. Some of the most prevalent desalination methods are reverse osmosis, multi-stage flash, and multiple effect distillation. Another method is humidification-dehumidification, which has potential to be one of the most efficient methods, and a horizontal geometry of humidification-dehumidification in particular, as discussed in this thesis, is particularly promising due to the ability to distill water with a high level of impurities, utilize low-temperature waste heat from other solar energy and other processes, and exhibit energy efficiencies far above other desalination methods.

2 BACKGROUND

Reverse Osmosis

RO utilizes osmotic pressure to drive water through a semi-permeable membrane, leaving salts behind the membrane. A diagram of the RO process is shown in Figure 1. RO is most effective when treating water with low concentrations of impurities and no suspended particles, since the frequency in which the membrane requires replacement is dependent on these factors. RO is the least expensive desalination technology currently available for treating brackish water (water with a moderate amount of dissolved salt) and seawater (water with a high amount of dissolved salt). The cost for treating brackish water for a relatively large facility is between \$0.26 and \$0.54 per m³ of clean water. The cost for treating seawater is between \$0.45 and \$0.56 per m³ [1].

Multi-Stage Flash

A diagram for multi-stage flash is shown in Figure 2. Multi-stage flash is a thermal desalination method that utilizes internal heat recovery to attain a high efficiency. Feed brine flows in a tube through the top of the system, where vapor inside the system condenses on the outside of the tube, heating up the brine. The brine flows through a

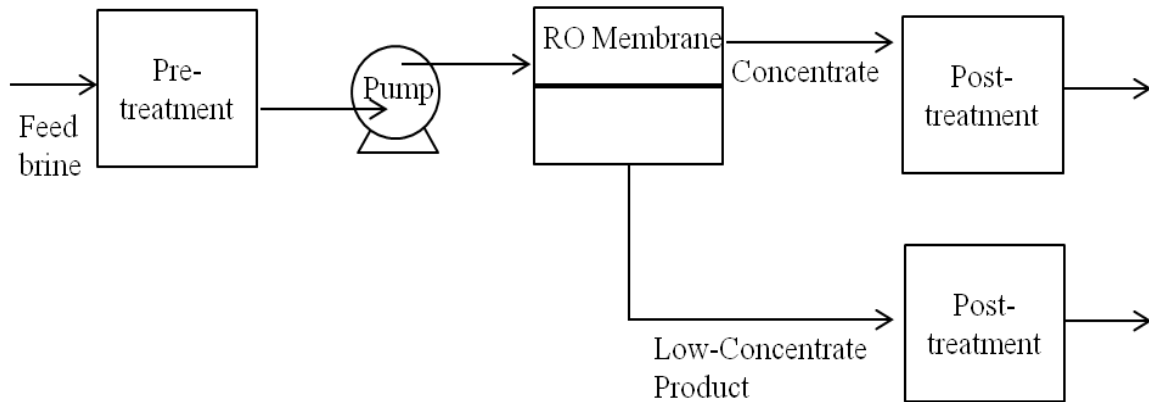


Figure 1. Diagram of the RO process [2].

heater and into the system where it evaporates as it flows through several stages, each consecutively decreasing in temperature, until the brine flows out at the end of stage 1. One consideration for multi-stage flash is that for every gallon of distillate produced, between 10-20 gallons of feed brine must be input, which poses advantages and disadvantages. One advantage is that the amount of scale formation in the system is low due to the large volume of liquid that passes through with relatively low concentration of salt. Since scale formation acts as a heat transfer barrier, the efficiency of multi-stage flash is maintained over time. One disadvantage is that the large volume of water needed prevents the technology from being useful in situations in which a large fraction of the feed brine is to be purified [3]. Multi-stage flash plants produce pure water at \$0.52-\$1.75 per m³ depending on various factors such as the size of the plant [1].

Multiple Effect Distillation

Multiple effect distillation functions by utilizing the latent heat of condensation of vapor condensation in one effect to generate steam for the following effect, as shown in Figure 3. The solid lines indicate brine flow, the dotted lines indicate steam flow, and the

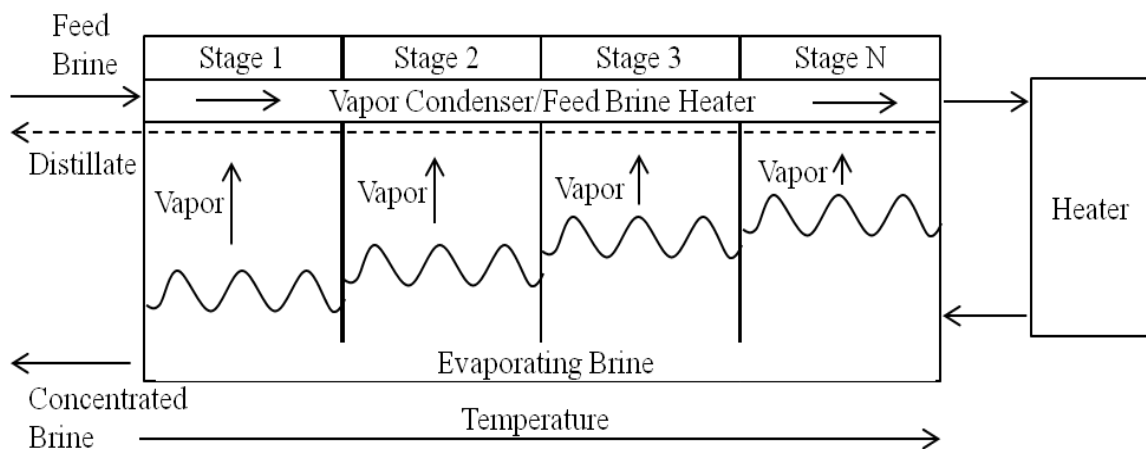


Figure 2. Diagram of the multi-stage flash process.

dash-dotted lines indicate condensate flow. In each effect, feed brine is sprayed onto a tube filled with steam, causing the steam to condense in the tube and the brine to evaporate, leaving behind concentrated brine. The steam condensate from the first effect is fed back into the steam generator to produce additional steam, and the brine and steam generated flow into the second effect. The process repeats in effects 2 through N, except the condensate from effects 2 through N flows out as product. The steam generated in effect N is condensed, generating additional product.

Multiple effect distillation is similar to multi-stage flash in that both methods rely on internal heat transfer to function. However, multiple effect distillation has two advantages in that the majority of the brine feed is converted to potable water at the outlet and the brine sprayed onto the tubes actually boils, increasing the heat transfer coefficient. One disadvantage is that scale forms quickly on the tubes since the volume of water involved is relatively low [3]. The cost for generating purified water with multiple effect distillation is between \$0.52 and \$1.01 per m³ for large plants [1].

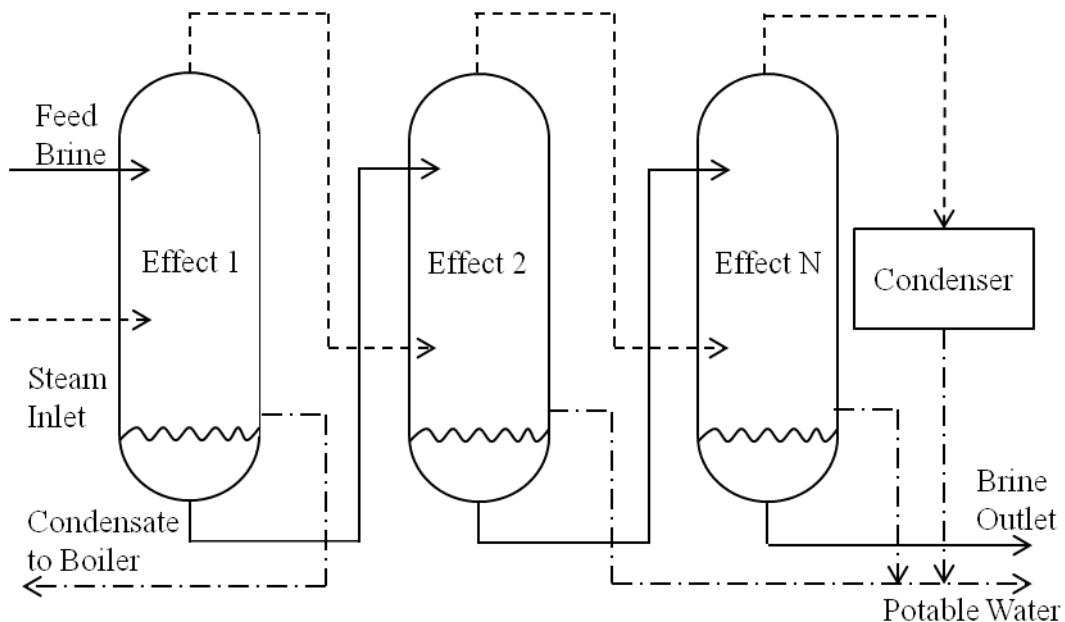


Figure 3. Diagram of the multiple effect distillation process.

Dewvaporation

Another desalination method, humidification-dehumidification, is particularly effective since the temperature required for heat input is less than the boiling temperature of water. Dewvaporation is a type of humidification-dehumidification and an ASU patented technology invented by Dr. James Beckman [4]. Dewvaporation accomplishes desalination of brine water with TDS as high as 200,000 ppm at atmospheric pressure by heating air, increasing the air's humidity, and cooling the air, causing purified distillate to condense inside polypropylene plastic sheets [5]. Since the heat required for the process is below the boiling temperature of water, heat from solar collectors or waste heat from other processes can be utilized to drive the separation. If heat is transferred effectively to the inlet air from the exit air, the energy reuse factor of Dewvaporation has potential to approach 5000, compared to 30 for current reverse osmosis technologies [6]. Dewvaporation is particularly useful for reducing waste water by separating pure water from waste streams [7]. By operating Dewvaporation towers in series, as much as 98% water recovery from waste streams can be obtained [5]. The New Mexico company Altela utilizes Dewvaporation to reduce brackish water waste from hydraulic fracking at low pressures, temperatures, and energy input [4]. The horizontal units described herein build upon the concept of Dewvaporation to accomplish a similar goal of purifying water at atmospheric pressure, low temperature, and low energy input.

Dewvaporation Characterization

An example of the Dewvaporation process is shown in Figure 4. Room-temperature air with vapor content, V_{e0} , of 0.025 lb-mol water per lb-mol air at

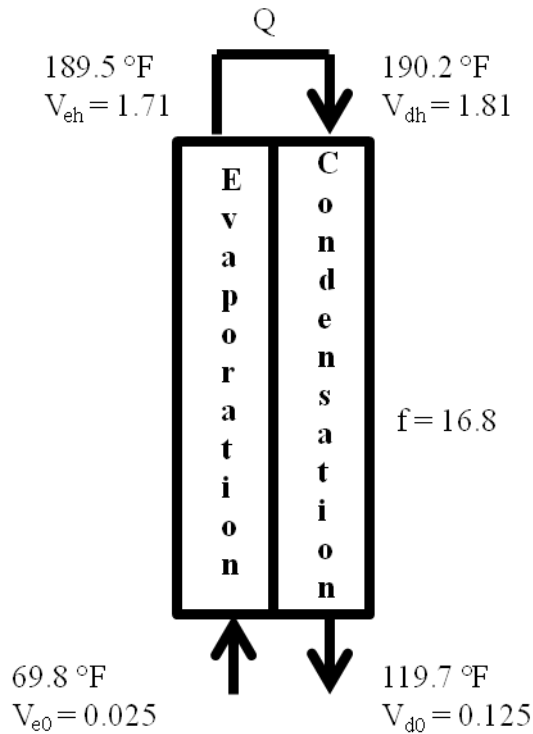


Figure 4. The NASVET tower, an example of a vertical Dewvaporation tower [8,9].

atmospheric pressure flows into the evaporation side of a unit, where it gradually increases in temperature through convective heat transfer from the evaporation-condensation heat-transfer wall. Since the evaporation side is covered in damp cloth, the air also absorbs moisture. Once the air reaches the end of the evaporation section, the air has been saturated with water vapor and reached a temperature of 189.5 °F. The vapor content of the air, V_{eh} , is 1.71 lb-mol water per lb-mol air. The air is mixed with saturated steam which increases the temperature and humidity content, V_{dh} , to 190.2 °F and 1.81 lb-mol water per lb-mol air. The high-temperature, saturated air enters the condensation side where it gradually releases its heat to the evaporation side and lowers in temperature. As the air temperature lowers, pure distillate is condensed on the walls of the condensation side, which flows out the bottom of the tower along with the cooled,

saturated air at 119.7 °F, and the air has a vapor content, V_{d0} , of 0.125 lb-mol water per lb-mol air. The energy reuse factor, f , a measure of the efficiency of the unit, can be calculated as follows:

$$f = \frac{V_{dh} - V_{d0}}{V_{dh} - V_{eh}} \quad (1)$$

Applying the energy reuse factor equation to the example in Figure 4, a factor of 16.8 is calculated. P_f , another factor that characterizes Dewvaporation towers and is defined by the flux of vapor in a tower per unit area can be calculated with the equation:

$$P_f = \frac{G}{A} (V_{dh} - V_{d0}) \quad (2)$$

The overall heat transfer coefficient can be calculated with the equation:

$$\frac{1}{U|_z} = \frac{1}{h_{fe}|_z} + \frac{1}{h_{fd}|_z} + \frac{t_{plastic}}{k_{plastic}} + \frac{\delta_e|_z}{k_{water}} + \frac{\delta_d|_z}{k_{water}} \quad (3)$$

h_f can be rewritten as:

$$h_f|_z = h_g|_z \cdot (1 + M|_z) \quad (4)$$

M is defined as:

$$M = \left(\frac{\lambda}{RT} \right)^2 \cdot \left(\frac{R}{c_p} \right) \cdot V \quad (5)$$

Since the air flow is laminar, h_g can be calculated from Nu :

$$Nu = \frac{h_g D}{k} = 4 \quad (6)$$

When analyzing Dewvaporation towers, minimizing the heat loss to the surrounding environment is crucial to attaining a high energy reuse factor. Heat loss is particularly relevant on the top of Dewvaporation towers where temperatures are high.

Applying insulating materials such as foams and plastics can reduce heat loss to a minimum. Performing an energy balance on towers identifies if heat loss to surroundings is a problem.

By applying the energy balance equation to the tower, the heat loss can be calculated:

$$\text{Energy Input} - \text{Energy Output} = \text{Accumulation} \quad (7)$$

For a continuous Dewvaporation setup, the energy input includes steam, air, and brine inlets. The energy output includes air, brine, distillate, and heat loss outlets. Accumulation is zero for continuous operation. The enthalpies of the air inlet and outlet can be calculated by the specific enthalpy:

$$\dot{H} = h\dot{m} \quad (8)$$

The enthalpies are calculated by the equation:

$$h = h_a + xh_w \quad (9)$$

The humidity ratio can be calculated by using the partial pressure of water vapor and atmospheric pressure [10]:

$$x = \frac{0.62198p_w}{p_a - p_w} \quad (10)$$

The partial pressure can be calculated using the relative humidity and the temperature of the air:

$$p_w = RH \frac{e^{\frac{77.3450 + 0.0057T - \frac{7235}{T}}{T^{8.2}}}}{T^{8.2}} \quad (11)$$

Figure 5 shows the relationship of humidity ratio on temperature by combining equations 10 and 11. By increasing temperature, the humidity ratio is increased drastically.

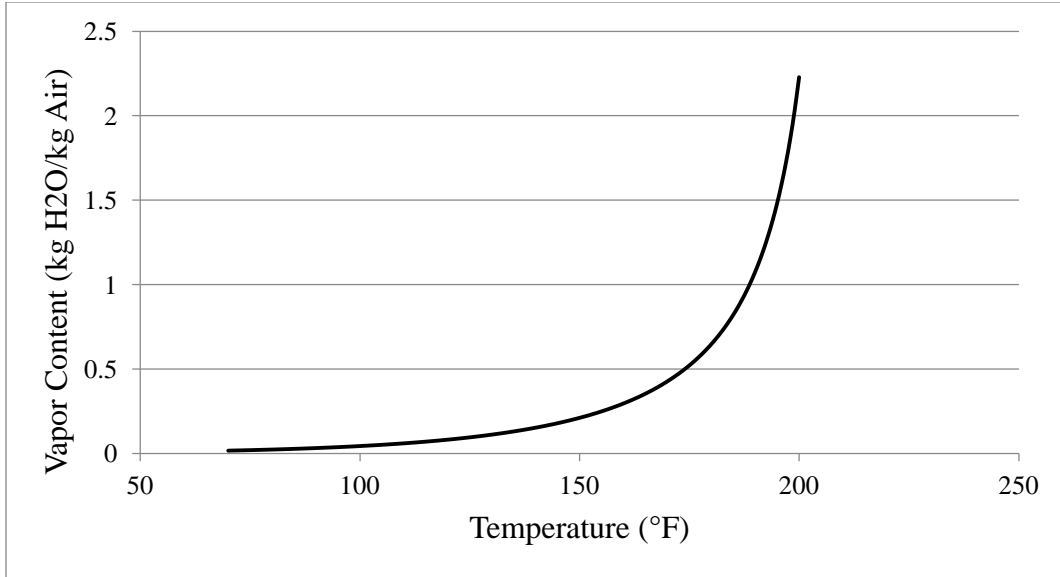


Figure 5. Vapor content of moist air at 100% RH.

The specific enthalpy of air can be expressed in terms of heat capacity and temperature:

$$h_a = c_{pa}T \quad (12)$$

The specific enthalpy of water vapor can be expressed using the specific evaporation enthalpy of water at 0 °C:

$$h_w = c_{pv}T + h_{we} \quad (13)$$

The energy of the steam input can be determined using steam tables [11]. The energy input and output of brine and distillate streams can be determined by the equation:

$$H_{bi} = c_{pw}T_w\dot{m}_w \quad (14)$$

If a dewvaporation system is not at steady state and water is accumulated or depleted, accumulation of energy occurs. The accumulation of energy can be taken into account by performing a mass balance on the water accounting for all input and output streams.

Inputs include the feed, steam, and air inputs, and outputs include brine, distillate and air

outputs, as well as evaporated water leaking out of the system if leakage is present. The mass balance can be calculated using the equation:

$$\text{Mass Input} - \text{Mass Output} = \text{Accumulation} \quad (15)$$

The resulting accumulation of water can be incorporated into the energy equation using the rate of accumulation of water mass and the average temperature of the system:

$$\dot{H}_{wa} = c_{pw} \bar{T} \dot{m}_{wa} \quad (16)$$

The heat loss of the system can be calculated by substituting all sources of heat input, output, and accumulation into equation 7 and solving for the heat loss term.

One interesting method to increase the efficiency of Dewvaporation towers is using a desiccant. Desiccants absorb water from air while simultaneously releasing heat. Since the temperature of heat necessary to drive the Dewvaporation process is low, desiccants can replace steam or another heat source at the top end of the tower. The desiccant can then be regenerated using solar energy. Water evaporation from lithium bromide liquid desiccant can occur at a rate as fast as 0.45 lb/ft² per day [12]. After regeneration, the desiccant can be reused as a heat source in Dewvaporation towers. Although lithium bromide is highly corrosive, Dewvaporation towers are unaffected due to the anti-corrosive polypropylene material [13]. Using lithium bromide in Dewvaporation towers has been shown to increase the energy reuse factor by as much as 50%. The cost of clean water using liquid desiccant in a Dewvaporation system was determined to be \$0.96 per m³ [14].

In previous experiments, Dewvaporation towers have demonstrated energy reuse factors ranging between 7-11 [15, 7]. Operating Dewvaporation towers in a cascade

series in a similar fashion as multiple effect distillation can increase the energy reuse factor to as high as 44 [16].

3 MATERIALS AND METHODS

The units were built from 2' x 4' sheets of twin-wall, polypropylene plastic as shown in Figure 6. The sheets were 4 mm wide from wall-to-wall and have a cost of \$0.05/sqft [6]. The plastic walls and the separations between each vein inside the walls measured 0.009". The cloth was One Tuff brand dropcloth and was plastic on one side and a paper-like absorbent material on the other side. The dropcloth thickness was 0.2 mm. TDS was measured by a Multi-Parameter Testr 35 Series probe. Feed water for all experiments was Phoenix tap water of approximately 580 ppm total dissolved solids. Adhesion and sealing was accomplished using 100% General Electric brand silicon II. The pumps used were Powercool water pumps rated at either 5000 or 7000 cfm. Reticulated foam, a porous, elastic foam, was used to restrict water and air movement.

4 UNIT CONSTRUCTION, RESULTS, AND DISCUSSION

The method of determining and increasing efficiency of a horizontal unit was twofold. First, an initial, batch unit called Unit 1 was built and tested. The design for Unit 1 is shown in Figure 7. Blue arrows indicate air flow through the hollow spaces created by the 2 mm gap between each of the 8 plastic sheets. Red arrows indicate air flow through the hollow veins inside each plastic sheet. Second, a continuous unit called Unit 2 was built taking into account lessons learned from Unit 1. Unit 1 functioned similarly to Dewvaporation towers. The unit consisted of eight 2' x 4' parallel plastic sheets, each



Figure 6. Twin-wall extruded polypropylene.

covered in cloth and separated from one another by a distance of 2 mm with the veins oriented horizontally. Air flowed in through the top of the front of the unit, and was redirected to flow horizontally by a sheet of plastic with horizontally oriented veins. The cloth was simultaneously dampened by pumps carrying liquid water to the top of the unit. The air flowed horizontally through the 2 mm wide open channel created by the space in between each plastic sheet and across the damp cloth until it reached the back of the unit, where it combined with steam and made a 180° turn back into the hollow veins inside

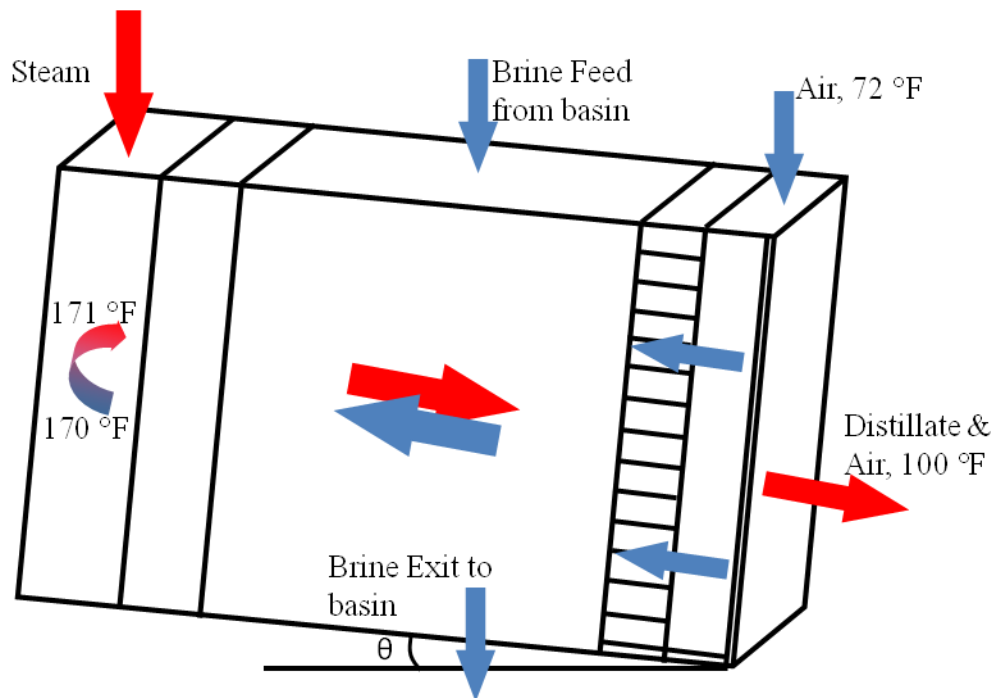


Figure 7. Design of Unit 1.

each sheet of plastic. The air decreased in temperature gradually and condensed liquid distillate until returning to the front of the unit, exiting along with the distillate. Unit 1 was set up with an adjustable angle from 0° to 22° to use gravity to drive distillate towards the front of the unit. An initial test before the unit was built determined that water would flow out of 4 mm veins in the plastic sheets at approximately 21° . Unit 1 was built at a variable angle because the angle at which water dripped was suspected to become shallower as air flow was introduced. The end of the unit was sealed except for a small opening for steam to enter.

A basin was built to house the water being pumped to the top of the unit, as shown in Figure 8. Blue arrows are brine exiting the basin to the unit, and green arrows are brine feed from unit. The basin was placed directly under the unit to catch the excess water flowing down the cloth. Ten pumps were placed in the basin. The pumps were spaced evenly in the basin in order to maintain a temperature gradient in which the back side of the basin was warmer than the front. Although mixing within the vicinity of each pump occurred, 10 pumps allowed for 10 different temperatures of water to be pumped to the top of the unit. A number of challenges existed that needed to be addressed before building the unit.

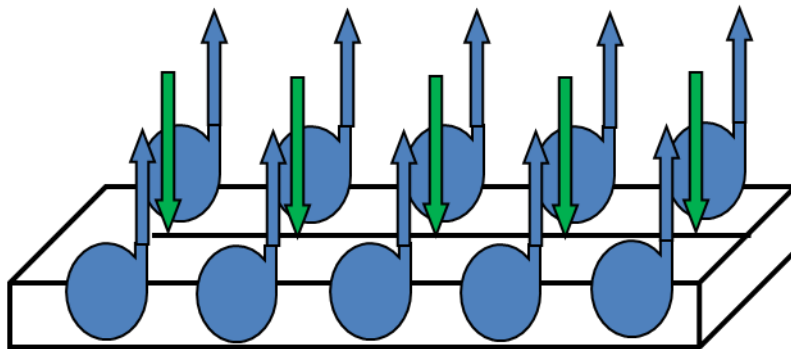


Figure 8. Diagram of basin.

Challenge 1: Air inlet velocity distribution. Since the air entered the unit at the top, the area near the top of the unit would receive higher air flow rates than the area at the bottom. Since a theoretical optimal air flow rate exists in which air has enough space time to absorb water and heat but is not too slow as to hinder distillate production, it was important to make the air velocity even at all heights in the unit. Several different geometries were independently tested outside the unit.

Geometry A is shown in Figure 9a, in which angled veins distribute air from the top of the unit to the entire length of the unit. Geometry A was proven to be ineffective in equalizing the velocity gradient as shown in Figure 10, because veins aimed at the bottom of the unit were longer and thus had a greater pressure drop, discouraging air to flow down the longer veins.

Geometry B is shown in Figure 9b, in which air flows down a cavity lined with horizontal holes in which the air can flow out of at any point in the cavity. Geometry B

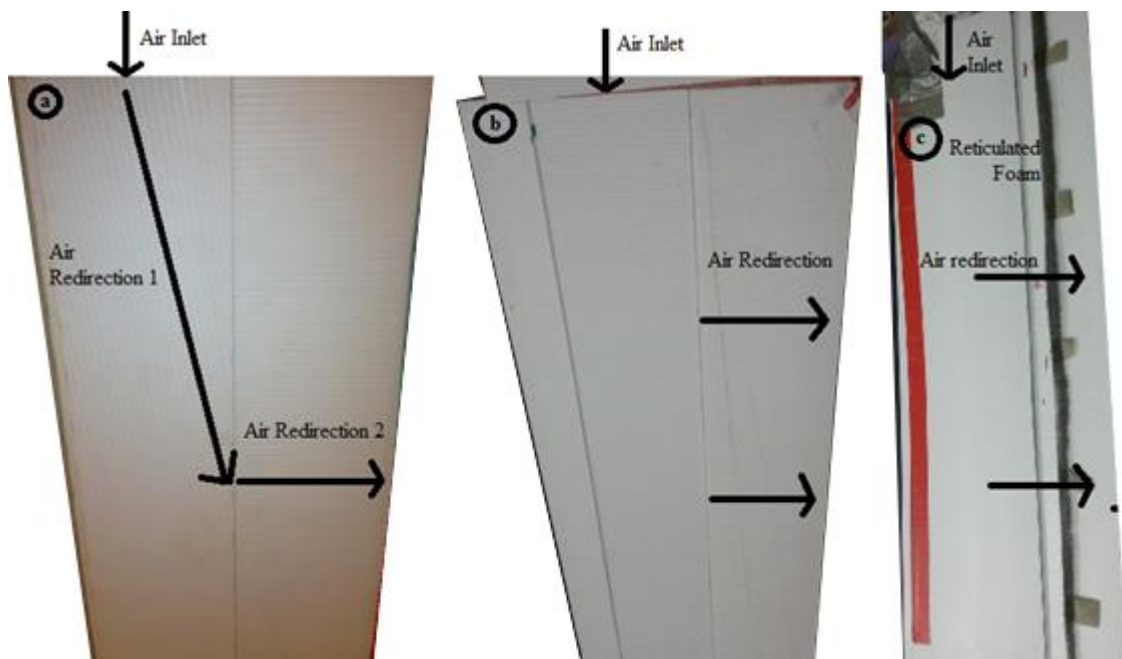


Figure 9. Geometries of different air inlet tests.

created a significantly equalized velocity gradient compared to Geometry A, as shown in Figure 10.

Geometry C is shown in Figure 9c which is similar to Geometry B, but with a thin strip of reticulated foam placed at the outlet of the horizontally aimed veins. Geometry C has the most equalized velocity gradient of any other geometry attempted. The velocity gradient is optimal because the addition of the reticulated foam increases the pressure drop associated with flowing out of the horizontal veins, and causes the air to explore the cavity adjacent to the horizontal veins more thoroughly, causing the air pressure at almost all vertical points in the cavity to be equalized. The equalized pressure causes the velocity at all points to be almost equivalent, as shown in Figure 10. Geometry C was the geometry implemented into Unit 1.

Challenge 2: Allowing brine to cycle vertically through the unit in order to wet the cloth but only allowing air to flow horizontally without escaping through the top and bottom of the unit. The challenge was addressed by tightly fitting sponges in the top and

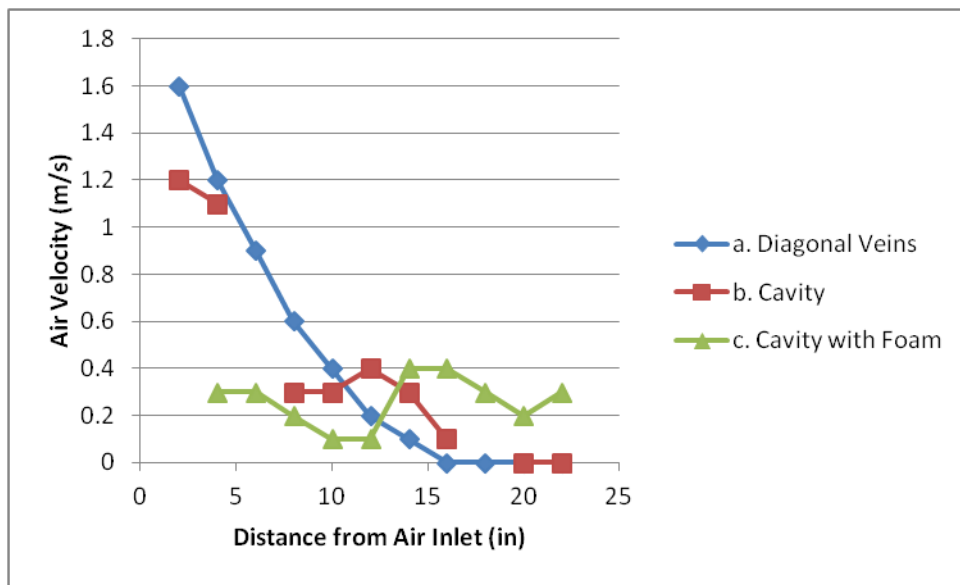


Figure 10. Air velocity distribution for each of the 3 test structures shown in Figure 9.

bottom of the unit as shown in Figure 11 such that water could completely wet the sponges, allowing water to permeate through but providing a significant enough barrier for air that the air would instead flow horizontally through the unit. The sponges on top also spread out incoming brine water across the sponge to drip brine more evenly.

Challenge 3: Preventing water from adhering and flowing horizontally along the bottom of sponges because of unit's downward angle. Dams were installed periodically between the sponges to prevent horizontal flow of brine water, as shown in Figure 12. 10 dams were built so that each of the 10 pumps deposited water onto one sponge segment, so even if there was some horizontal flow, the water would stop at the dam and the temperature gradient would not be affected.

Challenge 4: Preventing unit deformation and warping. Once all 8 pieces of polypropylene plastic were glued together with spacers in between, clear polycarbonate end pieces were put on the left and right sides of the unit. Although the clear polycarbonate end pieces provided an advantage in that they allowed the visual



Figure 11. Tightly fitting sponges to address Challenge 2.



Figure 12. Sponge dams to address Challenge 3.

inspection of the cloth to make sure it was wetting completely, the polycarbonate was also warped slightly, causing the unit to bow out on each side. To combat this, wooden side pieces were put in place to hold the unit tightly together and prevent further warping that might occur as the unit began operation, as shown in Figure 13.

Challenge 5: Overflow of water on sponges due to high pump flow rate. As the pumps brought brine to the top of the unit, the area between dams was overflowing as the flow rate of the pumps was too high for the sponges to permeate. By filling the pump tubing with reticulated foam and clamping the area with the reticulated foam as shown in Figure 14, the flow rate was reduced. By adjusting the clamp tightness, the flow rate of each pump was reduced to 200 mL/min, which prevented the sponge area from overflowing.

Results from Unit 1 with challenges 1-5 addressed are shown in Appendix A. The



Figure 13. Vertical wooden side braces to address Challenge 4.



Figure 14. Clamps and reticulated foam to address Challenge 5.

energy reuse factor of Unit 1 was 1.4. The results from Unit 1 brought to light a number of new challenges.

Challenge 6: Reducing heat loss. The heat loss was decreased by applying insulation to both the basin and the unit as shown in Figure 15. As shown in Appendix A, the increase in insulation increased the energy reuse factor to 1.5.

Challenge 7: Increasing the temperature gradient in the basin. A new segmented basin was built with compartments for each pump as shown in Figure 16. Without walls, the pumps agitate the water in the basin significantly, causing the water to mix significantly. With compartments, the mixing effect was eliminated. The new basin in conjunction with Unit 1 allowed the basin segment closest to the back of the unit to equilibrate to 115 °F while the segment closest to the front maintained a temperature of 72 °F, a significant improvement in temperature gradient over the non-segmented basin.

Challenge 8: Eliminating leaks in the steam box and elsewhere in the unit. The silicon used to seal the unit did not make a good seal with the polycarbonate outside walls of Unit 1, and small leaks often occurred that reduced efficiency and brought forth



Figure 15. Thermal insulation on the unit and basin to address Challenge 6. The insulation fits between the wooden beams, and the wooden beams also act as insulation.



Figure 16. Second basin built with segments to reduce mixing.

concerns that brine was leaking into the distillate. By using polypropylene for the outside walls in Unit 2, the silicon made a proper seal and eliminated all leaks in the unit.

Challenge 9: Determining the optimal angle of operation. At high angles of operation, a significantly large triangle of area at the back of the unit was not wetted

since the brine flowed straight down from the top of the unit and did not come into contact with the lower triangular area. At low angles of operation, distillate did not flow out of the veins in the plastic since there was not enough force in the horizontal direction to overcome the adhesive force of distillate droplets clinging to the inside of the veins. The optimal angle is the lowest possible angle in which distillate droplets fall with the forces of gravity and air flowing past. Through experimentation, the angle was determined to be approximately 0.1 radians.

Challenge 10: Equalizing brine flow rate from the unit to the basin. One sponge separated by dams on the bottom of Unit 1 was not depositing any brine into the basin, causing the segment of the basin directly underneath to dry up and other segments to overflow. The lack of flow was probably caused by water flowing over the sponge and dam into the sponge area adjacent. By lowering the sponges slightly in Unit 2 and keeping the dams at the same height, each sponge area had space to accumulate a small amount of brine to prevent all the brine for a particular sponge area to flow into the next sponge area. Other methods for addressing Challenge 10 included moving the entire basin forward or backward to make each pump approximately pump water back into the same pump. Loosening tube clamps for pumps whose basin segments were overflowing also helped, because a portion of the water that each pump sends to the top of the unit flows into neighboring basin segments. In addition, tightening tube clamps for pumps whose basin segments were too low helped, because less of that segment's water was pumped to neighboring basin segments.

Challenge 11: Preventing plastic sheets from touching each other. In Unit 1, some of the walls appeared to almost touch each other when viewing the top of the unit. Some

sheets may possibly have been touching each other inside the unit, but were not visible. Any area inside the unit in which sheets touch each other is wasted space because air cannot flow past. Increasing the spacer width from 2 mm to 4 mm in Unit 2 prevented the walls from touching each other. In addition, the pressure drop associated with air flow to the bottom of the cavities in the air inlet was lowered as a result, increasing the degree to which air will explore the cavity and normalizing the air distribution further past the improvements made through Challenge 1. To further decrease the chances of sheets touching each other internally, particularly around the edges of each sheet, a new pattern of spacers was designed.

As shown in Figure 17, spacer 1 has vertical veins so that air cannot escape the cavity created by spacers 1 and 2. Spacer 2 has horizontal veins so air in the cavity cannot pass. Spacers 3 and 6 have horizontal veins so air can pass through the blue cloth region. Spacers 4 and 5 have vertical veins so that water can pass through the blue cloth region. The small square spacers in the middle have horizontal veins so air can pass through.

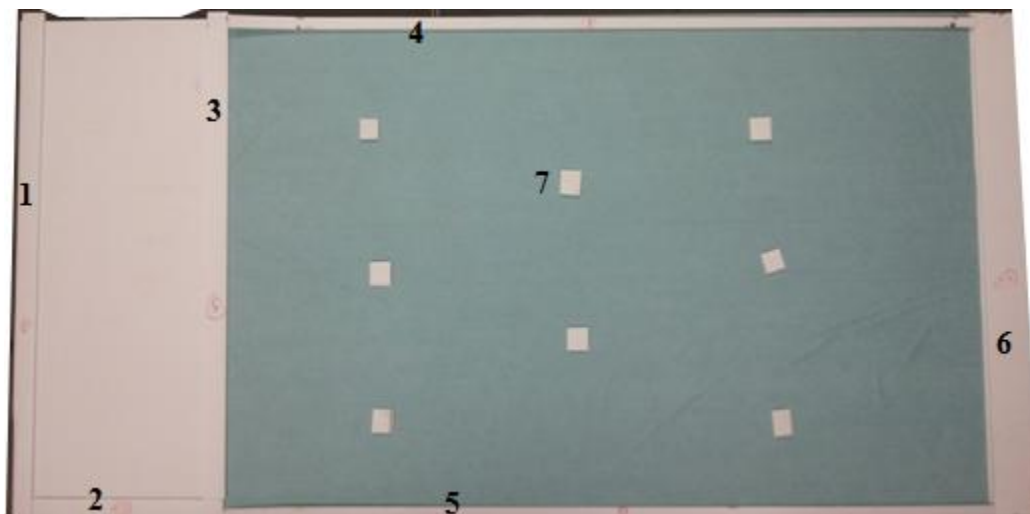


Figure 17. Spacer design on Unit 2 to address Challenge 11.

Challenge 12: Making the unit continuous rather than batch. An elevated bucket of water and a flow meter were set up such that a controlled amount of feed brine could be fed to the first compartment in the basin. Holes were drilled in each wall in the basin so that water could overflow from one compartment to another as feed water was fed in. A brine collection box was set up so concentrated brine from the last compartment could overflow into the box as needed.

In addition to the lessons learned from Challenges 6-11, several other features were modified when constructing Unit 2. Despite the improvements in the air velocity gradient in challenge 1 and 11, additional improvements in air velocity could increase efficiency further. Whereas the cavity in Unit 1 was only 2" wide, the cavity size of Unit 2 was increased to 8", further decreasing the pressure drop associated with exploring the cavity.

In Unit 2, the number of sheets was increased from 8 to 12. On the left and right sides of Unit 1, the two cavities on either side created by the polycarbonate walls and the first and eighth plastic sheets still had air input, but the air inside only had heat transfer from one wall because the polycarbonate walls did not have condensate in their veins. The result was a lowered overall efficiency because the two air streams did not have as high of a temperature or fraction of water vapor as the seven air streams in the middle of the unit. By increasing the number of plastic sheets to 12, the effect is lessened because a lower fraction of air streams ($2/13$ instead of $2/9$) are affected, increasing heat transfer efficiency.

Evaporation from the basin was a significant cause for concern in Unit 1 since much of the basin was uncovered. A shell of insulation was created for Unit 2 such that

the basin was covered to slow evaporation. The completed Unit 2 with the insulation over the basin and brine collection box is shown in Figure 18.

Unit 2 would benefit from increased insulation coverage in five different areas as shown in Figure 18. 1) The top insulation strip should be secured more tightly to the top of the unit. 2) The tubing leading from the basin to the top of the unit should be insulated. 3) The bottom of the unit should be tightly insulated such that the basin and unit are completely enclosed with insulation and no holes exist. 4) The front side of the insulation covering the pumps should be covered with insulation similarly to the back side. 5) The front-bottom corner of the unit should be insulated so no outside air can leak inside.

Unit 2 run data is shown in Appendix A. Unit 2 exhibited a much improved energy reuse factor over Unit 1 in two separate trials of 2.5. The low contaminant



Figure 18. Completed Unit 2 with insulation over the pumps and a short brine exit box beside the pumps.

concentration of the distillate of 13 ppm indicates that there were little or no leaks of brine into the distillate. A photo of dew formation in the front of Unit 2 is shown in Figure 19. Several new challenges were identified when running Unit 2 to improve the energy reuse factor further.

Challenge 13: Improve continuous operation. The pumps in Unit 2 operate by taking the water in a compartment and transporting it to the top of the unit, where the water trickles back down to all neighboring compartments. For example, the water in the pump segment in which the feed enters (the closest pump segment to the front on the right) deposits water to the feed segment, the second closest pump segment to the front, and the other pump closest to the front on the left, which is also the pump segment in which water exits the basin to the brine collection box. The amount of feed water needed to operate the unit could be vastly decreased if there were a center divider in the unit that prevented brine from passing from the right side to the left side of the unit and vice versa. By incorporating the center divider, the brine would then flow from the pump segment



Figure 19. Dew formation as seen through the front of the tower.

with the brine feed around the entire basin, increasing in salt concentration as it flowed, until the last pump segment, in which the salt concentration would be highest and the brine would exit into the brine collection box. In addition, adding a center divider in the unit would decrease the number of neighboring basin segments each pump delivers to, simplifying the process, decreasing the difficulty in balancing the pump flow rates, and decrease mixing of brine in order to establish a larger temperature gradient.

Challenge 14: Further reduce brine mixing. When the pumps are on, mixing occurs because the pumps deposit some brine to neighboring pump segments. By pulsing the pumps by turning the pumps on for a short period and then turning them off for a short period, mixing may be reduced while still keeping the cloth damp.

Challenge 15: Eliminate all remaining sources of leakage. The basin is made of polycarbonate, which is significantly more prone to cracking and leakage than polypropylene, particularly when high temperatures are present. Building a new basin made of polypropylene will prevent future leaks.

Challenge 16: Decreasing heat loss. The basin is a major source of heat loss since it consists of a large body of water. Warm air emanated out of every place in the basin and tower that was not completely covered with insulation. An energy balance was performed on Unit 2, and 493 W was lost as heat. 493 W was very significant, since 493 W was more than half of the heat input from the steamer (827 W of heat was input through the steamer). By containing the heat in the basin and unit more effectively with insulation, a significant increase in the equilibrium temperature gradient of the basin would most likely be attained. Since vapor content of moist air increases drastically with an increase in air temperature as shown in Figure 5, increasing the temperature on the

steam side of the unit would increase the multiple effect significantly. Areas in which insulation coverage could be improved are shown in Figure 18.

Challenge 17: Attaining a more uniform temperature vertically in the unit. On the back side of the unit, the temperature at the top of the unit is generally about 10 °F warmer than the bottom of the unit, most likely because the steam entrance is at the top of the unit. In addition, the exit air temperature is generally about 10 °F higher on top than on bottom as well. By installing a porous wall near the steam entrance to increase the pressure drop from the steam inlet to the rest of the unit, the steam may reach farther down in the unit and release steam more uniformly in a similar mechanism as the air inlet.

Challenge 18: Implementing a continuous longitudinal pump design. Instead of using 10 pumps, using a single continuous pump that maintained a temperature gradient would decrease mixing in the basin and create a more effective heat exchanger since, instead of there being 10 areas with different temperature, there would be a single area with a uniform temperature gradient similar to a cross-flow heat exchanger.

Challenge 19: Optimization. Tuning the air inlet flow rate could potentially increase efficiency by allowing more space-time for the air to more effectively absorb moisture and heat without cooling the unit as much if the air flow rate was lowered, or increase the amount of water evaporated if the air flow rate was raised. The steam inlet flow rate can also be tuned to examine the effects on the energy reuse factor.

5 CONCLUSION

A horizontal desalination-dehumidification unit has potential as a viable desalination technology due to the simplicity of operation, ease of construction, and potential for high efficiency. An energy reuse factor of 2.5 and a contaminant concentration of 13 ppm were achieved for unit 2. Using more plastic sheets in unit construction, normalizing the inlet air velocity gradient, decreasing basin mixing, and equilibrating compartment water input and output were improvements implemented over the course of the research. Sponges, dams, reticulated foam, clamps, and insulation proved to be valuable tools in unit operation and increasing efficiency. Areas for improvement and future steps include more careful insulation placement, implementation of a continuous longitudinal pump, construction of a center barrier, and optimization.

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APPENDIX A
DATA FROM UNIT TESTING

Unit 1 test on 2/5/15:

Tower angle: 10.5°

Distillate product: 2.02 L/h

Temperature at each pump starting at the front of the tower and zig-zagging to the back of the tower:

Pump: 1 2 3 4 5 6 7 8 9 10

Temp (°F): 81 77 81 78 81 78 79 78 79 77

Temperature of bottom sponges starting from the front of tower and ending at the back:

Location: 1 2 3 4 5 6 7 8

Temp (°F): 80 86 78 84 89 90 96 95

Distillate temp: 74 °F

Air outlet temp: 81 °F

Steam box temp: 182.2 °F

Temp just outside steam box: 152.6 °F

Steam generator output per hour: 1478 mL/hr

Unit 2 Test on 4/5/15

The compartment in the basin furthest to the steam box acquired a maximum temperature of 111 F.

Tower angle: 0.1 radians

Steam box temperature: 170 F at the top and 160 F at the bottom

Air exit velocity: 1.7 m/s

Distillate: 3280 mL/hr, 21 ppm

Test length: 2 hours (plus prep time of 1.5 hours)

"Saline" feed: 1 gal/hr ("saline" is in quotes because I used tap water, not salt water)

Exit "saline:" Exit flow out of the basin varied from no flow to high flow during the 2 hour test, but about 3500 mL was collected from the entire test.

Unit 2 Test on 4/9/15

The compartment in the basin furthest to the steam box acquired a maximum temperature of 111.9 F.

Tower angle: 0.1 radians

Steam box temperature: 170.5 F at the top and 162.4 F at the bottom

Air exit velocity: 1.7 m/s

Air exit temperature: 101 F

Distillate: 3280 mL/hr (exactly the same as the previous test!), 19 ppm

Test length: 95 minutes (plus prep time of 1.5 hours)

"Saline" feed: 1.5 gal/hr

Exit "saline:" No saline exited the basin during the test through the last compartment (probably because there were basin leaks).

Ending basin compartment temperatures: (From closest to the steam box to furthest) (F)
104.8, 102.3, 103.8, 100.8, 98, 97, 91.9, 96.7, 82.4, 82.6

Steam generator boiling rate over the 95 minute test: 2340 mL/hr = 0.390 gal/hr = 3.25 lb/hr

Unit 2 Test on 4/12/15

Distillate flow rate: 3200 mL/hr

Distillate TDS: 13 ppm

Brine inlet flow rate: 1.5 gal/hr

Air speed through unit: 1.7 m/s

Temperature of top of steam box: 173 F

Temperature of bottom of steam box: 167 F

Distillate temp: 99 F

Equilibrium temperature of basin compartment closest to steam box: 121.9 F

Test time: 1 hour, 50 minutes.

Basin ending temperatures of each compartment from closest to steam box to furthest (F):
121.9, 116.0, 106.6, 106.7, 103.1, 103.1, 95.7, 99.1, 95.4, 87.1

The TDS in the compartment in which brine was flowing into was 387 ppm.

TDS of the brine outlet collection box was 400 ppm.

TDS of the two compartments closest to the steam box were 458 and 480 ppm.

TDS of the last compartment in the front leading into the brine collection box was 450 ppm.

5375 mL of brine was collected from the brine outlet collection box.

APPENDIX B

UNIT 2 ENERGY BALANCE

Energy Balance on Unit

Heat in: Steamer, air in, brine in

Heat out: Air out, distillate out, brine out, heat loss from basin and unit

Accumulation: Loss of water from basin

Assumptions:

The system chosen is shown in Figure 20.

Average basin temperature: 103.5 (39.7 °C)

Take reference state for water and air enthalpy calculations as $H = 0$ at 0 °C

No accumulation of air in system

No water evaporation into the environment – note: may lead to an underestimation of water depletion in the system, but may be neglected since the accumulation term is relatively small. However, the final heat loss number will be somewhat smaller than if this term were taken into account.

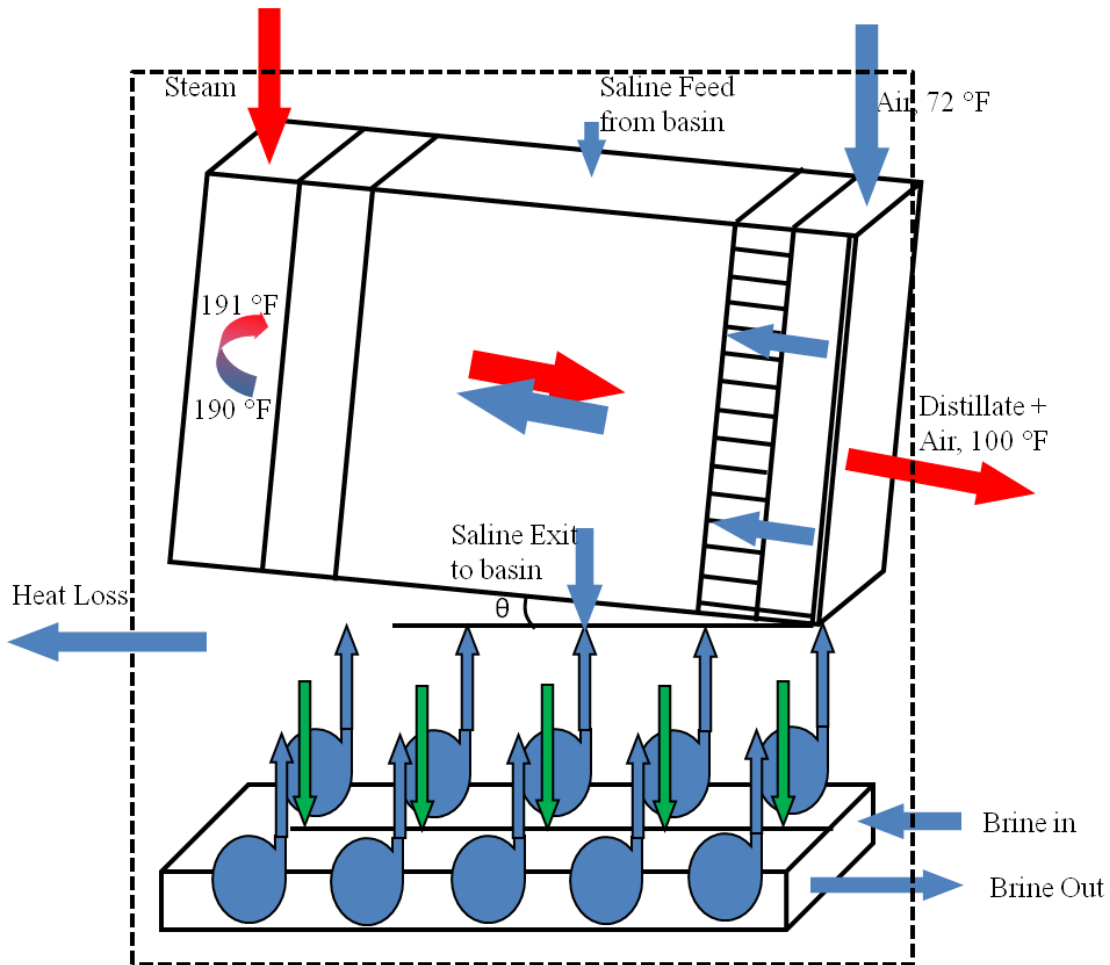


Figure 20. System chosen for mass and heat balances.

Heat in:

Steamer input: Energy from 0.349 gal/hr steam entering unit + 0.041 gal/hr condensed water:

Specific enthalpy of saturated steam at 100 °C: 2676 kJ/kg

$$0.349 \frac{\text{gal}}{\text{hr}} \left(\frac{1 \text{ hr}}{3600 \text{ s}} \right) \left(\frac{3.78 \text{ kg}}{1 \text{ gal water}} \right) \left(\frac{2676 \text{ kJ}}{1 \text{ kg}} \right) \left(\frac{1000 \text{ J}}{1 \text{ kJ}} \right) \\ + 0.041 \frac{\text{gal}}{\text{hr}} \left(\frac{1 \text{ hr}}{3600 \text{ s}} \right) \left(\frac{3.78 \text{ kg}}{1 \text{ gal water}} \right) \left(419 \frac{\text{kJ}}{\text{kg}} \right) \left(1000 \frac{\text{J}}{\text{kJ}} \right) = \mathbf{998 \text{ W}}$$

Air out: 99 °F (37.2 °C) at 100% humidity. Air in: 70.5 °F (21.4 °C) at 15% humidity.

Enthalpy of air in:

$$h = h_a + xh_w$$

Vapor content of Moist Air can be expressed as:

$$x = \frac{0.62198p_w}{p_a - p_w}$$

The partial pressures of water vapor can be calculated using:

$$p_w = RH \frac{e^{77.3450 + 0.0057T - \frac{7235}{T}}}{T^{8.2}}$$

Specific enthalpy of dry air can be expressed as

:

$$H_a = c_{pa}T$$

Specific Enthalpy of Water Vapor can be expressed as:

$$H_w = c_{pv}T + h_{we}$$

Substituting appropriate values in for p_w ,

$$p_w = RH \frac{e^{77.3450 + 0.0057T - \frac{7235}{T}}}{T^{8.2}} = 0.15 \frac{e^{77.3450 + 0.0057T - \frac{7235}{T}}}{T^{8.2}} = 381.2 \text{ Pa}$$

Using appropriate values for heat capacities and heat of vaporization, and combining these equations together:

$$H = c_{pa}T + \frac{0.62198p_w}{p_a - p_w}(c_{pw}T + h_{we})$$

$$\begin{aligned} H_0 &= \left(1.006 \frac{kJ}{kg^\circ C}\right) (21.4^\circ C) \\ &+ \frac{0.62198(381.2 \text{ Pa})}{101325 \text{ Pa} - (381.2 \text{ Pa})} \left[\left(1.84 \frac{kJ}{kg^\circ C}\right) (21.4^\circ C) + 2501 \frac{kJ}{kg} \right] \\ &= 27.50 \frac{kJ}{kg} \end{aligned}$$

Air flow through unit:

$$1.7 \frac{m}{s} * \pi * (1 \text{ in})^2 \left(\frac{0.0254 \text{ m}}{\text{in}}\right)^2 = 3.445 \times 10^{-3} \frac{m^3}{s}$$

Using ideal gas law:

$$\begin{aligned} Pv = nRT \rightarrow n &= \frac{PV}{RT} = \frac{101325 \text{ Pa} * 1.378 \times 10^{-2} \frac{m^3}{s}}{8.314 \frac{\text{Pa} * m^3}{\text{mol} * K} * 310.35 \text{ K}} = 0.541 \frac{\text{mol}}{s} \left(\frac{28.97 \text{ g}}{\text{mol}}\right) \\ &= 3.92 \frac{g}{s} \text{ air} \end{aligned}$$

Heat of air in:

$$15.67 \frac{g}{s} \left(\frac{1 \text{ kg}}{1000 \text{ g}}\right) \left(27.50 \frac{kJ}{kg}\right) = 107.7 \text{ W}$$

Heat of brine in:

$$\begin{aligned} H_{bi} &= c_{pw}T_{bi}\dot{m}_{bi} \\ &= 4.18 \frac{kJ}{kg^\circ C} (21.4^\circ C) \left(1.5 \frac{\text{gal}}{\text{hr}}\right) \left(\frac{1 \text{ hr}}{3600 \text{ s}}\right) \left(\frac{3785.4 \text{ mL}}{\text{gal}}\right) \left(\frac{1 \text{ g}}{\text{mL}}\right) \left(\frac{1 \text{ kg}}{1000 \text{ g}}\right) \\ &= 142 \text{ W} \end{aligned}$$

Heat Out:

Enthalpy of air out:

Substituting appropriate values:

$$p_w = RH \frac{e^{77.3450+0.0057T-\frac{7235}{T}}}{T^{8.2}} = 1 \frac{e^{77.3450+0.0057(310.35 \text{ }^\circ\text{C})-\frac{7235}{310.35 \text{ }^\circ\text{C}}}}{(310.35 \text{ }^\circ\text{C})^{8.2}} = 6326.2 \text{ Pa}$$

$$\begin{aligned} h_1 &= c_{pa}T_1 + \frac{0.62198p_{w1}}{p_{a1} - p_{w1}} (c_{pw}T_1 + h_{we}) \\ &= \left(1.006 \frac{\text{kJ}}{\text{kg}^\circ\text{C}}\right) (37.2 \text{ }^\circ\text{C}) \\ &\quad + \frac{0.62198(6326.2 \text{ Pa})}{101325 \text{ Pa} - (6326.2 \text{ Pa})} \left[\left(1.84 \frac{\text{kJ}}{\text{kg}^\circ\text{C}}\right) (37.2 \text{ }^\circ\text{C}) + 2501 \frac{\text{kJ}}{\text{kg}} \right] \\ &= 143.8 \frac{\text{kJ}}{\text{kg}} \end{aligned}$$

Using the change of enthalpy of the air:

$$15.67 \frac{\text{g}}{\text{s}} \left(\frac{1 \text{ kg}}{1000 \text{ g}} \right) \left(143.8 \frac{\text{kJ}}{\text{kg}} \right) = 562.5 \text{ kW}$$

Enthalpy of brine out:

$$\begin{aligned} h_{bo} &= c_{pw}T_{bo}m_{bo} = 4.18 \frac{\text{kJ}}{\text{kg}^\circ\text{C}} (35.2 \text{ }^\circ\text{C}) \left(2932 \frac{\text{mL}}{\text{hr}} \right) \left(\frac{1 \text{ hr}}{3600 \text{ s}} \right) \left(\frac{1 \text{ g}}{\text{mL}} \right) \left(\frac{1 \text{ kg}}{1000 \text{ g}} \right) \\ &= 120 \text{ W} \end{aligned}$$

Enthalpy of distillate out:

$$\begin{aligned} h_{do} &= c_{pw}T_{do}m_{do} = 4.18 \frac{\text{kJ}}{\text{kg}^\circ\text{C}} (37.2 \text{ }^\circ\text{C}) \left(3200 \frac{\text{mL}}{\text{hr}} \right) \left(\frac{1 \text{ hr}}{3600 \text{ s}} \right) \left(\frac{1 \text{ g}}{\text{mL}} \right) \left(\frac{1 \text{ kg}}{1000 \text{ g}} \right) \\ &= 127 \text{ W} \end{aligned}$$

Energy Accumulation:

Mass balance on Water

Water in:

1.5 gal/hr feed = 5678 mL/hr

0.390 gal/hr steam = 1476 mL/hr

Air in:

$$\begin{aligned}
 m_{wi} = x_i m_{ai} &= \frac{0.62198(381.2 \text{ Pa})}{101325 \text{ Pa} - (381.2 \text{ Pa})} 15.67 \frac{\text{g}}{\text{s}} \\
 &= 0.002349 \left(15.67 \frac{\text{g}}{\text{s}} \right) \left(\frac{3600 \text{ s}}{\text{hr}} \right) \left(\frac{1 \text{ mL}}{\text{g}} \right) = 132.5 \frac{\text{mL}}{\text{hr}}
 \end{aligned}$$

Water out:

2932 mL/hr brine outlet,

3200 mL/hr distillate

Air out:

$$\begin{aligned}
 m_{wo} = x_o m_{ao} &= \frac{0.62198(6326.2 \text{ Pa})}{101325 \text{ Pa} - (6326.2 \text{ Pa})} 15.67 \frac{\text{g}}{\text{s}} \\
 &= 0.04142 \left(15.67 \frac{\text{g}}{\text{s}} \right) \left(\frac{3600 \text{ s}}{\text{hr}} \right) \left(\frac{1 \text{ mL}}{\text{g}} \right) = 2336 \frac{\text{mL}}{\text{hr}}
 \end{aligned}$$

Water accumulation:

$$\begin{aligned}
 \text{accumulation} &= m_i - m_o \\
 &= 5678 \frac{\text{mL}}{\text{hr}} + 1476 \frac{\text{mL}}{\text{hr}} + 132.5 \frac{\text{mL}}{\text{hr}} - 2932 \frac{\text{mL}}{\text{hr}} - 3200 \frac{\text{mL}}{\text{hr}} - 2335 \frac{\text{mL}}{\text{hr}} \\
 &= -1180.5 \frac{\text{mL}}{\text{hr}}
 \end{aligned}$$

Heat of water accumulation:

$$\begin{aligned}
 \dot{H}_{wa} = c_{pw} \bar{T} \dot{m}_{wa} &= 4.18 \frac{\text{kJ}}{\text{kg}^\circ\text{C}} (39.7^\circ\text{C}) \left(-1180.5 \frac{\text{mL}}{\text{hr}} \right) \left(\frac{1 \text{ hr}}{3600 \text{ s}} \right) \left(\frac{1 \text{ g}}{\text{mL}} \right) \left(\frac{1 \text{ kg}}{1000 \text{ g}} \right) \\
 &= -54.3 \text{ W}
 \end{aligned}$$

Complete heat balance:

$$\begin{aligned}
 \text{accumulation} &= \text{heat in} - \text{heat out} \\
 &= \text{Steamer} + \text{air in} + \text{brine in} - \text{air out} - \text{distillate out} \\
 &\quad - \text{brine out} - \text{heat loss}
 \end{aligned}$$

$$\begin{aligned}
 \text{heat loss} &= \text{Steamer} + \text{air in} + \text{brine in} - \text{air out} - \text{distillate out} - \text{brine out} \\
 &\quad - \text{accumulation} \\
 &= 998 \text{ W} + 107.7 \text{ W} + 142 \text{ W} - 562 \text{ W} - 127 \text{ W} - 120 \text{ W} - (-54.3 \text{ W}) = \mathbf{493 \text{ W}}
 \end{aligned}$$