AN ADVANCED VAPOR-COMPRESSION DESALINATION SYSTEM

A Dissertation

by

JORGE HORACIO JUAN LARA RUIZ

Submitted to the Office of Graduate Studies of
Texas A&M University
in partial fulfillment of the requirements for the degree of

DOCTOR OF PHILOSOPHY

December 2005

Major Subject: Interdisciplinary Engineering

AN ADVANCED VAPOR-COMPRESSION DESALINATION SYSTEM

A Dissertation

by

JORGE HORACIO JUAN LARA RUIZ

Submitted to the Office of Graduate Studies of
Texas A&M University
in partial fulfillment of the requirements for the degree of

DOCTOR OF PHILOSOPHY

Approved by:

Chair of Committee,	Mark T. Holtzapple
Committee members,	Mahmoud El-Halwagi
	Je C. Han
	John T. Baldwin
Head of Department	N. K. Anand

December 2005

Major Subject: Interdisciplinary Engineering

ABSTRACT

An Advanced Vapor-Compression Desalination System. (December 2005)

Jorge Horacio Juan Lara Ruiz, B.S., National Polytechnic Institute, Mexico;
M.S., Lamar University;
M.Eng., Texas A&M University
Chair of Advisory Committee: Dr. Mark Holtzapple

Currently, the two dominant desalination methods are reverse osmosis (RO) and multi-stage flash (MSF). RO requires large capital investment and maintenance, whereas MSF is too energy intensive.

An innovative vapor-compression desalination system is developed in this study. A comprehensive mathematical model for the heat exchanger/evaporator is described. The literature indicates that extraordinarily high overall heat transfer coefficients for the evaporator are possible at selected operating conditions that employ dropwise condensation in the steam side and pool boiling in the liquid side. A smooth titanium surface is chosen to promote dropwise condensation and to resist corrosion.

To maximize energy efficiency, a combined-cycle cogeneration scheme is employed composed of a gas turbine, a heat recovery boiler, and a steam turbine that drive a compressor. The combined-cycle power source is oversized relative to the needs of the compressor. The excess power is converted to electricity and sold to the open market. A three-effect evaporator is employed. It is fed with seawater, assumed to be 3.5% salt. Boiling brine (7% salt) is in the low pressure side of the heat exchanger and condensing steam is in the high-pressure side of the heat exchanger. The condensing steam flows at 1.52 m/s (5 ft/s), which maximizes the heat transfer coefficient. The plant is sized to produce 37,854 m^3/d (10 mill gal/day) and is assumed to be financed with a 5%, 30-yr municipal bond.

Two economic cases were emphasized: the United States and the Middle East.

For the United States, the fuel costs \$5/GJ (\$5.27/mill Btu) with the latent heat exchanger at $\Delta T = 1.11\,\mathrm{K}\left(2.00^\circ\mathrm{F}\right)$. The required compressor energy is 14 MJ/m^3 (14.7 kW h/thous gal). The capital cost for the U.S. is \$884 d/m^3 (\$3,342/thous gal) and the delivered water selling price is \$0.47/m^3 (\$1.79/thous/gal).

For the Middle East, the fuel costs \$0.5/GJ (\$0.53/mill Btu) with the latent heat exchanger at $\Delta T = 3.33K\ \left(6.00^\circ\mathrm{F}\right)$. The required compressor energy is 26 MJ/m^3 (27.3 kW h/thous gal).). The capital cost for the Middle East is \$620 d/m^3 (\$2,344/thous gal), and the delivered water selling price is \$0.25/m^3 (\$0.95/thous/gal).

In all cases, the water selling price is attractive relative to competing technologies.

To my wife Nelida

Thank you for sharing your life with me.

I am in great debt to you…………

ACKNOWLEDGMENTS

I would like to thank Dr. Scott Wellington, Dr. J. M. (Jack) Hirsch, Dr. Renata Karlin, Mr. J. L. (Jerry) Morris, and Dr. Tom Riddle from the Shell Exploration and Production Game Changer Team for the financial support to make this project possible.

I also appreciate Mr. David Carraba of Gooseneck Manufacturing Co. for his help in analyzing the feasibility to manufacture the novel sheet-shell heat exchanger/evaporator.

I thank the faculty and staff of the Chemical Engineering Department of Texas A&M University for the professionalism and help provided. I also thank my student colleagues for their support.

I want to deeply thank Dr. Mark Holtzapple for his encouraging and knowledgeable advice. His very supportive enthusiasm has been inspirational.

I thank my committee members (Dr. John Baldwin, Dr. Mahmoud El-Halwagi, and Dr. Je Han) for their advice and continuous support.

But most of all I want to thank my family Raniero, Jorge, Nelida, and Fremiet for their unconditional moral support; I am especially in debt with my wife Nelida.

TABLE OF CONTENTS

Page

CHAPTER

LIST OF FIGURES

FIGURE Page

FIGURE Page

LIST OF TABLES

CHAPTER I

INTRODUCTION

Desalination recovers potable water from seawater, brackish water, or treated wastewater. Among water treatment technologies, distillation is the most effective at reducing the widest range of water contaminants.

Mechanical vapor compression is a modern method that produces potable water at acceptable energy efficiency [1]. Because the most critical cost parameters are the fixed charges (amortization) and energy costs [2], the focus of this proposed method is to reduce the energy requirements by innovating the multi-effect evaporator train and by developing a novel latent heat exchanger. Holtzapple and Noyes [3] conceived the system described in their invention disclosure (see Fig. 1-1). It uses a “combined cycle” engine, which consists of a gas turbine (Brayton cycle) and a steam turbine (Rankine cycle). Waste heat from the gas turbine is used to make steam, which powers the steam turbine. Both the gas turbine and the steam turbine power the compressor.

A multi-effect evaporator train operates at successively lower pressures and temperatures. Steam from a high-pressure evaporator boils water in an adjacent low-pressure evaporator. The compressor pulls vapors from the low-pressure evaporator, compresses them, and returns them to the high-pressure evaporator.

The salt water must be degassed prior to being fed to the evaporator.

This dissertation follows the style and format of Desalination.

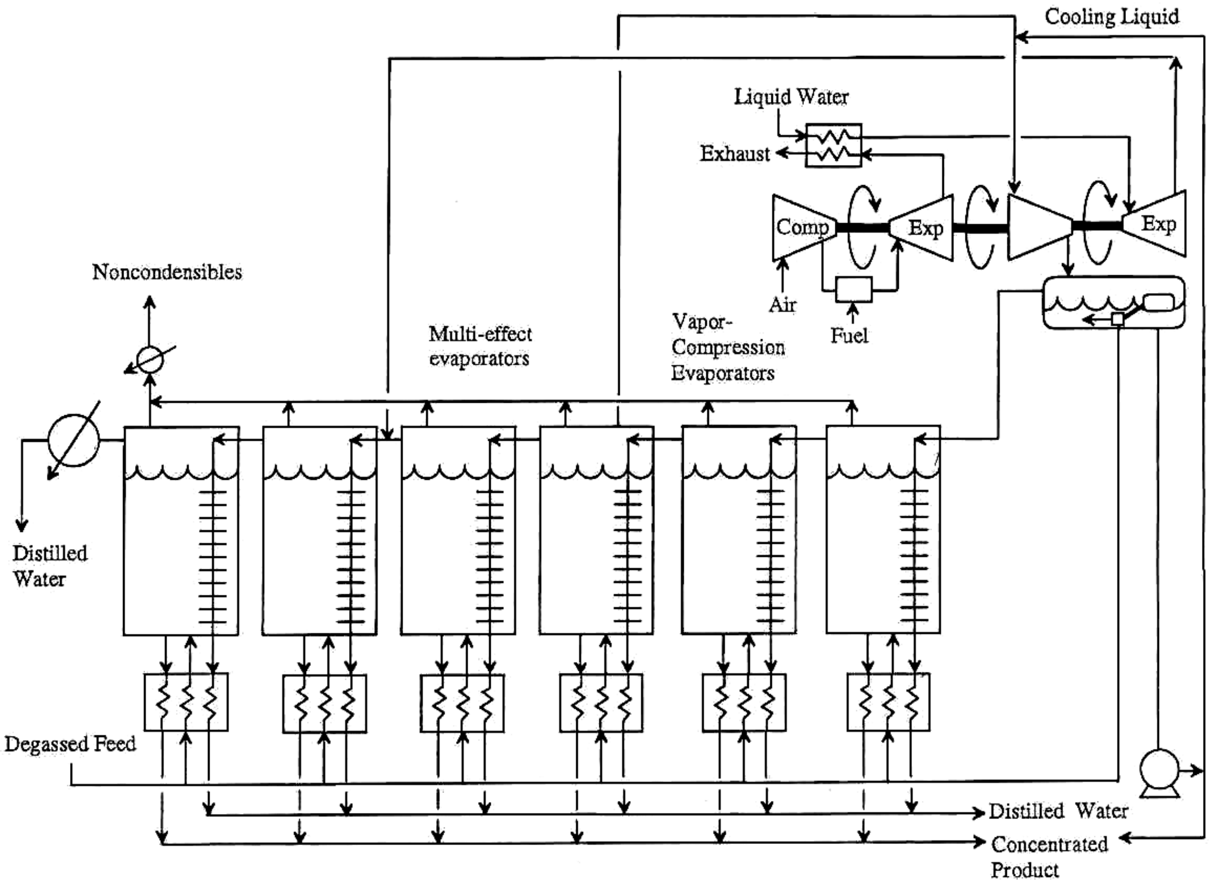


Figure 1-1. Advanced vapor-compression desalination system [3].

Low-pressure steam from the steam turbine exhaust may be added to the evaporator train where the pressures of the exhaust steam and evaporators most closely match.

Project Description

The goals of this project follow:

- Find the optimal operating conditions in the heat exchanger. To achieve this goal, an extensive review of the literature has been performed.
- In the heat exchanger, find a model that describes dropwise condensation, the most efficient heat transfer mechanism. To achieve this goal, the selected mathematical model has been supported with experimental data.
- Select the best material to construct the heat exchanger. It required an extensive review of the available materials and their ability to produce dropwise condensation on the steam side, and to avoid corrosion on both the steam and the boiling seawater side.
- Determine the condensing and boiling heat transfer coefficients.
- Determine the cost per unit area of heat exchanger by contacting manufacturers of the selected material to request both technical information and price quotes.
- Estimate the capital and operating costs of the whole seawater desalination system to determine the product selling cost.

Objectives

The specific objectives for this project follow:

1. Develop a model for the proposed sheet-shell heat exchanger for the system. This is a core objective because once the overall heat transfer coefficient is obtained, the area of the heat exchanger can be known and consequently its cost.
2. Find the recommended operating conditions of the system including compressor, gas turbine, and steam turbine.
3. Develop a model to determine the capital cost and operating cost of saturated steam compression.
4. Find the cost to produce a unit ($\$/m^3$) of potable water

Previous Studies and Results

There are five basic techniques that remove salt and other dissolved solids from seawater [4]: distillation, reverse osmosis (RO), electrodialysis (ED), ion exchange (IX), and freeze desalination.

Distillation. Feed water is heated and then evaporated to separate out dissolved minerals. The most common distillation methods include multi-stage flash (MSF), multiple-effect distillation MED, and vapor compression (VC). In MSF, liquid water is heated at a high pressure and then the pressure is lowered so the water “flashes” into steam. This process constitutes one stage of a number of stages in series, each at a lower pressure. In MED, the feed water passes through a number of evaporators in series. Vapor from the highest pressure evaporator evaporates water from the adjacent evaporator, which operates at a lower pressure. Vapor compression (VC) involves evaporating the feed water, compressing the resulting vapor, and then using the pressurized vapor as a heat source to evaporate additional feed water. Some distillation plants are a hybrid of more than one desalination technology. The waste product from these processes is a solution with a high salt concentration. The heat exchangers typically last 30 – 40 years and are fairly robust.

Reverse osmosis. Feed water is pumped at high pressure through semi-permeable membranes, separating salts from the water. The feed water is pretreated to remove particles that would clog the membranes. The quality of the water product depends on the pressure, the salt concentration in the feed water, and the salt permeation constant of the membranes. Product quality can be improved by adding a second pass of membranes, whereby product water from the first pass is fed to the second pass. RO membranes must be replaced every 5 – 7 years which is expensive. Also, the water must be very clean to prevent fouling.

Electrodialysis. The major energy requirement is the direct current used to separate the ionic substances in the membrane stack. The salts dissolved in water are ionic. These ions are attracted to electrodes with an opposite electric charge. Membranes can be constructed to permit selective passage of either anions or cations. The dissolved ionic constituents in a saline solution such as sodium (+), calcium(++), chloride (-), and carbonate(--) are dispersed in water, effectively neutralizing their individual charges. When electrodes connected to an outside source of direct current are placed in a container of saline water, electrical current is carried through the solution, with the ions tending to migrate to the electrode with the opposite charge. Typical freshwater recovery rates of ED range from 80 to 90% of the feed water volume.

Ion exchange. Undesirable ions in the feed water are exchanged for desirable ions as the water passes through granular ion exchange resins. The higher the concentration of dissolved solids in the feed water, the more often the resins need to be replaced or regenerated.

Freeze desalination. When salt water freezes, ice crystals form pure water leaving the dissolved salt and other minerals in pockets of high-salinity brine. Freeze desalination concentrates a variety of waste streams to higher concentrations with less energy than any distillation process. Freezing involves pre-cooling the feed water, crystallizing ice into slush, separating ice from the brine, washing the ice, and melting the ice.

CHAPTER II

SHEET-SHELL HEAT EXCHANGER

The sheet-shell heat exchanger is a key component of the desalination system. It consists of a novel assembly, which combines features of a conventional plate-and-frame and a shell-and-tube heat exchanger. It can be employed to exchange either sensible heat or latent heat. As a latent heat exchanger, it can be part of a multi-effect evaporator train that operates at successively lower pressures and temperatures. Steam from a high-pressure evaporator boils water in an adjacent low-pressure evaporator. Alternatively, it can be part of a vapor-compression evaporator.

Holtzapple and Noyes [3] conceived the sheet-shell heat exchanger depicted in Figures 2-1 and 2-2. In the high-pressure chamber (top view of Figure 2-2) the baffles cause the flow area to progressively reduce, which allows a relatively constant velocity through the heat exchanger. The flow finally pushes any noncondensible gases to the exit. The number of channels per pass is selected to achieve a low pressure drop. In the low-pressure chamber (side view of Figure 2-2), liquid enters from the bottom and boils due to heat transfer through the wall. The produced vapors exit from the top.

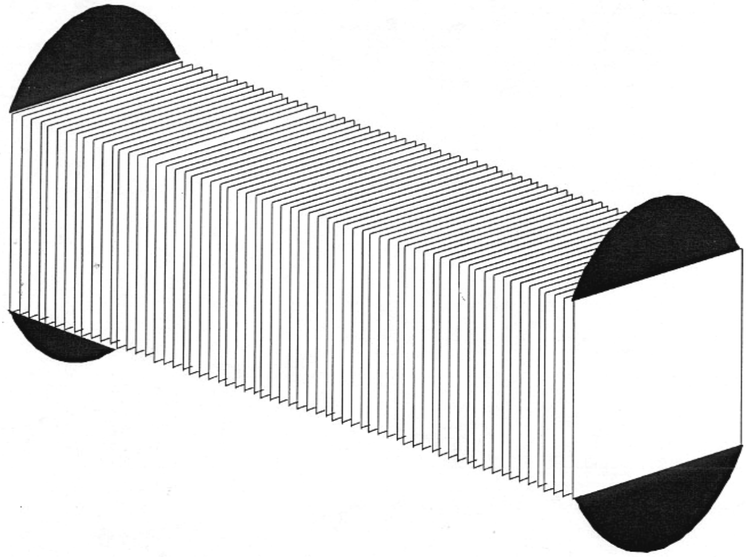


Figure 2-1. Perspective view of heat exchanger plates with top and bottom baffles for the boiling side [3].

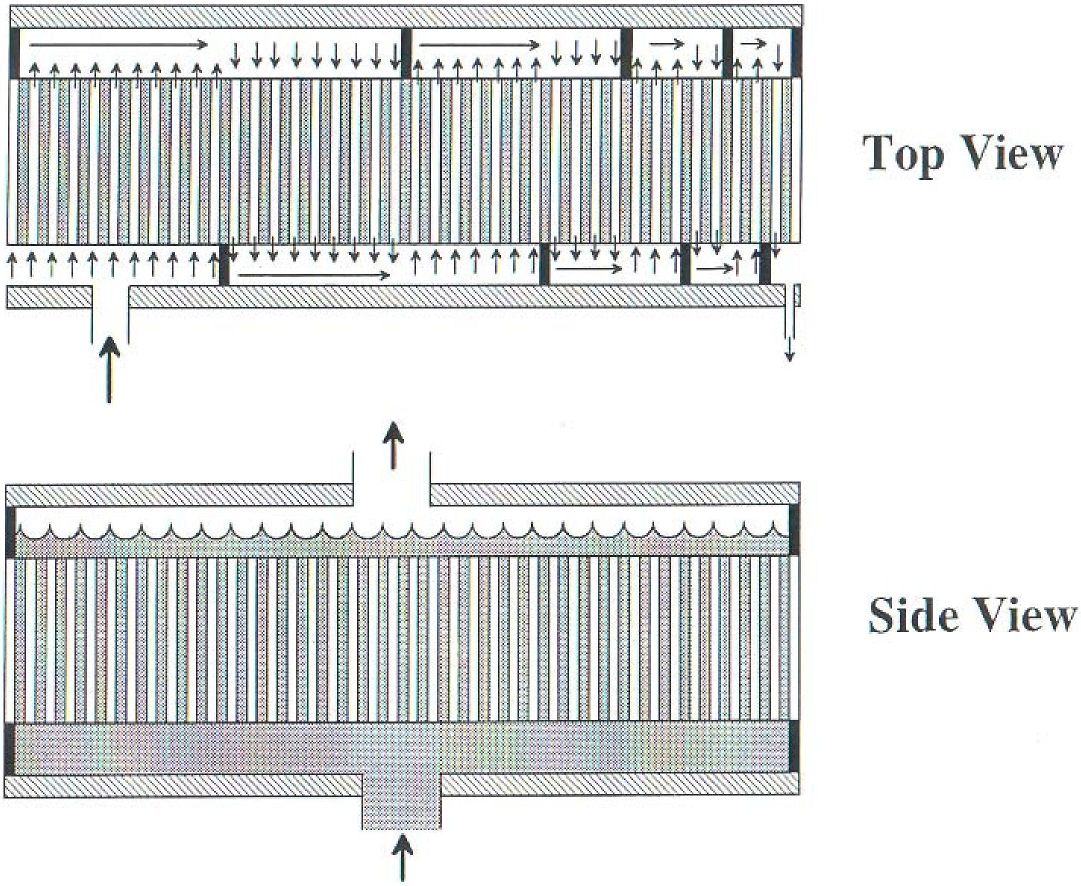


Figure 2-2. Top view and side view for latent heat exchanger. Note the baffle pattern in the high-pressure side shown in the top view [3].

For the high-pressure side of the sheet-shell heat exchanger, the goal is to produce dropwise condensation. This can be promoted using titanium cladding over carbon steel (see Figure 2-3), a gold plated surface [5], or a titanium plate [6]. Also, a titanium surface reduces fouling and avoids corrosion by seawater [6].

Figure 2-4 shows an example of the temperature drops through a titanium-clad sheet. The total ΔT is the sum of ΔT_s (steam side), ΔT_m (metal), and ΔT_b (boiling side).

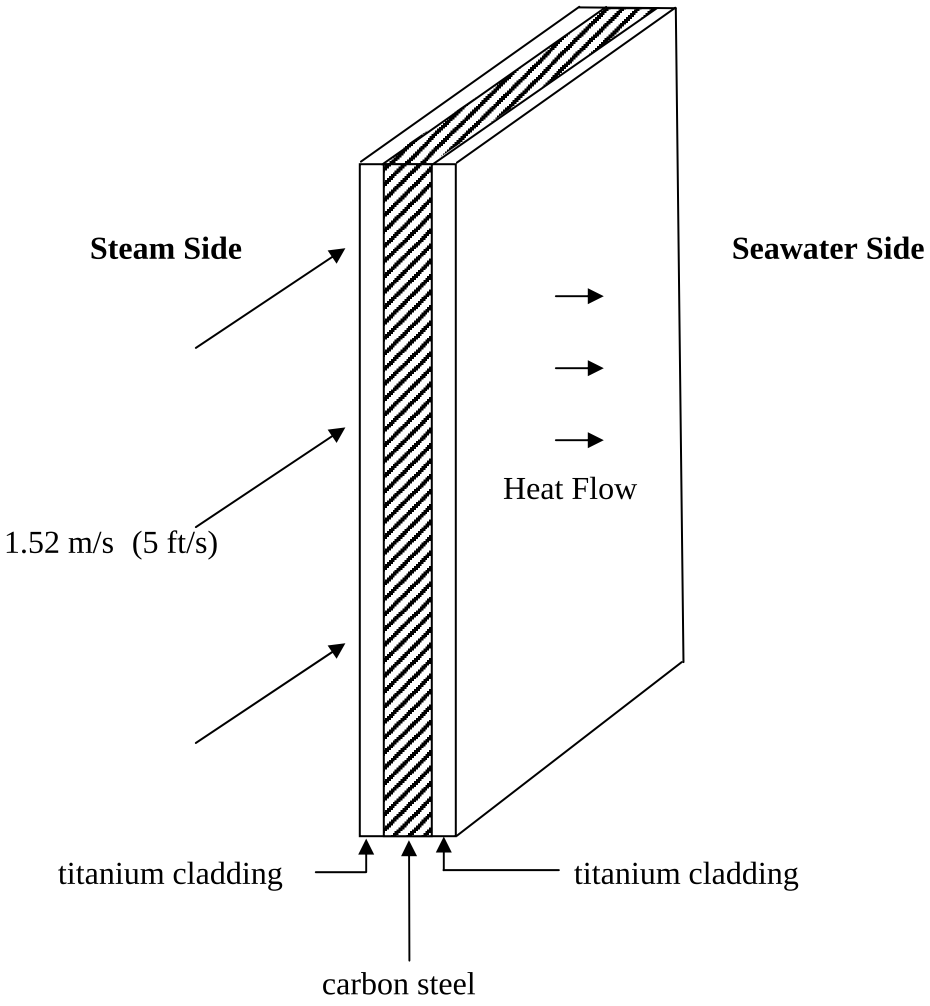
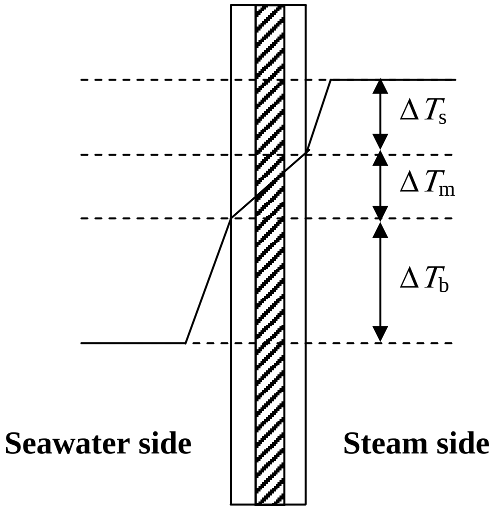


Figure 2-3. Proposed titanium-clad plate for sheet-shell heat exchanger.



$$\Delta T = \Delta T_b + \Delta T_m + \Delta T_s$$

Figure 2-4. Temperature differences at different locations near the heat exchanger sheet.

Holtzapple and Noyes [3] targeted $\Delta T = 5.56\,^{\circ}\mathrm{C}$ ($10^{\circ}\,\mathrm{F}$) as the temperature difference in each heat exchanger of a vapor compression evaporator train. An extensive review on the subject [4-14] indicates that this is a reasonable temperature difference. Further, the literature [15] recommends a velocity of approximately 1.52 m/s (5 ft/s) on the high-pressure steam side.

Experiments performed by Dollof, Metzger, and Roblee [16] show the effect of steam pressure (and temperature) on the steam-side heat transfer coefficient. O'Bara, Killian, and Roblee [15] show that at $\Delta T_s = 2.3\,^{\circ}\mathrm{C}$ ($4^{\circ}\,\mathrm{F}$), large heat transfer coefficients are obtained for a vapor velocity of about 1.52 m/s (5 ft/s).

Based on the information in Figures 2-5 and 2-6, we will employ the following preliminary conditions:

$\Delta T = 5.56°\text{C}\ (10°\text{F})$

$\Delta T_s = 2.3\,°\text{C}\ (4\,°\text{F})$

$P_{steam} = 827.4\,\text{kPa}\ (120\,\text{psig})$

$T_{steam} = 177\,°\text{C}\ (350°\text{F})$

$V_{steam} = 1.52\,\text{m/s}\ (5\,\text{ft/s})$

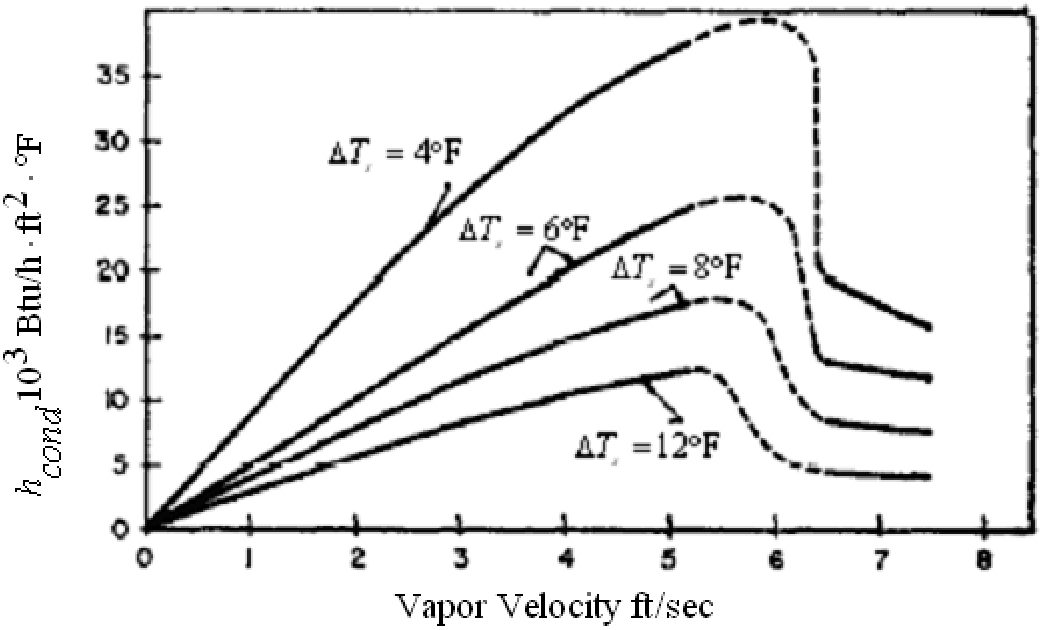


Figure 2-5. Influence of vapor velocity and temperature difference on steam-side heat transfer coefficients [15].

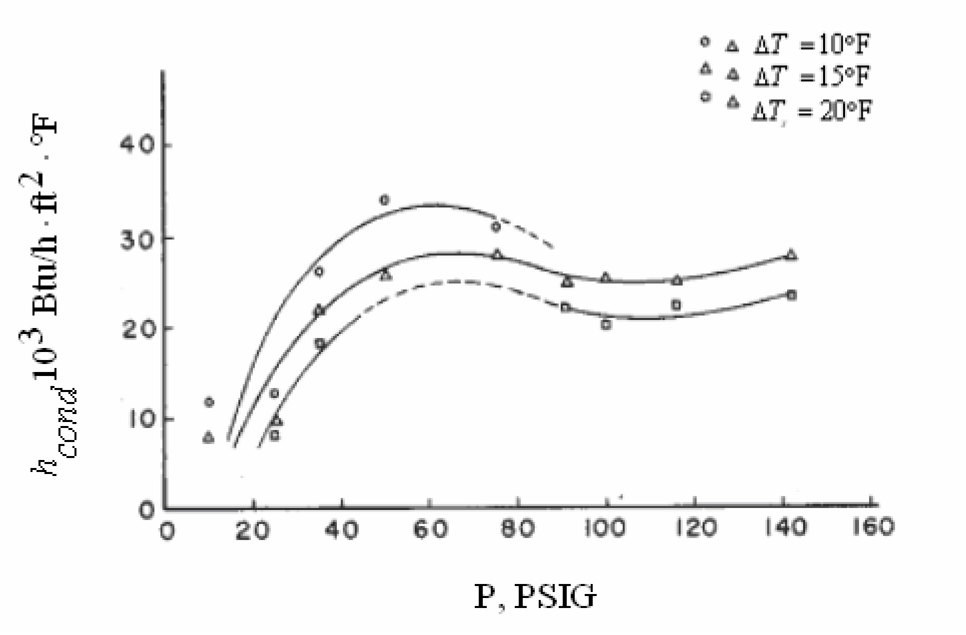


Figure 2-6. Variation of steam-side heat transfer coefficient with pressure for dropwise condensation corresponding to different ΔT_s values [16].

Steam Condensation

O'Bara, Killian, and Roblee [15] state that steam-side heat transfer is a significant part of the total resistance; thus, increasing the steam-side coefficient can measurably improve the overall system performance. Also, they found dropwise steam condensation at $\Delta T_s = 2.3\,^\circ\mathrm{C}\ (4\,^\circ\mathrm{F})$ has a heat transfer coefficient h_{cond} of 198,744 $\mathrm{W/(m^2 \cdot K)}$ (35,000 $\mathrm{Btu/(h \cdot ft^2 \cdot {}^\circ F)}$) at a steam velocity value of about 1.52 m/s (5 ft/s) (see Figure 2-5). Interestingly, they found that as the vapor velocity increased above 1.82 m/s (6 ft/s), h_{cond} abruptly dropped reflecting the transition from dropwise to mixed condensation, resulting from the greater vapor-liquid interfacial shear stress developed at the higher vapor velocities (see Figure 2-5).

It should be remarked that if the total ΔT between the bulk steam and the bulk seawater is about $\Delta T = 5.56^\circ\mathrm{C}\ (10^\circ\mathrm{F})$, and the steam side has a very high heat transfer coefficient h_{cond}, then ΔT_s will be only about 2.3 °C (4 °F). Figure 2-5 shows that the steam-side heat transfer coefficient improves at lower ΔT_s. At low ΔT_s, the heat transfer rate is lower, thus giving time for drops to coalesce and shed. The drops act as an insulator; their removal allows high heat transfer coefficients at the bare metal surface. Figure 6 also shows that regardless of ΔT_s the maximum vapor velocity at which dropwise condensation exists is about 1.52 m/s (5 ft/s). Photographs taken by Dolloff, Metzger, and Roblee [16] revealed that dropwise condensation was maintained for pressures up to $P_{steam} = 827.4\ \mathrm{kPa}\ (120\ \mathrm{psig})$, $T_{steam} = 177\,^\circ\mathrm{C}\ (350^\circ\mathrm{F})$.

As shown in Figure 2-6, Dolloff, Metzger, and Roblee [16] found experimentally that at $\Delta T_s = 5.56°\text{C}\,(10°\text{F})$, the heat transfer coefficient for dropwise condensation reached 193,066 $\text{W}/(\text{m}^2 \cdot \text{K})$ (34,000 $\text{Btu}/(\text{h} \cdot \text{ft}^2 \cdot {}°\text{F})$) at 345 kPa (50 psig). As the steam pressure increases, the heat transfer coefficient increases up to a critical pressure where the coefficient decreases slightly, and then gradually increases as the pressure increases.

Ruckenstein and Metiu [17] used a model to estimate the heat transfer coefficient for dropwise condensation on a solid surface. Their model considers the heat transferred through both the droplet and the film, which is dynamically breaking up on the solid surface.

$$h_{cond} = \frac{\left(\dfrac{S_f}{S}\right)}{\left(\dfrac{1}{h_{ev}} + \dfrac{\delta_c}{2k}\right)} \tag{2.1}$$

where:

k = liquid film thermal conductivity $= 0.3934\ \text{Btu}/(\text{h} \cdot \text{ft} \cdot {}°\text{F})$

$\frac{S_f}{S}$ = fraction of droplet to total area $= 0.25$ (An estimate from References 15 and 16, which used values of $\theta = 72° \pm 8°$ [14], the angle between the droplet and the solid surface shown in Figure 2-7.)

$\delta_c = 0.27\,\mu\text{m} = 8.86 \times 10^{-7}$ ft. [16]

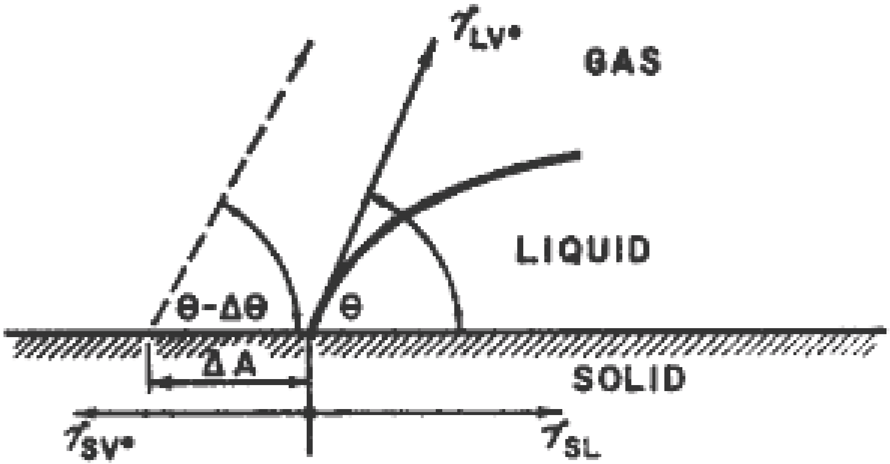


Figure 2-7. Surface tensions and contact angle θ for a drop that is forming on the solid surface of the plate [16].

Therefore,

$$\frac{\delta_c}{2k} = \frac{8.86\times10^{-7}\,\mathrm{ft}}{2\times0.3934\,\dfrac{\mathrm{Btu}}{\mathrm{h}\cdot\mathrm{ft}\cdot{}^\circ\mathrm{F}}} = 1.13\times10^{-6}\,\frac{\mathrm{ft}^2\cdot\mathrm{h}\cdot{}^\circ\mathrm{F}}{\mathrm{Btu}}$$

The heat transfer coefficient through the liquid-vapor interface between the droplet and the steam follows [16, 12, and 18]:

$$h_{ev} = 778\sigma_1\left(\frac{g_c M}{2\pi R T_{ave}}\right)^{\frac{1}{2}}\frac{h_{fg}^2}{T_s v_v} \qquad (2.2)$$

Nabavian and Bromley [19], wrote this equation in engineering units where

$$g_c \;=\; 4.17\times10^8\,\frac{\mathrm{lb_m}\cdot\mathrm{ft}}{\mathrm{lb_f}\cdot\mathrm{h}^2}$$

M = molecular weight $= 18.01 \dfrac{\mathrm{lb_m}}{\mathrm{lbmol}}$

R = gas constant $= 1545.35 \dfrac{\mathrm{ft \cdot lb_f}}{\mathrm{lbmol \cdot {}^\circ R}}$

$T_{ave} = \dfrac{1}{2}\left(T_s + T_v\right) = 794.67\ {}^\circ\mathrm{R}$

T_s = temperature = 171.11 °C = 799.67 °R

h_{fg} = latent heat of vaporization = 878 Btu/lb$_\mathrm{m}$

v_v = Specific volume of vapor under present working conditions = 3.8863 ft^3/lb$_\mathrm{m}$

$$\sigma_1 = \frac{2\sigma}{2-\sigma}$$

where

σ = mass transfer accommodation coefficient [20] = 0.05 (Ruckestein and Metiu [17])

Therefore,

$$\sigma_1 = \frac{2(0.05)}{2-0.05} = 0.051$$

Substituting into Equation 2.2:

$$h_{ev} = 778 \frac{\mathrm{lb_f \cdot ft}}{\mathrm{Btu}} \times 0.051 \times \left(\frac{\left(4.17\times10^8 \dfrac{\mathrm{lb_m \cdot ft}}{\mathrm{lb_f \cdot h^2}}\right)\left(18.01 \dfrac{\mathrm{lb_m}}{\mathrm{lbmol}}\right)}{2\pi\left(1545.35 \dfrac{\mathrm{ft \cdot lb_f}}{\mathrm{lbmol \cdot {}^\circ R}}\right)\left(794.67{}^\circ\mathrm{R}\right)} \right)^{\frac{1}{2}} \times \left(\frac{\left(878.49 \dfrac{\mathrm{Btu}}{\mathrm{lb_m}}\right)^2}{\left(799.67{}^\circ\mathrm{R}\right)\left(3.88 \dfrac{\mathrm{ft^3}}{\mathrm{lb_m}}\right)} \right)$$

$$= 307{,}488 \frac{\mathrm{Btu}}{\mathrm{h \cdot ft^2 \cdot {}^\circ F}}$$

Substituting into Equation 2.1:

$$h_{cond} = \frac{0.25}{\left(\dfrac{1}{307{,}488\,\dfrac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}}\right) + 1.13\times10^{-6}\,\dfrac{\text{ft}^2\cdot\text{h}\cdot{}^\circ\text{F}}{\text{Btu}}} = 57{,}049\,\frac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}$$

This calculated value is similar to the experimental measurements of $198{,}744\ \text{W}/\left(\text{m}^2\cdot\text{K}\right)\ \left(35{,}000\,\text{Btu}/\left(\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}\right)\right)$[15 and $193{,}065\,\text{W}/\left(\text{m}^2\cdot\text{K}\right)$ $\left(34{,}000\,\text{Btu}/\left(\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}\right)\right)$ [16].

Measurements of atmospheric-pressure steam made by Le Fevre and Rose [20–22], and by Tanner, Potter, Pope, and West [23, 24] were correlated by Rose [14] (see Figure 2-8). It shows that at $\Delta T_s = 4\,{}^\circ\text{C}\,(7.2{}^\circ\text{F})$, there is a measured heat flux Q'' of 1.2 MW/m^2 $\left(380{,}400\,\text{Btu}/\left(\text{h}\cdot\text{ft}^2\right)\right)$. The corresponding steam-side heat transfer coefficient is $299{,}820\ \ \text{W}/\left(\text{m}^2\cdot\text{K}\right)$ $\left(52{,}800\,\text{Btu}/\left(\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}\right)\right)$, which is nearly identical to the value calculated by Equation (2.1).

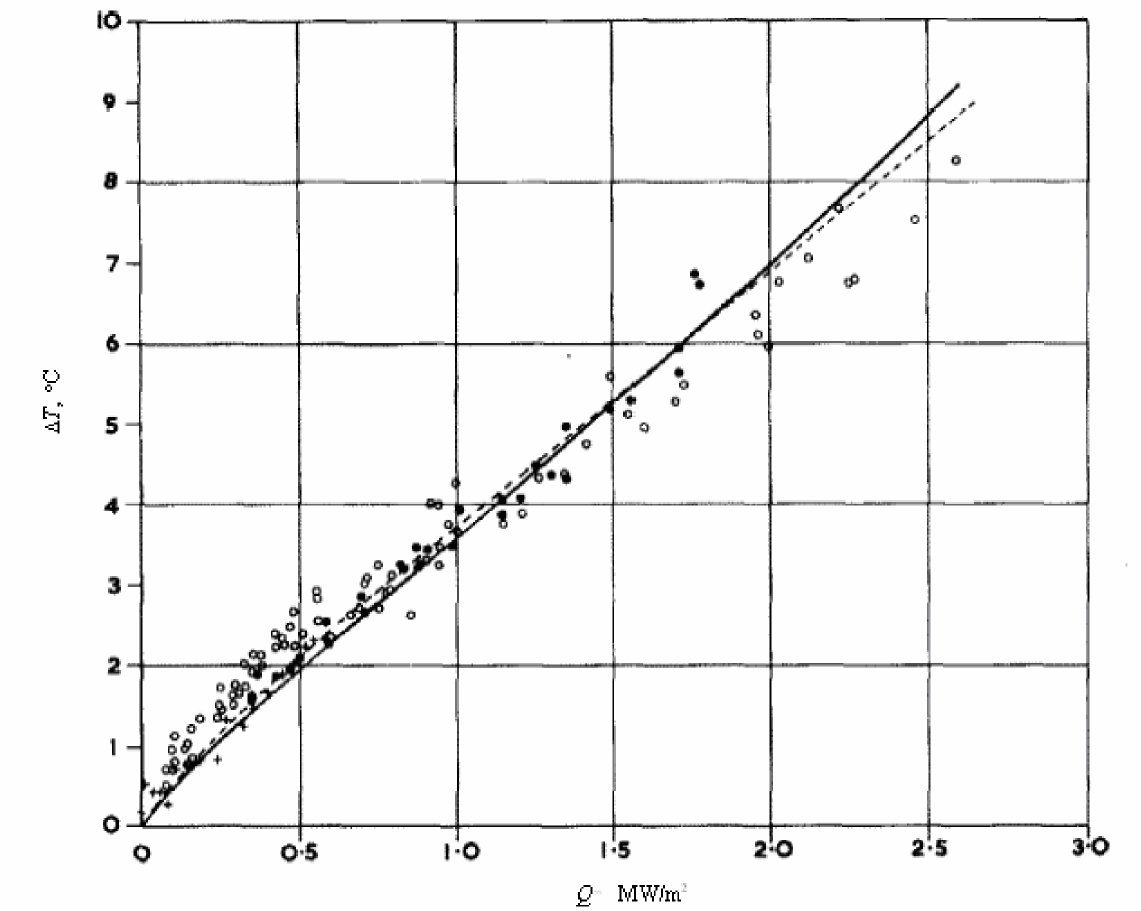


Figure 2-8. Steam-side temperature difference for condensing steam. Comparison between experiment [20–24] and theory [2]. (Note: ΔT on this figure corresponds to ΔT_s in our nomenclature and Q'' is the heat flux.)

Gebhart [9] shows that for atmospheric-pressure vapor, dropwise condensation has a heat flow that may be 10 to 20 times greater than filmwise condensation (see Figure 2-9). The use of different dropwise condensation promoters [19] and the presence of noncondensible gases may cause different measurements.

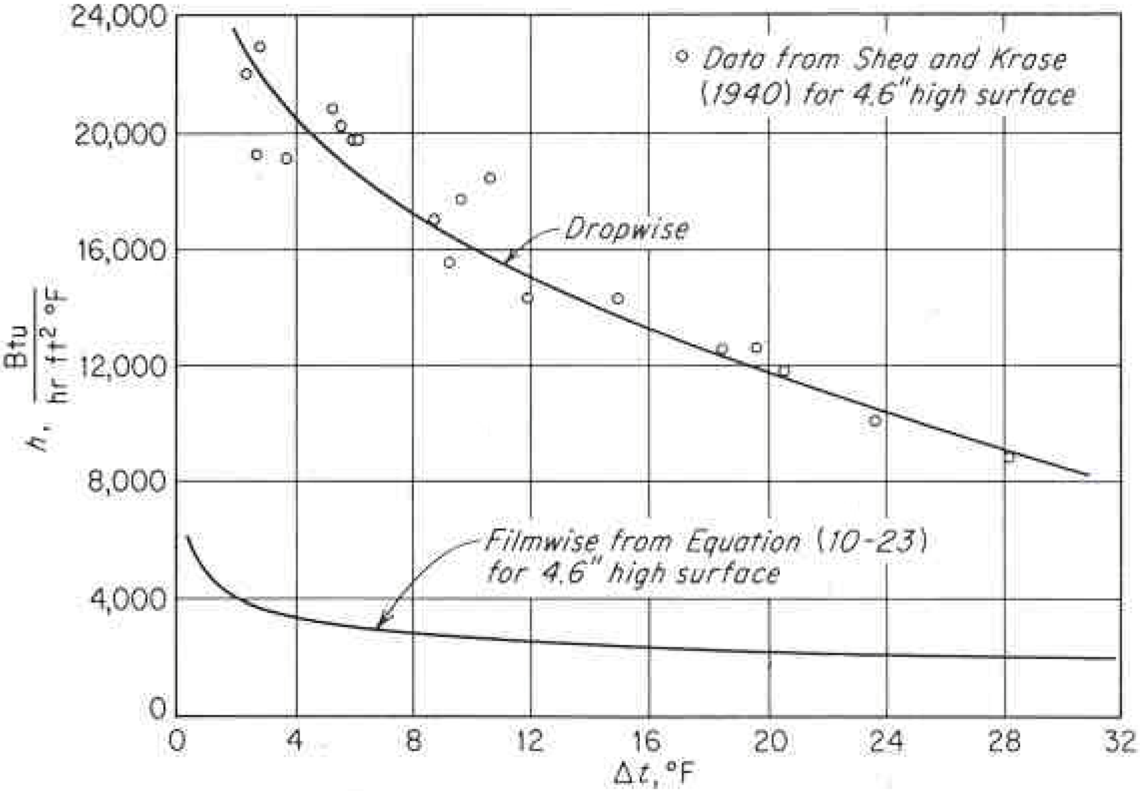


Figure 2-9. Comparison of the overall heat transfer coefficients h. The coeffiecients were obtained with filmwise condensation and dropwise condensation of atmospheric-pressure steam corresponding to different overall ΔT_s values [9].

Liquid-Side Boiling

For saturated pool boiling, ASHRAE Fundamentals [25] gives:

$$h_{boiling} = 55P_{\mathrm{r}}^{0.12\text{-}0.4343\ln R_{\mathrm{p}}}\left(-0.4343\ln P_{\mathrm{r}}\right)M^{-0.5}\left(\frac{q}{A}\right)^{0.67} \tag{2.3}$$

where the correlation requires the following units:

$\left(\frac{q}{A}\right) = \text{heat flux}\left(\frac{\mathrm{W}}{\mathrm{m}^2}\right)$

$h = \text{liquid-side heat transfer coefficient}\left(\frac{\mathrm{W}}{\mathrm{m}^2\cdot\mathrm{K}}\right)$

$R_p = \text{surface roughness } (\mu\mathrm{m})$

$P_r = \text{reduced pressure} = \frac{P}{P_{critical}}$

$P_{critical} = 22{,}063.4\ \mathrm{kPa}\ (3{,}200\ \mathrm{psia})$.

$P_r = \frac{(109.07\ \mathrm{psig} + 14.7\ \mathrm{psi})}{3{,}200\ \mathrm{psia}} = 0.038$

$M = \text{molecular weight}\left(\frac{\mathrm{g}}{\mathrm{gmol}}\right)$

The minimum heat exchanger area will result from a design that operates near the peak heat flux [9]. Boiling heat transfer is especially effective at high heat fluxes, so it cools nuclear reactors and rocket engines with heat transfer rates of the order of 9.4×10^3 to 9.4×10^4 kW/m^2 (3×10^6 to 3×10^7 Btu/h$\cdot$ft^2) [26]. Therefore, high heat transfer coefficients will be obtained from dropwise condensation and pool boiling.

Equation 2.3 requires the heat flux which is

$$\frac{q}{A} = h_{comp}\left(\Delta T_m + \Delta T_s\right) \tag{2.4}$$

For a titanium sheet, h_{comp} is

$$h_{comp} = \frac{1}{(\text{steam - side resistance}) + (\text{wall resistance})}$$

$$= \frac{1}{\left(\frac{1}{h_{cond}}\right) + \left(\frac{1}{\frac{k_{tit}}{x_{tit}}}\right)}$$

The properties for a 0.003-in-thick titanium sheet follow:

$$x_{tit} = 0.003\,\text{in} = 2.5\times10^{-4}\ \text{ft}$$

$$k_{tit} = 11.85\frac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}$$

Thus, from the condensing-side heat transfer coefficient and the resistance from the metal wall,

$$h_{comp} = \frac{1}{\dfrac{1}{57{,}049\dfrac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}} + \dfrac{1}{\dfrac{11.85\dfrac{\text{Btu}}{\text{h}\cdot\text{ft}\cdot{}^\circ\text{F}}}{2.5\times10^{-4}\ \text{ft}}}} = 25{,}889.9\frac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}$$

Therefore, for $\left(\Delta T_m + \Delta T_s\right) = 6\,{}^\circ\text{F}$,

$$\frac{q}{A} = 25{,}889.9\,\frac{\mathrm{Btu}}{\mathrm{h}\cdot\mathrm{ft}^2\cdot{}^\circ\mathrm{F}}\times 6\,{}^\circ\mathrm{F} = 155{,}340\,\frac{\mathrm{Btu}}{\mathrm{h}\cdot\mathrm{ft}^2}\times\frac{1\dfrac{\mathrm{W}}{\mathrm{m}^2}}{0.3171\dfrac{\mathrm{Btu}}{\mathrm{h}\cdot\mathrm{ft}^2}} = 489{,}875\,\frac{\mathrm{W}}{\mathrm{m}^2}$$

Substituting into Equation 2.3

$$h_{boiling} = 55\times(0.038)^{0.12-0.4343\ln 5}(-0.4343\ln 0.038)\left(18.01\frac{\mathrm{g}}{\mathrm{gmol}}\right)^{-0.5}\left(489{,}875\,\frac{\mathrm{W}}{\mathrm{m}^2}\right)^{0.67}$$

$$= 793{,}242.7\,\frac{\mathrm{W}}{\mathrm{m}^2}\times\frac{0.177612\,\dfrac{\mathrm{Btu}}{\mathrm{h}\cdot\mathrm{ft}^2\cdot{}^\circ\mathrm{F}}}{\dfrac{\mathrm{W}}{\mathrm{m}^2}} = 140{,}889.43\,\frac{\mathrm{Btu}}{\mathrm{h}\cdot\mathrm{ft}^2\cdot{}^\circ\mathrm{F}}$$

Farber and Scorah [10]performed experiments to measure heat transfer coefficients for boiling pressurized water. Figure 2-10 shows that $\Delta T_b = 3.3°\mathrm{C}\ (6\,°\mathrm{F})$ provides good values for the heat transfer coefficient $h_{boiling} = 0.85\,\mathrm{MW/(m^2\cdot K)} \approx 150{,}000\,\dfrac{\mathrm{Btu}}{\mathrm{h}\cdot\mathrm{ft}^2\cdot{}^\circ\mathrm{F}}$ for a Chromel A wire heater working under a pressure of 690 $\mathrm{kP_{gauge}}$ (100 psig). This value is very close to the one obtained with Formula 2.3.

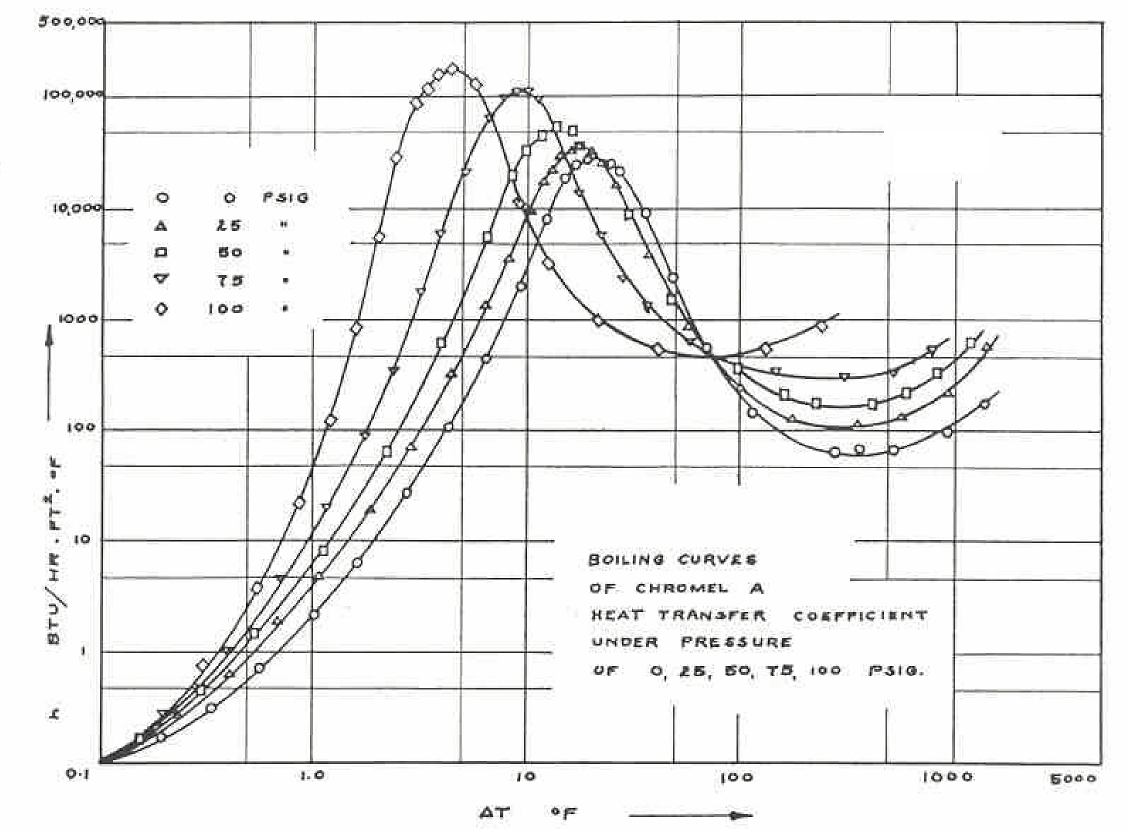


Figure 2-10. Boiling-side heat transfer coefficients. These were obtained under different pressures corresponding to different ΔT_b when the heated surface is a Chromel A wire [10].

Gebhart's Table 1 [9] summarizes data of experiments performed by Farber and Scorah [10], McAdams [11], and Cichelli and Bonilla [12]. At a pressure of 790.8 kPa (114.7 psia) and a liquid temperature of 169.4°C $(337\,^\circ\mathrm{F})$, they show it is possible to have a flux of up to 9.03 $\mathrm{MW/m^2}$ (1,590,000 Btu/$(\mathrm{h \cdot ft^2})$) when the heater surface is a horizontal Chromel C wire (see Table 2.1). Therefore, the calculated heat flux using Equation 2.3 is realistic and conservative.

Table 2-1. Observed peak heat transfer rates for pool boiling on submerged heaters concerning the pressures and temperature difference at which it occurs [7]

Liquid	Pressure (psia)	Liquid temperature $(°\mathrm{F})$	Peak flux $\frac{q}{A}$ $\left(\frac{\mathrm{Btu}}{\mathrm{h}\cdot\mathrm{ft}^2}\right)$	Critical temperature difference* $\Delta T_c, (°\mathrm{F})$	Heater surface and condition	Reference
Water	14.7	212	993,000	99.9	0.04-in-diameter horizontal Chromel C wire	[10]
	64.7	297	906,000	39.5		[10]
	114.7	337	1,590,000	12.8		[10]
	14.7	212	210,000	42	0.004-in-diameter clean platinum wire	[11]
	14.7	212	380,000	42	0.008-0.016-, and 0.024-in-diameter clean platinum wire	[12]

*The spots where bubbles originate on the heated surface become more numerous until a critical temperature is reached at which a maximum heat flux is attained [26]. At that point, the bubbles are so numerous that they interfere with each other. If the temperature is increased beyond the critical value by a few degrees, transition boiling begins.

Overall Heat Transfer Coefficient (U)

The overall heat transfer coefficient for a titanium-clad carbon steel heat exchanger is

$$U = \frac{1}{\frac{1}{h_{cond}} + \frac{1}{h_{boiling}} + \frac{1}{\frac{k_{steel}}{x_{steel}}} + 2\left(\frac{1}{\frac{k_{tit}}{x_{tit}}}\right)} \tag{2.5}$$

In one design option, both surfaces of the carbon steel plate are clad with titanium. The boiling waterside is clad to avoid corrosion and the steam condensing side produces dropwise condensation as well as avoiding fouling. Titanium Metals Corporation claims in its website [13] that on 60-day tests of titanium, hard smooth surfaces promote dropwise condensation and minimize buildup of external fouling films, making cleaning and maintenance easier and producing high rates of condensation.

In Equation 2.5

$h_{steam} = 57,049 \frac{\text{Btu}}{\text{h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}}$, calculated from Equation 2.1

$h_{boili\ln g} = 140,889 \frac{\text{Btu}}{\text{h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}}$, calculated from Equation 2.3

$k_{steel} =$ thermal conductivity for carbon steeel $= 29 \frac{\text{Btu}}{\text{h} \cdot \text{ft} \cdot {}^\circ\text{F}}$ at 340°F

$x_{steel} =$ carbon steel plate thickness $= 0.0375$ in $= 0.0031$ ft

$k_{tit} =$ thermal conductivity for titanium $= 11.8453 \frac{\text{Btu}}{\text{h} \cdot \text{ft} \cdot {}^\circ\text{F}}$ at 340 °F

$x_{tit} = 0.001\ \text{in} = 8.33\times10^{-5}\ \text{ft (first estimate)}$

Substituting into Equation 2.5:

$$U = \frac{1}{\frac{1}{57{,}049}\frac{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}{\text{Btu}} + \frac{1}{140{,}889}\frac{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}{\text{Btu}} + \frac{1}{\frac{29\frac{\text{Btu}}{\text{h}\cdot\text{ft}\cdot{}^\circ\text{F}}}{0.0031\,\text{ft}}} + 2\left(\frac{1}{\frac{11.89453\frac{\text{Btu}}{\text{h}\cdot\text{ft}\cdot{}^\circ\text{F}}}{8.33\times10^{-5}\ \text{ft}}}\right)}$$

$$= 7{,}234\ \frac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}$$

For a design option where the wall is 0.003-in-thick titanium sheet, the overall heat transfer coefficient becomes:

$$U = \frac{1}{\frac{1}{57{,}049}\frac{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}{\text{Btu}} + \frac{1}{140{,}889}\frac{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}{\text{Btu}} + \left(\frac{1}{\frac{11.89453\frac{\text{Btu}}{\text{h}\cdot\text{ft}\cdot{}^\circ\text{F}}}{8.33\times10^{-5}\ \text{ft}}}\right)}$$

$$= 31{,}647\ \frac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}$$

The single sheet of 0.003-in titanium has 4 times the heat transfer of a 0.0375-in carbon steel sheet with 0.001-in titanium cladding. This calculation shows that the metal heat transfer resistance is significant and that efforts should be taken to minimize metal thickness.

These overall heat transfer coefficients are very promising, but subject to the ideal working condition presented here. It would be desirable to obtain experimental data to verify these calculations.

Optimization

The previous sections showed how individual heat transfer coefficients could be calculated for specific conditions. Here, we will determine the overall heat transfer coefficient by adjusting the condensing-side temperature and the overall temperature difference. Once the relationship is known, an optimal value can be selected.

Step 1. Determine the quantity to be optimized (U) and define variables to be optimized. The quantity to be optimized, the overall heat transfer coefficient, is a mathematical function of:

1. steam pressure $P\left(\mathrm{psig}\right)$
2. steam temperature $T\left(°\mathrm{F}\right)$
3. steam velocity $V\left(\mathrm{ft/s}\right)$
4. steam dynamic viscosity $\mu_s\left(\mathrm{lb_m}/\left(\mathrm{h}\cdot\mathrm{ft}\right)\right)$
5. heat transfer surface thermal conductivity $k\left(\mathrm{Btu}/\left(\mathrm{h}\cdot\mathrm{ft}\cdot°\mathrm{F}\right)\right)$
6. liquid temperature $T\left(°\mathrm{F}\right)$
7. liquid pressure $P\left(\mathrm{psig}\right)$
8. liquid velocity $V\left(\mathrm{ft/s}\right)$
9. liquid dynamic viscosity $\mu_l\left(\mathrm{lb_m}/\left(\mathrm{h}\cdot\mathrm{ft}\right)\right)$
10. steam-side heat transfer coefficient $h_{cond}\left(\mathrm{Btu}/\left(\mathrm{h}\cdot\mathrm{ft}^2\cdot°\mathrm{F}\right)\right)$

11. liquid-side heat transfer coefficient $h_{boiling}\left(\mathrm{Btu}/\left(\mathrm{h}\cdot\mathrm{ft}^2\cdot{}^\circ\mathrm{F}\right)\right)$

12. overall heat transfer temperature differential $\Delta T\ \left({}^\circ\mathrm{F}\right)$

13. plate thickness Δx (ft)

14. heat flux $\frac{q}{A}\left(\mathrm{Btu}/\left(\mathrm{h}\cdot\mathrm{ft}^2\right)\right)$

$$U = f\left(P_{steam}, T_{steam}, V_{steam}, \mu_s, k_m, P_{liq}, T_{liq}, V_{liq}, \mu_{liq}, h_{cond}, h_{boiling}, \Delta T, \Delta x, \frac{q}{A}\right) \tag{2.6}$$

Step 2. Identify all stipulated restrictions, requirements, and limitations:

1. Promote dropwise condensation:

   Maximum allowable steam pressure = 120 psig

   Maximum allowable steam velocity = 5 ft/s

   Dropwise-condensation-promoting surface

2. Attain the maximum possible overall heat transfer coefficient:

   saturated condition on both steam and liquid side.

3. Pool boiling conditions.

4. Determine the lowest possible temperature and pressure drop to allow a maximum number of effects in the evaporator train.

Because the optimization process is performed under the severe restrictions described previously, the overall heat transfer coefficient is a function of the temperature drop, which itself is constrained to the parameters solely involved in the heat transfer mechanism

$$U = f_1\left[\Delta T \; f\left(h_{cond}, h_{boiling}, \Delta x, k_m, \frac{q}{A}\right)\right] \tag{2.7}$$

Figure 2-11 (Table B-2, Appendix B) shows the calculated overall heat transfer coefficients corresponding to different condensing temperatures and different heat transfer temperature differentials ΔT between the condensing steam side and the boiling liquid side. The overall heat transfer coefficient U reaches an optimal maximum value 278.3 $\mathrm{kW/\left(m^2 \cdot K\right)}$ $\left(\text{about } 49{,}000\ \mathrm{Btu/\left(h \cdot ft^2 \cdot {}^\circ F\right)}\right)$ at an optimal overall temperature differential $\Delta T = 3.3^\circ\mathrm{C}$ $\left(6\ {}^\circ\mathrm{F}\right)$.

It is worthwhile to mention at this point that although high values for the heat transfer coefficient can be attained in theory, it has been decided that a constant value of $24{,}900\ \mathrm{Btu/\left(h \cdot ft^2 \cdot {}^\circ F\right)}$ will be used in the engineering design.

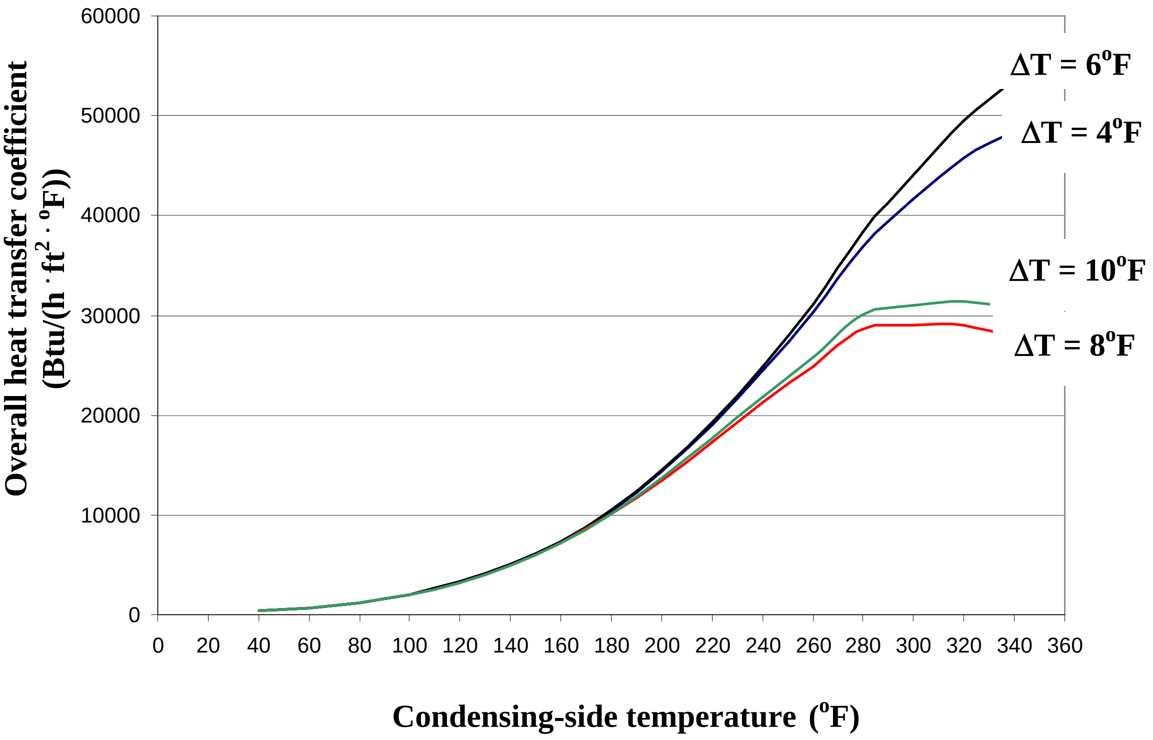


Figure 2-11. Calculated overall heat transfer coefficients. They correspond to specified overall temperature differentials ΔT and different condensing-side temperatures.

Titanium Metals Corporation [27] claims that in power generating plants, where saline, brackish, or polluted waters are used as the cooling medium, titanium thin-wall condenser tubing will last for the life of the condenser (with a 40-year warranty against failure under proper conditions) and will eliminate the need for a corrosion allowance.

Titanium Metals Corporation [13] also performed tests on titanium in a seawater environment. It showed excellent resistance to turbulence and erosion-corrosion, which permits the use of relatively high flow rates of 5.48–6.70 m/s (18–22 ft/s) (silt-laden seawater) or even up to 30.48 m/s (100 ft/s) (clean seawater) without damaging the passive oxide film. Tests in 26.7 °C (80 °F) seawater for 60 days at 7.6 m/s (25 ft/s) have shown titanium's corrosion-erosion resistance to be 80 times better than that of the next-best material, a copper-nickel alloy. Other tests in 29.4°C (85°F) seawater for 60 days at 8.2 m/s (27 ft/s) proved titanium to be almost 100 times better than stainless steel, the next best material.

The overall heat transfer coefficient of titanium in a desalination environment equaled that of 90-10 copper-nickel after a short operating period because titanium did not experience corrosion fouling. In this experiment, seawater moved at 1.52 m/s (5 ft/s) inside ¾ -in, 19-gauge tubes with steam condensing on the outside.

CHAPTER III

PRESSURE DROP IN THE HEAT EXCHANGER

For two-phase flow inside horizontal tubes and channels, the 2001 ASHRAE Fundamentals Handbook [25] presents a model where the pressure gradient is the sum of frictional and momentum terms. Thus,

$$\frac{dP}{dz} = \left(\frac{dP}{dz}\right)_{friction} + \left(\frac{dP}{dz}\right)_{momentum} \tag{3.1}$$

Dukkler, Wicks, and Cleveland [28] give a correlation for the frictional pressure gradient as

$$\left(\frac{dP}{dz}\right)_{friction} = -\frac{2G^2 f_{\circ}\alpha(\lambda)\beta}{D_h \rho_{NS}} \tag{3.2}$$

where,

D_h = hydraulic diameter (ft) $= \dfrac{4A}{P}$

A = cross-sectional area (ft^2)

P = wetted perimeter (ft)

G = mass flux $= \rho_v V$ $\left(\mathrm{lb_m}/\left(\mathrm{ft}^2 \cdot \mathrm{s}\right)\right)$

$f_{\circ}$ = Single-phase friction coefficient evaluated at two-phase Reynolds number

$$= 0.0014 + 0.125\left(\frac{4m_t\beta}{\pi D_i \mu_{NS}}\right)^{-0.32} \tag{3.3}$$

m_t = mass flow rate $= \rho_v VA$ ($\mathrm{lb_m}/\mathrm{s}$)

V = steam velocity (ft/s)

ρ_v = steam density (lb$_m$ / ft^3)

$$\alpha(\lambda) = \frac{1-(\ln\lambda)}{\left[1.281 + 0.478\ln\lambda + 0.444(\ln\lambda)^2 + 0.094(\ln\lambda)^3 + 0.00843(\ln\lambda)^4\right]} \tag{3.4}$$

$$\beta = \left(\frac{\rho_l}{\rho_{NS}}\right)\frac{\lambda^2}{(1-\Psi)} + \left(\frac{\rho_v}{\rho_{NS}}\right)\frac{(1-\lambda)^2}{\Psi} \tag{3.5}$$

$$\rho_{NS} = \rho_l\lambda + \rho_v(1-\lambda) \tag{3.6}$$

$$\mu_{NS} = \mu_l\lambda + \mu_v(1-\lambda) \tag{3.7}$$

$$\lambda = \frac{1}{\left(1 + \frac{x}{(1-x)}\frac{\rho_v}{\rho_l}\right)} \tag{3.8}$$

where,

ρ_v = vapor density $\left(\mathrm{lb_m / ft^3}\right)$

ρ_v = lquid density $\left(\mathrm{lb_m / ft^3}\right)$

μ_l = liquid dynamic viscosity $\left(\mathrm{lb_m /(h \cdot ft)}\right)$

μ_l = vapor dynamic viscosity $\left(\mathrm{lb_m /(h \cdot ft)}\right)$

x = mass fraction of vapor = 0.999

The void fraction Ψ is calculated from:

$$\Psi = \frac{1}{1 + A_l\left[\frac{(1-x)}{x}\right]^{q_l}\left(\frac{\rho_v}{\rho_l}\right)^{r_l}\left(\frac{\mu_l}{\mu_v}\right)^{s_l}} \tag{3.9}$$

where A_l, q_l, r_l, and s_l are constants and are listed for the various correlations in Table 3-1.

Table 3-1. Constants in Equation 3.9 for different void fraction correlations [25]

Model	A_l	q_l	r_l	s_l
Homogeneous (Collier [30])	1.0	1.0	1.0	0
Lockhart-Martinelli	0.28	0.64	0.36	0.07
Baroczy	1.0	0.74	0.65	0.13
Thom	1.0	1.0	0.89	0.18
Zivi	1.0	1.0	0.67	0
Turner-Wallis	1.0	0.72	0.40	0.08

The mass flow rate m_t is calculated from

$$m_t = \rho_v VA$$

where A is the cross sectional area between plates in the heat exchanger (see Fig. 3-1).

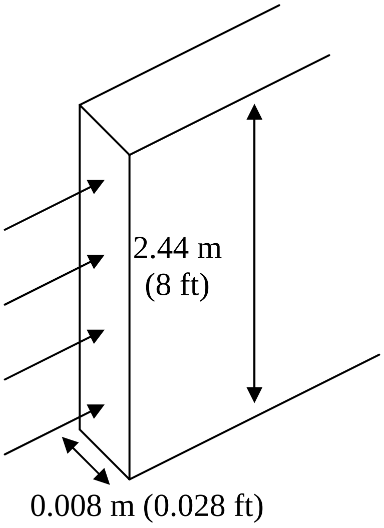


Figure 3-1. Cross sectional area representation for space between plates in the heat exchanger.

The hydraulic diameter D_h is

$$D_h = \frac{4A}{P}$$

where

$A =$ cross-sectional area $\left(\mathrm{ft}^2\right)$

$P =$ wetted perimeter $\left(\mathrm{ft}\right)$

The proposed steam velocity by Dolloff, Metzger, and Roblee [16] is 1.52 m/s (5 ft/s) whereas O'Bara, Killian and Roblee [15] suggested 1.98 m/s (6.5 ft/s).

The pressure drop due to momentum is calculated following reference [25] as

$$\left(\frac{dP}{dz}\right)_{momentum} = -G^2\left(\frac{dx}{dz}\right)\left\{\frac{2x}{\rho_v \Psi} - \frac{2(1-x)}{\rho_l(1-\Psi)} + q_l\left[\frac{\Psi(1-x)}{x(1-\Psi)\rho_l} - \frac{x(1-\Psi)}{\Psi(1-x)\rho_v}\right]\right\} \quad (3.10)$$

The heat exchanger geometry [29] (see Figures 2.1 and 2.2) suggests that there is a pressure drop ΔP due to the sudden channel enlargement when the steam flow exits the channel before it turns $180°$ and bends towards a sudden contraction when it enters to the plate channel.

Colliere [30] recommends the following scheme for $180°$ bends:

$$\frac{\Delta P_{TP}}{\Delta P_f} = 1 + CX + X^2 \quad (3.11)$$

This corresponds to pressure drop with vapor flowing alone, where

$\Delta P_{TP} =$ pressure drop of the two-phase flow

$\Delta P_f =$ pressure drop of the liquid-phase flow

$X =$ Martinelli parameter $= \dfrac{1}{\left(\dfrac{x}{1-x}\right)\left(\dfrac{v_g}{v_f}\right)^{0.5}}$

$v_g =$ vapor specific volume $\left(\mathrm{ft}^3/\mathrm{lb_m}\right)$

$v_f =$ liquid specific volume $\left(\mathrm{ft}^3/\mathrm{lb_m}\right)$

$$C = \text{Chisholm correlation coefficient } = \left[\lambda + \left(\mathrm{C}_2 - \lambda\right)\left(\frac{v_{fg}}{v_g}\right)^{0.5}\right]\left[\left(\frac{v_g}{v_f}\right)0.5 + \left(\frac{v_f}{v_g}\right)^{0.5}\right]$$

Chisholm [31] recommends the following values for C_2 and λ:

$$C_2 = 1 + 20\left(\frac{D}{L}\right)$$

$$\lambda = 1$$

From reference [28],

$$\Delta P_f = K\left(\frac{\rho_f}{g_c}\right)\left(\frac{V^2}{2}\right) \tag{3.12}$$

where

$K = 0.30$ for a flanged welded bend [25]

$V =$ fluid velocity (ft/s)

$\rho_f =$ fluid density $\left(\mathrm{lb_m} / \mathrm{ft}^3\right)$

$g_c = 32.2\ \mathrm{lb_m} \cdot \mathrm{ft}/\left(\mathrm{lb_f} \cdot \mathrm{s}^2\right)$

It should be remembered that this pressure drop corresponds to each bend that the flow encounters. Therefore, this value has to be multiplied times the number of bends required in the heat exchanger. The number of bends corresponds to the number of baffles. Consider for example that there are 10 plates per baffle.

Collier [30] derived the following equation to compute the pressure drop in

a sudden enlargement:

$$P_2 - P_1 = G_1^2 \sigma \left(1-\sigma\right) v_f \left[\frac{\left(1-x\right)^2}{\left(1-\alpha\right)} + \left(\frac{v_g}{v_f} \right) \frac{x^2}{\alpha} \right] \tag{3.13}$$

where

G_1 = steam mass velocity = $\rho_v V$ $\left(\mathrm{lb_m} / \left(\mathrm{ft}^2 \cdot \mathrm{s}\right)\right)$

α = void fraction = Ψ = 0.999

v_f = specific volume of saturated liquid $\left(\mathrm{ft}^3 / \mathrm{lb_m}\right)$

x = steam (vapor) quality (%)

V = velocity (ft/s)

σ = enlargement ratio = A_1 / A_2

where A_2 is estimated to be 10 times larger than A_1

Collier [30] proposes the following equation for a contraction:

$$P_2 - P_1 = \frac{G_2^2 v_f}{2} \left[\left(\frac{1}{C_c} - 1 \right)^2 + \left(1 - \frac{1}{\sigma^2} \right) \right] \left[1 + \left(\frac{v_{fg}}{v_f} \right) x \right] \tag{3.14}$$

where

C_c = coefficient of contraction is a function of σ (see Table 3-2)

G_2 = steam mass flux $\left(\mathrm{lb_m} / \left(\mathrm{ft}^2 \cdot \mathrm{s}\right)\right)$

v_f = specific volume of saturated liquid $\left(\mathrm{ft}^3 / \mathrm{lb_m}\right)$

σ = contraction ratio = $\dfrac{A_2}{A_1}$

Therefore, the total pressure drop per channel is the sum

$$\Delta P_{TOTAL} = \sum \left[(\text{channel}) + (180^\circ \text{ bend}) + (\text{enlargement}) + (\text{contraction}) \right]$$

As can be appreciated in the sample calculation in Appendix C, the pressure drop in the sheet-shell heat exchanger is not a major issue. The operating pressure in Stage 1 is 756.4 kPa (109.7 psia) and the calculated total pressure drop in the heat exchanger is 0.362 kPa (0.0525 psid), which is negligible. However, it should be noted that the liquid head as well as the boiling point elevation reduces the available temperature drop in each effect of the multiple-effect evaporator.

Table 3-2 shows correlated coefficient of contraction values corresponding to different contraction ratios as given by Collier [30].

Table 3-2. Correlated coefficient of contraction [30]

$\frac{1}{\sigma}$	0	0.2	0.4	0.6	0.8	1.0
C_c	0.586	0.598	0.625	0.686	0.790	1.0
$\left(\frac{1}{C_c}-1\right)^2$	0.5	0.45	0.36	0.21	0.07	0

CHAPTER IV

MASS AND ENERGY BALANCE

The steam latent heat of condensation is transferred through the hot surface to vaporize water from seawater. Assuming the heat exchanger is well insulated, the condensing-steam enthalpy change equals the seawater boiling-side enthalpy change. The evaporator mass and energy flow diagram is depicted in Figure 4-1. The following symbols are used:

$m_s =$ rate of steam flow and condensate $\left(\mathrm{lb_m / h}\right)$

$m_f =$ rate of seawater feed flow $\left(\mathrm{lb_m / h}\right)$

$m_v = m_f - m_b =$ rate of vapor flow to the next effect $\left(\mathrm{lb_m / h}\right)$

$m_b =$ rate of exiting brine flow $\left(\mathrm{lb_m / h}\right)$

$T_s =$ condensing temperature of the steam $\left(^\circ \mathrm{F}\right)$

$T_b =$ boiling temperature of the seawater $\left(^\circ \mathrm{F}\right)$

$T_f =$ temperature of the seawater feed $\left(^\circ \mathrm{F}\right)$

The enthalpy balance has the following assumptions [32]:

1. no leakage or entrainment
2. negligible flow of non-condensable gases
3. no heat losses from evaporator
4. small superheat of steam
5. small subcooling of condensate

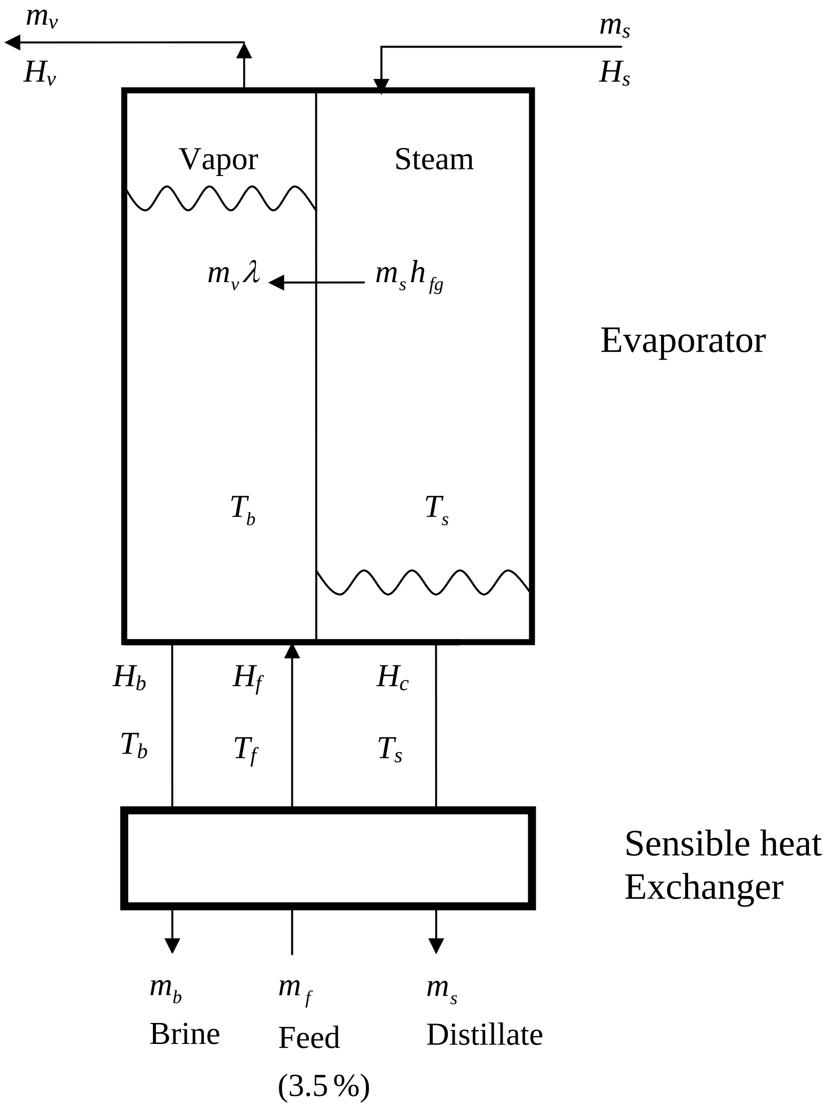


Figure 4-1. Evaporator mass and energy balance diagram.

The steam-side enthalpy balance is

$$q_s = m_s\left(H_s - H_c\right) = m_s h_{fg} \tag{4.1}$$

where

q_s = rate of heat transfer (Btu/h)

H_c = specific enthalpy of condensate (Btu/lb$_\text{m}$)

H_s = specific enthalpy of steam (Btu/lb$_\text{m}$)

h_{fg} = latent heat of evaporation (Btu/lb$_\text{m}$)

The seawater side enthalpy balance is:

$$\begin{aligned} q &= m_v H_v - m_f H_f + m_b H_b \\ &= \left(m_f - m_b\right) H_v - m_f H_f + m_b H_b \end{aligned} \tag{4.2}$$

where

H_v = specific enthalpy of vapor (Btu/lb$_\text{m}$)

H_f = specific enthalpy of feed (Btu/lb$_\text{m}$)

H_b = specific enthalpy of exiting brine (Btu/lb$_\text{m}$)

Therefore, for the heating surface shown in Figure 4-1

$$q_s = q$$

$$m_s h_{fg} = \left(m_f - m_b\right) H_v - m_f H_f + m_b H_b \tag{4.3}$$

Seawater does not have an appreciable heat of dilution [32], which simplifies calculations. For simplicity, liquid brine at T_b can be used as a reference condition for the enthalpy calculations. The enthalpy for the exiting brine is

$$H_b = C_p\left(T_b - T_b\right) = 0$$

The enthalpy H_f can be calculated from the specific heat of seawater

$$H_f = C_{pf}\left(T_b - T_f\right)$$

where

C_{pf} = specific heat of seawater feed (at 3.5% concentration) $\left(\mathrm{Btu}/\left(\mathrm{lb_m}\cdot{}^\circ\mathrm{F}\right)\right)$

Table A-4 gives the specific heat as a function of seawater concentration. Tables A-2 through A-6 [33] provide the seawater data used in the calculations.

At steady-state flow conditions in the evaporator, the seawater concentration has been set to 7%. Under these circumstances, there is an appreciable boiling point elevation. The vapor leaving the evaporator solution is superheated by about $1.5\,^\circ\mathrm{C}$ ($2.7\,^\circ\mathrm{F}$) (see Figure 5-1), which corresponds to the boiling point elevation. Using the boiling temperature as a reference, the specific enthalpy H_v of the leaving vapor equals the latent heat of vaporization plus the sensible superheat. However, the sensible superheat is small, so it is approximately true that H_v is the latent heat of vaporization, which is called λ.[32] (See Appendix D for numerical calculations.) With this simplifying assumption, the steady-state evaporator energy balance becomes:

$$\begin{aligned} m_s h_{fg} &= \left(m_f - m_b\right)\lambda - m_f H_f + m_b H_b \\ &= \left(m_f - m_b\right)\lambda - m_f C_{pf}\left(T_b - T_f\right) + 0 \end{aligned} \tag{4.4}$$

CHAPTER V

FACTORS AFFECTING EVAPORATOR PERFORMANCE

As was explained in the last part of Chapter IV, the evaporator energy balance must include the boiling point elevation. Increasing the seawater concentration elevates its boiling temperature and reduces its vapor pressure. These and other factors affect evaporator performance.

Boiling Point Elevation

Figure 5-1 shows the boiling point elevation as a function of salinity and temperature.

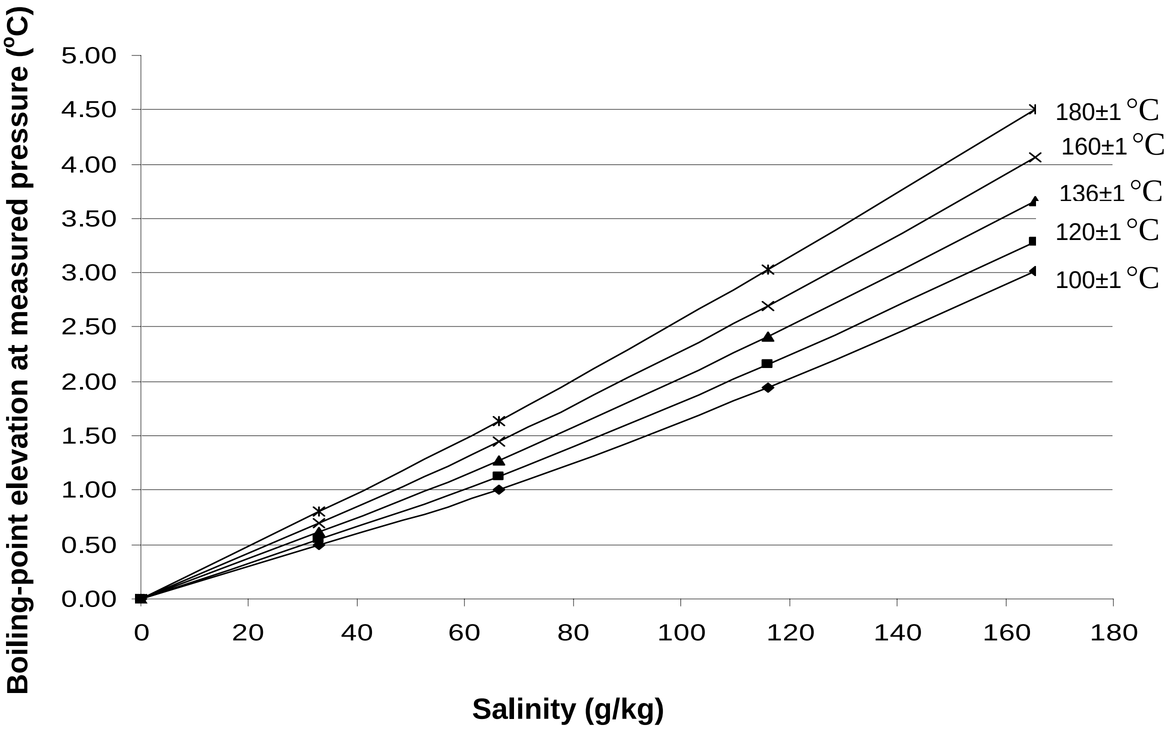


Figure 5-1. Boiling point elevation and salinity measurements at different temperatures. Data from Table A-7.

Seawater Vapor Pressure Calculation

The water vapor pressure of seawater and its concentrates has been measured in a temperature range $100\,°\mathrm{C}$ to $180\,°\mathrm{C}$ by Emerson and Jamieson [34]. The results of their measures are given in Table A-7. There is a close agreement with the analytical method described by The National Engineering Laboratory of England in Reference [35]. The vapor pressure of pure water at a measured temperature p_0 can be obtained from steam tables or it can be calculated from

$$\log_{10} p_0 = a + \frac{b}{z} + \frac{cx}{z}\left(10^{dx^2} - 1\right) + e10^{fy^{1.25}} \tag{5.1}$$

where

$p_0 = \text{pure water vapor pressure}\left(10^5\ \mathrm{N/m^2}\right)$

$x = z^2 - g$

$y = 344.11 - t$

$z = t + 273.16$

$t =$ measured temperature $°\mathrm{C}$

$a = 5.432368$

$b = -2.0051 \times 10^3$

$c = 1.3869 \times 10^{-4}$

$d = 1.1965 \times 10^{-11}$

$e = -4.4000 \times 10^{-3}$

$f = -5.7148 \times 10^{-3}$

$g = 2.9370 \times 10^{5}$

The activity p / p_0 fits an equation of the form

$$\log_{10}\left(p / p_0\right) = hS + jS^{2} \tag{5.2}$$

where

$p =$ vapor pressure of salt water at the same temperature (10^5 N/m^2)

$h = -2.1609 \times 10^{-4}$

$j = -3.5012 \times 10^{-7}$

$S =$ salinity (g salt/kg seawater)

The seawater activity is shown in Figure 5-2 and Appendix A.

Static Head

Because seawater depth in the evaporator is significant – 2.4 m (8 ft) in the present design – the boiling point corresponding to the pressure vapor is the boiling point of the surface layers only [32]. The liquid below the surface is under the pressure vapor plus a head of liquid and therefore has a slightly higher boiling point. This effect is overcome by pumping the seawater upwards so the warmed water at the bottom of the heat exchanger can readily vaporize.

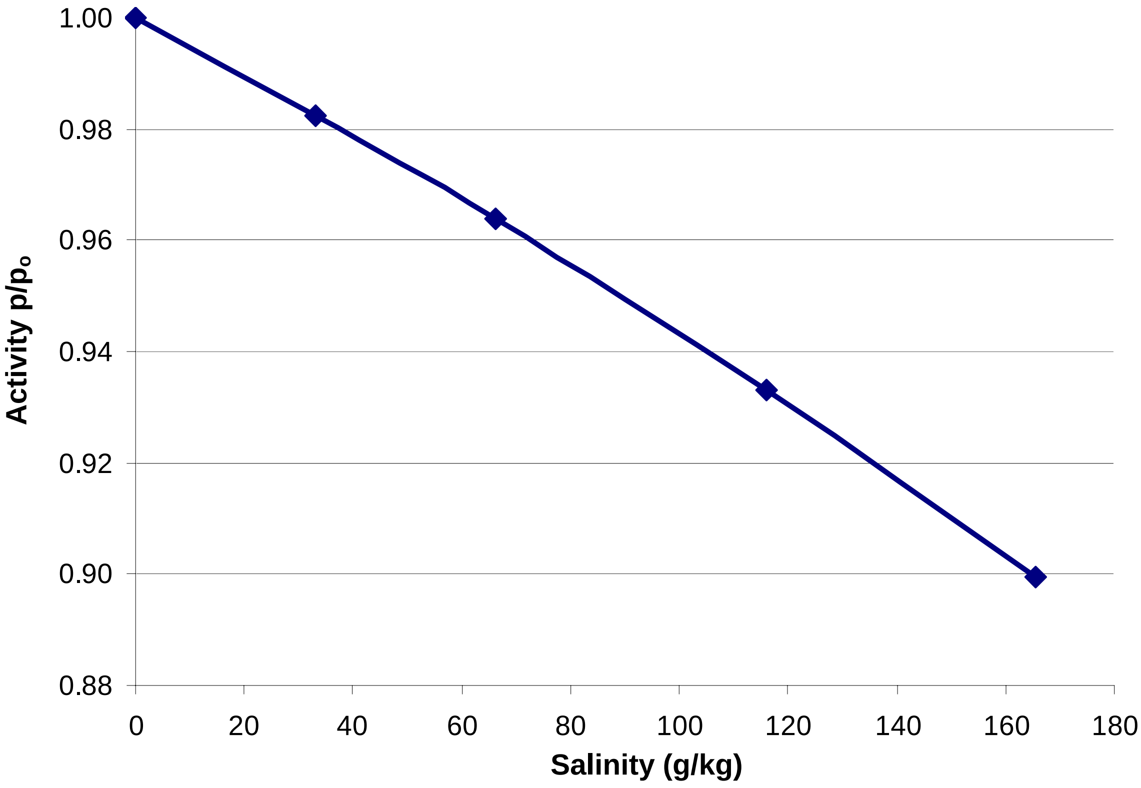


Figure 5-2. Activity of seawater due to salt concentration (g/kg). The line indicates average activity values of measurements at different temperatures for various salinity samples.

Corrosion

Seawater attacks ferrous metals. In addition, seawater deposits scale on heating surfaces. To avoid corrosion, it is suggested to construct the heat exchanger of titanium. It is known that scale forms at temperatures higher than $120\,^{\circ}\mathrm{C}$. If the working conditions are above this limit, small abrasive balls can be suspended in the flow to remove scale from the walls.

CHAPTER VI

POWER SYSTEM

Waste heat from the gas turbine is recovered in a heat recovery boiler (HRB) as shown in Figures 6.1(a) and 6.1(b). The combined-cycle power system consists of a gas turbine, a heat recovery boiler (HRB), and a steam turbine that drives a compressor and an electric generator. In the case of a centrifugal compressor, the exiting steam is superheated. The superheat is removed by contacting the superheated steam with liquid water in a packed column (Figure 6.1(b)). In the case of a gerotor compressor, liquid water can be injected directly into the compressor so it emerges saturated (Figure 6.1(a)). The generator shown in both Figures 6.1 (a) and 6.1(b) produce electricity used for pumps. Also, they allow any excess power produced by the gas turbine and steam turbine to be converted to to electricity and sold on the open market.

The gas turbine produces 11.19 MW, which corresponds to the output of a Mars 100 gas turbine produced by Solar Turbines. The steam turbine produce 5.032 MW and operates with the following inlet and outlet conditions:

$p_{in} = 4{,}240$ kPa (615 psia)

$p_{out} = 10.1$ kPa (1.47 psia)

$T_{in} = 400\,^{\circ}\mathrm{C}\left(752\,^{\circ}\mathrm{F}\right)$

$T_{out} = 46.1\,^{\circ}\mathrm{C}\left(115^{\circ}\mathrm{F}\right)$

The combined efficiency of this combined cycle is 51.5%, which is conservative. Some combined cycles have efficiencies of 57% [36].

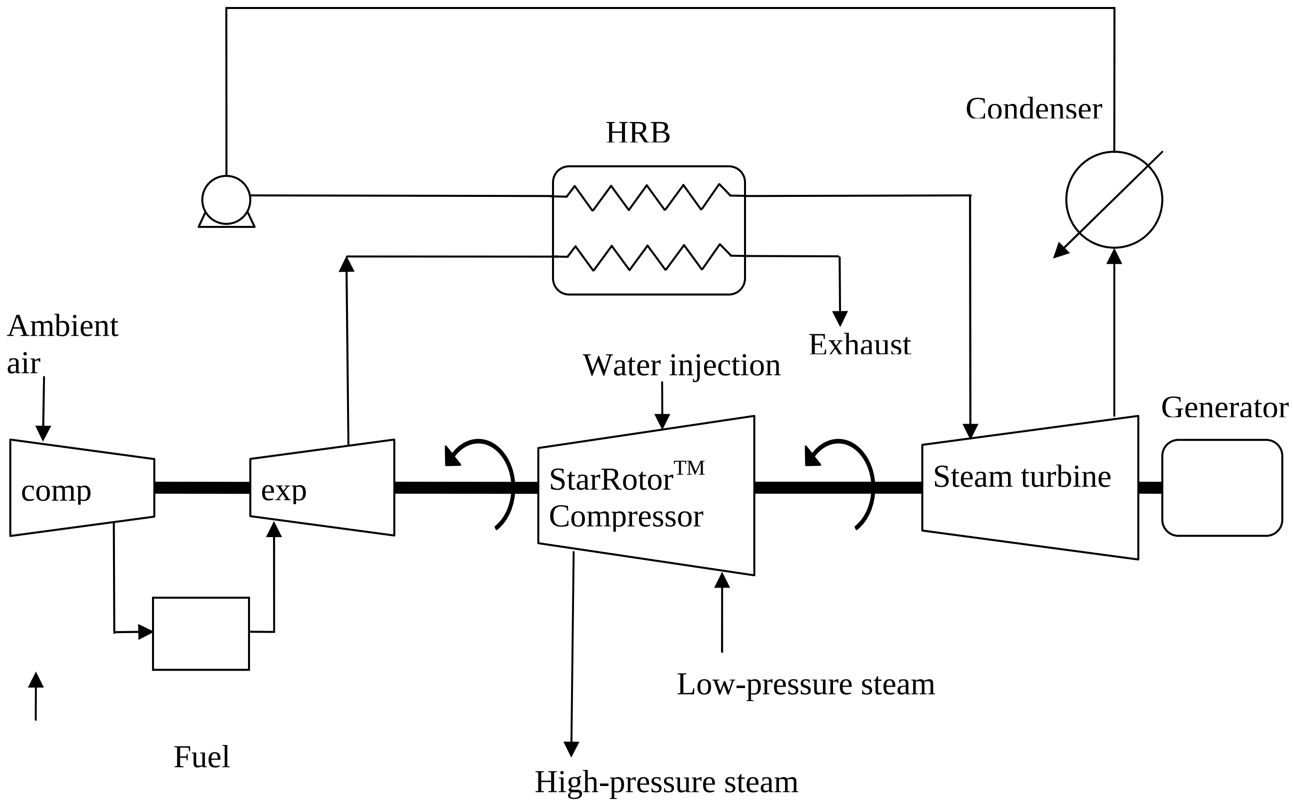


Figure 6-1(a). Combined-cycle power system with StarRotor gerotor compressor.

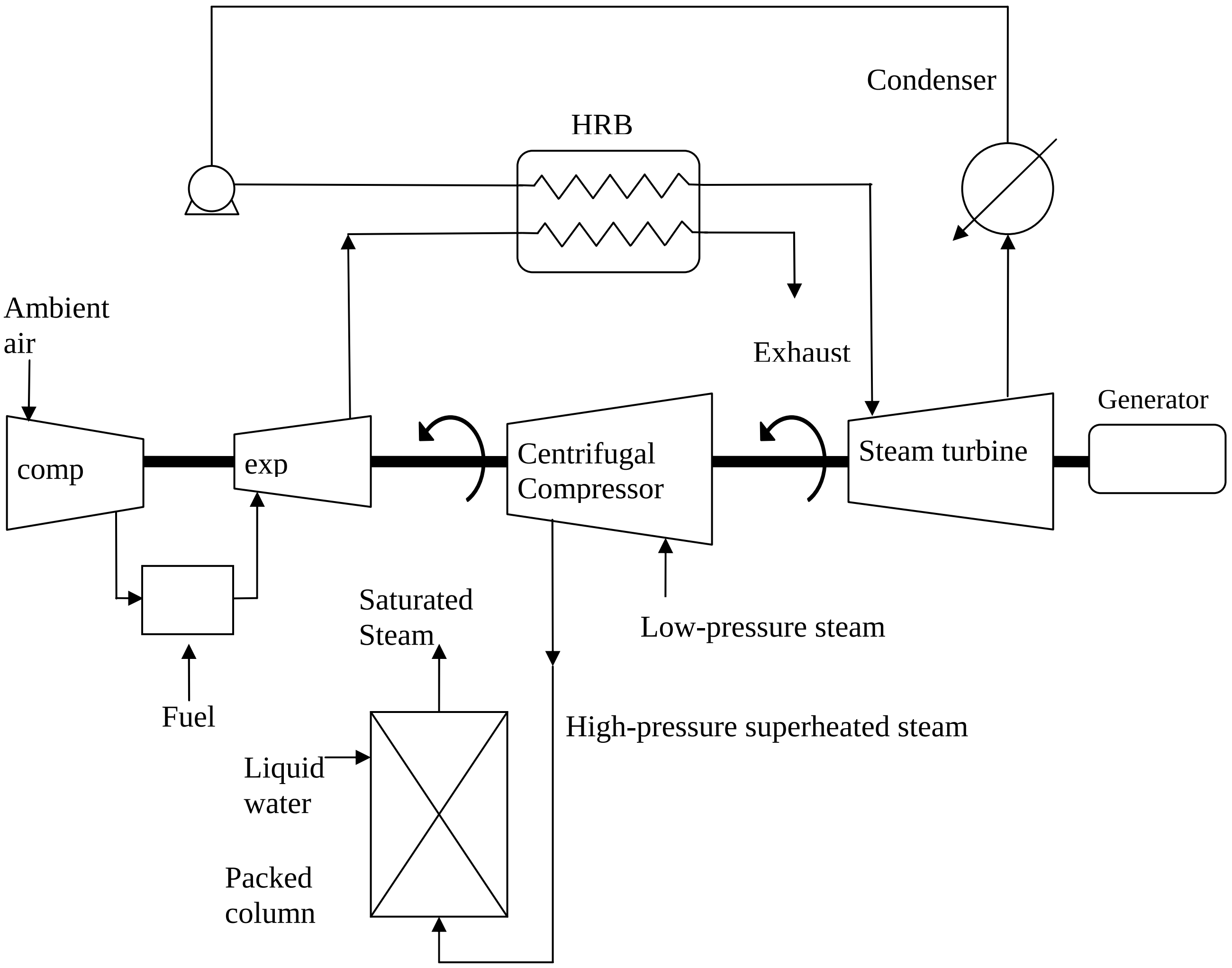


Figure 6-1(b). Combined-cycle power system with centrifugal compressor.

CHAPTER VII

EQUIPMENT DESIGN

Basic Process-design Variables

Fluids. Degassed seawater supplied to the evaporator train is connected in parallel to satisfy individual evaporator temperature needs. The flow is between 147.5 kg/s (324.5 lb_m/s) and 144.57 kg/s (318.05 lb_m/s)). Saturated steam is supplied to the first stage at 176.7 °C (350 °F) and recovered from the last stage at 163.5 °C (326 °F), assuming three evaporator stages, each with $\Delta T = 3.3333\,\mathrm{K}\,(6°\mathrm{F})$ and 7% brine.

Flow rates. The flow rates are individually calculated for each evaporator as required by production.

Entrance temperatures. The estimated seawater temperature range is: 21.1–26.2 °C (70–80 °F).

Amount of vaporization or condensation. The amount of condensation as required by flow rates or production is calculated by the corresponding mass balance.

Operating pressures. To ensure dropwise condensation, 827.4 kPa gauge (120 psig) is the maximum pressure on steam side

Fouling factors. None (surfaces are assumed to be cleaned by entrained scrubbing balls)

Heat transfer coefficient. About 0.14 $\mathrm{MW/(m^2 \cdot K)}$ $(24{,}900\,\mathrm{Btu/(h \cdot ft^2 \cdot °F)})$. To be conservative, the heat transfer coefficient is less than those presented in Figure 2-11.

Maximum steam velocity. 1.52 m/s (5 ft/s) to allow dropwise condensation

Suggested codes of construction API-ASME codes .

Material of construction. Smooth surface 0.007-in-thick titanium plates are expected to attain dropwise condensation, the evaporator is constructed with elastomer gaskets, and a carbon steel vessel coated with epoxy or titanium cladding to protect it from corrosion on seawater side.

Center-to-center distance between adjacent plates. 0.25 inches.

Standard plate dimension. 2.4 m × 2.4 m (8 ft × 8 ft).

Vessel shell thickness. 29.6 mm is recommended (see Appendix M).

Compressor

For a dry centrifugal compressor, the isentropic compressor work is

$$W = \frac{\frac{k}{k-1} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right]}{\eta_c} = \frac{H_2 - H_1}{\eta_c} \tag{7.1}$$

where

$W =$ compressor work $(\mathrm{J/kg})$

$p_1 =$ entering compressor pressure (Pa)

$p_2 =$ exiting compressor pressure (Pa)

$v_1 =$ steam specific volume at compressor entrance $(\mathrm{m^3/kg})$

$k =$ isentropic constant $= C_p / C_v = 1.3$ for steam

$H_2 =$ specific enthalpy of compressor outlet (J/kg)

$H_1 =$ specific enthalpy of compressor inlet (J/kg)

η_c = isentropic compressor efficiency = 0.85 (assumed)

The work supplied to the compressor [37] increases the gas pressure and temperature. The compressor is small relative to the flow through it, so it is essentially adiabatic, i.e., a small amount of energy is lost to heat transfer out of the compressor. The goal is to increase the pressure of the gas using the least possible work, which is accomplished by making the compression process as reversible as possible.

The work required in a reversible isothermal compression is less than that done in a reversible adiabatic compression. If the vapor temperature is maintained near the suction temperature by injecting liquid water, then the process becomes nearly isothermal.

Conventional centrifugal compressors do not allow water injection because the high-speed blades can be damaged from the impact with the droplets. In contrast, a StarRotor compressor operates at lower speeds and has robust components that can tolerate liquid injection. The StarRotor compressor uses gerotors, which are positive displacement devices.

For the case of liquid water injection, the compressor work W is evaluated [3] as

$$W = \frac{(1+x)H_2^{vap} - \left(H_1^{vap} + xH_1^{liq}\right)}{\eta_c} \tag{7.2}$$

where

H_2^{vap} = vapor enthalpy at compressor exit (2) (J/kg)

H_1^{vap} = vapor enthalpy at compressor inlet (1) (J/kg)

H_1^{liq} = liquid enthalpy at compressor inlet (J/kg)

η_c = compressor efficiency = 0.85 (assumed)

x = the amount of injection water that evaporates in the compressor

$$= \frac{S_1^{vap} - S_2^{vap}}{S_2^{vap} - S_1^{liq}} \qquad (7.3)$$

where

S_1^{liq} = entropy of liquid water at compressor inlet $(\mathrm{J}/(\mathrm{kg}\cdot\mathrm{K}))$

S_1^{vap} = entropy of steam at compressor inlet $(\mathrm{J}/(\mathrm{kg}\cdot\mathrm{K}))$

S_2^{vap} = entropy of steam at compressor exit $(\mathrm{J}/(\mathrm{kg}\cdot\mathrm{K}))$

Gas Turbine

In conjunction with the steam turbine, the gas turbine must supply the compressor and generator work requirements. From the manufacturer's chart [38] (see Appendix F):

a) gas turbine power output: 11,190 kW

b) gas turbine fuel requirements: 10,600 kJ/kW h

Steam Turbine

The steam turbine is designed according to the available steam working conditions. The enthalpy change defines the work output from the steam turbine. At the turbine inlet (1), the steam is superheated with the following properties:

$S_1 =$ superheated steam entropy $\left(\mathrm{kJ/(kg \cdot K)}\right)$

$H_1 =$ superheated steam enthalpy $\left(\mathrm{kJ/kg}\right)$

After an isentropic expansion, the steam properties at the turbine outlet (2) correspond to saturated steam

$S_{2l} =$ entropy of saturated liquid $\left(\mathrm{kJ/(kg \cdot K)}\right)$

$S_{2v} =$ entropy of saturated vapor $\left(\mathrm{kJ/(kg \cdot K)}\right)$

$H_{2l} =$ enthalpy of saturated liquid $\left(\mathrm{kJ/kg}\right)$

$H_{2v} =$ enthalpy of saturated vapor $\left(\mathrm{kJ/kg}\right)$

$$x = \text{saturated vapor quality} = \frac{S_1 - S_{2l}}{S_{2v} - S_{2l}}$$

$$H_2 = xH_{2v} + (1 - x)H_{2l}$$

The energy recovered from the steam turbine is

$$W = m(H_1 - H_2)\eta_{turb} \tag{7.4}$$

Therefore the total mechanical power available for the proposed combined-cycle system is

Total power = [Power from gas turbine] + [Power from steam turbine]

Heat Recovery Boiler

For a given steam capacity (kg/s) [39], a boiler plant can be selected using the methodology provided in Appendix B-7. The size and price information provided includes the complete boiler, feed water deaerator, boiler feed pumps, chemical injection system, and shop assembly labor [40].

The rate of heat available is

$$q = \dot{m} C_p \left(T_{in} - T_{out}\right)$$

where

$q =$ heat recovered in HRB (J/s)

$T_{in} =$ gas temperature at boiler inlet (K)

$T_{out} =$ gas temperature at boiler outlet (K)

$C_p =$ gas heat capacity $\left(\mathrm{J/}\left(\mathrm{kg} \cdot \mathrm{K}\right)\right)$

$\dot{m} =$ exhaust gas flow rate

$= 150{,}390\ \mathrm{kg/s}$ @ $485°\mathrm{C}\ \left(905°\mathrm{F}\right)$

Pumps

Pumps will handle a total of 0.876 m^3/s of seawater (3.5% concentrated feed). Three pumps are required, one per stage. The purchase cost shown in Table 9.1 includes the electrical motor. The total power requirements for pumping [41] are 704.8 kW.

Table 9.1 summarizes the individual feed pumping requirements at each stage in the vapor-compression train, and provides the estimated working conditions and purchase cost. The discharge pressure required corresponds to the sum of the pressure drop in the sensible heat exchanger plus the vapor pressure in the latent heat exchanger. The price is obtained from Figure 12-20 of Reference [40]. Pumps P-1, P-2, and P-3 receive a correction factor of 1.6 because the pressure required exceeds 1,035 kPa. In addition, there is a factor of 1.26 for bronze anti-corrosion material for all pumps used in the seawater environment.

CHAPTER VIII

VAPOR COMPRESSION TRADE-OFFS

Figure 8-1 shows the system boundary for an overall energy balance. Compressor work (W) enters the system and exits as thermal energy carried out by the distillate $\left(m_s C_{ps}\left(T_s - T_f\right)\right)$ and the brine $\left(m_b C_{pb}\left(T_b - T_f\right)\right)$.

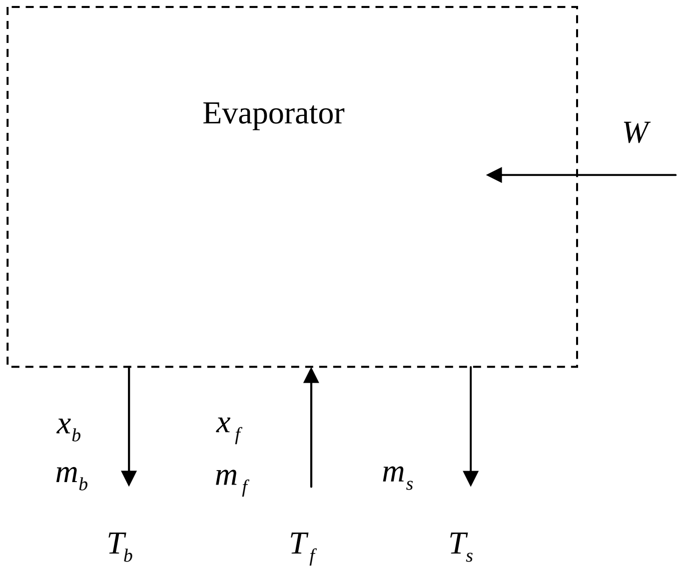


Figure 8-1. Overall energy balance.

Therefore, the energy balance is

$$W = m_b C_{pb}\left(T_b - T_f\right) - m_s C_{ps}\left(T_s - T_f\right)$$

$$W = m_b C_{pb} \Delta T_b - m_s C_{ps} \Delta T_s$$

Let $\Delta T = \Delta T_b = \Delta T_s$

$$W = m_b C_{pb} \Delta T - m_s C_{ps} \Delta T$$

$$= \left(m_b C_{pb} - m_s C_{ps} \right) \Delta T$$

$$\frac{W}{m_s} = \left(\frac{m_b}{m_s} C_{pb} - C_{ps} \right) \Delta T$$

Total mass balance: $m_s + m_b = m_f$

Salt mass balance: $m_f x_f = m_b x_b$

$$m_f = m_b \frac{x_b}{x_f}$$

$$m_s + m_b = m_b \frac{x_b}{x_f}$$

$$m_s = m_b \frac{x_b}{x_f} - m_b$$

$$m_s = m_b \left(\frac{x_b}{x_f} - 1 \right)$$

$$1 = \frac{m_b}{m_s} \left(\frac{x_b}{x_f} - 1 \right)$$

$$\frac{m_b}{m_s} = \frac{1}{\left(\frac{x_b}{x_f} - 1 \right)}$$

$$\frac{W}{m_s} = \left(\frac{1}{\left(\frac{x_b}{x_f} - 1 \right)} C_{pb} + C_{ps} \right) \Delta T$$

$$\Delta T = \frac{1}{\left(\frac{1}{\left(\frac{x_b}{x_f} - 1 \right)} C_{pb} + C_{ps} \right)} \left(\frac{W}{m_s} \right) \qquad (8.1)$$

This ΔT represents the temperature rise of the distillate and brine. In addition, it is the approach temperature in the sensible heat exchangers. The seawater feed is 3.5% salt $\left(x_f = 35\,\text{g/kg}\right)$ and the following brine concentrations $\left(x_b\right)$ are employed: 50, 70, 100 and 150 g/kg. The ΔT is proportional to W / m_s; the proportionality constant is shown in Table 8.1. The results are plotted on Figure 8.2. It is observed that at 7% concentration and low ΔT $\left(\text{between}\,1.5\,\text{and}\,3.5\,\text{K}\right)$ the compressor shaft work requirements are between 12 and 27 MJ/m^3, which is an acceptable range.

Table 8-1. Proportionality constant in Equation 8.1

x_f (g/kg)	x_b (g/kg)	$C_{pb}\left(\mathrm{kJ}/\left(\mathrm{kg}\cdot\mathrm{K}\right)\right)$	$C_{ps}\left(\mathrm{kJ}/\left(\mathrm{kg}\cdot\mathrm{K}\right)\right)$	$\dfrac{1}{\left(\dfrac{1}{\dfrac{x_b}{x_f}-1}\right)C_{pb}+C_{ps}}\left(\dfrac{\mathrm{kg}\cdot\mathrm{K}}{\mathrm{kJ}}\right)$
35	50	3.917	4.209	0.07491
35	70	3.822	4.209	0.1245
35	100	3.649	4.209	0.16197
35	150	3.521	4.209	0.18937

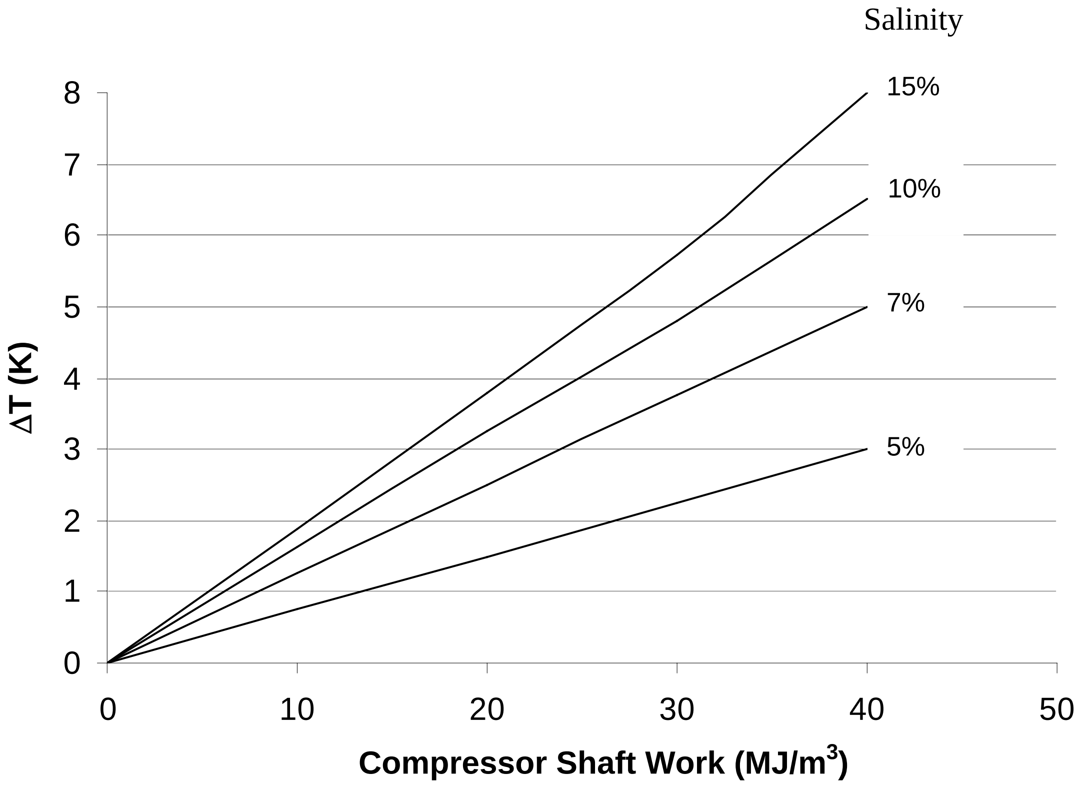


Figure 8-2. Temperature rise of both brine and distillate. This ΔT is also the approach temperature in the sensible heat exchanger.

Table 8-2. Cases used in vapor-compression trade-off

Case	A	B	C
ΔT_{cond}	$3.333\ \text{K}\ (6°\text{F})$	$2.222\ \text{K}\ (4°\text{F})$	$1.111\ \text{K}\ (2°\text{F})$
ΔT_{sens}	$3.25\ \text{K}\ (5.85°\text{F})$	$2.5\ \text{K}\ (4.5°\text{F})$	$1.75\ \text{K}\ (3.15°\text{F})$
Salinity (g/kg)	70	70	70
No. of Stages	3	3	3
Compressor energy ($\text{MJ/m}^3_{\text{liquid product}}$)	26	20	14

Three trade-off cases will be studied as shown in Table 8.2. Appendix D provides a detailed thermodynamic evaluation of each case. Tables from D.3 to D.6 summarize the results. The analysis follows:

Case A. Figure 8-3 shows the compressor inlet volume as a function of number of stages. Figure 8-4 shows the compressor work for a dry compressor. Figure 8-5 shows the compressor work for a wet compressor, such as a gerotor compressor. In all cases, the wet compressor had significantly less work requirements, so only wet compressors were evaluated in Cases B and C.

Case B. Figure 8-6 shows the compressor inlet volume and Figure 8-7 shows the compressor work.

Case C. Figure 8-8 shows the compressor inlet volume and Figure 8-9 shows the compressor work.

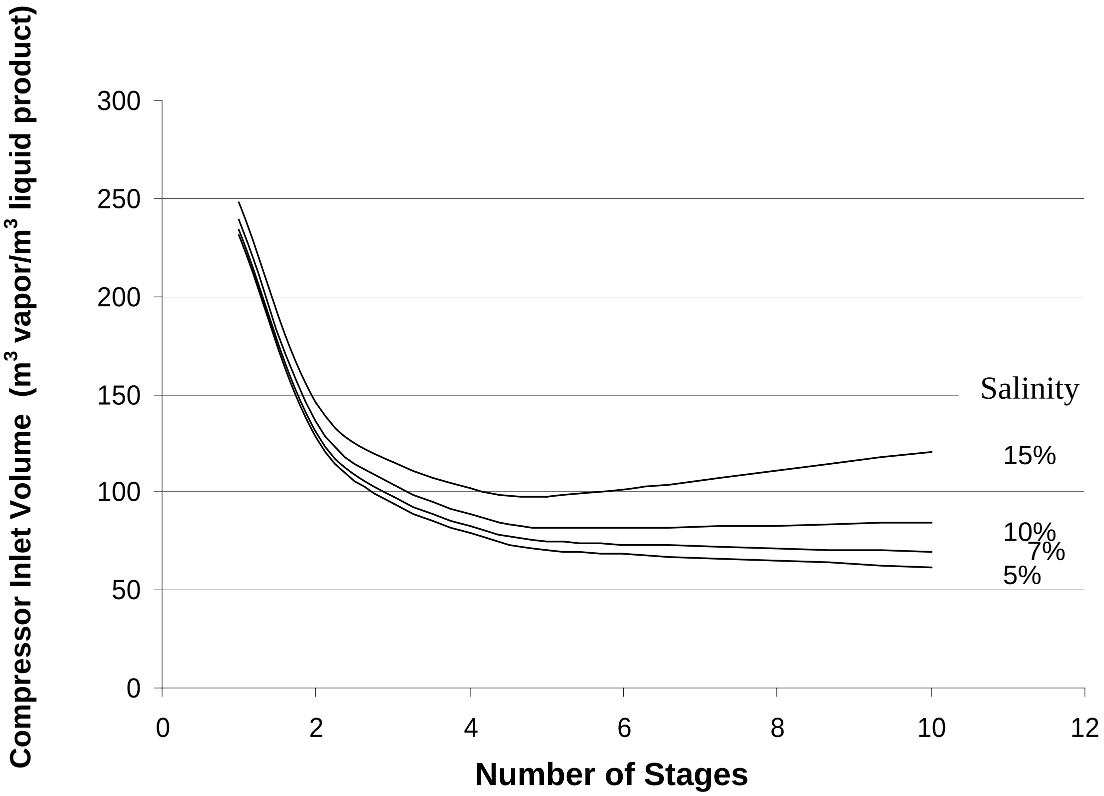


Figure 8-3. Case A. Gas flow at compressor inlet. (Latent heat exchanger ΔT = 3.333 K $(6°\text{F})$, Seawater feed = 3.5%, Heat exchanger inlet $T = 446.53$ K (173 °C, 344 °F)).

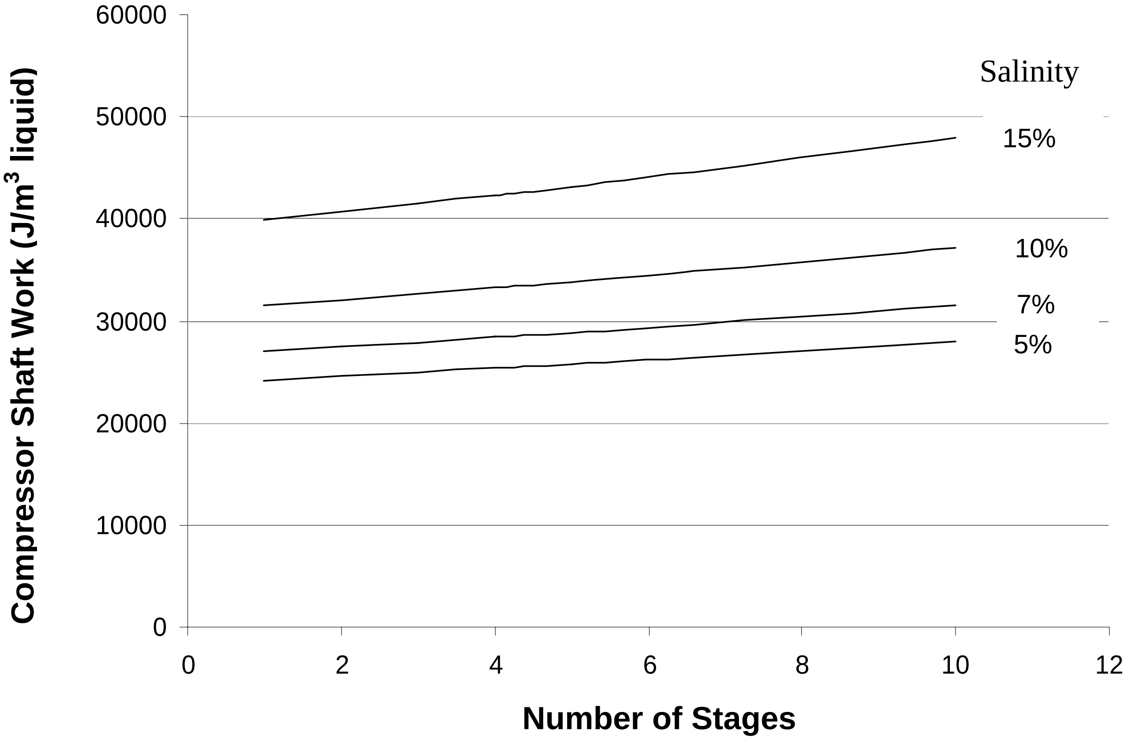


Figure 8-4. Case A. Compressor shaft work for dry compressor. (Latent heat exchanger $\Delta T = 3.333\,\mathrm{K}\,(6°\mathrm{F})$, Seawater feed = 3.5%, Heat exchanger inlet $T =$ 446.53 K (173 °C, 344.08 °F). Compressor efficiency = 85%).

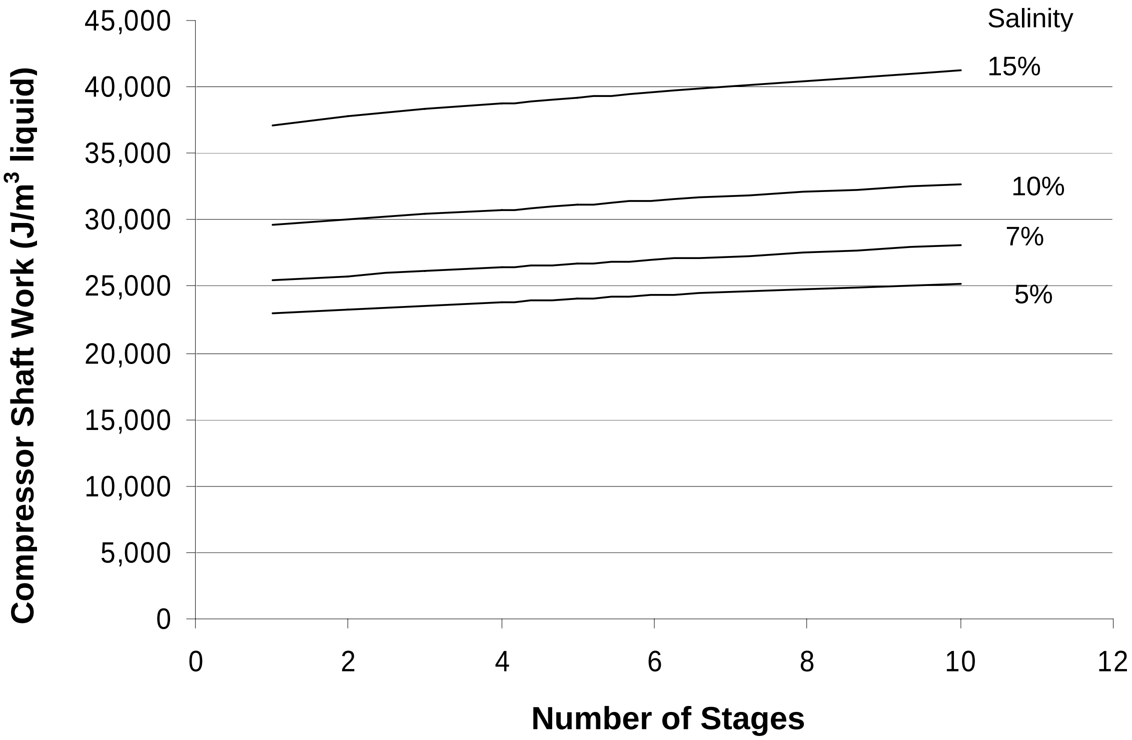


Figure 8-5. Case A. Compressor shaft work for wet compressor. (Latent heat exchanger $\Delta T = 3.333\,\mathrm{K}\,(6°\mathrm{F})$, Seawater feed = 3.5%, Heat exchanger inlet $T =$ 446.53 K (173 °C, 344.08 °F). Compressor efficiency = 85%).

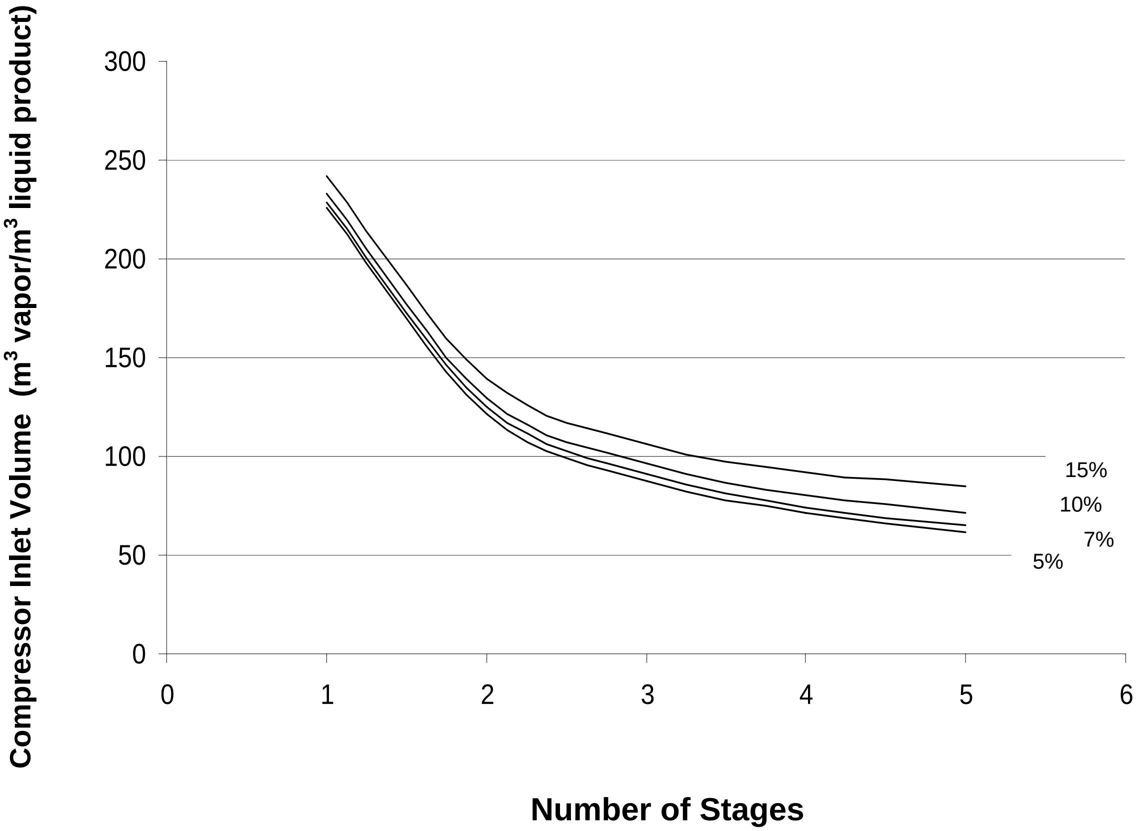


Figure 8-6. Case B. Gas flow at compressor inlet. (Latent heat exchanger $\Delta T =$ 2.222 K $(4°\text{F})$, Seawater feed = 3.5%, Heat exchanger inlet $T = 447.63$ K (174.4 °C, 346.06 °F)).

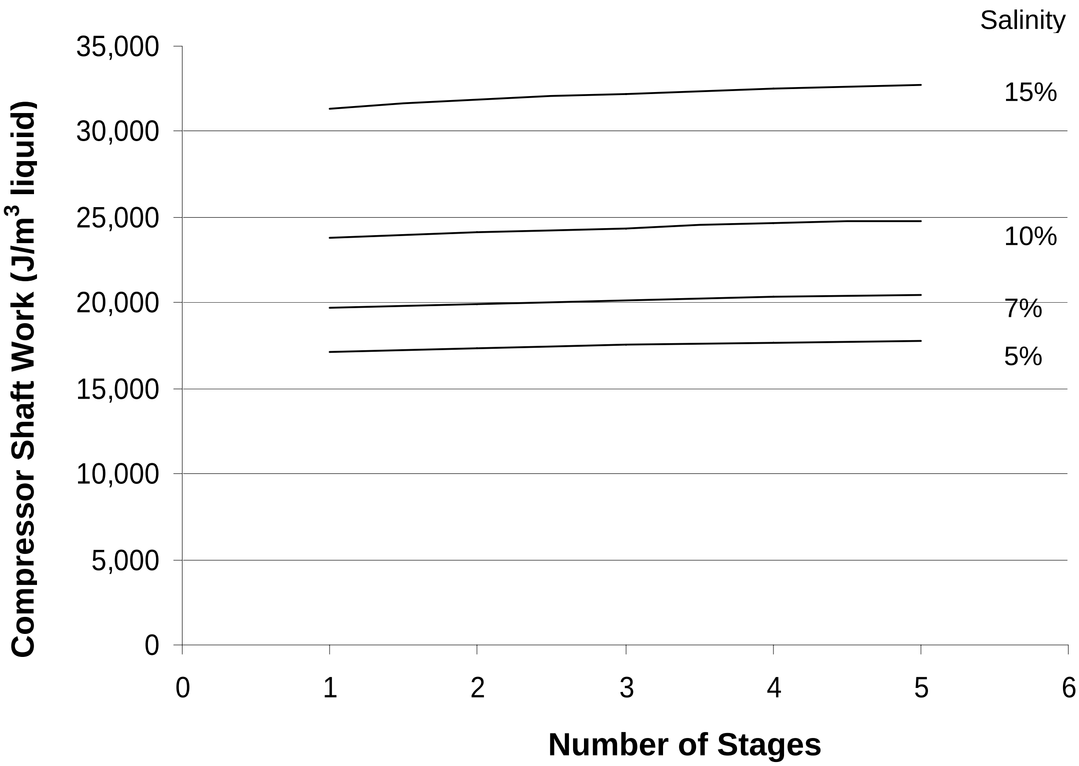


Figure 8-7. Case B. Compressor shaft work for wet compressor. (Latent heat exchanger $\Delta T = 2.222\ \mathrm{K}\ (4°\mathrm{F})$, Seawater feed = 3.5%, Heat exchanger inlet $T =$ 447.63 K (174 °C, 346.06 °F). Compressor efficiency = 85%).

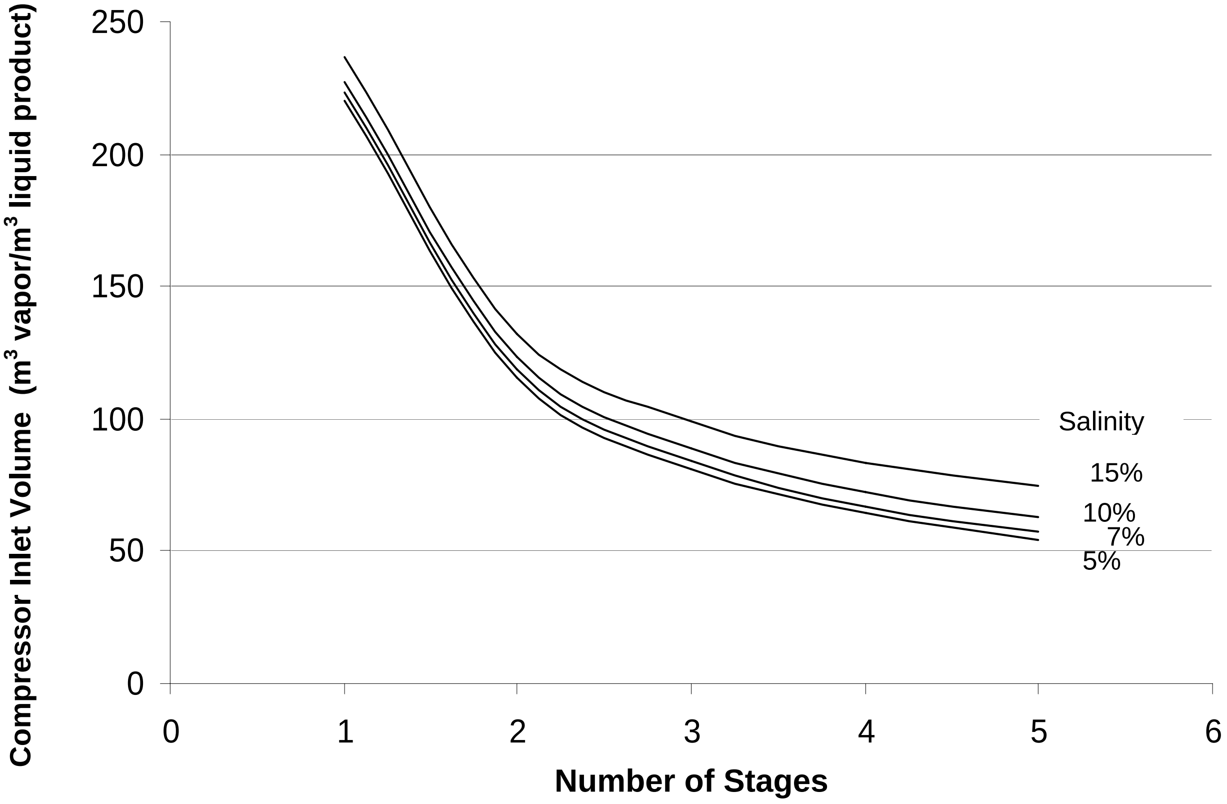


Figure 8-8. Case C. Gas flow at compressor inlet. (Latent heat exchanger $\Delta T =$ 1.111 K $(2°\text{F})$, Seawater feed = 3.5%, Heat exchanger inlet $T = 448.7$ K (175.5 °C, 348 °F)).

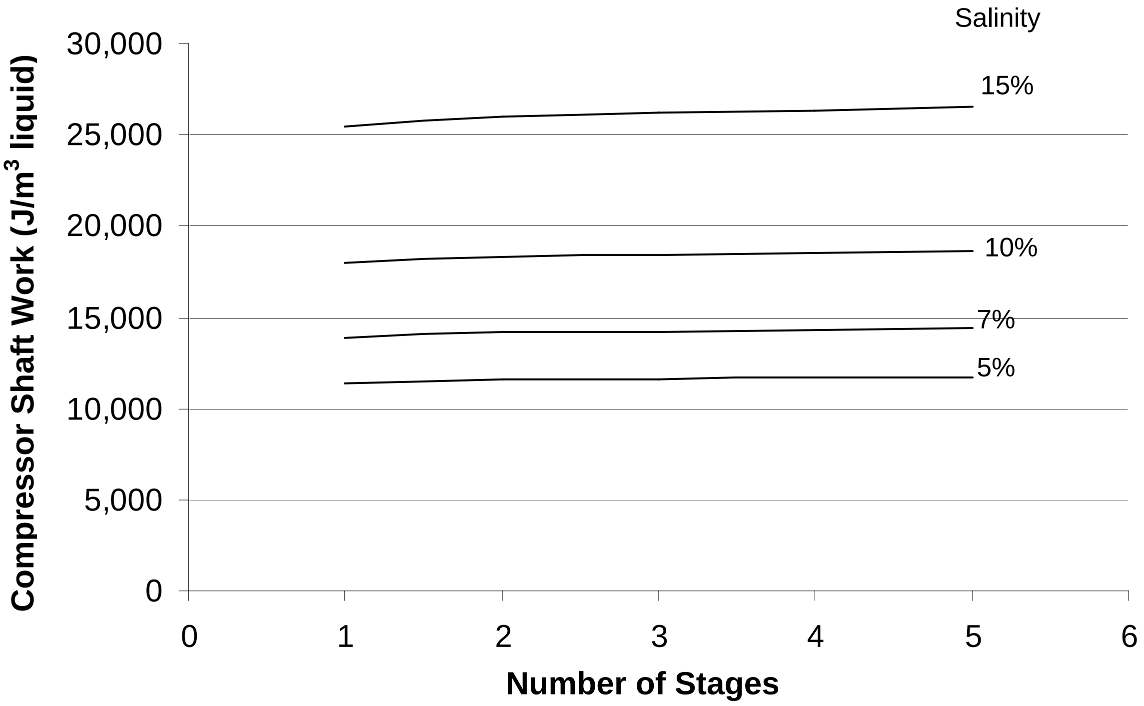


Figure 8-9. Case C. Compressor shaft work for wet compressor. (Latent heat exchanger $\Delta T = 1.111\,\mathrm{K}\ (2°\mathrm{F})$, Seawater feed = 3.5%, Heat exchanger inlet $T =$ 446.53 K (175 °C, 348 °F). Compressor efficiency = 85%).

.

CHAPTER IX

DESALINATION PLANT COST ANALYSIS

The elements of cost analysis for desalination processes [40] are depicted in the flowchart shown in Figure 9-1.

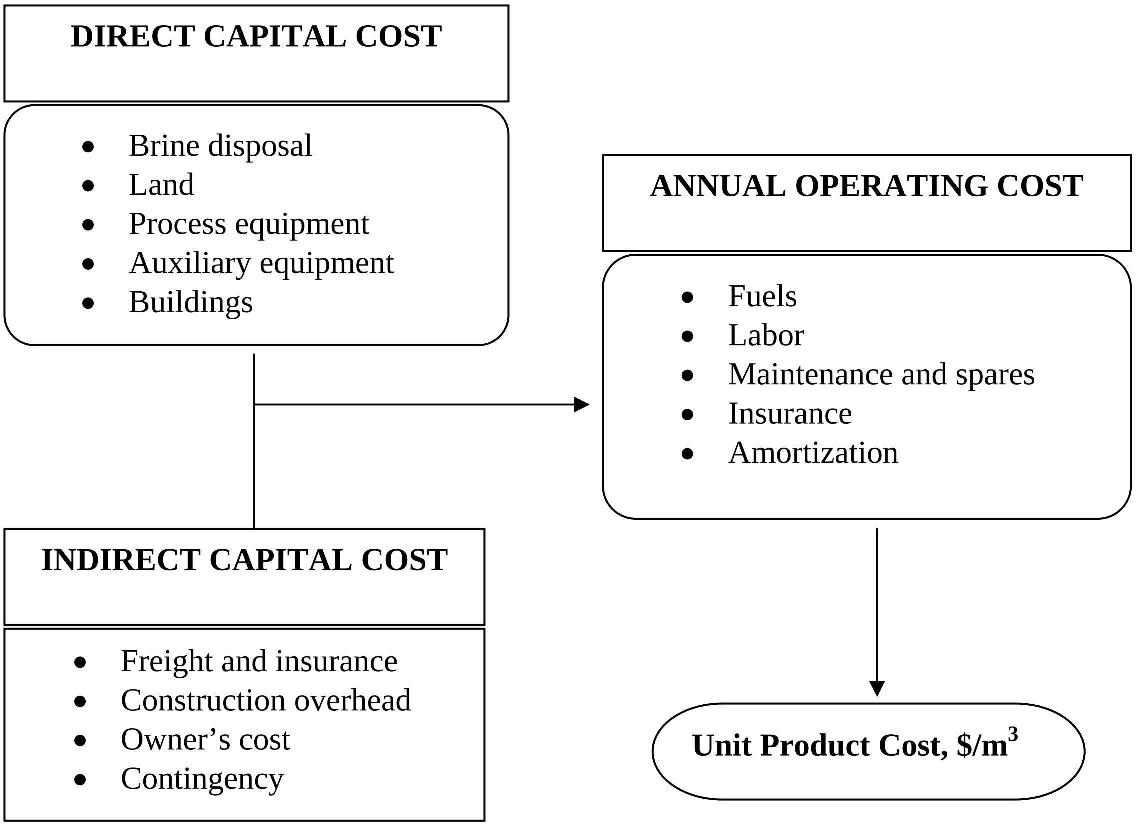


Figure 9-1. Elements of cost analysis for the process [40].

Tables 9.1, 9.2 and 9.3 summarize the equipment needed in the advanced vapor-compression desalination plant for Cases A, B, and C respectively. Also these tables provide utility requirements for operating the plant. Table 9.4 shows the components of the Lang Factor used to estimates the total plant cost from the equipment cost.

Table 9.5 shows the unitary cost of desalinated water produced for the three cases studied at two energy costs: \$0.5/GJ and \$5/GJ. For \$0.5/GJ, Cases A and B were the least expensive (\$0.252/m^3, \$0.95/thousand gal). For \$5/GJ, Case C was the least expensive (\$0.474/m^3, \$1.79/thousand gal).

Table 9.6 compares this cost to other water desalination methods. Vapor-compression technology has a significantly lower capital cost and selling price. Figures 9-2, 9-3, and 9-4 show the water cost for a variety of fuel prices and interest rates for each case.

Table 9-1. Case A. Summary of equipment and utility requirements for a 10 million gallons/day desalination plant

code	Name	Description	Purchase cost ($)	Utilities		
				Nat Gas	Electricity	Cooling water
L1	Latent Heat Exchanger 1	6,895 ft^2	48, 748			
L2	Latent Heat Exchanger 2	6,874 ft^2	48, 600			
L3	Latent Heat Exchanger 3	6,846 ft^2	48,401			
B1	Brine Sensible Heat Exchanger 1	42,358 ft^2	299,471			
B2	Brine Sensible Heat Exchanger 2	40,957 ft^2	289,566			
B3	Brine Sensible Heat Exchanger 3	42,550 ft^2	300,829			
D1	Distillate Sensible Heat Exchanger 1	46,458 ft^2	328,458			
D2	Distillate Sensible Heat Exchanger 2	43,313 ft^2	306,223			
D3	Distillate Sensible Heat Exchanger 3	46,067 ft^2	325,699			
P1	Pump 1 Seawater feed. Stage 1	1,480 kPa 147.5 kg/s	20,000		251.3 kW	
P2	Pump 2 Seawater feed. Stage 2	1,480 kPa 146.03 kg/s	20,000		248.81 kW	
P3	Pump 3 Seawater Feed. Stage 3	1,480 kPa 0.0584 m^3/s	20,000		246.32 kW	
GT	Gas Turbine		800,000	10,600 kJ/kW-h	11,190 kW	
C	Compressor		450,000		11,390 kW	
ST	Steam Turbine		300,000		5,032 kW	
G	Generator		100,000		5,086 kW	
C	Condenser		19,000			0.198 m^3/s
HRB	Heat recovery boiler (unfired)		800,000			
Total Equipment			4,524,995			
Fixed Capital Investment = 5.19 x 4,524,995 = $23,484,724						

Table 9-2. Case B. Summary of equipment and utility requirements for a 10 million gallons/day desalination plant

code	Name	Description	Purchase cost ($)	Utilities		
				Nat Gas	Electricity	Cooling water
L1	Latent Heat Exchanger 1	10,311 ft^2	72,899			
L2	Latent Heat Exchanger 2	10,279 ft^2	72,673			
L3	Latent Heat Exchanger 3	10,237 ft^2	72,376			
B1	Brine Sensible Heat Exchanger 1	55,065 ft^2	389,310			
B2	Brine Sensible Heat Exchanger 2	53,244 ft^2	376,435			
B3	Brine Sensible Heat Exchanger 3	55,315 ft^2	391,078			
D1	Distillate Sensible Heat Exchanger 1	60,395 ft^2	426,993			
D2	Distillate Sensible Heat Exchanger 2	58,906 ft^2	416,466			
D3	Distillate Sensible Heat Exchanger 3	59,888 ft^2	423,409			
P1	Pump 1 Seawater feed. Stage 1	1,480 kPa 147.5 kg/s	20,000		251.3 kW	
P2	Pump 2 Seawater feed. Stage 2	1,480 kPa 146.03 kg/s	20,000		248.81 kW	
P3	Pump 3 Seawater Feed. Stage 3	1,480 kPa 0.0584 m^3/s	20,00		246.32 kW	
GT	Gas Turbine		800,000	10,600kJ/kW-h	11,190 kW	
C	Compressor		450,000		8,760 kW	
ST	Steam Turbine		300,000		5,032 kW	
G	Generator		110,000		7,714 kW	
C	Condenser		19,000			0.198 m^3/s
HRB	Heat recovery boiler (unfired)		800,000			
Total Equipment			5,180,639			
Fixed Capital Investment = 5.19 x 5,180,639 = $26,887,517						

Table 9-3. Case C. Summary of equipment and utility requirements for a 10 million gallons/day desalination plant

code	Name	Description	Purchase cost ($)	Utilities		
				Nat Gas	Electricity	Cooling water
L1	Latent Heat Exchanger 1	20,551 ft^2	145, 296			
L2	Latent Heat Exchanger 2	20,499 ft^2	144,928			
L3	Latent Heat Exchanger 3	20,417 ft^2	144,349			
B1	Brine Sensible Heat Exchanger 1	78,665 ft^2	556,130			
B2	Brine Sensible Heat Exchanger 2	76,059 ft^2	537,738			
B3	Brine Sensible Heat Exchanger 3	79,018 ft^2	558,655			
D1	Distillate Sensible Heat Exchanger 1	86,275 ft^2	609,960			
D2	Distillate Sensible Heat Exchanger 2	84,147 ft^2	594,922			
D3	Distillate Sensible Heat Exchanger 3	85,550 ft^2	604,840			
P1	Pump 1 Seawater feed. Stage 1	1,480 kPa 147.5 kg/s	20,000		251.3 kW	
P2	Pump 2 Seawater feed. Stage 2	1,480 kPa 146.03 kg/s	20,000		248.81 kW	
P3	Pump 3 Seawater Feed. Stage 3	1,480 kPa 0.0584 m^3/s	20,00		246.32 kW	
GT	Gas Turbine		800,000	10,600 kJ/kW-h	11,190 kW	
C	Compressor		450,000		6,130 kW	
ST	Steam Turbine		300,000		5,032 kW	
G	Generator		120,000		10,340 kW	
C	Condenser		19,000			0.198 m^3/s
HRB	Heat recovery boiler (unfired)		800,000			
Total Equipment			6,445,818			
Fixed Capital Investment = 5.19 x 6,445,818					**= $33,453,796**	

Table 9-4. Lang Factor for a fluid processing plant [41]

Item	Fraction of delivered equipment (Fluid processing plant)
Purchased equipment	1.00
Delivery	0.10
Installation of purchased equipment	0.47
Instrumentation and controls (installed)	0.36
Piping (installed)	0.68
Electrical systems (installed)	0.11
Buildings (including services)	0.18
Yard improvements	0.10
Services facilities (installed)	0.70
Engineering and supervision	0.33
Construction expenses	0.41
Legal expenses	0.04
Contractor’s fee	0.22
Contingency	0.44
Working capital*	0.05
Total FCI	5.19

*Industrial plants use 0.20, but this municipal plant is assumed to have lower working capital requirements.

Table 9-5. Executive summary of unitary costs of the desalination plant components for the three design cases presented

		Fuel cost $0.5/GJ			Fuel cost $5/GJ		
Case A	Variable Costs	$\$/\mathrm{m}^3$	$/day	$/yr	$\$/\mathrm{m}^3$	$/day	$/yr
	Fuel	0.037	1,423	519,650	**0.376**	14,237	5,196,505
	Labor[a]	0.089	3,367	1,228,955	**0.089**	3,367	1,228,955
	Cooling water[b]	0.031	1,200	438,000	**0.031**	1,200	438,000
	Electricity[c]	-0.032	-1,220	-445,534	**-0.097**	-3,661	-1,336,600
	Fixed costs						
	Debt Service[d]	0.109	4,146	1,513,046	**0.109**	4,146	1,513,046
	Maintenance[e]	0.015	568.83	207,622.00	**0.015**	568.83	207,622.00
	Insurance[f]	0.003	142.2	51,906	**0.003**	142.2	51,906
	TOTAL	**0.252**			**0.526**		
Case B	Variable Costs						
	Fuel	0.037	1,423	519,650	**0.376**	14,237	5,196,505
	Labor[a]	0.089	3,367	1,228,955	**0.089**	3,367	1,228,955
	Cooling water[b]	0.031	1,200	438,000	**0.031**	1,200	438,000
	Electricity[c]	-0.048	-1,851	-675,746	**-0.146**	-5,554	-2,027,240
	Fixed Costs						
	Debt Service[d]	0.125	4,745	1,732,056	**0.125**	4,745	1,732,056
	Maintenance[e]	0.015	568.83	207,622.00	**0.015**	568.83	207,622.00
	Insurance[f]	0.003	142.2	51,906	**0.003**	142.2	51,906
	TOTAL	**0.252**			**0.493**		
Case C	Variable Costs						
	Fuel	0.037	1,423	519,650	**0.376**	14,237	5,196,505
	Labor[a]	0.089	3,367	1,228,955	**0.089**	3,367	1,228,955
	Cooling water[b]	0.031	1,200	438,000	**0.031**	1,200	438,000
	Electricity[c]	-0.065	-2,482	-905,784	**-0.196**	-7,445	-2,717,352
	Fixed Costs						
	Debt Service[d]	0.156	5,904	2,155,046	**0.156**	5,904	2,155,046
	Maintenance[e]	0.015	568.83	207,622.00	**0.015**	568.83	207,622.00
	Insurance[f]	0.003	142.2	51,906	**0.003**	142.2	51,906
	TOTAL	**0.266**			**0.474**		

[a] $33.67/h [34]
[b] $\$0.07/\mathrm{m}^3$ (tower)
[c] $0.03/kWh. Fuel cost @ $5/GJ
$0.01/kWh. Fuel cost @ $0.5/GJ*
[d] 30-year bond at 5% interest rate
[e] Maintenance: 4% FCI
[f] Insurance: 1% FCI
*www.sciencemag.org May/27/2005
Vol. 308

Table 9-6. Comparison of water desalination methods

	Water cost ($\$/m^3$)	Capital cost ($\$/(m^3/day)$)	Heat (MJ/m^3)		Work (MJ/m^3)
Theoretical (see Appendix N)					2.55
Advanced Vapor-compression					
$0.50/GJ (Case A)	0.252	620	54.3	or	27.7**
$5.00/GJ (Case C)	0.474	884	30.8	or	15.7**
Conventional Vapor-compression*	0.46 to 2.50	894 to 1322	-		21.6 to 36.0
Reverse Osmosis*	0.64 to 1.98	1035 to 1665	-		21.6 to 36.0
Multi-stage flash*	0.77 to 1.84	1598 to 2269	145 to 290	and	14.4

*Economics of Desalination, www.cepmagazine.org 12,2002

**Includes both compressor and pump work.

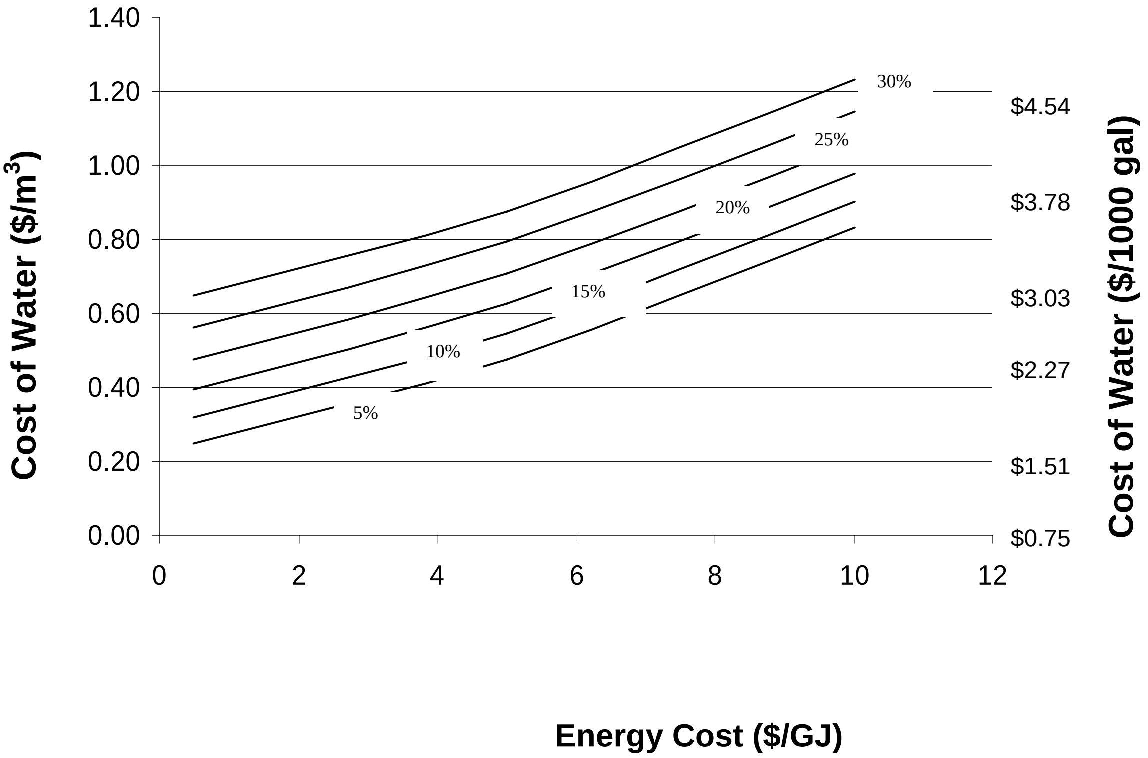


Figure 9-2. Case A. Cost of water for a variety of energy cost and interest rates. The price of electricity sold was extrapolated to -\$0.168/m^3 for a cost of gas of \$10/GJ.

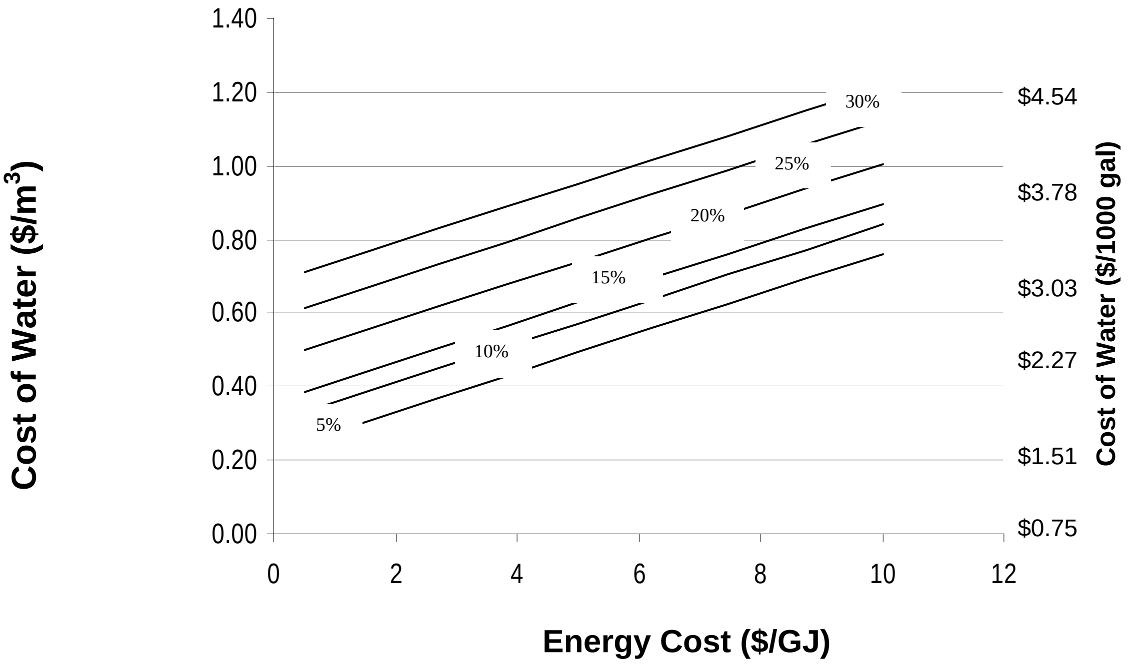


Figure 9-3. Case B. Cost of water for a variety of energy costs and interest rates. The price of electricity sold was extrapolated to -\$0.254/m^3 for a cost of gas of \$10/GJ.

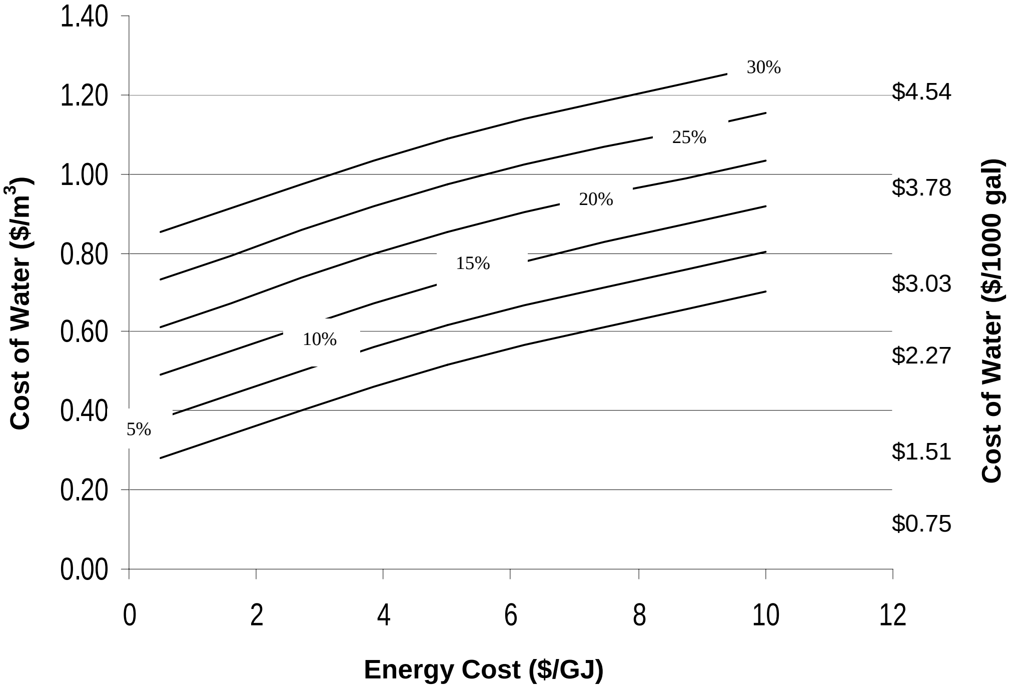


Figure 9-4. Case C. Cost of water for a variety of energy costs and interest rates. The price of electricity sold was extrapolated to -\$0.345/m^3 for a cost of gas of \$10/GJ.

CHAPTER X

CONCLUSIONS

A novel heat exchanger/evaporator design was analyzed. According to literature data, dropwise condensation on the steam side and pool boiling on the liquid side makes it possible to obtain overall heat transfer coefficients of up to $0.28\,\mathrm{MW/m^2}\left(49{,}000\,\mathrm{Btu/}\left(\mathrm{h}\cdot\mathrm{ft}^2\cdot{}^\circ\mathrm{F}\right)\right)$ under specific working conditions. To allow dropwise condensation, the recommended steam velocity should not exceed 1.52 m/s (5 ft/s), and steam pressure should not exceed 828 kPa gauge (120 psig). Smooth-surface titanium thin sheets were the heat transfer surfaces. Pressure drops due to vapor flow across the evaporator surface were shown to be minor. Figure 2-11 shows that the maximum heat transfer coefficient ocurrs at an overall temperature drop across the heat exchanger of $3.33\,{}^\circ\mathrm{C}$ $(6{}^\circ\mathrm{F})$. Compared to a conventional centrifugal compressor, the StarRotor gerotor compressor has the following advantages technical advantages:

- StarRotor gerotor compressor is less expensive ($450,000 vs $3,000,000).
- The StarRotor gerotor compressor can tolerate liquid water injection making the compression nearly isothermal requiring less work.
- The StarRotor gerotor compressor can be easily sized to the specific compression needs.
- The StarRotor gerotor compressor is efficient over a wide range of operating conditions.

CHAPTER XI

FUTURE WORK

The present study establishes solid ground basis for future optimization work. The following variables should be explored:

- Use of waste heat from steam turbine
- Use of waste heat exiting the heat recovery boiler
- Evaporator operating pressure and temperature

The study presented here was somewhat simplified and did not address some minor pieces of equipment that degas the feedwater, remove noncondensibles from the evaporator, and recover the abrasive scrubbing balls. These items should be included in future studies.

REFERENCES

1. G. Ejjeh, Desalination, A Reliable Source of New Water, International Desalination Association, Brussels, Belgium, 2001.
2. International Atomic Energy Agency, Examining the Economics of Seawater Desalination Using the DEEP Code, Technical Document 1186, International Atomic Energy Agency, Vienna, Austria 2000.
3. M. Holtzapple and G. Noyes, Advanced Vapor-Compression Evaporator and Heat Exchanger System, Disclosure of Invention, Department of Chemical Engineering, Texas A&M University, January 2004.
4. Princeton University, For Desalination Technologies Overview, October 2004, http://www.wws.princeton.edu/cgi-bin/byteserv.prl/~ota/disk2/1988/884204.PDF.
5. P. Griffith and M. S. Lee, The Effect of Surface Thermal Propieties and Finish on Dropwise Condensation. Int. J. Heat Mass Transfer. 10 (1967) 697.
6. J. W. Rose, On the Mechanism of Dropwise Condensation. Int. J. Heat Mass Transfer 10 (1967) 755.
7. C. Haslego and G. Polley, Compact Heat Exchanger Part 1: Designing Plate and Frame Heat Exchangers, CEPmagazine, http://cepmagazine.org, September 2002.
8. B. B. Mikic, On the Mechanism of Dropwise Condensation. Int. J. Heat Mass Transfer 12 (1969) 1311.
9. B. Gebhart, Heat Transfer, 2nd ed., McGraw Hill, New York, 1971.

10. E. A. Farber and R. L. Scorah, Heat Transfer to Water Boiling Under Pressure, Trans. ASME, 70 (1948) 369.
11. W. H. McAdams, Heat Transmission. 3rd ed., McGraw Hill, New York, 1954.
12. M. T. Cichelli and C. F. Bonilla, Heat Transfer to Liquids Boiling Under Pressure, AIChE. Trans. 42 (1946) 2.
13. Titanium Metals Corporation, Titanium, Products and Applications. November 2004, http://www.timet.com/fab-p09.htm.
14. C. Graham and P. Griffith, Drop Size Distributions and Heat Transfer in Dropwise Condensation. Int. J. Heat Mass Transfer 16 (1972) 337.
15. J. T. O’Bara, E.S. Killian and L. H. S. Roblee, Jr., Dropwise Condensation of Steam at Atmospheric and Above Pressures, Chem. Eng. Sci., 22 (1967) 1305.
16. J. B. Dolloff, N. H. Metzger and L. H. S. Roblee, Jr., Dropwise Condensation of Steam at Elevated Pressures, Chem. Eng. Sci., 24 (1969) 571.
17. E. Ruckestein and H. Metiu, On Dropwise Condensation on a Solid Surface, Chem. Eng. Sci., 20 (1965) 173.
18. C. Graham and P. Griffith, Drop Size Distributions and Heat Transfer in Dropwise Condensation, Int. J. Heat Mass Transfer, 16 (1972) 337.
19. K. Nabavian and L. A. Bromley, Condensation Coefficient of Water, Chem. Eng. Sci., 18 (1963) 651.
20. E. J. Le Fevre and J. W. Rose, A Theory of Heat Transfer by Dropwise Condensation, in: Proceedings of the Third International Heat Transfer Conference, Chicago, Vol. 2, 1966, p. 362.

21. E. J. Le Fevre and J. W. Rose, Heat-transfer Measurements During Dropwise Condensation of Steam, Int. J. Heat Mass Transfer, 7 (1964) 272.
22. E. J. Le Fevre and J. W. Rose, An Experimental Study of Heat Transfer by Dropwise Condensation, Int. J. Heat Mass Transfer, 8 (1965) 1117.
23. D. W. Tanner, C. J. Potter, D. Pope and D. West, Heat Transfer in Dropwise Condensation- Part I: The Effects of Heat Flux, Steam Velocity and Non-Condensable Gas Concentration, Int. J. Heat Mass Transfer, 8 (1965) 419.
24. D. W. Tanner, C. J. Potter, D. Pope and D. West, Heat Transfer in Dropwise Condensation- Part II: Surface Chemistry, Int. J. Heat Mass Transfer, 8 (1965) 427.
25. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., ASHRAE Fundamentals Handbook, Atlanta, GA, 2001.
26. Z. Novak, On the Stability of Boiling Heat Transfer, Paper Presented at the ASME-AIChE Heat Transfer Conference, University Park, PA., August 11, 1957.
27. Titanium Metals Corporation, Design and Fabrication, October 2004, http://www.timet.com/.
28. A. E. Dukler, Wicks, III and R. G. Cleveland, Frictional Pressure Drop in Two-Phase Flow: An Approach Through Similarity Analysis. AIChE Journal, 10 (1964) 44.
29. T. Kuppan, Heat Exchanger Design Handbook. Mechanical Engineering; Marcel Dekker, Inc., New York, 2000.

30. J. G. Collier and J. R. Thome, Convective Boiling and Condensation, 2nd ed., McGraw Hill, New York, 1981.
31. D. Chisholm, and L. A. Sutherland, Prediction of Pressure Gradients in Pipeline Systems during Two Phase Flow, Paper Presented at Symposium of Fluid Mechanics and Measurements in Two-phase Flow Systems, Leeds, September 24-25, 1969.
32. W. L. Mc Cabe and J. C. Smith, Unit Operations of Chemical Engineering, 3rd ed., McGraw Hill, New York, 1976.
33. G. F. Hewitt, Ed., Handbook of Heat Exchanger Design, Beggel House Inc., New York, 1992.
34. W. H. Emerson and D. T. Jamieson, Some Physical Properties of Sea Water in Various Concentrations, Desalination, 3 (1967) 213.
35. The National Engineering Laboratory, NEL Steam Tables, Her Majesty's Stationary Office, Edinburg, United Kingdom, 1964.
36. M. Lucas and B. Tabourier, The Mechanical Vapor-Compression Process Applied to Seawater Desalination, Desalination, 52 (1985) 123.
37. P. C. Hanlon, Compressor Handbook, McGraw-Hill, New York, 2000.
38. Solar Turbines, Mars 100 Compressor Set, April 2005, http://mysolar.cat.com/cda/layout?m=41428&x=7.
39. W. C. Turner, Energy Management Handbook, 4th ed., Prentice Hall Inc., Upper Saddle River, NJ, 2001.

40. H. M. Ettouney, H. T. El-Dessouky and P. J. Gowing. Evaluating the Economics of Desalination, CEPmagazine, http://cepmagazine.org, December 2002.

41. M. S. Peters, K. D. Timmerhaus, and R. E. West, Plant Design and Economics for Chemical Engineers, 5th ed., McGraw Hill, New York, 2003.

42. MacDonald Company, - Calculate Total Dynamic Head (TDH).htm. November (2004).
http://www.rfmacdonald.com/techtalk/headcalc.html

43. US Government, Cost of Minerals, June 2005,
http://minerals.usgs.gov/minerals/pubs/commodity/titanium/670798.pdf.

44. R. Rautenbach and B. Arzt, Gas Turbine Waste Heat Utilization for Desalination, Desalination, 52 (1985) 105.

Suplemental Sources Consulted

Alpha Knife Supply, Titanium Sheets, January 2005, http://www.alphaknifesupply.com/ti-small.htm.

Aquara, Costs of Desalination, November 2004, www.uara.com/Merchant2/merchant.mv?Screen=CTGY&Store_Code=22e&Category_Code=Sea+Water+R.O.+Membranes++2-8.

Matche. Compressor Costs, March 2005, http://matche.com/EquipCost/Compressor.htm.

StarRotor Corporation, Engine Development, January 2005 http://www.starrotor.com/indexflash.htm

Perry, J. H. Chemical Engineers' Handbook, 5th ed., McGraw-Hill, New York, 1973.

Texas A&M University, Mixalco Process, February 2005 http://64.233.187.104/u/tamu?q=cache:3AAasMqj9yEJ:www.ct.ornl.gov/symposium/20th/Abstracts/Paper26.htm+Mixalco+process+at+Texas+A%26M&hl=en&ie=UTF-8

Urila, Desalination, November 2004, http://urila.tripoid.com/Seawater.htm.

US Department of Energy, Federal Energy Management Program, June 2005, http://www.eere.energy.gov/femp/technologies/derchp_fedrequire.cfm.

Utah Net Metering & Small Electrical Generators, January 2005, http:// www.utahpower.net/File/File3880.pdf.

Water Properties, June 2005, http//www.Isbu.ac.uk/water/data.html.

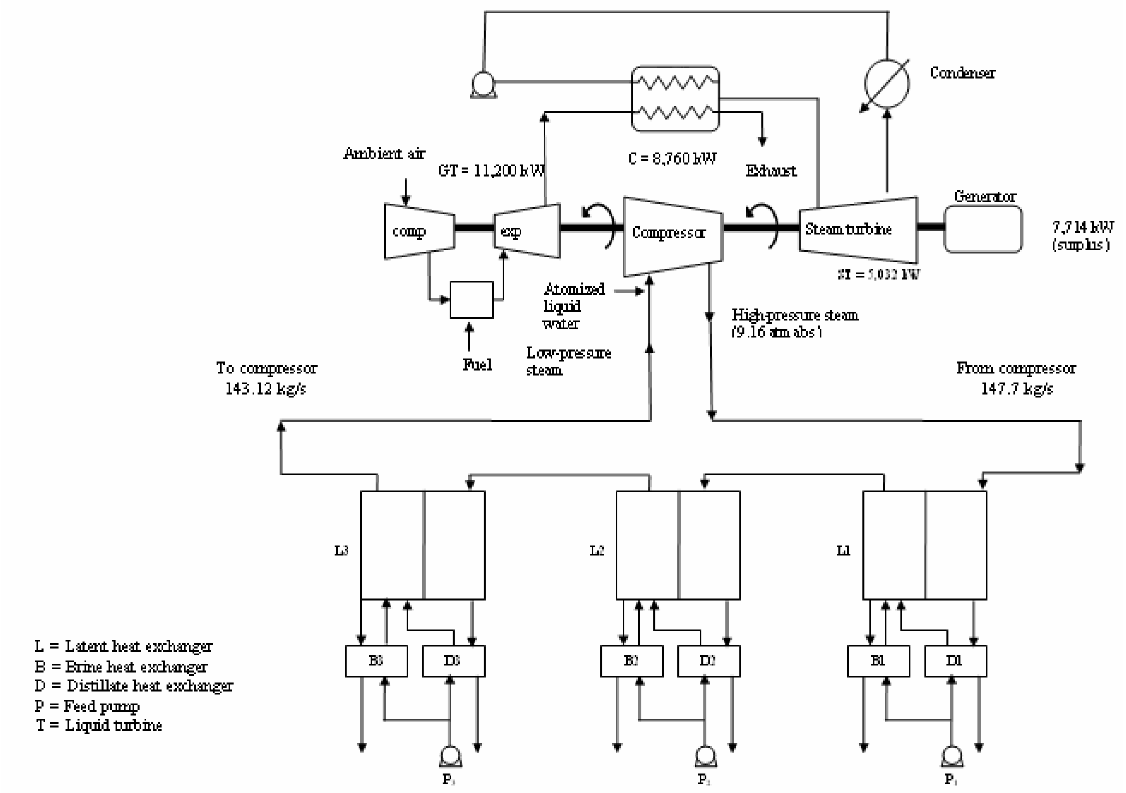
WebDopc, Boiling Point Problem Solutions, June 2005,

http://dbhs.wvusd.k12.ca.us/webdocs/Solutions/BP-Elev-Probs.html

APPENDIX A

SALT WATER PROPERTIES

The following data corresponds to an advanced vapor-compression desalination plant designed to produce 37,854 m^3/day (10 million gallons /day) of potable water.



L = Latent heat exchanger
B = Brine heat exchanger
D = Distillate heat exchanger
P = Feed pump
T = Liquid turbine

Figure A-1. Case B. Advanced vapor compressor desalination concept and combined-cycle power source.

Table A-1. Composition of seawater [33]

Ion	**Concentration in seawater (g/kg)**
Chloride	**19.344**
Sodium	**10.773**
Sulfate	**2.712**
Magnesium	**1.294**
Calcium	**0.412**
Potassium	**0.399**
Bicarbonate	**1.142**
Bromide	**0.067**
Strontium	**0.008**
Boron	**0.004**
Fluoride	**0.0013**
Total	**35.00**

Table A-2. Thermal conductivity of seawater and its concentrates (mW/(m · K)) [33]

	Salinity (g/kg)																
t (°C)	0[a]	10	20	30	35[b]	40	50	60	70	80	90	100	110	120	130	140	150
0	572	570	569	567	566	565	563	562	560	558	556	554	552	550	548	546	544
10	589	587	586	584	584	583	581	580	578	577	575	573	571	570	568	566	564
20	604	603	602	600	600	599	598	597	595	594	592	591	589	588	586	585	583
30	618	617	616	615	614	614	613	612	611	609	608	607	606	604	603	602	600
40	630	629	629	628	628	627	626	626	625	624	623	622	621	620	618	617	616
50	641	641	640	640	639	639	639	638	637	637	636	635	634	633	632	631	630
60	651	651	650	650	650	650	649	649	649	648	648	647	646	646	645	644	644
70	659	659	659	659	659	659	659	659	658	658	658	658	657	657	656	656	655
80	666	666	667	667	667	667	667	667	667	667	667	667	667	666	666	666	666
90	672	672	673	673	673	674	674	674	674	675	675	675	675	675	675	675	675
100	676	677	678	678	679	679	680	680	681	681	681	682	682	682	682	682	683
110	680	681	682	683	683	683	684	685	685	686	687	687	688	688	688	689	689
120	682	683	684	685	686	686	687	688	689	690	691	691	692	693	693	694	694
130	683	685	686	687	688	688	690	691	692	693	694	695	695	696	697	698	699
140	684	685	687	688	689	689	691	692	693	694	696	697	698	699	700	701	702
150	683	684	686	688	688	689	691	692	694	695	696	698	699	700	701	702	703
160	681	683	684	686	687	688	690	691	693	694	696	697	699	700	701	703	704
170	678	680	682	684	685	686	687	689	691	693	694	696	698	699	701	702	704
180	674	676	678	680	681	682	684	686	686	690	692	694	695	697	699	700	702

[a] While these values for pure water are within the claimed accuracy, more accurate values are available in the appropriate ESDU Data Item.

[b] “Normal” seawater

Table A-3. Dynamic viscosity of seawater and its concentrates (10^{-3}Ns/m^2) [33]

	Salinity (g/kg)															
t (°C)	**0[a]**	**10**	**20**	**30**	**40**	**50**	**60**	**70**	**80**	**90**	**100**	**110**	**120**	**130**	**140**	**150**
0	**1.775**	**1.802**	**1.831**	**1.861**	**1.893**	**1.928**	**1.965**	**2.005**	**2.049**	**2.096**	**2.147**	**2.202**	**2.261**	**2.326**	**2.395**	**2.470**
10	**1.304**	**1.327**	**1.350**	**1.375**	**1.401**	**1.429**	**1.459**	**1.491**	**1.526**	**1.563**	**1.603**	**1.646**	**1.693**	**1.743**	**1.797**	**1.855**
20	**1.002**	**1.021**	**1.041**	**1.061**	**1.083**	**1.106**	**1.131**	**1.157**	**1.185**	**1.216**	**1.248**	**1.283**	**1.321**	**1.361**	**1.404**	**1.451**
30	**0.797**	**0.814**	**0.830**	**0.848**	**0.866**	**0.886**	**0.906**	**0.929**	**0.952**	**0.977**	**1.004**	**1.033**	**1.064**	**1.098**	**1.133**	**1.171**
40	**0.653**	**0.667**	**0.681**	**0.696**	**0.712**	**0.729**	**0.747**	**0.765**	**0.786**	**0.807**	**0.830**	**0.845**	**0.880**	**0.908**	**0.938**	**0.970**
50	**0.546**	**0.559**	**0.571**	**0.585**	**0.599**	**0.613**	**0.629**	**0.645**	**0.662**	**0.681**	**0.700**	**0.721**	**0.744**	**0.768**	**0.793**	**0.821**
60	**0.466**	**0.477**	**0.488**	**0.500**	**0.512**	**0.525**	**0.539**	**0.533**	**0.568**	**0.584**	**0.602**	**0.620**	**0.639**	**0.660**	**0.682**	**0.706**
70	**0.404**	**0.414**	**0.424**	**0.434**	**0.445**	**0.457**	**0.469**	**0.481**	**0.495**	**0.509**	**0.524**	**0.540**	**0.558**	**0.576**	**0595**	**0.616**
80	**0.355**	**0.364**	**0.373**	**0.382**	**0.392**	**0.402**	**0.413**	**0.424**	**0.436**	**0.449**	**0.463**	**0.477**	**0.492**	**0.508**	**0.525**	**0.544**
90	**0.315**	**0.323**	**0.331**	**0.340**	**0.349**	**0.358**	**0.368**	**0.378**	**0.389**	**0.400**	**0.412**	**0.425**	**0.439**	**0.453**	**0.469**	**0.485**
100	**0.282**	**0.290**	**0.397**	**0.305**	**0.313**	**0.322**	**0.331**	**0.340**	**0.350**	**0.360**	**0.371**	**0.383**	**0.395**	**0.408**	**0.422**	**0.436**
110	**0.255**	**0.262**	**0.269**	**0.276**	**0.284**	**0.291**	**0.300**	**0.308**	**0.317**	**0.326**	**0.336**	**0.347**	**0.358**	**0.370**	**0.382**	**0.395**
120	**0.232**	**0.239**	**0.245**	**0.252**	**0.259**	**0.266**	**0.273**	**0.281**	**0.289**	**0.298**	**0.307**	**0.317**	**0.327**	**0.337**	**0.349**	**0.361**
130	**0.213**	**0.219**	**0.225**	**0.231**	**0.237**	**0.244**	**0.251**	**0.258**	**0.266**	**0.273**	**0.282**	**0.291**	**0.300**	**0.310**	**0.320**	**0.331**
140	**0.196**	**0.201**	**0.207**	**0.213**	**0.219**	**0.225**	**0.231**	**0.238**	**0.245**	**0.252**	**0.260**	**0.268**	**0.277**	**0.286**	**0.295**	**0.305**
150	**0.181**	**0.187**	**0.192**	**0.197**	**0.203**	**0.208**	**0.214**	**0.221**	**0.227**	**0.234**	**0.241**	**0.249**	**0.256**	**0.265**	**0.273**	**0.283**
160	**0.169**	**0.173**	**0.178**	**0.183**	**0.189**	**0.194**	**0.200**	**0.205**	**0.211**	**0.218**	**0.224**	**0.231**	**0.239**	**0.246**	**0.254**	**0.263**
170	**0.157**	**0.162**	**0.167**	**0.171**	**0.176**	**0.181**	**0.186**	**0.192**	**0.198**	**0.203**	**0.210**	**0.216**	**0.223**	**0.230**	**0.237**	**0.245**
180	**0.147**	**0.152**	**0.156**	**0.161**	**0.165**	**0.170**	**0.175**	**0.180**	**0.185**	**0.191**	**0.196**	**0.202**	**0.209**	**0.215**	**0.222**	**0.230**

Table A-3. Continued

	Salinity (g/kg)									
t (^{o}C)	30	31	32	33	34	35	36	37	38	39	40
0	1.861	1.864	1.867	1.871	1.874	1.877	1.880	1.883	1.887	1.890	1.893
10	1.375	1.377	1.380	1.382	1.385	1.388	1.390	1.393	1.396	1.398	1.401
20	1.061	1.063	1.065	1.068	1.070	1.072	1.074	1.076	1.078	1.081	1.083
30	0.848	0.850	0.851	0.853	0.855	0.857	0.859	0.861	0.862	0.864	0.866
40	0.696	0.698	0.699	0.701	0.702	0.704	0.706	0.707	0.709	0.710	0.712
50	0.585	0.586	0.587	0.589	0.590	0.592	0.593	0.594	0.596	0.597	0.599
60	0.500	0.501	0.503	0.504	0.505	0.506	0.507	0.509	0.510	0.511	0.512
70	0.434	0.435	0.437	0.438	0.439	0.440	0.441	0.442	0.443	0.444	0.445
80	0.382	0.383	0.384	0.385	0.386	0.387	0.388	0.389	0.390	0.391	0.392
90	0.340	0.341	0.342	0.343	0.343	0.344	0.345	0.349	0.347	0.348	0.349
100	0.305	0.306	0.307	0.308	0.308	0.309	0.310	0.311	0.312	0.312	0.313
110	0.276	0.277	0.278	0.278	0.279	0.280	0.281	0.281	0.282	0.283	0.284
120	0.252	0.252	0.253	0.254	0.254	0.255	0.256	0.257	0.257	0.258	0.259
130	0.231	0.231	0.232	0.233	0.233	0.234	0.235	0.235	0.236	0.237	0.237
140	0.213	0.213	0.214	0.215	0.215	0.216	0.216	0.217	0.218	0.218	0.219
150	0.197	0.198	0.198	0.199	0.199	0.200	0.200	0.201	0.202	0.202	0.203
160	0.183	0.184	0.184	0.185	0.186	0.186	0.187	0.187	0.188	0.188	0.189
170	0.171	0.172	0.172	0.173	0.173	0.174	0.174	0.175	0.175	0.176	0.176
180	0.161	0.161	0.161	0.162	0.162	0.163	0.163	0.164	0.164	0.165	0.165

[a] While these values for pure water are within the claimed accuracy, more accurate values are available in the appropriate ESDU Data Item

Table A-4. Heat capacity of seawater and its concentrates (kJ/(kg · K)) [33]

	Salinity (g/kg)															
t (°C)	**0[a]**	**10**	**20**	**30**	**40**	**50**	**60**	**70**	**80**	**90**	**100**	**110**	**120**	**130**	**140**	**150**
0	4.209	4.143	4.081	4.021	3.964	3.910	3.858	3.809	3.763	3.720	3.679	3.641	3.606	3.573	3.543	3.516
10	4.198	4.136	4.077	4.020	3.965	3.913	3.863	3.815	3.770	3.727	3.686	3.648	3.612	3.579	3.547	3.518
20	4.189	4.131	4.074	4.020	3.967	3.917	3.868	3.822	3.777	3.735	3.649	3.656	3.619	3.584	3.552	3.521
30	4.184	4.128	4.074	4.021	3.971	3.922	3.874	3.829	3.785	3.743	3.702	3.663	3.626	3.591	3.557	3.525
40	4.180	4.127	4.075	4.024	3.975	3.927	3.881	3.836	3.793	3.751	3.710	3.671	3.633	3.597	3.562	3.529
50	4.180	4.128	4.078	4.029	3.981	3.934	3.888	3.844	3.801	3.759	3.719	3.679	3.641	3.604	3.568	3.533
60	4.181	4.131	4.082	4.034	3.987	3.941	3.896	3.853	3.810	3.768	3.727	3.687	3.649	3.611	3.574	3.538
70	4.186	4.137	4.088	4.041	3.995	3.950	3.905	3.861	3.819	3.777	3.736	3.696	3.657	3.618	3.581	3.544
80	4.192	4.144	4.096	4.050	4.004	3.959	3.914	3.871	3.828	3.786	3.745	3.704	3.665	3.626	3.588	3.551
90	4.202	4.154	4.106	4.059	4.014	3.968	3.924	3.880	3.837	3.795	3.754	3.713	3.673	3.634	3.595	3.558
100	4.213	4.165	4.118	4.071	4.025	3.979	3.934	3.891	3.847	3.805	3.763	3.722	3.82	3.642	3.603	3.565
110	4.228	4.179	4.131	4.083	4.037	3.991	3.946	3.901	3.857	3.815	3.772	3.731	3.690	3.651	3.612	3.573
120	4.245	4.195	4.146	4.097	4.050	4.003	3.957	3.912	3.868	3.825	3.782	3.740	3.700	3.659	3.620	3.582
130	4.264	4.213	4.162	4.113	4.064	4.016	3.970	3.924	3.879	3.835	3.792	3.750	3.709	3.669	3.629	3.591
140	4.286	4.233	4.181	4.129	4.079	4.030	3.982	3.936	3.890	3.845	3.802	3.760	3.718	3.678	3.639	3.601
150	4.311	4.255	4.201	4.148	4.096	4.045	3.996	3.948	3.902	3.856	3.812	3.769	3.728	3.688	3.649	3.611
160	4.338	4.279	4.222	4.167	4.113	4.061	4.010	3.961	3.913	3.867	3.823	3.780	3.738	3.698	3.659	3.622
170	4.367	4.306	4.246	4.188	4.132	4.078	4.025	3.974	3.926	3.878	3.833	3.790	3.748	3.708	3.670	3.634
180	4.399	4.334	4.271	4.210	4.152	4.095	4.041	3.988	3.938	3.890	3.844	3.800	3.758	3.719	3.681	3.646

Table A-4. Continued

	Salinity (g/kg)										
t (°C)	**30**	**31**	**32**	**33**	**34**	**35**	**36**	**37**	**38**	**39**	**40**
0	**4.021**	**4.015**	**4.010**	**4.004**	**3.998**	**3.992**	**3.987**	**3.981**	**3.975**	**3.970**	**3.964**
10	**4.020**	**4.014**	**4.009**	**4.003**	**3.998**	**3.992**	**3.987**	**3.981**	**3.976**	**3.971**	**3.965**
20	**4.020**	**4.015**	**4.009**	**4.004**	**3.999**	**3.993**	**3.988**	**3.983**	**3.978**	**3.973**	**3.967**
30	**4.021**	**4.016**	**4.011**	**4.006**	**4.001**	**3.996**	**3.991**	**3.986**	**3.981**	**3.976**	**3.971**
40	**4.024**	**4.019**	**4.014**	**4.009**	**4.004**	**4.000**	**3.995**	**3.990**	**3.985**	**3.980**	**3.975**
50	**4.029**	**4.024**	**4.019**	**4.014**	**4.009**	**4.004**	**4.000**	**3.995**	**3.990**	**3.985**	**3.981**
60	**4.034**	**4.029**	**4.025**	**4.020**	**4.015**	**4.011**	**4.006**	**4.001**	**3.997**	**3.992**	**3.987**
70	**4.041**	**4.037**	**4.032**	**4.027**	**4.023**	**4.018**	**4.013**	**4.009**	**4.004**	**4.000**	**3.995**
80	**4.050**	**4.045**	**4.040**	**4.036**	**4.031**	**4.027**	**4.022**	**4.017**	**4.013**	**4.008**	**4.004**
90	**4.059**	**4.055**	**4.050**	**4.046**	**4.041**	**4.036**	**4.032**	**4.027**	**4.023**	**4.018**	**4.014**
100	**4.071**	**4.066**	**4.061**	**4.057**	**4.052**	**4.048**	**4.043**	**4.038**	**4.034**	**4.029**	**4.025**
110	**4.083**	**4.079**	**4.074**	**4.069**	**4.065**	**4.060**	**4.055**	**4.051**	**4.046**	**4.041**	**4.037**
120	**4.097**	**4.092**	**4.088**	**4.083**	**4.078**	**4.073**	**4.069**	**4.064**	**4.059**	**4.054**	**4.050**
130	**4.113**	**4.108**	**4.103**	**4.098**	**4.093**	**4.088**	**4.083**	**4.078**	**4.074**	**4.069**	**4.064**
140	**4.129**	**4.124**	**4.119**	**4.114**	**4.109**	**4.104**	**4.099**	**4.094**	**4.089**	**4.084**	**4.079**
150	**4.148**	**4.142**	**4.137**	**4.132**	**4.127**	**4.121**	**4.116**	**4.111**	**4.106**	**4.101**	**4.096**
160	**4.167**	**4.162**	**4.156**	**4.151**	**4.145**	**4.140**	**4.135**	**4.129**	**4.124**	**4.119**	**4.113**
170	**4.188**	**4.182**	**4.177**	**4.171**	**4.165**	**4.160**	**4.154**	**4.149**	**4.143**	**4.137**	**4.132**
180	**4.120**	**4.204**	**4.198**	**4.192**	**4.187**	**4.181**	**4.175**	**4.169**	**4.163**	**4.157**	**4.152**

[a] **While these values for pure water are within the claimed accuracy, more accurate values are available in the appropriate ESDU Data Item**

Table A-5. Density of seawater and its concentrates (kg/m^3) [33]

	Salinity (g/kg)															
t (^{o}C)	**0[a]**	**10**	**20**	**30**	**40**	**50**	**60**	**70**	**80**	**90**	**100**	**110**	**120**	**130**	**140**	**150**
0	999.8	1008.1	1016.2	1024.2	1032.0	1039.8	1047.6	1055.5	1063.5	1071.6	1079.7	1088.0	1096.2	1104.4	1112.5	1120.4
10	999.7	1007.7	1015.5	1023.2	1.030.2	1038.4	1046.0	1053.8	1061.6	1669.6	1077.6	1085.7	1093.9	1102.0	1110.1	1118.0
20	998.4	1005.8	1013.3	1020.8	1028.3	1035.9	1043.5	1051.2	1058.9	1066.7	1074.5	1082.4	1090.3	1098.2	1106.2	1114.2
30	995.4	1002.8	1010.2	1017.6	1025.1	1032.6	1040.2	1047.8	1055.4	1063.1	1070.8	1078.5	1086.3	1094.1	1102.0	1109.9
40	991.9	999.2	1006.6	1013.9	1021.4	1028.8	1036.3	1043.8	1051.4	1059.0	1066.6	1074.2	1081.9	1089.6	1091.4	1105.2
50	987.7	995.0	1002.3	1009.7	1017.1	1024.5	1031.9	1039.4	1046.9	1054.4	1062.0	1069.5	1077.1	1084.8	1092.4	1100.1
60	982.9	990.2	997.5	1004.9	1012.2	1019.6	1027.0	1034.5	1041.9	1049.4	1056.9	1064.4	1072.0	1079.5	1087.1	1094.8
70	977.6	984.9	992.2	999.5	1006.9	1014.3	1021.7	1029.1	1036.5	1043.9	1051.4	1058.9	1066.4	1074.0	1081.5	1089.1
80	971.7	979.0	986.4	993.7	1001.1	1008.4	1015.8	1023.2	1030.6	1038.1	1045.5	1053.0	1060.5	1068.0	1075.6	1083.1
90	965.3	972.7	980.0	987.4	994.7	1002.1	1009.5	1017.0	1024.4	1031.8	1039.3	1046.8	1054.3	1061.8	1069.3	1076.8
100	958.4	965.8	973.2	980.6	988.0	995.4	1002.8	1010.3	1017.7	1025.2	1032.7	1040.2	1047.7	1055.2	1062.7	1070.3
110	951.0	958.5	965.9	973.3	980.8	988.3	995.7	1003.2	1010.7	1018.2	1025.7	1033.2	1040.8	1048.3	1055.9	1063.4
120	943.2	950.7	958.2	965.7	973.2	980.7	988.2	995.8	1003.3	1010.9	1018.4	1026.0	1033.6	1041.2	1048.7	1056.3
130	934.8	942.4	950.0	957.6	965.2	972.8	980.4	988.0	995.6	1003.2	1010.8	1018.5	1026.1	1033.7	1041.3	1049.0
140	926.1	933.8	941.4	949.1	956.8	964.5	972.2	979.9	987.6	995.2	1002.9	1010.6	1018.3	1026.0	1033.7	1041.4
150	916.9	924.7	932.5	940.3	948.1	955.9	963.7	971.4	979.2	987.0	994.8	1002.5	1010.3	1018.0	1025.8	1033.6
160	907.3	915.2	923.2	931.1	939.0	946.9	954.8	962.7	970.6	978.5	986.3	994.2	1002.0	1009.9	1017.7	1025.5
170	897.3	905.4	913.5	921.6	929.6	937.7	945.7	953.7	961.7	969.7	977.6	985.6	993.5	1001.4	1009.3	1017.2
180	887.0	895.3	903.5	911.7	919.9	928.1	936.3	944.4	952.6	960.7	968.7	976.8	984.8	992.8	1000.8	1008.7

Table A-5. Continued

	Salinity (g/kg)										
t (^{o}C)	**30**	**31**	**32**	**33**	**34**	**35**	**36**	**37**	**38**	**39**	**40**
0	**1024.2**	**1024.9**	**1025.7**	**1026.5**	**1027.3**	**1028.1**	**1028.9**	**1029.6**	**1030.4**	**1031.2**	**1032.0**
10	**1023.2**	**1023.9**	**1024.7**	**1025.4**	**1026.2**	**1027.0**	**1027.7**	**1028.5**	**1029.3**	**1030.0**	**1030.8**
20	**1020.8**	**1021.5**	**1022.3**	**1023.0**	**1023.8**	**1024.5**	**1025.3**	**1026.0**	**1026.8**	**1027.5**	**1028.3**
30	**1017.6**	**1018.4**	**1019.1**	**1019.9**	**1020.6**	**1021.4**	**1022.1**	**1022.9**	**1023.6**	**1024.4**	**1025.1**
40	**1013.9**	**1014.7**	**1015.4**	**1016.2**	**1016.9**	**1017.7**	**1018.4**	**1019.1**	**1019.9**	**1020.6**	**1021.4**
50	**1009.7**	**1010.4**	**1011.2**	**1011.9**	**1012.6**	**1013.4**	**1014.1**	**1014.8**	**1015.6**	**1016.3**	**1017.1**
60	**1004.9**	**1005.6**	**1006.3**	**1007.1**	**1007.8**	**1008.6**	**1009.3**	**1010.0**	**1010.8**	**1011.5**	**1012.2**
70	**999.5**	**1000.3**	**1001.0**	**1001.7**	**1002.5**	**1003.2**	**1003.9**	**1004.7**	**1005.4**	**1006.2**	**1006.9**
80	**993.7**	**994.4**	**995.2**	**995.9**	**996.6**	**997.4**	**998.1**	**998.8**	**999.6**	**1000.3**	**1001.1**
90	**987.4**	**988.1**	**988.8**	**989.6**	**990.3**	**991.1**	**991.8**	**992.5**	**993.3**	**994.0**	**994.7**
100	**980.6**	**981.3**	**982.1**	**982.8**	**983.5**	**984.3**	**985.0**	**985.8**	**986.5**	**987.2**	**988.0**
110	**973.3**	**974.1**	**974.8**	**975.6**	**976.3**	**977.1**	**997.8**	**978.6**	**979.3**	**980.0**	**980.8**
120	**965.7**	**966.4**	**967.2**	**967.9**	**968.7**	**969.4**	**970.2**	**970.9**	**971.7**	**972.4**	**973.2**
130	**957.6**	**958.4**	**959.1**	**959.9**	**960.6**	**961.4**	**962.1**	**962.9**	**963.7**	**964.4**	**965.2**
140	**949.1**	**949.9**	**950.7**	**951.4**	**952.2**	**953.0**	**953.7**	**954.5**	**955.3**	**956.0**	**956.8**
150	**940.3**	**941.1**	**941.8**	**942.6**	**943.4**	**944.2**	**945.0**	**945.7**	**946.5**	**947.3**	**948.1**
160	**931.1**	**931.9**	**932.7**	**933.5**	**934.4**	**935.1**	**935.8**	**936.6**	**937.4**	**938.2**	**939.0**
170	**921.6**	**922.4**	**923.7**	**924.0**	**924.8**	**925.6**	**926.4**	**927.2**	**928.0**	**928.8**	**929.6**
180	**911.7**	**912.6**	**913.4**	**914.2**	**915.0**	**915.8**	**916.7**	**917.5**	**918.3**	**919.1**	**919.9**

[a] While these values for pure water are within the claimed accuracy, more accurate values are available in the appropriate ESDU Data Item

Table A-6. Prandtl number of seawater and its concentrates [33]

Salinity (g/kg)																
t (^{o}C)	**0[a]**	**10**	**20**	**30**	**35[b]**	**40**	**50**	**60**	**70**	**80**	**90**	**100**	**110**	**120**	**130**	**140**	**150**
0	**13.1**	**13.1**	**13.1**	**13.2**	**13.2**	**13.3**	**13.4**	**13.5**	**13.6**	**13.8**	**14.0**	**14.3**	**14.5**	**14.8**	**15.2**	**15.5**	**16.0**
10	**9.29**	**9.35**	**9.39**	**9.46**	**9.49**	**9.53**	**9.62**	**9.72**	**9.84**	**9.97**	**10.1**	**10.3**	**10.5**	**10.7**	**11.0**	**11.2**	**11.6**
20	**6.95**	**6.99**	**7.04**	**7.11**	**7.13**	**7.17**	**7.24**	**7.33**	**7.43**	**7.53**	**7.67**	**7.80**	**7.96**	**8.13**	**8.32**	**8.52**	**8.76**
30	**5.40**	**5.45**	**5.49**	**5.54**	**5.58**	**5.60**	**5.67**	**5.74**	**5.82**	**5.92**	**6.01**	**6.12**	**6.24**	**6.39**	**6.54**	**6.69**	**6.88**
40	**4.33**	**4.38**	**4.41**	**4.46**	**4.48**	**4.51**	**4.57**	**4.63**	**4.70**	**4.78**	**4.86**	**4.95**	**5.05**	**5.16**	**5.28**	**5.42**	**5.56**
50	**3.56**	**3.60**	**3.64**	**3.68**	**3.71**	**3.73**	**3.77**	**3.83**	**3.89**	**3.95**	**4.02**	**4.10**	**4.18**	**4.28**	**4.38**	**4.48**	**4.60**
60	**2.99**	**3.03**	**3.06**	**3.10**	**3.12**	**3.14**	**3.19**	**3.24**	**3.28**	**3.34**	**3.40**	**3.47**	**3.54**	**3.61**	**3.69**	**3.78**	**3.88**
70	**2.57**	**2.60**	**2.63**	**2.66**	**2.68**	**2.70**	**2.74**	**2.78**	**2.82**	**2.87**	**2.92**	**2.98**	**3.04**	**3.11**	**3.18**	**3.25**	**3.33**
80	**2.23**	**2.26**	**2.29**	**2.32**	**2.34**	**2.35**	**2.39**	**2.42**	**2.46**	**2.50**	**2.55**	**2.60**	**2.65**	**2.71**	**2.77**	**2.83**	**2.90**
90	**1.97**	**2.00**	**2.02**	**2.05**	**2.06**	**2.08**	**2.11**	**2.14**	**2.18**	**2.21**	**2.25**	**2.29**	**2.34**	**2.39**	**2.44**	**2.50**	**2.56**
100	**1.75**	**1.78**	**1.80**	**1.83**	**1.84**	**1.86**	**1.88**	**1.92**	**1.94**	**1.98**	**2.01**	**2.05**	**2.09**	**2.13**	**2.18**	**2.23**	**2.28**
110	**1.59**	**1.61**	**1.63**	**1.65**	**1.66**	**1.68**	**1.70**	**1.73**	**1.75**	**1.78**	**1.81**	**1.84**	**1.88**	**1.92**	**1.96**	**2.00**	**2.05**
120	**1.44**	**1.47**	**1.49**	**1.51**	**1.51**	**1.53**	**1.55**	**1.57**	**1.60**	**1.62**	**1.65**	**1.68**	**1.71**	**1.75**	**1.78**	**1.82**	**1.86**
130	**1.33**	**1.35**	**1.37**	**1.38**	**1.39**	**1.40**	**1.42**	**1.44**	**1.46**	**1.49**	**1.51**	**1.54**	**1.57**	**1.60**	**1.63**	**1.66**	**1.70**
140	**1.23**	**1.24**	**1.26**	**1.28**	**1.29**	**1.30**	**1.31**	**1.33**	**1.35**	**1.37**	**1.39**	**1.42**	**1.44**	**1.47**	**1.50**	**1.53**	**1.56**
150	**1.14**	**1.16**	**1.18**	**1.19**	**1.20**	**1.21**	**1.22**	**1.24**	**1.26**	**1.27**	**1.30**	**1.32**	**1.34**	**1.36**	**1.39**	**1.42**	**1.45**
160	**1.08**	**1.08**	**1.10**	**1.11**	**1.12**	**1.13**	**1.14**	**1.16**	**1.17**	**1.19**	**1.21**	**1.23**	**1.25**	**1.28**	**1.30**	**1.32**	**1.35**
170	**1.01**	**1.03**	**1.04**	**1.05**	**1.06**	**1.06**	**1.07**	**1.09**	**1.10**	**1.12**	**1.132**	**1.16**	**1.17**	**1.20**	**1.22**	**1.24**	**1.26**
180	**0.959**	**0.975**	**0.983**	**0.997**	**1.00**	**1.00**	**1.02**	**1.03**	**1.04**	**1.06**	**1.07**	**1.09**	**1.10**	**1.13**	**1.14**	**1.17**	**1.19**

[a] While these values for pure water are within the claimed accuracy, more accurate values are available in the appropriate ESDU Data Item

[b] "Normal" seawater

Factors affecting the evaporator performance

A-2 Boiling point elevation

Having measured the vapor pressure of seawater at a variety of concentrations and temperatures, Emerson and Jamieson [32] used the NEL steam tables (1964) to evaluate the boiling point elevation (see Table A-7). In Table A-7, the boiling point elevation increases with salinity (g/kg) and temperature.

It should be remarked that the vapor coming from the boiling solution in the evaporator is at the solution temperature [32] and is therefore superheated by the amount of the boiling point elevation.

A-3 Seawater vapor-pressure

Table A-6 shows the measured vapor pressures of saline solutions as given by Emerson and Jamieson [34]. The last column shows values of the calculated boiling point elevation. Note that the values correspond to a salinity of up to 8 times the normal water salinity.

Table A-7. Measured seawater vapor pressure and boiling point elevation [34]

MEASUREMENT OF VAPOUR PRESSURE OF SALINE SOLUTIONS								
Sample No.	**Salinity (g/kg)**	**Temperature (^oC)**	**Measured pressure (p) (10^5 N/m^2)**	**Pressure of pure water (p_0) (10^5 N/m^2)**	**Activity p/p_0**	**Pressure (10^5 N/m^2)**	**$\frac{p' - p}{p}$ (%)^a**	**Boiling-point elevation at measured pressure (^oC)**
1	33.13	100.392	1.0098	1.0275	0.98277	1.0098	0.002	0.49
	33.21	119.686	1.9317	1.9657	0.98270	1.9317	0.002	0.55
	33.32	136.705	3.2318	3.2890	0.98261	3.2320	0.007	0.61
	33.56	159.998	6.0698	6.1801	0.98215	6.0723	0.041	0.70
	33.88	180.315	9.9134	10.1000	0.98152	9.9219	0.086	0.81
2	33.43	99.887	0.9922	1.0092	0.98315	0.9918	-0.041	0.48
	33.21	120.155	1.9608	1.9951	0.98281	1.9607	-0.007	0.55
	33.32	136.542	3.2162	3.2736	0.98247	3.2169	0.001	0.61
	33.54	159.808	6.0410	6.1503	0.98223	6.0430	0.002	0.70
	33.84	177.948	9.3850	9.5624	0.98145	9.3940	0.009	0.81
3	66.26	101.970	1.0484	1.0866	0.96484	1.0477	-0.001	1.01
	66.41	119.422	1.8806	1.9493	0.96476	1.8792	-0.001	1.13
	66.66	137.635	3.2570	3.3781	0.96415	3.2562	-0.001	1.27
	67.09	158.705	5.7608	5.9795	0.96343	5.7622	0.001	1.45
	67.72	179.031	9.4414	9.8055	0.96287	9.4456	0.004	1.63
4	115.97	101.776	1.0074	1.0792	0.93347	1.0077	0.030	1.94
	116.24	119.748	1.8385	1.9695	0.93349	1.8387	0.012	2.16
	116.66	138.059	3.1907	3.4194	0.93312	3.1933	0.020	2.41
	117.44	159.583	5.7069	6.1152	0.93323	5.7043	-0.046	2.69
	118.61	180.531	9.4616	10.1502	0.93216	9.4606	-0.011	3.03
5	116.27	121.250	1.9289	2.0652	0.93400	1.9280	-0.045	2.16
	116.67	138.354	3.2181	3.4483	0.93324	3.2183	0.008	2.39
	117.49	160.463	5.8357	6.2536	0.93317	8.8332	-0.042	2.71
	118.69	181.240	9.6170	10.3164	0.93321	9.6150	-0.021	3.04
6	165.68	102.800	1.0059	1.1188	0.89909	1.0077	0.182	3.01
	166.03	119.030	1.7322	1.9251	0.89980	1.7335	0.075	3.29
	166.68	137.520	3.0285	3.3670	0.89947	3.0304	0.062	3.66
	167.70	158.330	5.3304	5.9222	0.90007	5.3260	-0.082	4.06
	169.27	178.760	8.7678	9.7442	0.89980	8.7526	-0.174	4.50

[a] Standard deviation 0.069 per cent.

APPENDIX B

EQUIPMENT DESIGN

Sample calculation

Figure B-1 shows the nomenclature used to identify each stage of the evaporator. Calculations showing the temperature and pressure of Points A, B, and I are shown on the next few pages. Temperatures and pressure drops for the sensible heat exchanger are obtained from Appendix C. The enthalpy of vaporization from Points A to B is obtained from steam tables.

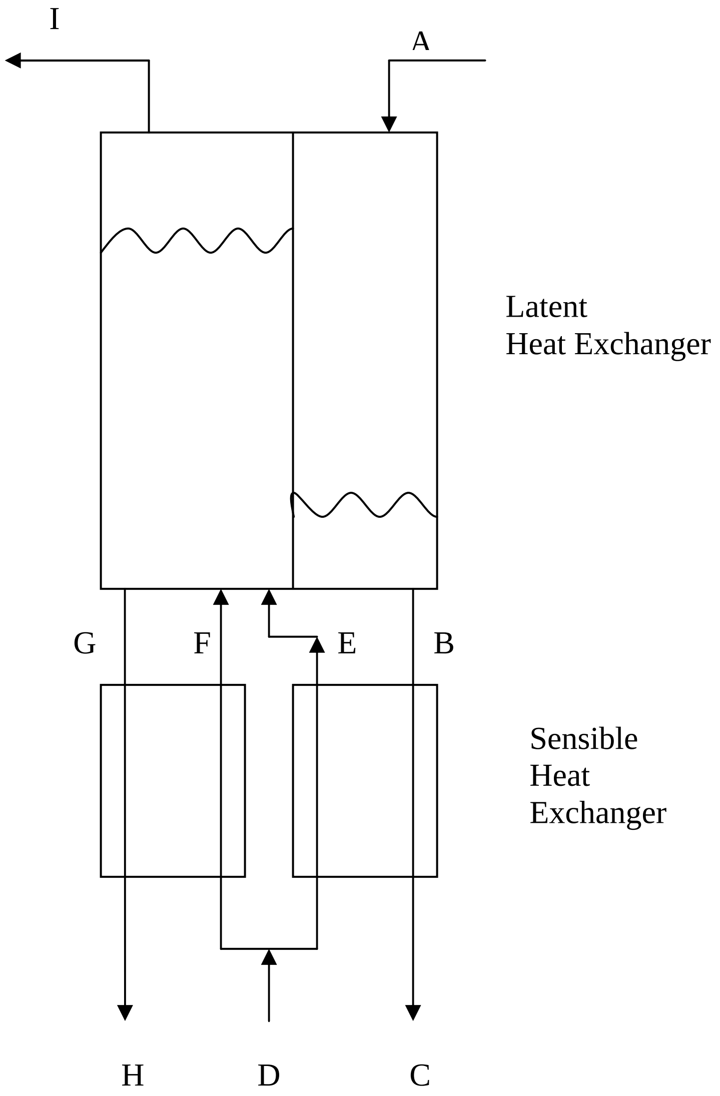


Figure B-1. Exiting and entering flows nomenclature of the sensible heat exchanger.

Table B-1. Case B. Individual temperatures and pressures along the evaporator train

Stage	Point (see Figure (B-1)	Flow	Calculated Vapor Pressure		Temperature
		(kg/s)	(atm)$_{abs}$	(kN/m^2)	(K)
1	A	147.5	9.16	928.36	449.86
	B	147.5	9.16	928.36	449.86
	C	147.5			
	D	295			
	E	147.5			
	F	147.5			
	G	147.5	8.3658		447.638
	H	147.5			301
	I	146.03	8.3658	691.8	447.638
2	A	146.03	8.3658	691.8	447.638
	B	146.03	8.3658	691.8	446.004
	C	146.03			
	D	292.06			
	E	146.03			
	F	146.03			
	G	146.03	7.6303	641.8	443.782
	H	146.03			
	I	144.57	7.6303	641.8	443.782
3	A	144.57	7.6303	641.8	443.782
	B	144.57	7.6303	641.8	442.179
	C	144.57			
	D	289.14			
	E	144.57			
	F	144.57			
	G	144.57	6.9523	704.6	439.95
	H	144.57			
	I	143.12	6.9523	704.6	439.95

Table B-2 summarizes the results of a numerical trade-off performed to find the best heat transfer coefficient corresponding to the optimal temperature drop across the latent heat exchanger.

Table B-2. Calculated overall heat transfer coefficients $\left(\mathrm{Btu}/\left(\mathrm{h} \cdot \mathrm{ft}^2 \cdot {}^\circ\mathrm{F}\right)\right)$. It Corresponds to different condensing temperatures and different heat transfer temperature differentials ΔT between the condensing steam side and the boiling liquid side

Condensing Temperature (°F)	Overall heat transfer coefficient. ΔT = 4°F	Overall heat transfer coefficient ΔT = 6°F	Overall heat transfer coefficient ΔT = 8°F	Overall heat transfer coefficient ΔT=10°F
40	372	372	371	371
60	695	695	694	694
80	1224	1224	1221	1221
100	2046	2048	2037	2039
120	3265	3269	3238	3244
140	4995	5004	4924	4939
160	7351	7374	7178	7214
180	10440	10493	10044	10126
200	14331	14448	13483	13654
220	19020	19261	17322	17656
240	24415	24888	21251	21850
260	30275	31150	24814	25799
280	36852	38253	28636	30054
300	41586	44041	28945	30935
320	45804	49478	28957	31402
340	48484	53585	27771	30532

APPENDIX C

PRESSURE DROP IN THE LATENT HEAT EXCHANGER

(First Stage)

For saturated steam at $340\,^{\circ}\mathrm{F}$ $(178^{\circ}\mathrm{C})$:

$$\rho_v = \text{ vapor density } = 0.264\,\frac{\mathrm{lb_m}}{\mathrm{ft^3}}$$

$$\rho_l = \text{ liquid density} = 55.95\,\frac{\mathrm{lb_m}}{\mathrm{ft^3}}$$

$$\mu_l = \text{ liquid dynamic viscosity = 0.393 } \mathrm{lb_m}/(\mathrm{h}\cdot\mathrm{ft})$$

$$\mu_v = \text{ vapor dynamic viscosity= 0.066 } \mathrm{lb_m}/(\mathrm{h}\cdot\mathrm{ft})$$

$$x = \text{mass fraction of vapor} = 0.999$$

Then, substituting into Equation 3.8:

$$\lambda = \frac{1}{\left(1+\dfrac{0.999}{(1-0.999)}\left(\dfrac{0.264\,\dfrac{\mathrm{lb_m}}{\mathrm{ft^3}}}{55.95\,\dfrac{\mathrm{lb_m}}{\mathrm{ft^3}}}\right)\right)} = 0.999$$

From Equation 3.4:

$$\alpha(\lambda) = 1 - \frac{\ln 0.999}{\left[1.281 + 0.478\ln 0.999 + 0.444(\ln 0.999)^2 + 0.094(\ln 0.999)^3 + 0.00843(\ln 0.999)^4\right]}$$
$$= 1.00031$$

Substituting the homogeneous model constants into Equation 3.9:

$$\Psi = \frac{1}{1+1\times\left(\frac{1-0.999}{0.999}\right)^{1}\left(\frac{0.264\,\frac{\mathrm{lb_m}}{\mathrm{ft^3}}}{55.95\,\frac{\mathrm{lb_m}}{\mathrm{ft^3}}}\right)^{1}\left(\frac{0.393\,\frac{\mathrm{lb}}{\mathrm{h\cdot ft}}}{0.066\,\frac{\mathrm{lb}}{\mathrm{h\cdot ft}}}\right)^{0}} = 0.9999$$

From Equations 3.6 and 3.9:

$$\rho_{NS} = \left(55.95\,\frac{\mathrm{lb_m}}{\mathrm{ft^3}}\right)(0.999) + 0.264\,\frac{\mathrm{lb_m}}{\mathrm{ft^3}}\,(1-0.9996) = 55.92\,\frac{\mathrm{lb_m}}{\mathrm{ft^3}}$$

$$\mu_{NS} = \left(0.393\,\frac{\mathrm{lb_m}}{\mathrm{h\cdot ft}}\right)(0.999) + 0.066\,\frac{\mathrm{lb_m}}{\mathrm{h\cdot ft}}\,(1-0.393) = 0.3928\,\frac{\mathrm{lb_m}}{\mathrm{h\cdot ft}}$$

From Equation 3.5:

$$\beta = \frac{55.95\,\frac{\mathrm{lb_m}}{\mathrm{ft^3}}}{55.92\,\frac{\mathrm{lb_m}}{\mathrm{ft^3}}}\times\frac{(0.999)^2}{(1-0.999)} + \frac{0.264\,\frac{\mathrm{lb_m}}{\mathrm{ft^3}}}{55.92\,\frac{\mathrm{lb_m}}{\mathrm{ft^3}}}\times\frac{(1-0.999)^2}{0.9457} = 18.3999$$

The hydraulic diameter D_h is

$$D_h = \frac{4A}{P}$$

where

$A = \text{cross sectional area} = 0.028\,\mathrm{ft}\times 8\,\mathrm{ft} = 0.224\,\mathrm{ft}^2$

$P = \text{wetted perimeter} = 16.06\,\mathrm{ft}$

Therefore

$$D_h = \frac{4\times 0.224\,\mathrm{ft}^2}{16.04\,\mathrm{ft}} = 0.055\,\mathrm{ft}$$

Then, the mass flow rate m_t is:

$$m_t = 0.264\frac{\mathrm{lb_m}}{\mathrm{ft}^3}\times 5\frac{\mathrm{ft}}{\mathrm{s}}\times 0.224\,\mathrm{ft}^2 = 0.295\frac{\mathrm{lb_m}}{\mathrm{s}}$$

From Equation 3.3:

$$f_\circ = 0.0014 + 0.125\left(\frac{4\times 0.295\times 18.4}{\pi\times 0.055\times 0.393}\right)^{-0.32} = 0.02114$$

From Equation 3.2

$$\left(\frac{dP}{dz}\right)_f = -\frac{2G^2 f_\circ \alpha\left(\lambda\right)\beta}{D_h\rho_{NS}}$$

= Pressure gradient per unit length

$$= \frac{2\rho_v^2 V^2 f_o \alpha(\lambda)\beta}{D_k \rho_{NS}}\left(\frac{\mathrm{lb_f}}{\mathrm{ft}^2}\right) \text{ per ft}$$

$$= \frac{2\times\left(5\frac{\mathrm{ft}}{\mathrm{s}}\right)^2\times\left(0.264\frac{\mathrm{lb_m}}{\mathrm{ft}^3}\right)^2\times 0.02114\times 1.00031\times 18.4}{0.055\,\mathrm{ft}\times 32.2\frac{\mathrm{lb_m}\cdot\mathrm{ft}}{\mathrm{lb_f}\cdot\mathrm{s}^2}\times 55.95\frac{\mathrm{lb_m}}{\mathrm{ft}^3}}$$

$$= 0.013\frac{\mathrm{lb_f}}{\mathrm{ft}^2}\times\frac{1\,\mathrm{ft}^2}{144\,\mathrm{in}^2}\text{ per ft of length} = 9.502\times 10^{-5}\frac{\mathrm{lb_f}}{\mathrm{in}^2}\frac{1}{\mathrm{ft}}$$

Then, for a 8-ft plate:

$$\left(\frac{dP}{dz}\right)_f = 0.0008 \text{ psi}$$

The pressure drop due to momentum is calculated following Reference [25] as

$$= \left(0.264\frac{\mathrm{lb_m}}{\mathrm{ft^3}}\right)^2 \times \left(5\frac{\mathrm{ft}}{\mathrm{s}}\right)^2 \times \frac{1}{\mathrm{ft}} \times \left\{ \begin{array}{l} \dfrac{2\times 0.999}{0.264\dfrac{\mathrm{lb_m}}{\mathrm{ft^3}}\times 0.999} - \dfrac{2(1-0.999)}{55.95\dfrac{\mathrm{lb_m}}{\mathrm{ft^3}}(1-0.999)} + 1\times \\ \left[\dfrac{0.9994\times(1-0.999)}{0.9994-0.999\times 55.95\dfrac{\mathrm{lb_m}}{\mathrm{ft^3}}}\right] - \dfrac{0.999(1-0.999)}{0.999(1-0.999)\times 0.264\dfrac{\mathrm{lb_m}}{\mathrm{ft^3}}} \end{array} \right\}$$

$$= 6.83\frac{\mathrm{lb_m}}{\mathrm{ft}\cdot\mathrm{s}^2} \times \frac{1}{32.2\dfrac{\mathrm{lb_m}\cdot\mathrm{ft}}{\mathrm{lb_f}\cdot\mathrm{s}^2}} \times \frac{1}{\mathrm{ft}} = 0.2123\frac{\mathrm{lb_f}}{\mathrm{ft}^2} \times \frac{1\,\mathrm{ft}^2}{144\,\mathrm{in}^2} \times \frac{1}{\mathrm{ft}} = 0.0014\,\frac{\mathrm{psi}}{\mathrm{ft}}$$

Then for a 8-ft plate:

$$\left(\frac{dP}{dz}\right)_m = 0.011 \text{ psi}$$

Substituting into Equation 3.1:

$$\frac{dP}{dz} = \left(\frac{dP}{dz}\right)_{friction} + \left(\frac{dP}{dz}\right)_{momentum} \begin{array}{l} = 0.0008\,\mathrm{psi} + 0.011\,\mathrm{psi} \\ = 0.011\,\mathrm{psi\ per\ plate\ channel} \end{array}$$

ΔP due to a sudden channel enlargement when the steam flow is coming out of the channel before it turns 180° and bends towards a sudden contraction when it comes back to the plate channel.

v_g = vapor specific volume $= 3.7878\,\mathrm{ft^3/lb_m}$

v_f = fluid specific volume $= 0.017878\,\mathrm{ft^3/lb_m}$

Chisholm [31] recommends the following values for C_2 and λ:

$$C_2 = 1 + 20\left(\frac{D}{L}\right)$$

$\lambda = 1$

$D = 0.0282 \text{ ft}$

$L = 8 \text{ ft}$

$\frac{D}{L} = 0.0035$

$C_2 = 1.07$

$v_{fg} = 3.7699 \frac{\text{ft}^3}{\text{lb}_\text{m}}$

Then

$$C = \left[1 + \left(1.07 - 1\right)\left(\frac{3.7699 \frac{\text{ft}^3}{\text{lb}_\text{m}}}{3.7878 \frac{\text{ft}^3}{\text{lb}_\text{m}}}\right)^{0.5}\right] \cdot \left[\left(\frac{3.7878 \frac{\text{ft}^3}{\text{lb}_\text{m}}}{0.01787 \frac{\text{ft}^3}{\text{lb}_\text{m}}}\right)^{0.5} 0.5 + \left(\frac{0.01787 \frac{\text{ft}^3}{\text{lb}_\text{m}}}{3.7878 \frac{\text{ft}^3}{\text{lb}_\text{m}}}\right)^{0.5}\right]$$

$$= 17.38$$

$$X = \left[\left(\frac{0.999}{1 - 0.999}\right)\left(\frac{3.7878}{0.01787}\right)0.5\right]^{-1} = 6.9 \times 10^{-4}$$

Substituting into Equation 3.11:

$$\frac{\Delta P_{TP}}{\Delta P_f} = 1 + \left(17.38\right)\left(6.9 \times 10^{-4}\right) + \left(\times 6.9 \times 10^{-4}\right)^2 = 1.011$$

$$\Delta P_f = K\left(\frac{\rho_f}{g_c}\right)\left(\frac{V^2}{2}\right)$$

where

$K = 0.30$ for a flanged welded bend [25]

$V =$ fluid velocity $= 5$ ft/s

ρ_f = fluid density $= 55.95\ \mathrm{lb_m/ft^3}$

$g_c = 32.2\ \mathrm{lb_m \cdot ft/(lb_{f.} \cdot s^2)}$

$$\Delta P_f = \frac{0.30 \times 55.95 \frac{\mathrm{lb_m}}{\mathrm{ft}^3} \times \left(5 \frac{\mathrm{ft}}{\mathrm{s}}\right)^2}{32.174 \frac{\mathrm{lb_m \cdot ft}}{\mathrm{lb_f \cdot s^2}} \times 2} = 6.52 \frac{\mathrm{lb_f}}{\mathrm{ft}^2} \times \frac{1\ \mathrm{ft}^2}{144\ \mathrm{in}^2} = 0.045\ \mathrm{psi}$$

Therefore, substituting in Equation 3.11

$$\Delta P_{TP} = 1.011 \times 0.045 = 0.0457\ \mathrm{psi}$$

From Equation 3.13

$$P_2 - P_1 = G_1^2 \sigma (1-\sigma) v_f \left[\frac{(1-x)^2}{(1-\alpha)} + \left(\frac{v_g}{v_f} \right) \frac{x^2}{\alpha} \right]$$

where

G_1 = steam mass velocity $= \rho_v V = 0.264 \frac{\mathrm{lb_m}}{\mathrm{ft}^3} \times 5 \frac{\mathrm{ft}}{\mathrm{s}} = 1.32 \frac{\mathrm{lb_m}}{\mathrm{ft}^2 \cdot \mathrm{s}}$

α = void fraction $= \Psi = 0.999$

v_f = specific volume of the saturated liquid =0.01787 $\mathrm{ft^3/lb_m}$

$x = 0.999$ (@340°F)

$V = 5$ ft/s

σ = enlargement ratio A_1 / A_2

where

A_2 is estimated 10 times larger than A_1

$$\sigma = \frac{(0.02083\,\text{ft})^2}{(0.2083\,\text{ft})^2} = 0.1$$

then

$$P_2 - P_1 = \left(\frac{1.32\,\text{lb}_\text{m}}{\text{ft}^2 \cdot \text{s}}\right)^2 \times 0.1 \times (1-0.1) \times 0.01787\,(\text{lb}_\text{m}/\text{ft}^3)$$
$$\left[\frac{(1\text{-}0.999)}{1\text{-}0.999} + \left(\frac{0.37878\,\dfrac{\text{ft}^3}{\text{lb}_\text{m}}}{0.1787\,\dfrac{\text{ft}^3}{\text{lb}_\text{m}}}\right)\left(\frac{(0.999)^2}{0.999}\right)\right]$$

$$= \frac{0.5933\,\dfrac{\text{lb}_\text{m}}{\text{ft}\cdot\text{s}^2}}{32.2\,\dfrac{\text{lb}_\text{m}\cdot\text{ft}}{\text{lb}_\text{f}\cdot\text{s}^2}} = 0.1842\,\frac{\text{lb}_\text{f}}{\text{ft}^2} \times \frac{1\,\text{ft}^2}{144\,\text{in}^2} = 1.27 \times 10^{-4}\ \text{psi}$$

For a contraction in the homogeneous model:

$$p_2 - p_1 = \frac{G_2^2 v_f}{2}\left[\left(\frac{1}{C_c} - 1\right)^2 + \left(1 - \frac{1}{\sigma^2}\right)\right]\left[1 + \left(\frac{v_{fg}}{v_f}\right)x\right]$$

G_2 = steam mass flux = 1.32 $\dfrac{\text{lb}_\text{m}}{\text{ft}^2 \cdot \text{s}}$

v_f = specific volume of saturated liquid = 0.01787 $\text{ft}^3/\text{lb}_\text{m}$

$$\sigma = \frac{(0.2083\,\text{ft})^2}{(0.02083\,\text{ft})^2} = 10$$

$$v_{fg} = 3.7699\,\frac{\text{ft}^3}{\text{lb}_\text{m}}\,(@\ 340\ °\text{F})$$

Then

$$P_2 - P_1 = \frac{\left(\frac{1.32\ \mathrm{lb_m}}{\mathrm{ft}^2 \cdot \mathrm{s}}\right)^2 \times 0.017 \frac{\mathrm{lb_m}}{\mathrm{ft}^3}}{2}\left[\left(\frac{1}{0.592}-1\right)^2 + \left(1+\frac{1}{100}\right)\right]$$

$$\left[1+\left(\frac{3.7699 \frac{\mathrm{lb_m}}{\mathrm{ft}^3}}{0.01787 \frac{\mathrm{lb_m}}{\mathrm{ft}^3}}\right)\times 0.999\right]$$

$$= \left(\frac{0.5933 \frac{\mathrm{lb_m}}{\mathrm{ft}\cdot \mathrm{s}^2}}{32.2 \frac{\mathrm{lb_m}\cdot \mathrm{ft}}{\mathrm{lb_f}\cdot \mathrm{s}^2}}\right) = 0.0184\ \frac{\mathrm{lb_f}}{\mathrm{ft}^2} \times \frac{1\ \mathrm{ft}^2}{144\ \mathrm{in}^2} = 1.2796 \times 10^{-4}\,\mathrm{psi}$$

Therefore, the total pressure drop per channel is the sum

$$\Delta \mathrm{P_{TOTAL}} = \sum\left[\left(\text{ channel}\right) + \left(180^\circ \text{ bend}\right) + \left(\text{ enlargement}\right) + \left(\text{contraction}\right)\right]$$

$$= \left(0.011\ \mathrm{psi}\right) + \left(0.045\ \mathrm{psi}\right) + \left(1.27 \times 10^{-4}\ \mathrm{psi}\right) + \left(1.279634 \times 10^{-4}\ \mathrm{psi}\right)$$

$$= 0.0525\ \mathrm{psi}$$

As can be appreciated, the pressure drop in this sheet-shell heat exchanger is not a major issue.

APPENDIX D

VAPOR COMPRESSION TRADE-OFFS

Case A. $\Delta T_{cond} = 3.333\ \mathrm{K}$ = (6 °F)

15% Salt Activity Calculation

For S = 150 g/kg, substitute in the activity formula given in Chapter V.

$$\log_{10}\left(p/p_0\right) = \left(-2.1609\times10^{-4}\right)\left(150\right) + \left(-3.5012\times10^{-7}\right)\left(150\right)^2$$

$$= -0.04028$$

$$p/p_0 = 10^{-0.04028} = 0.9114$$

The nomenclature follows Figure B-1 (Appendix B)

Stage 1

A $P = 9.16$ atm (Maximum pressure for dropwise condensation, 120 psig)

$T = 449.86$ K (Saturated, steam table)

B $P = 9.16$ atm

$T = 449.86$ K

I $T = 449.89 - 3.333 = 446.53$ K

$P_0 = 8.47123$ atm

$P = 0.9114\ (8.47123) = 7.7203$ atm (15% salt)

Stage 2

A $P = 7.7207$ atm

$T = 446.53$ K

B $P = 7.7207$ atm

$T = 442.664$ K (saturated steam table)

I $T = 442.664$ K – 3.333 K = 439.331 K

P_0 = 7.11672 atm (saturated steam table)

$P = 0.9114$ (7.11672) = 6.486 atm (15% salt)

Stage 3

A $P = 6.486$ atm

$T = 439.331$ K

B $P = 6.486$ atm

$T = 435.601$ K (saturated, steam table)

I $T = 435.601$ – 3.333 = 432.263 K

P_0 = 5.96001 atm

$P = 0.9114$ (5.96001) = 5.4320 atm (15% salt)

Stage 4

A $P = 5.4320$ atm

$T = 432.263$ K

B $P = 5.4320$ atm

$T = 428.667$ K

I $T = 428.667$ – 3.333 = 425.334 K

P_0 = 4.97665 atm (saturated, steam table)

$P = 0.9114$ (4.97665) = 4.536 atm

Stage 5

A $P = 4.536$ atm

$T = 425.334$ K

B $P = 4.536$ atm

$T = 421.87$ K (saturated, steam table)

I $T = 421.87$ K – 3.333 K = 418.54 K

P_0 = 4.143 atm (saturated steam table)

$P = 0.9114$ (4.143) = 3.776 atm

Stage 6

A $P = 3.776$ atm

$T = 418.54$ K

B $P = 3.776$ atm

$T = 445.195$ K (Saturated, steam table)

I $T = 415.195$ – 3.333 = 411.86 K

P_0 = 3.4366 atm (Saturated, steam table)

$P = 0.9114$ (3.4366 atm) = 3.132 atm (15% salt)

Stage 7

A $P = 3.132$ atm

$T = 411.86$ K

B $P = 3.132$ atm

$T = 408.632$ K (saturated steam table)

I $T = 408.632$ K – 3.333 K = 405.30 K

P_0 = 2.8409 atm (saturated steam table)

$P = 0.9114$ (2.8403 atm) = 2.589 atm (15% salt)

Stage 8

A $P = 2.589$ atm

$T = 405.30$ K

B $P = 2.589$ atm

$T = 402.186$ K (saturated, steam table)

I $T = 402.186$ K– 3.333 K = 398.85 K

P_0 = 2.3396 atm (saturated, steam table)

$P = 0.9114$ (2.3396) = 2.132 atm (15% salt)

Stage 9

A $P = 2.132$ atm

$T = 398.85$ K

B $P = 2.132$ atm

$T = 395.844$ K (saturated, steam table)

I $T = 395.844$ – 3.333 = 392.51 K

P_0 = 1.9195 atm (saturated, steam table)

$P = 0.9114$ (1.9195) = 1.749 atm (15% salt)

Stage 10

A $P = 1.749$ atm

$T = 392.51$ K

B $P = 1.749$ atm

$T = 389.61$ K (saturated, steam table)

I $T = 389.61$ K – 3.333 K =386.28 K

P_0 = 1.5689 atm (saturated steam table)

$P = 0.9114$ (1.5689) = 1.430 atm (15% salt)

10% Salt Activity Calculation

For S = 100 g/kg, substitute in the activity formula given in Chapter V.

$$\log_{10}\left(p / p_0\right) = \left(-2.1609 \times 10^{-4}\right)\left(100\right) + \left(-3.5012 \times 10^{-7}\right)\left(100\right)^2$$

$$= -0.02251102$$

$$p / p_0 = 10^{-0.0251102} = 0.94382$$

10 % Salt $\Delta T_{cond} = 3.333$ K

Stage 1

A $P = 9.16$ atm (Maximum pressure for dropwise condensation, 120 psig)

$T = 449.86$ K (Saturated, steam table)

B $P = 9.16$ atm

$T = 449.86$ K

I $T = 449.89 – 3.333 = 446.53$ K

$P_0 = 8.47123$ atm

$P = 0.94382 (8.47123) = 7.9953$ atm (10% salt)

Stage 2

A $P = 7.9953$ atm

$T = 446.53$ K

B $P = 7.9953$ atm

T = 444.112 K (saturated steam table)

I $T = 444.112$ K – 3.333 K = 440.779 K

$P_0 = 7.37430$ atm (saturated steam table)

$P = 0.94382 (7.37430) = 6.96001$ atm (10% salt)

Stage 3

A $P = 6.96001$ atm

$T = 440.779$ K

B $P = 6.96001$ atm

$T = 438.430$ K (saturated, steam table)

I $T = 438.430 – 3.333 = 435.097$ K

P_0 = 6.40429 atm (saturated, steam table)

$P = 0.94382$ (6.40429 atm) = 6.0445 atm (10% salt)

Stage 4

A $P = 6.0445$ atm

$T = 435.097$ K

B $P = 6.0445$ atm

$T = 432.815$ K

I $T = 432.815 – 3.333 = 429.482$ K

P_0 = 5.54827 atm (saturated, steam table)

$P = 0.94382$ (5.54827 atm) = 5.2366 atm (10% salt)

Stage 5

A $P = 5.2366$ atm

$T = 429.482$ K

B $P = 5.2366$ atm

$T = 427.265$ K (saturated, steam table)

I $T = 427.165$ K – 3.333 K = 423.93 K

P_0 = 4.79444 atm (saturated steam table)

$P = 0.94382$ (4.79444 atm) = 4.5251 atm (10% salt)

Stage 6

A $P = 4.5251$ atm

$T = 423.93$K

B $P = 4.5251$ atm

$T = 421.778$ K (Saturated, steam table)

I $T = 421.778 – 3.333 = 418.445$ K

P_0 = 4.13219 atm (Saturated, steam table)

$P = 0.94382$ (4.13219 atm) = 3.900 atm (10% salt)

Stage 7

A $P = 3.900$ atm

$T = 418.445$ K

B $P = 3.900$ atm

$T = 416.353$ K (saturated steam table)

I $T = 416.353$ K – 3.333 K = 413.02 K

P_0 = 3.55174 atm (saturated steam table)

$P = 0.94382$ (3.55174 atm) = 3.3522 atm (10% salt)

Stage 8

A $P = 3.3522$ atm

$T = 413.02$ K

B $P = 3.3522$ atm

$T = 410.989$ K (saturated, steam table)

I $T = 410.989$ K– 3.333 K = 407.656 K

P_0 = 3.0444 atm (saturated, steam table)

P = 0.94382 (3.04433 atm) = 2.8733 atm (10% salt)

Stage 9

A P = 2. 8733 atm

T = 407.656 K

B P = 2.8733 atm

T = 405.685 K (saturated, steam table)

I T = 405.685 – 3.333 = 402.35 K

P_0 = 2.60194 atm (saturated, steam table)

P = 0.94382 (2.60194) = 2.45579 atm (10% salt)

Stage 10

A P = 2.45576 atm

T = 402.351 K

B P = 2.45576 atm

T = 400.438 K (saturated, steam table)

I T = 400.438 K – 3.333 K =397.105 K

P_0 = 2. 21716 atm (saturated, steam table)

P = 0.94382 (2.21716 atm) = 2.093 atm (10% salt)

7% Salt Activity calculation

For S = 70 g/kg, substitute in the activity formula given in Chapter V.

$$\log_{10}\ p/p_0 = \left(-2.1609\times10^{-4}\right)\left(70\right)+\left(-3.5012\times10^{-7}\right)\left(70\right)^2$$

$$= -0.0168419$$

$$p/p_0 = 10^{-0.6168419} = 0.961962$$

7% Salt $\Delta T_{cond} = 3.333\ \text{K}\ (6°\text{F})$

Stage 1

A $P = 9.16$ atm (Maximum pressure for dropwise condensation, 120 psig)

$T = 449.86$ K (Saturated, steam table)

B $P = 9.16$ atm

$T = 449.86$ K

I $T = 449.89 – 3.333 = 446.53$ K

$P_0 = 8.47123$ atm

$P = 0.96196\ (8.47123) = 8.149$ atm (7% salt)

Stage 2

A $P = 8.149$ atm

$T = 446.53$ K

B $P = 8.149$ atm

$T = 444.905$ K (saturated steam table)

I $T = 444.905$ K – 3.333 K = 441.572 K

$P_0 = 7.51847$ atm (saturated steam table)

$P = 0.94382\ (7.51847) = 7.2325$ atm (7% salt)

Stage 3

A $P = 7.2325$ atm

$T = 441.572$ K

B $P = 7.2325$ atm

$T = 439.987$ K (saturated, steam table)

I $T = 439.987 – 3.333 = 436.654$ K

$P_0 = 6.65935$ atm (saturated, steam table)

$P = 0.96196\ (6.65935$ atm$) = 6.4060$ atm (7% salt)

Stage 4

A $P = 6.4060$ atm

$T = 436.654$ K

B $P = 6.4060$ atm

$T = 435.108$ K

I $T = 435.108 – 3.333 = 431.775$ K

$P_0 = 5.88605$ atm (saturated, steam table)

$P = 0.96196\ (5.88605$ atm$) = 5.66214$ atm (7% salt)

Stage 5

A $P = 5.66214$ atm

$T = 431.775$ K

B $P = 5.66214$ atm

$T = 430.267$ K (saturated, steam table)

I $T = 430.267$ K $– 3.333$ K $= 426.934$ K

$P_0 = 5.19127$ atm (saturated steam table)

$P = 0.94382\ (5.19127$ atm$) = 4.9938$ atm (7% salt)

Stage 6

A $P = 4.9938$ atm

$T = 426.934$K

B $P = 4.9938$ atm

$T = 425.464$ K (Saturated, steam table)

I $T = 425.464 – 3.333 = 422.131$ K

P_0 = 4.56834 atm (Saturated, steam table)

$P = 0.96196$ (4.56834 atm) = 4.39456 atm (7% salt)

Stage 7

A $P = 4.39456$ atm

$T = 422.131$ K

B $P = 4.39456$ atm

$T = 420.698$ K (saturated steam table)

I $T = 420.698$ K – 3.333 K = 417.365 K

P_0 = 4.10095 atm (saturated steam table)

$P = 0.96196$ (4.10095 atm) = 3.8584 atm (7% salt)

Stage 8

A $P = 3.8584$ atm

$T = 417.365$ K

B $P = 3.8584$ atm

$T = 415.968$ K (saturated, steam table)

I $T = 415.968$ K– 3.333 K = 412.635 K

P_0 = 3.15319 atm (saturated, steam table)

P = 0.96196 (3.15319 atm) = 3.3795 atm (7% salt)

Stage 9

A P = 3 3795 atm

T = 412.635 K

B P = 3.3795 atm

T = 411.273 K (saturated, steam table)

I T = 411.273 – 3.333 = 407.94 K

P_0 = 3.06964 atm (saturated, steam table)

P = 0.94382 (3.06964) = 2.95287 atm (7% salt)

Stage 10

A P = 2.95287 atm

T = 407.94 K

B P = 2.95287 atm

T = 406.613 K (saturated, steam table)

I T = 406.613 K – 3.333 K = 403.28 K

P_0 = 2. 67532 atm (saturated, steam table)

P = 0.96196 (2.67532 atm) = 2.5736 atm (7% salt)

5% Salt. Activity calculation

For $S = 50$ g/kg, substitute in the activity formula given in Chapter V.

$$\log_{10} p/p_0 = \left(-2.1609 \times 10^{-4}\right)\left(50\right) + \left(-3.5012 \times 10^{-7}\right)\left(50\right)^2$$

$$= -0.0116798$$

$$p/p_0 = 0.97346$$

5% Salt

Stage 1

A $P = 9.16$ atm

$T =$ 449.86 K

B $P = 9.16$ atm

$T = 449.86$ K (saturated, steam table)

I $T = 449.86 – 3.333 = 446.53$ K

P_0 = 8.47123 atm (saturated, steam table)

$P = 0.97346$ (8.47123) = 8.246 atm

Stage 2

A $P = 8.2464$ atm

$T = 446.53$ K

B $P = 8.$ 2464 atm

$T = 445.401$ K (saturated, steam table)

I $T = 444.401 – 3.333 = 442.07$ K

P_0 = 7.61014 atm (saturated, steam table)

$P = 0.973465$ (7.61014) = 7.4082 atm (5% salt)

Stage 3

A $P = 7.4082$ atm

$T = 442.07$ K

B $P = 7.4082$ atm

$T = 440.967$ K (saturated, steam table)

I $T = 440.967$ K – 3.333 K = 437.63 K

P_0 = 6.82399 atm

$P = 0.97346$ (6.82399) = 6.643 atm

Stage 4

A $P = 6.643$ atm

$T = 437.63$ K

B $P = 6.643$ atm

$T = 436.556$ K (saturated, steam table)

I $T = 436.556$ – 3.333 = 433.223 K

P_0 = 6.0766 atm (saturated, steam table)

$P = 0.97346$ (6.0766) = 5.9456 atm (5% salt)

Stage 5

A $P = 5.9456$ atm

$T = 433.223$ K

B $P = 5.9456$ atm

$T = 432.168$ K (saturated, steam table)

I $T = 432.168$ – 3.333 = 428.835 K

P_0 = 5.45582 atm (saturated, steam table)

$P = 0.97346$ (5.45582) = 5.3110 atm

Stage 6

A $P = 5.3110$ atm

$T = 428.835$K

B $P = 5.3110$ atm

$T = 427.804$ K (saturated, steam table)

I $T = 427.804 – 3.333 = 424.47$ K

P_0 = 4.86374 atm (saturated, steam table)

$P = 0.97346$ (4.86374) = 4.7347 atm (5% salt)

Stage 7

A $P = 4.7347$ atm

$T = 424.47$ K

B $P = 4.7347$ atm

$T = 423.46$ K (saturated, steam table)

I T = 423.46 – 3.333 = 420.13 K

P_0 = 4.32686 atm (saturated, steam table)

$P = 0.973465$ (4.32686) = 4.212 atm (5% salt)

Stage 8

A $P = 4.2122$ atm

T = 420.13 K

B $P = 4.2122$ atm

T = 419.142 K (saturated, steam table)

I T = 419.142 K – 3.333 K = 415.81 K

P_0 = 3.84142 atm (saturated, steam table)

$P = 0.97346$ (3.84142) = 3.7395 atm

Stage 9

A $P = 3.7395$ atm

$T = 415.81$ K

B $P = 3.7395$ atm

$T = 414.848$ K (saturated, steam table)

I $T = 414.848 - 3.333 = 411.515$ K

$P_0 = 3.40296$ atm (saturated, steam table)

$P = 0.97346\ (3.40296) = 3.3127$ atm (5% salt)

Stage 10

A $P = 3.3127$ atm

$T = 411.515$ K

B $P = 3.3127$ atm

$T = 410.576$ K (saturated, steam table)

I $T = 410.576 - 3.333 = 407.24$ K

$P_0 = 3.00784$ atm (saturated, steam table)

$P = 0.97346\ (3.00784) = 2.928$ atm (5%)

Case B. $\Delta T_{cond} = 2.222\,\mathrm{K}\left(4°\mathrm{F}\right)$

The nomenclature follows Figure B-1 (Appendix B)

15% Salt

Stage 1

A $P = 9.16$ atm

T = 449.86 K

B $P = 9.16$ atm

T = 449.86 K

I T = 449.89 – 2.222 = 447.638 K

P_0 = 8.69665 atm

$P = 0.9114$ (8.69665) = 7.93613 atm

Stage 2

A $P = 7.92613$ atm

T = 447.638 K

B $P = 7.92613$ atm

T = 443.751 K

I T = 443.751 K – 2.222 K = 441.529 K

P_0 = 7.51059 atm

$P = 0.9114$ (7.71059) = 6.8452 atm

Stage 3

A $P = 6.8452$ atm

T = 441.529 K

B $P = 6.8452$ atm

$T = 437.759$ K

I $T = 437.759 – 2.222 = 435.537$ K

$P_0 = 6.47556$ atm

$P = 0.9114\ (6.47556) = 5.9018$ atm

Stage 4

A $P = 5.9018$ atm

$T = 435.537$ K

B $P = 5.9018$ atm

$T = 431.879$ K

I $T = 431.879 – 2.222 = 429.657$ K

$P_0 = 5.57349$ atm

$P = 0.9114\ (5.57359) = 5.0797$ atm

Stage 5

A $P = 5.0797$ atm

$T = 429.657$ K

B $P = 5.0797$ atm

$T = 426.109$ K

I $T = 426.109 – 2.222 = 423.887$ K

$P_0 = 4.78872$ atm

$P = 0.9114\ (4.78872) = 4.3644$ atm

10% Salt $\Delta T_{cond} = 2.222\ \mathrm{K}\ (4°\mathrm{F})$

Stage 1

A $P = 9.16$ atm

$T = 449.86$ K

B $P = 9.16$ atm

$T = 449.86$ K

I $T = 449.86 - 2.222 = 447.638$ K

$P_0 = 8.69665$ atm

$P = 0.94382\ (8.69665) = 8.2081$ atm

Stage 2

A $P = 8.2081$ atm

$T = 447.6381$ K

B $P = 8.2081$ atm

$T = 445.207$ K

I $T = 445.207 - 2.222 = 442.985$ K

$P_0 = 7.78088$ atm

$P = 0.94382\ (7.78088) = 7.3438$ atm

Stage 3

A $P = 7.3438$ atm

$T = 442.985$ K

B $P = 7.3438$ atm

$T = 440.609$ K

I $T = 440.609 - 2.222 = 438.387$ K

$P_0 = 6.95268$ atm

$P = 0.94382\ (6.95268) = 6.5621$ atm

Stage 4

A $P = 6.5621$ atm

$T = 438.387$ K

B $P = 6.5621$ atm

$T = 436.066$ K

I $T = 436.066 - 2.222 = 433.844$ K

$P_0 = 6.20472$ atm

$P = 0.94382\ (6.20472) = 5.8561$ atm

Stage 5

A $P = 5.8561$ atm

$T = 433.844$K

B $P = 5.8561$ atm

$T = 431.576$ K

I $T = 431.576 - 2.222 = 429.3549$ K

$P_0 = 5.52988$ atm

$P = 0.94382\ (5.52988) = 5.2192$ atm

7% Salt $\Delta T_{cond} = 2.222\ \text{K}\ (4°\text{F})$

Stage 1

A $P = 9.16$ atm

$T = 449.86$ K

B $P = 9.16$ atm

$T = 449.86$ K

I $T = 449.86 - 2.222 = 447.638$ K

$P_0 = 8.69665$ atm

$P = 0.961062\ (8.69665) = 8.3658$ atm

Stage 2

A $P = 8.3658$ atm

$T = 447.638$ K

B $P = 8.3658$ atm

$T = 446.004$ K

I $T = 446.004 - 2.222 = 443.782$ K

$P_0 = 7.93206$ atm

$P = 0.961962\ (7.93206) = 7.63034$ atm

Stage 3

A $P = 7.63034$ atm

$T = 443.782$ K

B $P = 7.63034$atm

$T = 442.179$ K

I T = 442.179 – 2.222 = 439.957 K

P_0 = 7.22719 atm

P = 0.94382 (7.22719) = 6.9523 atm

Stage 4

A P = 6.9522 atm

T = 439.957 K

B P = 6.9522 atm

T = 438.385 K

I T = 438.385 – 2.222 = 436.163 K

P_0 = 6.57888 atm

P = 0.961962 (6.57888) = 6.3286 atm

Stage 5

A P = 6.3286 atm

T = 436.163K

B P = 6.3286 atm

T = 434.626 K

I T = 434.626 – 2.222 = 432.4 K

P_0 = 5.98152 atm

P = 0.94382 (5.98152) = 5.75399 atm

5% Salt $\Delta T_{cond} = 2.222\ \mathrm{K}\ (4°\mathrm{F})$

Stage 1

A $P = 9.16$ atm

T = 449.86 K

B $P = 9.16$ atm

T = 449.86 K

I T = 449.86 – 2.222 = 447.638 K

P_0 = 8.69665 atm

P = 0.97346 (8.69665) = 8.4658 atm

Stage 2

A $P = 8.4658$ atm

T = 447.638 K

B $P = 8.4658$ atm

T = 446.504 K

I T = 446.503 – 2.222 = 444.281 K

P_0 = 8.02786 atm

P = 0.97346 (8.02789) = 7.8148 atm

Stage 3

A $P = 7.8148$ atm

T = 444.281` K

B $P = 7.8148$atm

T = 443.165 K

I T = 443.165 – 2.222 = 440.943 K

P_0 = 7.40394 atm

P = 0.94382 (7.40394) = 7.2074 atm

Stage 4

A P = 7.2074 atm

T = 440.943 K

B P = 7.2074 atm

T = 439.845 K

I T = 439.845 – 2.222 = 437.623 K

P_0 = 6.82213 atm

P = 0.97346 (6.82213) = 6.6411 atm

Stage 5

A P = 6.6411 atm

T = 437.623K

B P = 6.6411 atm

T = 436.544 K

I T = 436.544 – 2.222 = 434.322 K

P_0 = 6.28026 atm

P = 0.97346 (6.28206) = 6.1136 atm

Case C. $\Delta T_{cond} = 1.111\,\mathrm{K}(2°\mathrm{F})$

The nomenclature follows Figure B-1 (Appendix B)

15% salt

Stage 1

A $P =$ 9.16 atm

$T =$ 449.86 K

B $P =$ 9.16 atm

$T =$ 449.86 K

I $T =$ 449.89 – 1.111 = 448.749 K

$P_0 =$ 8.92741 atm

$P =$ 0.9114 (8.92741) = 8.1364 atm

Stage 2

A $P =$ 8.1364 atm

$T =$ 448.74 K

B $P =$ 8.1364 atm

$T =$ 444.841 K

I $T =$ 444.84 – 1.111 = 443.73 K

$P_0 =$ 7.92212 atm

$P =$ 0.9114 (7.92212) = 7.2202 atm

Stage 3

A $P =$ 7.2202 atm

$T =$ 443.73 K

B $P = 7.2202$ atm

$T = 439.918$ K

I $T = 439.918 – 1.111 = 438.807$ K

$P_0 = 7.02529$ atm

$P = 0.9114\ (7.02529) = 6.4028$ atm

Stage 4

A $P = 6.4028$ atm

$T = 438.807$ K

B $P = 6.4028$ atm

$T = 435.088$ K

I $T = 435.088 – 1.111 = 433.977$ K

$P_0 = 6.22566$ atm

$P = 0.9114\ (6.22566) = 5.6741$ atm

Stage 5

A $P = 5.6741$ atm

$T = 433.977$ K

B $P = 5.6741$ atm

$T = 430.349$ K

I $T = 430.349 – 1.111 = 429.238$ K

$P_0 = 5.51254$ atm

$P = 0.9114\ (5.51254) = 5.024$ atm

10% Salt $\Delta T_{cond} = 1.111\,\text{K}\left(2°\text{F}\right)$

Stage 1

A $P = 9.16$ atm

$T = 449.86$ K

B $P = 9.16$ atm

$T = 449.86$ K

I $T = 449.86 – 1.111 = 448.749$ K

$P_0 = 8.92741$atm

$P = 0.94382\ (8.92741) = 8.4259$ atm

Stage 2

A $P = 8.4259$ atm

$T = 448.749$ K

B $P = 8.4259$ atm

$T = 446.304$ K

I $T = 446.304 – 1.111 = 445.193$ K

$P_0 = 8.20540$ atm

$P = 0.94382\ (8.20540) = 7.7444$ atm

Stage 3

A $P = 7.7444$ atm

$T = 445.193$ K

B $P = 7.7444$ atm

$T = 442.791$ K

I $T = 442.791 - 1.111 = 441.68$ K

$P_0 = 7.53827$ atm

$P = 0.94382\ (7.53827) = 7.114$ atm

Stage 4

A $P = 7.1148$ atm

$T = 441.68$ K

B $P = 7.1148$ atm

$T = 439.320$ K

I $T = 439.320 - 1.111 = 438.21$ K

$P_0 = 6.92209$ atm

$P = 0.94382\ (6.92209) = 6.5332$ atm

Stage 5

A $P = 6.5332$ atm

$T = 438.21$ K

B $P = 6.5332$ atm

$T = 435.89$ K

I $T = 435.89 - 1.111 = 434.779$ K

$P_0 = 6.35316$ atm

$P = 0.94382\ (6.35316) = 5.9962$ atm

7% Salt $\Delta T_{cond} = 1.111\,\text{K}(2°\text{F})$

Stage 1

A $P = 9.16$ atm

$T = 449.86$ K

B $P = 9.16$ atm

$T = 449.86$ K

I $T = 449.86 – 1.111 = 434.779$ K

$P_0 = 8.92741$ atm

$P = 0.961962\ (8.9274) = 8.5878$ atm

Stage 2

A $P = 8.5878$ atm

$T = 448.749$ K

B $P = 8.5878$ atm

$T = 447.106$ K

I $T = 447.10 – 1.111 = 445.995$ K

$P_0 = 8.36405$ atm

$P = 0.961962\ (8.36405) = 8.0459$ atm

Stage 3

A $P = 8.0459$ atm

$T = 445.99$ K

B $P = 8.0459$ atm

$T = 444.374$ K

I $T = 444.37 – 1.111 = 443.263$ K

$P_0 = 7.83335$ atm

$P = 0.961962$ (7.83335) = 7.53539 atm

Stage 4

A $P = 7.53539$ atm

$T = 443.263$ K

B $P = 7.53539$ atm

$T = 441.66$ K

I $T = 441.66 – 1.111 = 440.553$ K

$P_0 = 7.33362$ atm

$P = 0.961962$ (7.3336) = 7.0547 atm

Stage 5

A $P = 7.0547$ atm

$T = 440.553$ K

B $P = 7.0547$ atm

$T = 438.97$ K

I $T = 438.976 – 1.111 = 437.865$ K

$P_0 = 6.86327$ atm

$P = 0.961962$ (6.86327) = 6.6022 atm

5% Salt $\Delta T_{cond} = 1.111\,\text{K}(2°\text{F})$

Stage 1

A $P = 9.16$ atm

$T = 449.86$ K

B $P = 9.16$ atm

$T = 449.86$ K

I $T = 449.86 – 1.111 = 448.749$ K

$P_0 = 8.92741$ atm

$P = 0.97346\ (8.92741) = 8.6905$ atm

Stage 2

A $P = 8.6905$ atm

$T = 448.749$ K

B $P = 8.\ 6905$ atm

$T = 447.\ 608$ K

I $T = 447.608 – 1.111 = 446.\ 497$ K

$P_0 = 8.46459$ atm

$P = 0.97346\ (8.46459) = 8.2399$ atm

Stage 3

A $P = 8.2399$ atm

$T = 446.497$ K

B $P = 8.2399$ atm

$T = 445.369$ K

I $T = 445.369 - 1.111 = 444.258$ K

$P_0 = 8.02345$ atm

$P = 0.97346\,(8.02345) = 7.8105$ atm

Stage 4

A $P = 7.8105$ atm

$T = 444.258$ K

B $P = 7.8105$ atm

$T = 443.142$ K

I $T = 443.142 - 1.111 = 442.031$ K

$P_0 = 7.60293$ atm

$P = 0.97346\,(7.60293) = 7.4011$ atm

Stage 5

A $P = 7.4011$ atm

$T = 442.031$ K

B $P = 7.4011$ atm

$T = 440.928$ K

I $T = 440.928 - 1.111 = 439.817$ K

$P_0 = 7.20237$ atm

$P = 0.97346\,(7.20237) = 7.0112$ atm

Equations used in the thermodynamic tables

The dry compressor work is calculated with Equation 7.1. The wet compressor work is calculated with Equation 7.2. Results in the last column of Tables D.1 to D.4 are given per unit of water product. The amount of water injected into the wet compressor is calculated with Equation 7.3

The thermodynamic values are read from the steam tables with data from the trade-offs.

Table D.1. Case A Thermodynamic calculations for dry compressor. $\Delta T = 3.333\,\text{K}\,(6°\text{F})$

Salt	n	m_1 (kg)	P_1 (atm)	T_1 (K)	V_1 (m^3/kg)	$\hat{H}_1^{vap}$ (kJ/kg)	$\hat{S}_1^{vap}$ (kJ/(kg · K))	P_2 (atm)	T_2 (K)	H_2 (kJ/kg)	V_1 (m^3/kg)	$*W_1$ (kJ/m^3)
5%	1	1000	8.246	446.53	0.231381	27772.34	6.65045	9.16	457.348	2792.91	231.38	24,200
	2	500	7.4082	442.07	0.256122	2767.85	6.68683	9.16	464.092	2809.67	128.06	24,600
	3	33	6.643	437.63	0.283956	2763.16	6.72369	9.16	471.221	2826.90	94.56	24,971
	4	250	5.9456	433.22	0.315326	2758.30	6.76107	9.16	478.737	2844.66	78.83	25,400
	5	200	5.3110	428.84	0.350757	2753.29	6.79904	9.16	486.650	2862.99	70.15	25,812
	10	100	2.928	407.24	0.613771	2725.90	6.99777	9.16	532.176	2964.11	61.38	28,025
7%	1	1000	8.149	446.53	0.234344	2773.12	6.65733	9.16	458.601	2796.06	234.34	26,988
	2	500	7.233	441.57	0.262370	2768.04	6.69768	9.16	466.161	2814.71	131.19	27,456
	3	33	6.406	436.65	0.294289	2762.75	6.73863	9.16	474.192	2833.97	98.0	27,901
	4	250	5.662	431.78	0.330666	2757.28	6.78016	9.16	482.681	2853.83	82.67	28,397
	5	200	4.994	426.93	0.372173	2751.57	6.82213	9.16	491.593	2874.28	74.43	28,873
	10	100	2.574	403.28	0.693329	2720.77	7.04281	9.16	543.377	2988.33	69.33	31,477
10%	1	1000	7.995	446.53	0.239194	2774.36	6.66837	9.16	460.633	2801.13	239.19	31,494
	2	500	6.960	440.78	0.272726	2768.32	6.71509	9.16	469.532	2822.86	136.36	32,082
	3	33	6.045	435.10	0.311546	2762.03	6.76238	9.16	479.006	2845.29	103.74	32,618
	4	250	5.237	429.48	0.356646	2755.48	6.81031	9.16	489.051	2868.48	89.16	33,235
	5	200	4.525	423.93	0.409208	2748.73	6.85904	9.16	499.693	2892.57	81.84	33,845
	10	100	2.093	397.11	0.842965	2712.44	7.11493	9.16	561.946	3028.19	84.29	37,147
15%	1	1000	7.721	446.53	0.248296	2776.53	6.68838	9.16	464.386	2810.39	248.30	39,835
	2	500	6.486	439.33	0.292721	2768.65	6.74666	9.16	475.807	2837.78	146.36	40,665
	3	33	5.432	432.26	0.345937	2760.41	6.80566	9.16	488.058	2866.21	115.20	41,449
	4	250	4.536	425.33	0.409810	2751.90	6.86543	9.16	501.119	2895.77	102.45	42,315
	5	200	3.776	418.54	0.486769	2743.17	6.92614	9.16	515.014	2926.61	97.25	43,162
	7	143	2.589	405.30	0.69315	2725.07	7.05084	9.16	545.406	2992.70	99.02	45,025
	10	100	1.430	386.28	1.20728	2696.88	7.24723	9.16	597.980	3104.89	120.73	48,001

$*\eta_{compressor} = 0.85$

Table D.2. Case A. Thermodynamic calculations for wet compressor. $\Delta T_{cond} = 3.333\,\text{K}\,(6\,°\text{F})$

	$T_1^{liq} = 300$ (K)		$H_1^{liq} = 111.916$ (kJ/kg)		$S_1^{liq} = 0.390357$ (kJ/(kg · K))	$P_2 = 9.16$ (atm)	$T_2^{sat} = 449.85$ (K)		$H_2^{vap} = 2773.58$ (kJ/kg)		$S_2^{vap} = 6.60784$ (kJ/(kg · K))
Salt	n	m_1 (kg)	P_1 (atm)	T_1 (K)	V_1 (m^3/kg)	$\hat{H}_1^{vap}$ (kJ/kg)	$\hat{S}_1^{vap}$ (kJ/(kg · K))	V_1 $\left(\frac{\text{m}^3\text{vapor}}{\text{m}^3\text{liquid}}\right)$	x	$*\hat{W}$ (kJ/kg)	$*\hat{W}$ (kJ/m^3)
5%	1	1,000	8.246	446.53	0.231381	2772.34	6.65045	231.38	0.00685	22.91	22,919
	2	500	7.408	442.07	0.256122	2767.85	6.68683	128.06	0.01270	46.54	23,273
	3	333	6.643	437.63	0.283956	2763.16	6.72369	94.56	0.01863	70.59	23,508
	4	250	5.945	433.22	0.315326	2758.30	6.76107	78.83	0.02460	95.01	23,752
	5	200	5.311	428.84	0.350757	2753.29	6.79904	70.15	0.03075	120.17	24,033
	10	100	2.928	407.24	0.613771	2925.90	6.99777	61.38	0.06272	252.48	25,248
7%	1	1,000	8.149	446.53	0.234344	2773.12	6.65733	234.34	0.0079	25.46	25,466
	2	500	7.233	441.57	0.262370	2768.04	6.69768	131.19	0.0144	51.61	25,805
	3	333	6.406	436.65	0.294289	2762.75	6.73863	98.0	0.0210	78.50	26,140
	4	250	5.662	431.78	0.330666	2757.28	6.78016	82.67	0.0277	105.91	26,478
	5	200	4.994	426.93	0.372173	2751.57	6.82213	74.43	0.0344	133.83	26,766
	10	100	2.574	403.28	0.693329	2720.77	7.04281	69.33	0.0699	281.23	28,122
10%	1	1,000	7.995	446.53	0.239194	2774.36	6.66837	239.19	0.0097	29.56	29,569
	2	500	6.960	440.78	0.272726	2768.32	6.71509	136.36	0.0172	67.47	30,027
	3	333	6.045	435.10	0.311546	2762.03	6.76238	103.74	0.0248	91.29	30,402
	4	250	5.237	429.48	0.356646	2755.48	6.81031	89.16	0.0325	123.07	30,768
	5	200	4.525	423.93	0.409208	2748.73	6.85904	81.84	0.0404	155.74	31,149
	10	100	2.093	397.11	0.842965	2712.44	7.11493	84.29	0.0815	327.32	32,732
15%	1	1,000	7.721	446.53	0.248296	2776.53	6.68838	248.30	0.0129	37.10	37,103
	2	500	6.486	439.33	0.292721	2768.65	6.74666	146.36	0.0223	75.63	37,818
	3	333	5.432	432.26	0.345937	2760.41	6.80566	115.20	0.0318	115.08	38,322
	4	250	4.536	425.33	0.409810	2751.90	6.86543	102.45	0.0414	155.24	38,812
	5	200	3.776	418.54	0.486769	2743.17	6.92614	97.25	0.0511	195.99	39,198
	10	100	1.430	386.28	1.20728	2696.88	7.24723	120.73	0.1028	412.26	41,226

$*\eta_{compressor} = 0.85$

Table D.3 Case B Thermodynamic calculations for wet compressor. $\Delta T_{cond} = 2.222\ \mathrm{K}(4\ ^\circ\mathrm{F})$

$T_1^{liq} = 300$ (K)	$H_1^{liq} = 111.916$ $(\mathrm{kJ/kg})$	$S_1^{liq} = 0.390357$ $(\mathrm{kJ/(kg \cdot K)})$	$P_2 = 9.16$ (atm)	$T_2^{sat} = 449.85$ (K)	$H_2^{vap} = 2773.58$ $(\mathrm{kJ/kg})$	$S_2^{vap} = 6.60784$ $(\mathrm{kJ/(kg \cdot K)})$

Salt	n	m_1 (kg)	P_1 (atm)	T_1 (K)	V_1 (m^3/kg)	$\hat{H}_1^{vap}$ (kJ/kg)	$\hat{S}_1^{vap}$ (kJ/(kg · K))	V_1 $\left(\frac{\mathrm{m^3 vapor}}{\mathrm{m^3 liquid}}\right)$	x	$*\hat{W}$ (kJ/kg)	$*\hat{W}$ (kJ/m^3)
5%	1	1,000	8.4658	447.64	0.225673	2773.42	6.64149	225.67	0.00541	17.14	17,136
	2	500	7.8148	444.28	0.243474	2770.10	6.66870	121.74	0.00978	34.71	17,360
	3	333	7.2074	440.94	0.262873	2766.67	6.69613	87.54	0.01420	52.59	17,514
	4	250	6.6411	437.62	0.284034	2763.15	6.72379	71.01	0.01864	70.63	17,660
	5	200	6.1136	434.32	0.307139	2759.53	6.75170	61.43	0.02314	88.98	17,797
7%	1	1,000	8.3658	447.64	0.228579	2774.22	6.64842	228.58	0.00652	19.68	19,685
	2	500	7.6303	443.782	0.249419	2770.34	6.67962	124.71	0.01154	39.94	19,974
	3	333	6.9523	439.96	0.272388	2766.35	6.71109	90.71	0.01660	60.48	20,142
	4	250	6.3286	436.16	0.297684	27620	6.74270	74.42	0.02169	81.31	20,329
	5	200	5.7540	432.40	0.325668	2757.98	6.77469	65.13	0.026836	102.39	20,477
10%	1	1,000	8.2081	447.638	0.233303	2775.48	6.65946	233.30	0.00830	23.76	23,763
	2	500	7.3438	442.99	0.259231	2770.69	6.69705	129.62	0.01434	48.30	24,152
	3	333	6.5621	438.39	0.288326	2765.72	6.73488	96.01	0.02043	73.22	24,382
	4	250	5.8561	433.84	0.321011	2760.58	6.77299	80.25	0.02656	98.46	24,616
	5	200	5.2192	429.35	0.357792	2755.33	6.81146	71.56	0.03275	124.02	24,804
15%	1	1,000	7.926	447.64	0.242221	2777.70	6.67961	242.22	0.01154	31.30	31,299
	2	500	6.845	441.53	0.278217	2771.11	6.72869	139.11	0.01943	63.74	31,874
	3	333	5.902	435.54	0.319946	2764.30	6.77810	106.54	0.02738	96.65	32,186
	4	250	5.080	429.66	0.368441	2757.28	6.82792	92.11	0.03539	129.99	32,499
	5	200	4.364	423.89	0.424974	2750.10	6.87830	84.99	0.0435	163.84	32,768

$*\eta_{compressor} = 0.85$

Table D.4. Case C Thermodynamic calculations for wet compressor. $\Delta T_{cond} = 1.111\,\mathrm{K}\,(2\,°\mathrm{F})$

$T_1^{liq} = 300$ (K)	$H_1^{liq} = 111.916$ $(\mathrm{kJ/kg})$	$S_1^{liq} = 0.390357$ $(\mathrm{kJ/(kg \cdot K)})$	$P_2 = 9.16$ (atm)	$T_2^{sat} = 449.85$ (K)	$H_2^{vap} = 2773.58$ $(\mathrm{kJ/kg})$	$S_2^{vap} = 6.60784$ $(\mathrm{kJ/(kg \cdot K)})$

Salt	n	m_1 (kg)	P_1 (atm)	T_1 (K)	V_1 (m^3/kg)	$\hat{H}_1^{vap}$ (kJ/kg)	$\hat{S}_1^{vap}$ (kJ/(kg · K))	V_1 $\left(\frac{\mathrm{m^3 vapor}}{\mathrm{m^3 liquid}}\right)$	x	$*\hat{W}$ (kJ/kg)	$*\hat{W}$ (kJ/m^3)
5%	1	1,000	8.6905	448.75	0.220125	2774.49	6.63255	220.13	0.00397	11.374	11,374
	2	500	8.2399	446.50	0.231544	2772.31	6.65071	115.77	0.00689	23.07	11,535
	3	333	7.8105	444.26	0.243603	2770.08	6.66890	81.12	0.00982	34.87	11,615
	4	250	7.4011	442.03	0.256354	2767.81	6.68716	64.09	0.01275	46.72	11,680
	5	200	7.0112	439.82	0.269836	2765.51	6.70550	53.97	0.01571	58.68	11,736
7%	1	1,000	8.5878	448.75	0.22966	2775.31	6.63951	222.97	0.00509	13.91	13,915
	2	500	8.0459	446.00	0.237194	2772.60	6.66168	118.60	0.00866	28.28	14,140
	3	333	7.5353	443.26	0.252391	2769.80	6.68385	84.05	0.01222	42.71	14,223
	4	250	7.0544	440.55	0.268642	2766.97	6.70615	67.16	0.01581	57.28	14,320
	5	200	6.6022	437.87	0.286027	2764.10	6.72852	57.21	0.01941	71.93	14,386
10%	1	1,000	8.4259	448.75	0.227582	2776.59	6.65060	227.58	0.00687	17.99	17,994
	2	500	7.7444	445.19	0.246516	2772.97	6.67910	123.26	0.01146	36.60	18,301
	3	333	7.1148	441.68	0.267112	2769.29	6.70769	88.95	0.01605	55.30	18,416
	4	250	6.5332	438.21	0.289529	2765.51	6.73635	72.38	0.02066	74.18	18,547
	5	200	5.9962	434.80	0.313957	2761.71	6.76521	62.79	0.02531	93.22	18,645
15%	1	1,000	8.1364	448.75	0.236288	2778.86	6.67081	236.29	0.01012	25.50	25,502
	2	500	7.2202	443.73	0.264544	2773.52	6.71085	132.27	0.01656	51.92	25,963
	3	333	6.4028	438.81	0.296310	2768.06	6.75098	98.67	0.02302	78057	26,167
	4	250	5.6741	433.98	0.332033	2762.47	6.79120	83.01	0.02949	105.41	26,354
	5	200	5.0240	429.24	0.372309	2756.77	6.83164	74.46	0.03600	132.51	26,501

$*\eta_{compressor} = 0.85$

APPENDIX E

SENSIBLE HEAT EXCHANGER CALCULATION

The sensible heat exchanger for the evaporator exiting flow and the entering seawater feed is divided in two parts: one corresponds to the sensible heat transfer between the exiting distillate and 50% of the seawater feed, and the other corresponds to the sensible heat transfer between the exiting brine and 50% of the seawater feed.

The sensible heat exchangers are plate-and-frame type and are to be sized to produce the following:

Distilled flow = 10,000,000 gallons/day

= 1,568,182 kg/h

$= 3.45 \times 10^6 \text{ lb}_\text{m} / \text{h}$

= 0.4356 m^3/s

To design the sensible heat exchanger, first a mass balance and an enthalpy balance are performed for each evaporator. The data are used to compute both the sensible heat exchanger area, and the latent heat exchanger area.

The design methodology is that suggested by Cooper and Usher [32]. A separate design is performed for each of the effects.

The calculation method is based on an iterative process where the goal is to establish the number of plates (n) and the number of passes (H), such that the pressure drop and the heat transfer requirements of the heat exchanger are satisfied. Let

m_f = required seawater flow (kg/s)

$\Delta T =$ temperature difference between the evaporator exit flow and the seawater feed (K)

$\Phi =$ required seawater temperature change (K)

$\Delta p =$ permissible pressure drop $\left(\text{kN/m}^2\right)$

$n/2H =$ passages per pass

$2m_f H/n =$ plate flow rate (kg/s)

The methodology follows:

1. Select from the plate area/flow rate graph (Figure E-1), the approximate area (a) of the plate to be used, based on the higher of the two flow rates, the seawater and the corresponding distillate or brine.

2. Find the number of plates (n) and the number of passes (H) according to the requirements as follows:

(a) Select a whole number of passes (H) and determine the available pressure drop per pass on each side as given by

$$\Delta p = \frac{\text{allowable pressure}}{H}$$

(b) Select (α) values from Figure E-2 corresponding to Δp values for each side.

(c) Obtain the overall (U) value excluding metal resistance by combining the two (α) values obtained above.

(d) Determine (q) corresponding to the allowable pressure drop per pass on each side (Figure E-3).

(e) Calculate $n = 2HV/q$ where (V) is the higher of the two liquid flow rates and select the higher of the two values (q) found in (d).

(f) Calculate $Una\Delta T$ and compare with the required heat transfer rate. Repeat, if necessary, with different values of (H) until acceptable agreement is found.

The nomenclature used in the following sensible heat exchanger design is depicted in Figure B-1 for the exiting and entering flows. Figure A-1 depicts the identifying nomenclature corresponding to the individual sensible heat exchangers.

D1: Distillate/Seawater Sensible Heat Exchanger (Case B, Effect 1)

Distillate	seawater
$T_B = 449.86\,\text{K}$	$T_E = 447.4\,\text{K}$
$T_C = 301\,\text{K}$	$T_D = 297\,\text{K}$
$C_p = 4.36\dfrac{\text{kJ}}{\text{kg K}}$	$C_p = 4.04\dfrac{\text{kJ}}{\text{kg K}}$

The log mean heat exchanger temperature difference is

$$\Delta T = LMTD = 2.5\,\text{K}$$

Mass balance:

Required steam input = $m_{steam} = 147.5\,\dfrac{\text{kg}}{\text{s}}$

Calculate m_D, the seawater feed flow required in the sensible heat exchanger

$$m_D = \left(\frac{1}{1-\dfrac{3.5\%}{7\%}}\right) \times m_{steam} = 2 \times 147.5\,\frac{\text{kg}}{\text{s}} = 295\,\frac{\text{kg}}{\text{s}}$$

This feed flow is divided into two parts. 50 % of the flow goes to the distillate/feed flow sensible heat exchanger.

$$m_E = 0.5 \times 295 = 147.5\,\text{kg/s}$$

50% of the flow goes to the brine/feed flow sensible heat exchanger

$$m_F = 0.5 \times 295 = 147.5\,\text{kg/s}$$

Energy balance:

$$Q = m_B C_{p\,B} \Delta T_{distillate} = m_E C_{p\,E} \Delta T_{seawater}$$

Therefore, the rate of heat required in the distillate/feed sensible heat exchanger is

$$Q_R = 147.5\,\frac{\text{kg}}{\text{s}} \times 4.04\,\frac{\text{kJ}}{\text{kg}\cdot\text{K}} \times \left(447.4 - 297\right)\text{K} = 89{,}623.36\ \text{kW}$$

2) Heat exchanger design

2-1) Given the mass flow on each side of the heat exchanger:

$$m_B = 147.5\,\text{kg/s}$$

$$m_E = 147.5\,\text{kg/s} = 0.1475\,\text{m}^3\text{/s}$$

Select the area using Figure E-2

$$a = 1.4\,\text{m}^2$$

2-2) Define the pressure drop allowed on each side of the heat exchanger and number of passes (H)

Let:

$$\Delta P_1 = 800\,\text{kN/m}^2 \ \text{(distillate side)}$$

$$\Delta P_2 = 800\,\text{kN/m}^2 \ \text{(seawater side)}$$

$$H = 20$$

Therefore

$\Delta P_1 = 800 / 20 \text{ kN/m}^2 = 40 \text{ kN/m}^2$

$\Delta P_2 = 800 / 20 \text{ kN/m}^2 = 40 \text{ kN/m}^2$ (Use a correction factor due to seawater properties different from pure water = 0.82)

2-3) Find the heat transfer coefficients from Figure E-3

$\alpha_1 = 10{,}000 \text{ W/}\left(\text{m}^2 \cdot \text{K}\right)$

$\alpha_2 = 11{,}200 \text{ W/}\left(\text{m}^2 \cdot \text{K}\right)$ (Use a correction factor due to seawater properties different from pure water = 1.22)

2-4) Calculate the overall heat transfer coefficient U

$$U = \frac{1}{\frac{1}{\alpha_1} + \frac{1}{\alpha_2}} = \frac{1}{\frac{1}{10{,}000} + \frac{1}{11{,}200}} = 5{,}283 \text{ W/}\left(\text{m}^2 \cdot \text{K}\right)$$

2-5) Estimate the individual flow rate q (use Figure E-4)

$q_1 = 1.3 \times 10^{-3} \text{ m}^3\text{/s}$

$q_2 = 1.2 \times 10^{-3} \text{ m}^3\text{/s}$

2-6) Calculate the number of plates (n)

$$n = \frac{2HV}{q} = \frac{2 \times 20 \times 147.5 \times 10^{-3}}{1.3 \times 10^{-3}} = 4{,}538$$

2-7) Find the rate of heat achieved with present design, and compare with required rate of heat

$$Q = Uan\Delta T = 5{,}055 \times 1.4 \times 4{,}538 \times 2.5 \times 10^{-3} = 83{,}910\,\mathrm{W}$$

Consider this number close to the heat rate required. Therefore the iteration stops.

2-8) calculate the heat exchanger area

$$A = n \times a = 4{,}538 \times 1.4\,\mathrm{m}^2 = 5{,}353\,\mathrm{m}^2 = 68{,}387\,\mathrm{ft}^2$$

B1: Brine/Seawater Sensible Heat Exchanger (Case B, Effect 1)

Brine	seawater
$T_G = 447.638\,\text{K}$	$T_F = 445.17\,\text{K}$
$T_H = 301\,\text{K}$	$T_D = 297\,\text{K}$
$C_p = 3.891\frac{\text{kJ}}{\text{kg K}}$	$C_p = 4.04\frac{\text{kJ}}{\text{kg K}}$

The log mean heat exchanger temperature difference is

$\Delta T = LMTD = 2.5\,\text{K}$

Mass balance:

$$m_G = m_F$$

The feed seawater flow is

$$m_G = 147.5\,\text{kg/s}$$

$$m_F = 147.5\,\text{kg/s}$$

1) Calculate the required rate of heat on the seawater to achieve the temperatures expected

Energy balance:

$$Q_{brine} = Q_{feed}$$

The rate of heat required is

$$m_G C_{p\,brine}(T_G - T_H) = m_F C_{p\,feed}\left(T_F - T_D\right)$$

$$Q = m_F \times C_{p\,feed} \times \left(T_F - T_D\right)$$

$$Q = 147.5\,\frac{\text{kg}}{\text{s}} \times 4.04\,\frac{\text{kJ}}{\text{kg}\cdot\text{K}} \times \left(445.17\text{ - }297\right)\text{K} = 88{,}294.5\,\text{kJ/s}$$

2) Heat exchanger design

2-1) given the mass flow on each side of the heat exchanger

$$m_G = 147.5\,\text{kg/s}$$

$$m_F = 147.5\,\text{kg/s}$$

Select the area from Figure E-2,

$$a = 1.4\,\text{m}^2$$

2-2) Define the pressure drop allowed on each side of the heat exchanger, and the number of passes.

Let $H = 20$

Find pressure drop:

$\Delta P_1 = \dfrac{740}{20} = 37\,\text{kN/m}^2$ (brine side) (use a correction factor = 0.86 due to seawater concentration compared to pure water)

$\Delta P_2 = \dfrac{740}{20} = 37\,\text{kN/m}^2$ (seawater side) (use a correction factor = 0.82 due to seawater concentration compared to pure water)

2-3) Find the heat transfer coefficients corresponding to each side. From Figure E-3

$$\alpha_1 = 10{,}800\ \text{W/}\left(\text{m}^2 \cdot \text{K}\right)$$

$$\alpha_2 = 11{,}040\ \text{W/}\left(\text{m}^2 \cdot \text{K}\right)$$

2-4) Calculate the overall heat transfer coefficient U

$$U = \frac{1}{\dfrac{1}{10{,}800} + \dfrac{1}{11{,}040}} = 5{,}462.1\ \text{W/}\left(\text{m}^2 \cdot \text{K}\right)$$

2-5) Estimate the individual flow rate q from Figure E-4

$$q_1 = 1.1 \times 10^{-3}\ \text{m}^3\text{/s}$$

$$q_2 = 1.2 \times 10^{-3}\ \text{m}^3\text{/s}$$

2-6) Calculate the number of plates (n)

$$n = \frac{2 \times H \times V}{q} = \frac{2 \times 20 \times 147.5 \times 10^{-3}}{1.2 \times 10^{-3}} = 4{,}917$$

2-7) Calculate the heat rate obtained with this arrangement

$$Q = U \times a \times n \times \Delta T = 5{,}462.1 \times 1.4 \times 4{,}917 \times 2.5 \times 10^{-3} = 94{,}000\ \text{kW}$$

Reduce the number of plates to 4,618 giving a heat rate value close to the calculated rate of heat required. Therefore the iteration is closed.

Find area

$$A = 1.4\ \text{m}^2 \times 4{,}618 = 6{,}466\ \text{m}^2 = 69{,}601\ \text{ft}^2$$

Total area for the sensible heat exchanger:

$$69{,}601\ \text{ft}^2 + 55{,}065\ \text{ft}^2 = 137{,}988\ \text{ft}^2$$

D2: Distillate/Seawater Sensible Heat Exchanger (Case B, Effect 2)

Distillate	seawater
$T_B = 447.638\ \text{K}$	$T_E = 445.17\ \text{K}$
$T_C = 301\ \text{K}$	$T_D = 297\ \text{K}$
$C_p = 4.36 \frac{\text{kJ}}{\text{kg K}}$	$C_p = 4.04 \frac{\text{kJ}}{\text{kg K}}$

The log mean heat exchanger temperature difference is

$$\Delta T = LMTD = 2.5\ \text{K}$$

Mass balance:

Required steam input = $m_{steam} = 146.03 \frac{\text{kg}}{\text{s}}$

Calculate m_D, the seawater feed flow required in the sensible heat exchanger

$$m_D = \left(\frac{1}{1 - \frac{3.5\%}{7\%}} \right) \times m_{steam} = 2 \times 146.03\ \frac{\text{kg}}{\text{s}} = 292.06 \frac{\text{kg}}{\text{s}}$$

This feed flow is divided into two parts. 50 % of the flow goes to the distillate/feed flow sensible heat exchanger.

$$m_E = 0.5 \times 292.06 = 146.03\ \text{kg/s}$$

50% of the flow goes to the brine/feed flow sensible heat exchanger

$$m_F = 0.5 \times 292.06 = 146.03 \text{ kg/s}$$

Energy balance:

$$Q = m_B C_{p\,B} \Delta T_{distillate} = m_E C_{p\,E} \Delta T_{seawater}$$

Therefore, the rate of heat required in the distillate/feed sensible heat exchanger is

$$Q_R = 146.03 \frac{\text{kg}}{\text{s}} \times 4.04 \frac{\text{kJ}}{\text{kg} \cdot \text{K}} \times \left(445.17 - 297\right) \text{K} = 87{,}414 \text{ kW}$$

2) Heat exchanger design

2-1) Given the mass flow on each side of the heat exchanger:

$$m_B = 146.03 \text{ kg/s}$$

$$m_E = 146.03 \text{ kg/s} = 0.146 \text{ m}^3\text{/s}$$

Select the area using Figure E-2

$$a = 1.4 \text{ m}^2$$

2-2) Define the pressure drop allowed on each side of the heat exchanger and number of passes $\left(H\right)$

Let:

$$\Delta P_1 = 800 \text{ kN/m}^2 \text{ (distillate side)}$$

$$\Delta P_2 = 800 \text{ kN/m}^2 \text{ (seawater side)}$$

$$H = 20$$

Therefore

$$\Delta P_1 = 800 / 20 \text{ kN/m}^2 = 40 \text{ kN/m}^2$$

$\Delta P_2 = 800/20\,\text{kN/m}^2 = 40\,\text{kN/m}^2$ (Use a correction factor due to seawater properties different from pure water = 0.82)

2-3) Find the heat transfer coefficients from Figure E-3

$\alpha_1 = 10{,}000\,\text{W/}\left(\text{m}^2 \cdot \text{K}\right)$

$\alpha_2 = 11{,}200\,\text{W/}\left(\text{m}^2 \cdot \text{K}\right)$ (Use a correction factor due to seawater properties different from pure water = 1.22)

2-4) Calculate the overall heat transfer coefficient U

$$U = \frac{1}{\dfrac{1}{\alpha_1} + \dfrac{1}{\alpha_2}} = \frac{1}{\dfrac{1}{10{,}000} + \dfrac{1}{110200}} = 5{,}283\,\text{W/}\left(\text{m}^2 \cdot \text{K}\right)$$

2-5) Estimate the individual flow rate q (use Figure E-4)

$q_1 = 1.3 \times 10^{-3}\ \text{m}^3\text{/s}$

$q_2 = 1.2 \times 10^{-3}\ \text{m}^3\text{/s}$

2-6) Calculate the number of plates (n)

$$n = \frac{2HV}{q} = \frac{2 \times 20 \times 146.03 \times 10^{-3}}{1.3 \times 10^{-3}} = 4{,}494$$

2-7) Find the rate of heat achieved with present design, and compare with required rate of heat

$Q = Uan\Delta T = 5{,}055 \times 1.4 \times 4{,}494 \times 2.5 \times 10^{-3} = 83{,}097\,\text{kW}$

Consider this value close to the required heat rate. Therefore the iteration stops.

2-8) calculate the heat exchanger area

$A = n \times a = 4{,}494 \times 1.4\,\text{m}^2 = 6{,}291\,\text{m}^2 = 67{,}724\,\text{ft}^2$

B2: Brine/Seawater Sensible Heat Exchanger (Case B, Effect 2)

Brine	seawater
$T_G = 443.782\,\mathrm{K}$	$T_F = 441.322\,\mathrm{K}$
$T_H = 301\,\mathrm{K}$	$T_D = 297\,\mathrm{K}$
$C_p = 3.891\dfrac{\mathrm{kJ}}{\mathrm{kg\ K}}$	$C_p = 4.04\dfrac{\mathrm{kJ}}{\mathrm{kg\ K}}$

The log mean heat exchanger temperature difference is

$$\Delta T = LMTD = 2.5\,\mathrm{K}$$

Mass balance:

$$m_G = m_F$$

The feed seawater flow is

$$m_G = 146.03\,\mathrm{kg/s}$$

$$m_F = 146.03\,\mathrm{kg/s}$$

1) Calculate the required rate of heat on the seawater to achieve the temperatures expected

Energy balance:

$$Q_{brine} = Q_{feed}$$

The rate of heat required is

$$m_G C_{p\,brine}(T_G - T_H) = m_F C_{p\ feed}\left(T_F - T_D\right)$$

$$Q = m_F \times C_{p\ feed} \times \left(T_F - T_D\right)$$

$$Q = 146.03\,\frac{\mathrm{kg}}{\mathrm{s}} \times 4.04\,\frac{\mathrm{kJ}}{\mathrm{kg\cdot K}} \times \left(441.322 - 297\right)\mathrm{K} = 85{,}144.38\ \mathrm{kJ/s}$$

2) Heat exchanger design

2-1) given the mass flow on each side of the heat exchanger

$$m_G = 146.03\ \mathrm{kg/s}$$

$$m_F = 146.03\ \mathrm{kg/s}$$

Select the area from Figure E-2,

$$a = 1.4\ \mathrm{m}^2$$

2-2) Define the pressure drop allowed on each side of the heat exchanger, and the number of passes.

Let $H = 20$

Find pressure drop:

$\Delta P_1 = \frac{740}{20} = 37\ \mathrm{kN/m^2}$ (brine side) (correction factor = 0.86 due to brine concentration with respect to pure water)

$\Delta P_2 = \frac{740}{20} = 37\ \mathrm{kN/m^2}$ (seawater side) (correction factor = 0.82 due to seawater feed concentration with respect to pure water)

2-3) Find the heat transfer coefficients corresponding to each side. From Figure E-3

$$\alpha_1 = 10{,}920\ \mathrm{W/\left(m^2 \cdot K\right)}$$

$$\alpha_2 = 10{,}800\ \mathrm{W/\left(m^2 \cdot K\right)}$$

2-4) Calculate the overall heat transfer coefficient U

$$U = \frac{1}{\frac{1}{10{,}800} + \frac{1}{10{,}920}} = 5{,}447.36\ \mathrm{W/\left(m^2 \cdot K\right)}$$

2-5) Estimate the individual flow rate q from Figure E-4

$$q_1 = 1.1 \times 10^{-3}\ \mathrm{m^3/s}$$

$$q_2 = 1.0 \times 10^{-3}\ \mathrm{m^3/s}$$

2-6) Calculate the number of plates (n)

$$n = \frac{2 \times H \times V}{q} = \frac{2 \times 20 \times 146.03 \times 10^{-3}}{1.1 \times 10^{-3}} = 5{,}310$$

2-7) Calculate the heat rate obtained with this arrangement

$$Q = U \times a \times n \times \Delta T = 5{,}447.36 \times 1.4 \times 5{,}310 \times 2.5 \times 10^{-3} = 101{,}239\ \mathrm{kW}$$

Reduce the number of plates to 4,463giving a heat rate value close to the calculated rate of heat required. Therefore the iteration is closed.

Find area

$$A = 1.4\ \mathrm{m}^2 \times 4{,}463 = 6{,}248\ \mathrm{m}^2 = 67{,}255\ \mathrm{ft}^2$$

Total area for the sensible heat exchanger in stage 2 is

67,724 ft^2 + 67,255 ft^2 = 134,979 ft^2

D3: Distillate/Seawater Sensible Heat Exchanger (Case B, Effect 3)

Distillate	seawater
$T_B = 443.782\ \text{K}$	$T_E = 441.322\ \text{K}$
$T_C = 301\ \text{K}$	$T_D = 297\ \text{K}$
$C_p = 4.36\,\frac{\text{kJ}}{\text{kg K}}$	$C_p = 4.04\,\frac{\text{kJ}}{\text{kg K}}$

The log mean heat exchanger temperature difference is

$$\Delta T = LMTD = 2.5\ \text{K}$$

Mass balance:

Required steam input = $m_{steam} = 144.57\,\frac{\text{kg}}{\text{s}}$

Calculate m_D, the seawater feed flow required in the sensible heat exchanger

$$m_D = \left(\frac{1}{1-\frac{3.5\%}{7\%}}\right)\times m_{steam} = 2\times 144.57\ \frac{\text{kg}}{\text{s}} = 289.14\,\frac{\text{kg}}{\text{s}}$$

This feed flow is divided into two parts. 50 % of the flow goes to the distillate/feed flow sensible heat exchanger.

$$m_E = 0.5\times 289.14 = 144.57\ \text{kg/s}$$

50% of the flow goes to the brine/feed flow sensible heat exchanger

$$m_F = 0.5 \times 289.14 = 144.57 \text{ kg/s}$$

Energy balance:

$$Q = m_B C_{p\,B} \Delta T_{distillate} = m_E C_{p\,E} \Delta T_{seawater}$$

Therefore, the rate of heat required in the distillate/feed sensible heat exchanger is

$$Q_R = 144.57 \frac{\text{kg}}{\text{s}} \times 4.04 \frac{\text{kJ}}{\text{kg} \cdot \text{K}} \times \left(441.322 - 297\right) \text{K} = 84.293 \text{ kW}$$

2) Heat exchanger design

2-1) Given the mass flow on each side of the heat exchanger:

$$m_B = 144.57 \text{ kg/s}$$

$$m_E = 144.57 \text{ kg/s} = 0.144 \text{ m}^3\text{/s}$$

Select the area using Figure E-2

$$a = 1.4 \text{ m}^2$$

2-2) Define the pressure drop allowed on each side of the heat exchanger and number of passes $\left(H\right)$

Let:

$$\Delta P_1 = 660 \text{ kN/m}^2 \text{ (distillate side)}$$

$$\Delta P_2 = 660 \text{ kN/m}^2 \text{ (seawater side)}$$

$$H = 20$$

Therefore

$$\Delta P_1 = 660 / 20 \text{ kN/m}^2 = 33 \text{ kN/m}^2$$

$\Delta P_2 = 660/20\,\text{kN/m}^2 = 33\,\text{kN/m}^2$ (Use a correction factor due to seawater properties different from pure water = 0.82)

2-3) Find the heat transfer coefficients from Figure E-3

$\alpha_1 = 9{,}500\ \text{W/}\left(\text{m}^2\cdot\text{K}\right)$

$\alpha_2 = 10{,}200\ \text{W/}\left(\text{m}^2\cdot\text{K}\right)$ (Use a correction factor due to seawater properties different from pure water = 1.22)

2-4) Calculate the overall heat transfer coefficient U

$$U = \frac{1}{\frac{1}{\alpha_1}+\frac{1}{\alpha_2}} = \frac{1}{\frac{1}{9{,}500}+\frac{1}{10{,}200}} = 4{,}925\ \text{W/}\left(\text{m}^2\cdot\text{K}\right)$$

2-5) Estimate the individual flow rate q (use Figure E-4)

$q_1 = 1.2\times 10^{-3}\ \text{m}^3\text{/s}$

$q_2 = 1.8\times 10^{-4}\ \text{m}^3\text{/s}$

2-6) Calculate the number of plates (n)

$$n = \frac{2HV}{q} = \frac{2\times 20\times 144.57\times 10^{-3}}{1.2\times 10^{-3}} = 4{,}819$$

2-7) Find the rate of heat achieved with present design, and compare with required rate of heat

$$Q = U\times a\times n\times \Delta T = 4{,}925\times 1.4\times 4{,}819\times 2.5\times 10^{-3} = 83{,}067\ \text{kW}$$

Consider this values close to the rate of heat required. The iteration stops.

Find the area

$$A = 1.4\ \mathrm{m}^2 \times 4{,}819 = 6{,}746\ \mathrm{m}^2\ = 72{,}615\ \mathrm{ft}^2$$

B3: Brine/Seawater Sensible Heat Exchanger (Case B, Effect 3)

Brine	seawater
$T_G = 439.957\ \text{K}$	$T_F = 437.497\ \text{K}$
$T_H = 301\ \text{K}$	$T_D = 297\ \text{K}$
$C_p = 3.891 \frac{\text{kJ}}{\text{kg K}}$	$C_p = 4.04 \frac{\text{kJ}}{\text{kg K}}$

The log mean heat exchanger temperature difference is

$\Delta T = LMTD = 2.5\ \text{K}$

Mass balance:

$$m_G = m_F$$

The feed seawater flow is

$$m_G = 144.57\ \text{kg/s}$$

$$m_F = 144.57\ \text{kg/s}$$

1) Calculate the required rate of heat on the seawater to achieve the temperatures expected

Energy balance:

$$Q_{brine} = Q_{feed}$$

The rate of heat required is

$$m_G C_{p_{brine}} (T_G - T_H) = m_F C_{p_{feed}} \left(T_F - T_D\right)$$

$$Q = m_F \times C_{p_{feed}} \times \left(T_F - T_D\right)$$

$$Q = 144.57 \frac{\mathrm{kg}}{\mathrm{s}} \times 4.04 \frac{\mathrm{kJ}}{\mathrm{kg} \cdot \mathrm{K}} \times \left(437.497 - 297\right) \mathrm{K} = 82{,}059.07 \ \mathrm{kJ/s}$$

2) Heat exchanger design

2-1) given the mass flow on each side of the heat exchanger

$$m_G = 146.03 \ \mathrm{kg/s}$$

$$m_F = 146.03 \ \mathrm{kg/s}$$

Select the area from Figure E-2,

$$a = 1.4 \ \mathrm{m}^2$$

2-2) Define the pressure drop allowed on each side of the heat exchanger, and the number of passes.

Let $H = 20$

Find pressure drop:

$\Delta P_1 = \frac{740}{20} = 37 \ \mathrm{kN/m}^2$ (brine side) (correction factor = 0.86 due to brine concentration with respect to pure water)

$\Delta P_2 = \frac{740}{20} = 37 \ \mathrm{kN/m}^2$ (seawater side) (correction factor = 0.82 due to seawater feed concentration with respect to pure water)

2-3) Find the heat transfer coefficients corresponding to each side. From Figure E-3

$$\alpha_1 = 10{,}920 \ \mathrm{W/\left(m^2 \cdot K\right)}$$

$$\alpha_2 = 10{,}800 \ \mathrm{W/\left(m^2 \cdot K\right)}$$

2-4) Calculate the overall heat transfer coefficient U

$$U = \frac{1}{\frac{1}{10{,}800} + \frac{1}{10{,}920}} = 5{,}447.36 \ \mathrm{W/\left(m^2 \cdot K\right)}$$

2-5) Estimate the individual flow rate q from Figure E-4

$$q_1 = 1.1 \times 10^{-3} \ \mathrm{m^3/s}$$

$$q_2 = 1.0 \times 10^{-3} \ \mathrm{m^3/s}$$

2-6) Calculate the number of plates (n)

$$n = \frac{2 \times H \times V}{q} = \frac{2 \times 20 \times 146.03 \times 10^{-3}}{1.1 \times 10^{-3}} = 5{,}310$$

2-7) Calculate the heat rate obtained with this arrangement

$$Q = U \times a \times n \times \Delta T = 5{,}447.36 \times 1.4 \times 5{,}310 \times 2.5 \times 10^{-3} = 101{,}239.1 \ \mathrm{kW}$$

Reduce the number of plates to 4,300 giving a heat rate value close to the calculated rate of heat required. Therefore the iteration is closed.

Find area

$$A = 1.4 \ \mathrm{m^2} \times 4{,}300 = 6{,}020 \ \mathrm{m^2} = 64{,}800 \ \mathrm{ft^2}$$

Total area for the sensible heat exchanger in stage 3 is

$$72{,}615 \ \mathrm{ft^2} + 64{,}800 \ \mathrm{ft^2} = 137{,}415 \ \mathrm{ft^2}$$

Table E-1 summarizes the calculated individual sensible heat exchanger areas.

Table E-1. Calculated sensible heat exchanger area (ft^2)

Stage	Distillate/seawater	Brine/seawater	Stage area
1	68,387	69,601	137,988
2	67,724	67,255	134,979
3	72,615	64,800	137,415
TOTAL			**410,382**

To scale up the sensible heat exchanger area for Case C (S_c) a temperature relationship has been used as follows:

$$\text{Area Case C } (S_c) = \left(\frac{2.5}{1.75}\right) S_B$$

where

$(S_B) =$ sensible heat exchanger area for Case B

The corresponding scaling ratio for Case A is

$$\text{Area Case A } (S_A) = \left(\frac{2.5}{3.25}\right) S_B$$

Results are summarized in Tables 9.1 and 9.2

LATENT HEAT EXCHANGER CALCULATION

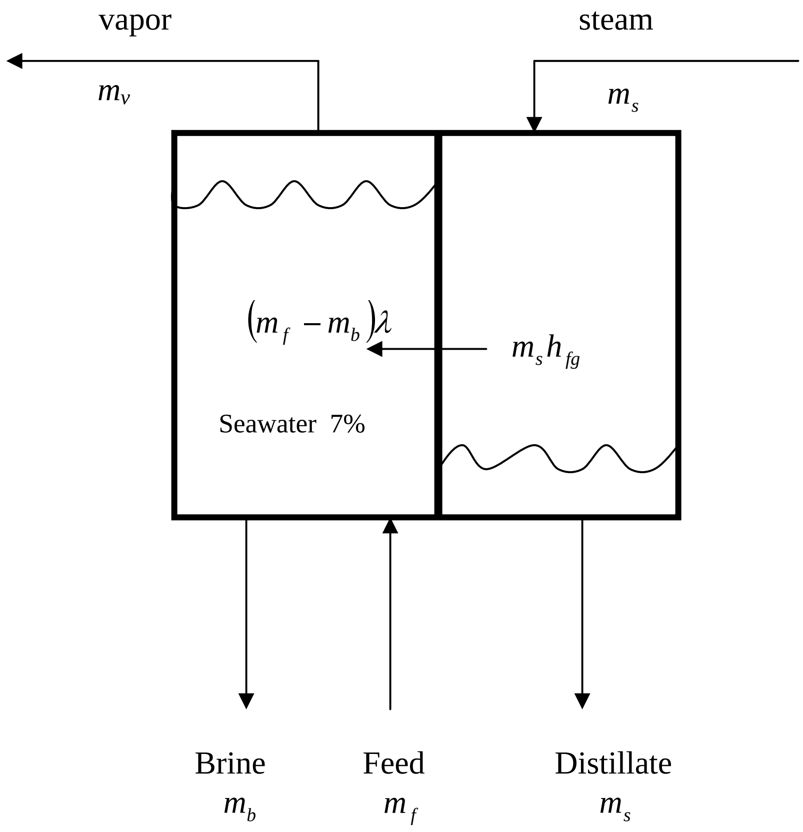


Figure E-1. Heat transfer arrangement in the latent heat exchanger.

Figure E.1 shows nomenclature and flows direction in the latent heat exchanger. In steady-state flow, the seawater concentration on the liquid side is considered to be 7%.

Before sizing the heat exchanger, the following must be resolved:

- Mass and energy balance (Chapter IV).
- Factors affecting the evaporator performance (Chapter VI and Appendix A-2).

Although the optimization process performed in Chapter V delivers a heat transfer coefficient of $278{,}242\,\mathrm{W/\left(m^2 \cdot K\right)}$ $\left(49{,}000\,\mathrm{Btu/\left(h \cdot ft^2 \cdot {}^\circ F\right)}\right)$ the latent heat exchanger design will use a conservative value $\left(24{,}900\,\mathrm{Btu/\left(h \cdot ft^2 \cdot {}^\circ F\right)}\right)$ for all Cases A, B and C

L1: Latent Heat Exchanger (Case B, Effect 1)

From the relation

$$q = AU\Delta T$$

The required heat transfer area is calculated as

$$A = \frac{m_s \left[h_{fg_s} + \left(\frac{1}{1 - \frac{3.5\%}{7\%}} \right) \times C_p \left(\Delta T_{sens} \right) \right]}{U_1 \Delta T_{HX}}$$

where

m_s = entering steam = 147.5 kg/s $\left(324{,}5\ \mathrm{lb_m/s} = 1{,}168{,}200\ \mathrm{lb_m/h}\right)$

h_{fg} = enthalpy of vaporization of steam at entering conditions = 870.5 $\mathrm{Btu/lb_m}$

$C_{p\,f}$ = seawater heat capacity at 3.5% = 4.04 $\mathrm{kJ/(kg \cdot K)}$ $\left(0.9651\ \mathrm{Btu/(lb_m \cdot {}^\circ F)}\right)$

ΔT_{sens} = sensible temperature increment = 2.5 °K (5 °F) (required, see Appendix A)

U_1 = calculated overall heat transfer coefficient = $24{,}900 \dfrac{\mathrm{Btu}}{\mathrm{h \cdot ft^2 \cdot {}^\circ F}}$

ΔT_{HX} = optimum overall heat exchanger temperature difference = 2.222 K (4 °F)

Substituting

$$A = \frac{1{,}168{,}200\,\frac{\text{lb}_\text{m}}{\text{h}}\left[870.5\,\frac{\text{Btu}}{\text{lb}_\text{m}} + (2)\left(0.9651\,\frac{\text{Btu}}{\text{lb}_\text{m}\cdot{}^\circ\text{F}}\right)(4.44\,^\circ\text{F})\right]}{\left(24{,}900\,\frac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}\right)(4\,^\circ\text{F})}$$

$$= 10{,}311\ \text{ft}^2$$

(Case A, Effect 1)

From Table 8.2

$\Delta T_{sens} = 3.25\,^\circ\text{K}\ (5.85\,^\circ\text{F})$

$\Delta T_{sens} = 3.333\,^\circ\text{K}\ (6\,^\circ\text{F})$

Therefore

$$A = \frac{1{,}168{,}200\,\frac{\text{lb}_\text{m}}{\text{h}}\left[870.5\,\frac{\text{Btu}}{\text{lb}_\text{m}} + (2)\left(0.9651\,\frac{\text{Btu}}{\text{lb}_\text{m}\cdot{}^\circ\text{F}}\right)(5.85\,^\circ\text{F})\right]}{\left(24{,}900\,\frac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}\right)(6\,^\circ\text{F})}$$

$$= 6{,}895\ \text{ft}^2$$

(Case C, Effect 1)

From Table 8.2

$\Delta T_{sens} = 1.75\,^\circ\text{K}\ (3.15\,^\circ\text{F})$

$\Delta T_{sens} = 1.111\,^\circ\text{K}\ (2\,^\circ\text{F})$

Therefore

$$A = \frac{1{,}168{,}200\,\frac{\text{lb}_\text{m}}{\text{h}}\left[870.5\,\frac{\text{Btu}}{\text{lb}_\text{m}} + (2)\left(0.9651\,\frac{\text{Btu}}{\text{lb}_\text{m}\cdot{}^\circ\text{F}}\right)(3.15\,^\circ\text{F})\right]}{\left(24{,}900\,\frac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}\right)(2\,^\circ\text{F})}$$

$$= 20{,}551\ \text{ft}^2$$

L2: Latent Heat Exchanger (Case B, Effect 2)

The required amount of steam supplied to this stage is

(0.99)(147.5 kg/s) = 146.03 kg/s = 321.26 lb_m/s = 1,156,557.6 lb_m/h

The required heat transfer area is calculated as

$$A = \frac{m_s \left[h_{fg} + \left(\dfrac{1}{1 - \dfrac{3.5\%}{7\%}} \right) \times Cp_f \left(\Delta T_{sens} \right) \right]}{U_1 \Delta T_{HX}}$$

where

m_s = entering steam = 1,156,557.6 lb_m / h

$h_{fg_s} = \lambda$ = enthalpy of vaporization of steam at entering conditions = 876.6 Btu/lb_m

$C_{p\,f}$ = seawater heat capacity at 3.5% = 0.9651 Btu/(lb_m · °F)

ΔT_{sens} = sensible temperature increment = 4.44 °F (required)

U_1 = calculated overall heat transfer coefficient = 24,900 $\dfrac{\text{Btu}}{\text{h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}}$

ΔT_{HX} = optimum overall heat exchanger temperature difference = 4 °F

Substituting

$$A = \frac{1{,}156{,}557.6 \ \dfrac{\text{lb}_\text{m}}{\text{h}} \left[876.6 \dfrac{\text{Btu}}{\text{lb}_\text{m}} + (2) \left(0.9651 \dfrac{\text{Btu}}{\text{lb}_\text{m} \cdot {}^\circ\text{F}} \right) (4.44\ {}^\circ\text{F}) \right]}{\left(24{,}900 \dfrac{\text{Btu}}{\text{h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}} \right) (4\ {}^\circ\text{F})}$$

$$= 10{,}279\ \text{ft}^2$$

(Case A, Effect 2)

From Table 8.2

$\Delta T_{sens} = 3.25\,°\mathrm{K}\ (5.85\,°\mathrm{F})$

$\Delta T_{sens} = 3.333\,°\mathrm{K}\ (6\,°\mathrm{F})$

Therefore

$$A = \frac{1{,}156{,}557.6\,\frac{\mathrm{lb_m}}{\mathrm{h}}\left[876.6\,\frac{\mathrm{Btu}}{\mathrm{lb_m}} + (2)\left(0.9651\,\frac{\mathrm{Btu}}{\mathrm{lb_m}\cdot°\mathrm{F}}\right)(5.85\,°\mathrm{F})\right]}{\left(24{,}900\,\frac{\mathrm{Btu}}{\mathrm{h}\cdot\mathrm{ft}^2\cdot°\mathrm{F}}\right)(6\,°\mathrm{F})}$$

$$= 6{,}874\ \mathrm{ft}^2$$

(Case C, Effect 2)

From Table 8.2

$\Delta T_{sens} = 1.75\,°\mathrm{K}\ (3.15\,°\mathrm{F})$

$\Delta T_{sens} = 1.111\,°\mathrm{K}\ (2\,°\mathrm{F})$

Therefore

$$A = \frac{1{,}156{,}557.6\,\frac{\mathrm{lb_m}}{\mathrm{h}}\left[876.6\,\frac{\mathrm{Btu}}{\mathrm{lb_m}} + (2)\left(0.9651\,\frac{\mathrm{Btu}}{\mathrm{lb_m}\cdot°\mathrm{F}}\right)(3.15\,°\mathrm{F})\right]}{\left(24{,}900\,\frac{\mathrm{Btu}}{\mathrm{h}\cdot\mathrm{ft}^2\cdot°\mathrm{F}}\right)(2\,°\mathrm{F})}$$

$$= 34{,}347\ \mathrm{ft}^2$$

L3: Latent Heat Exchanger (Case B, Effect 3)

The required amount of steam supplied to this stage is

(0.99) (146.03 kg/s) = 144.57 kg/s = 318.05 lb_m/s = 1,144,994.4 lb_m/h

The required heat transfer area is calculated as

$$A = \frac{m_s\left[h_{fg} + \left(\frac{1}{1-\frac{3.5\%}{7\%}}\right)\times C_{p\,f}\left(\Delta T_{sens}\right)\right]}{U_1 \Delta T_{HX}}$$

where

$$m_s = \text{entering steam} = 1{,}144{,}994.4\,\frac{\text{lb}_\text{m}}{\text{h}}$$

$$h_{fg} = \lambda = \text{enthalpy of vaporization of steam at entering conditions} = 881.9\,\frac{\text{Btu}}{\text{lb}_\text{m}}$$

$$C_{p\,f} = \text{seawater heat capacity at 3.5\%} = 4.04\ \text{kJ/}(\text{kg}\cdot\text{K})\left(0.9651\frac{\text{Btu}}{\text{lb}_\text{m}\,°\text{F}}\right)$$

$$\Delta T_{sens} = \text{sensible temperature increment} = 2.46\ \text{K}\ (4.44\ °\text{F})\ \ \text{(Required)}$$

$$U_1 = \text{calculated overall heat transfer coefficient} = 24{,}900\frac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot°\text{F}}$$

$$\Delta T_{HX} = \text{optimum overall heat exchanger temperature difference} = 2.222\ \text{K}\ (4\ °\text{F})$$

Substituting

$$A = \frac{1{,}144{,}994.4\ \dfrac{\text{lb}_\text{m}}{\text{h}}\left[881.90\dfrac{\text{Btu}}{\text{lb}_\text{m}} + (2)\left(0.9651\dfrac{\text{Btu}}{\text{lb}_\text{m}\cdot{}^\circ\text{F}}\right)(4.44\,{}^\circ\text{F})\right]}{\left(24{,}900\dfrac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}\right)(4\,{}^\circ\text{F})}$$

$$= 10{,}237\ \text{ft}^2$$

(Case A, Effect 3)

From Table 8.2

$\Delta T_{sens} = 3.25\,{}^\circ\text{K}\ (5.85\,{}^\circ\text{F})$

$\Delta T_{sens} = 3.333\,{}^\circ\text{K}\ (6\,{}^\circ\text{F})$

Therefore

$$A = \frac{1{,}144{,}994.4\dfrac{\text{lb}_\text{m}}{\text{h}}\left[881.90\dfrac{\text{Btu}}{\text{lb}_\text{m}} + (2)\left(0.9651\dfrac{\text{Btu}}{\text{lb}_\text{m}\cdot{}^\circ\text{F}}\right)(5.85\,{}^\circ\text{F})\right]}{\left(24{,}900\dfrac{\text{Btu}}{\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F}}\right)(6\,{}^\circ\text{F})}$$

$$= 6{,}846\ \text{ft}^2$$

(Case C, Effect 3)

From Table 8.2

$\Delta T_{sens} = 1.75\,{}^\circ\text{K}\ (3.15\,{}^\circ\text{F})$

$\Delta T_{sens} = 1.111\,{}^\circ\text{K}\ (2\,{}^\circ\text{F})$

Therefore

$$A = \frac{1{,}144{,}994.4\,\frac{\mathrm{lb_m}}{\mathrm{h}}\left[881.9\,\frac{\mathrm{Btu}}{\mathrm{lb_m}} + (2)\left(0.9651\,\frac{\mathrm{Btu}}{\mathrm{lb_m}\cdot{}^\circ\mathrm{F}}\right)(3.15\,{}^\circ\mathrm{F})\right]}{\left(24{,}900\,\frac{\mathrm{Btu}}{\mathrm{h}\cdot\mathrm{ft}^2\cdot{}^\circ\mathrm{F}}\right)(2\,{}^\circ\mathrm{F})}$$

$$= 20{,}417\ \mathrm{ft}^2$$

Table E-2. Calculated individual latent heat exchanger area (ft^2)

Case B	
Effect	**Area**
1	10,311
2	10,279
3	10,237
Total latent heat exchanger area	30,827
Case A	
1	6,895
2	6,874
3	6,846
Total latent heat exchanger area	20,615
Case C	
1	20,551
2	20,499
3	20,417
Total latent heat exchanger area	61,467

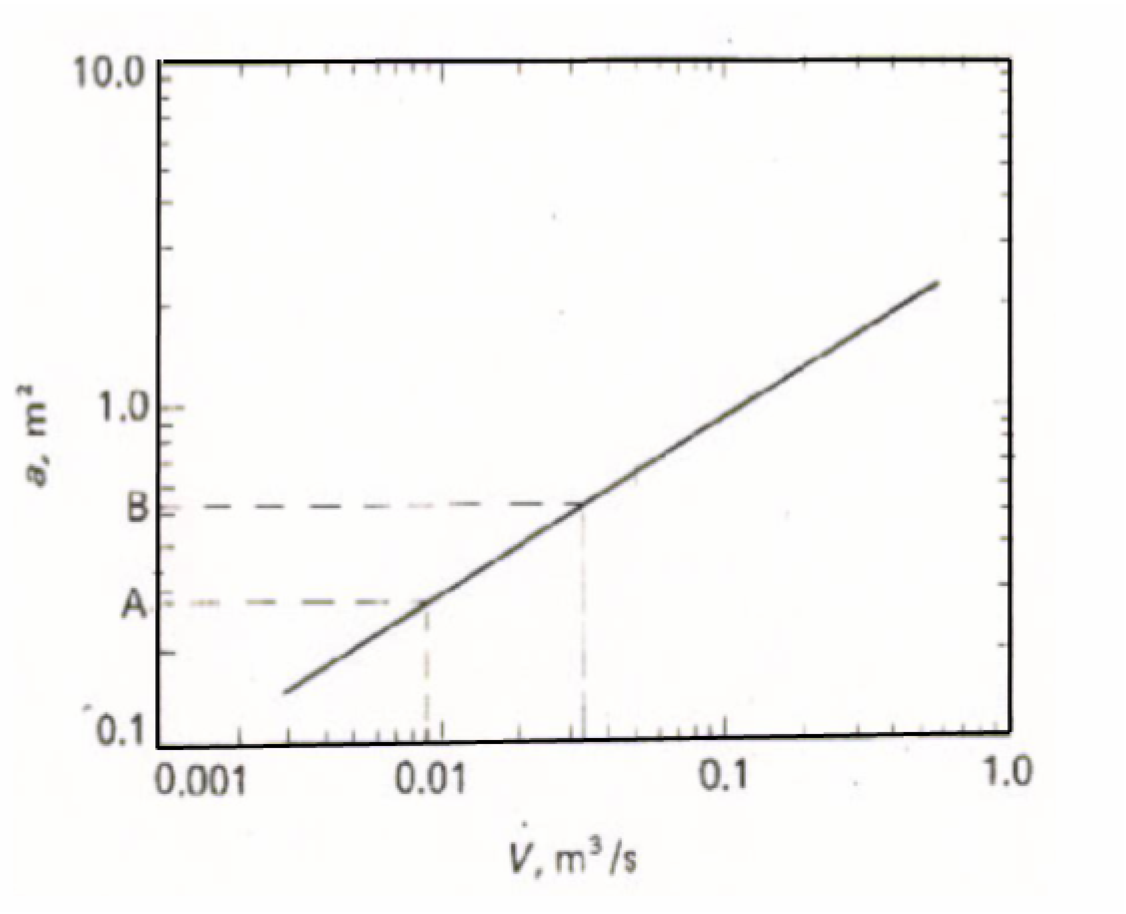


Figure E-2. Plate area/total flow rate [32].

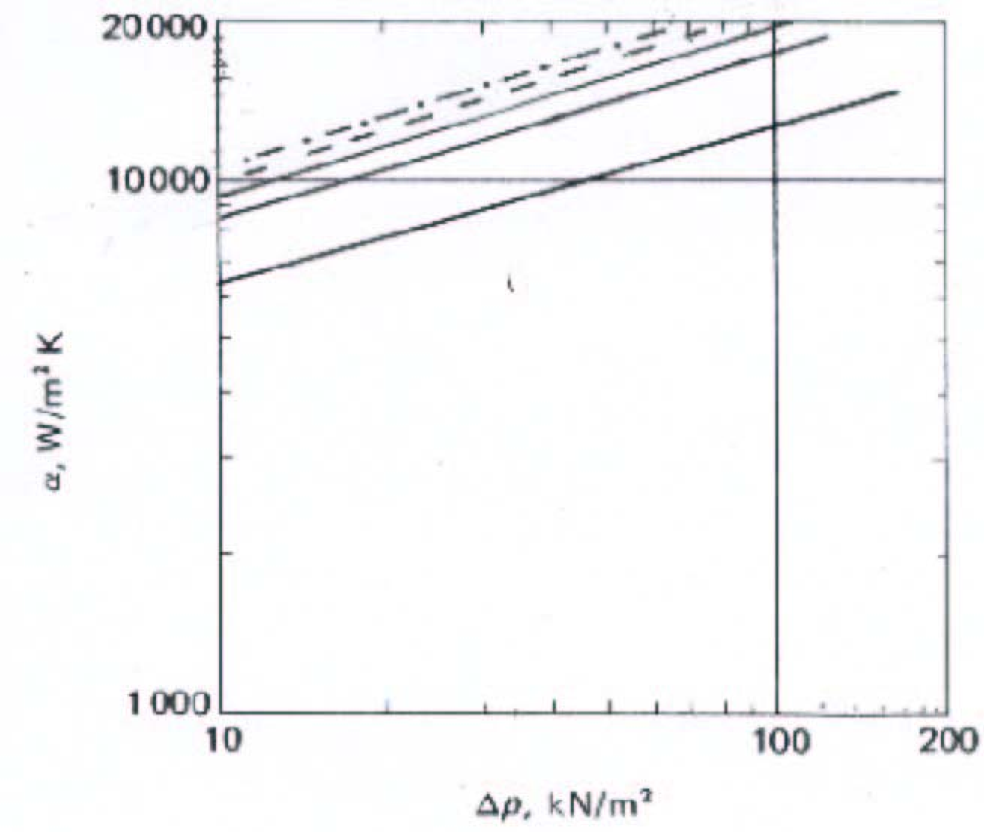


Figure E-3. Film heat transfer coefficient α (W/m^2) / pressure drop [32].

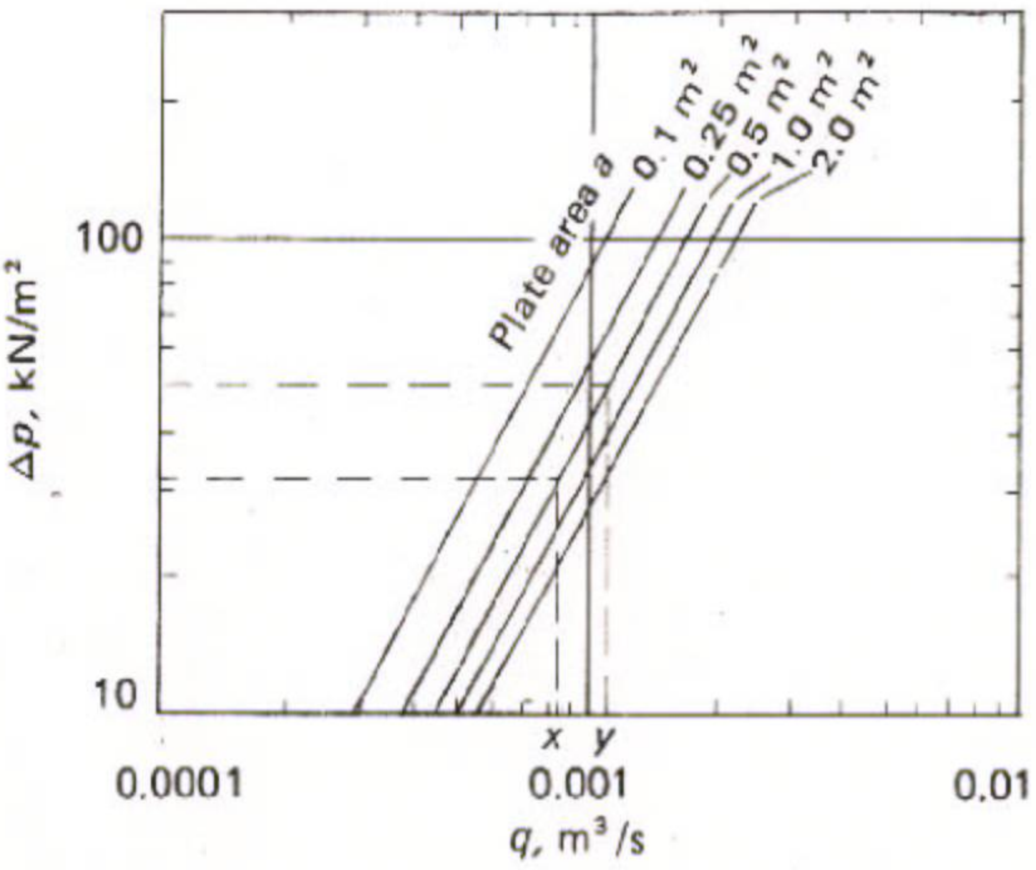


Figure E-4. Pressure drop/plate flow rate for different plate areas on water [32].

APPENDIX F

DRIVING POWER SCHEME

Waste heat from the gas turbine exhaust is recovered in the heat recovery boiler HRB which is an unfired boiler (see Appendix G) that generates steam to drive the steam turbine. The gas turbine and the steam turbine drive the compressor. Excess shaft power is used in the generator to make electricity used for pumps, or to be sold to the grid.

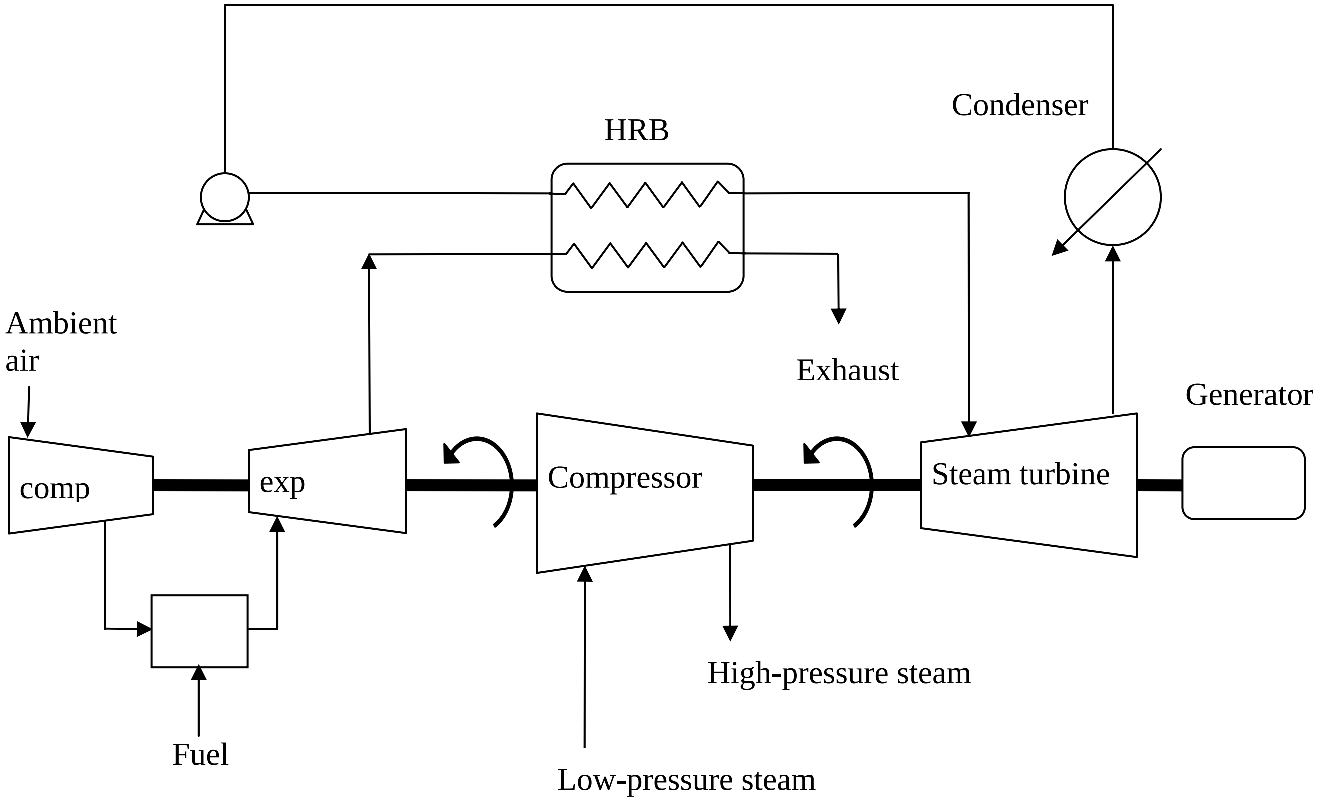
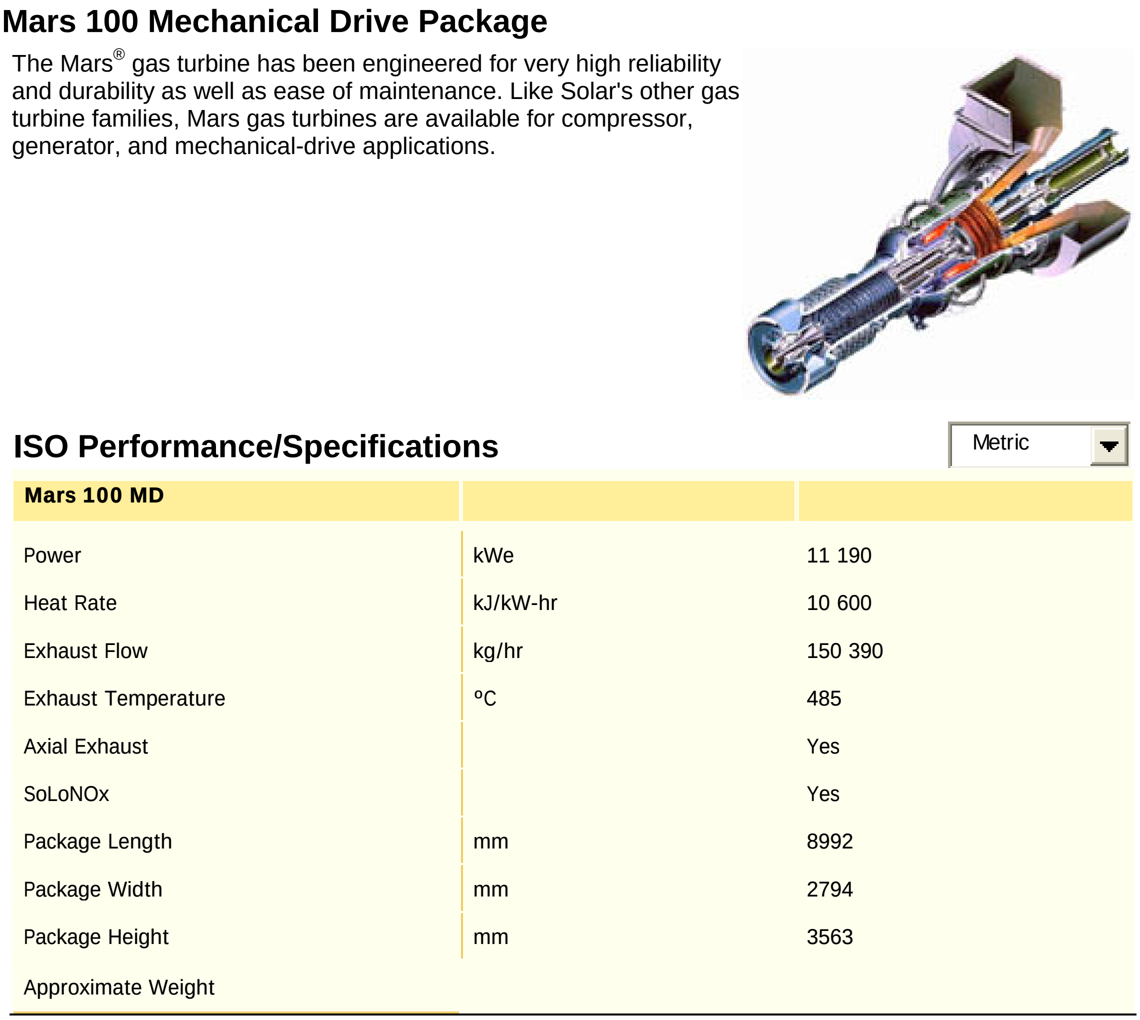


Figure F-1. Combined-cycle power system.

GT-1: Gas Turbine



Mars 100 Mechanical Drive Package

The Mars® gas turbine has been engineered for very high reliability and durability as well as ease of maintenance. Like Solar's other gas turbine families, Mars gas turbines are available for compressor, generator, and mechanical-drive applications.

ISO Performance/Specifications

Metric

Mars 100 MD		
Power	kWe	11 190
Heat Rate	kJ/kW-hr	10 600
Exhaust Flow	kg/hr	150 390
Exhaust Temperature	ºC	485
Axial Exhaust		Yes
SoLoNOx		Yes
Package Length	mm	8992
Package Width	mm	2794
Package Height	mm	3563
Approximate Weight		

Figure F-2. Gas turbine delivered power and performance specifications (from Solar Turbines Manufacturing Co.).

Figure F-2 shows the performance specifications for the selected gas turbine. The gas turbine delivers 11,190 kW at its maximum capacity.

APPENDIX G

UNFIRED WASTE-HEAT-STEAM GENERATION

Guidelines from Reference [39] provide a criterion on the amount of steam that could be reasonably generated using an unfired waste-heat-steam generator. The gas turbine exhaust flue-gas temperature drop could practically be reduced to 65 or to $100\,°\mathrm{F}$ above the boiler feedwater temperature. It will be assumed that feedwater is supplied at $115\,°\mathrm{F}$ from the condenser. Using an approach temperature of $65\,°\mathrm{F}$ yields an exit-flue gas temperature of $180\,°\mathrm{F}$.

An energy balance on the flue-gas side based on the temperature drop provides the total heat available for steam generation.

The sensible heat of the flue-gas is derived from Figures G-1 and G-2 [39] based on the flue-gas temperature and percent moisture.

The percentage moisture in the flue-gas is a function of type of fuel fired in the gas turbine and percentage of excess air operation in its combustor. A typical value of percentage moisture for natural gas [39] with 10% excess air is 12.1%. Then, from Figure G-1 considering that the waste heat gas exits at $905\,°\mathrm{F}$, the total sensible heat available from the flue gas for steam generation becomes

(330,858 lb/h W.G.) (230 Btu/lb W.G.) = 76,097,340 Btu/h

where W.G. stands for waste gas.

The amount of steam generated is determined by an energy balance on the steam side.

To generate superheated steam @ $752\,^{\circ}\mathrm{F}$, it will be assumed [39] that boiler blowdown is 10% of steam flow. So, feedwater flow through the boiler drum is 1.10 times the steam outflow from the boiler drum.

Let $X =$ steam outflow

$h_1 = \text{enthalpy of steam} = 1377.47\ \text{Btu/lb}_\text{m}$

$h_3 = \text{enthalpy of feedwater} = 82.97\ \text{Btu/lb}_\text{m}$

Then, the energy balance equals the heat absorbed by the waste-heat-steam generator to the heat available from reducing the flue-gas temperature from 905 to $180\,^{\circ}\mathrm{F}$

$$(1.10)(X)(h_1) + (X)(h_3) = 76{,}097{,}340\,\frac{\text{Btu}}{\text{h}}$$

Therefore, solving for X

$$(1.10)(X)\left(1377.47\,\frac{\text{Btu}}{\text{lb}_\text{m}}\right) + (X)\left(82.97\,\frac{\text{Btu}}{\text{lb}_\text{m}}\right) = 76{,}097{,}340\,\frac{\text{Btu}}{\text{h}}$$

$$X = \frac{76{,}097{,}340\,\frac{\text{Btu}}{\text{h}}}{(1.10)\left(1377.47\,\frac{\text{Btu}}{\text{lb}_\text{m}}\right) + \left(82.97\,\frac{\text{Btu}}{\text{lb}_\text{m}}\right)} = 47{,}615\,\frac{\text{lb}_\text{m}}{\text{h}}$$

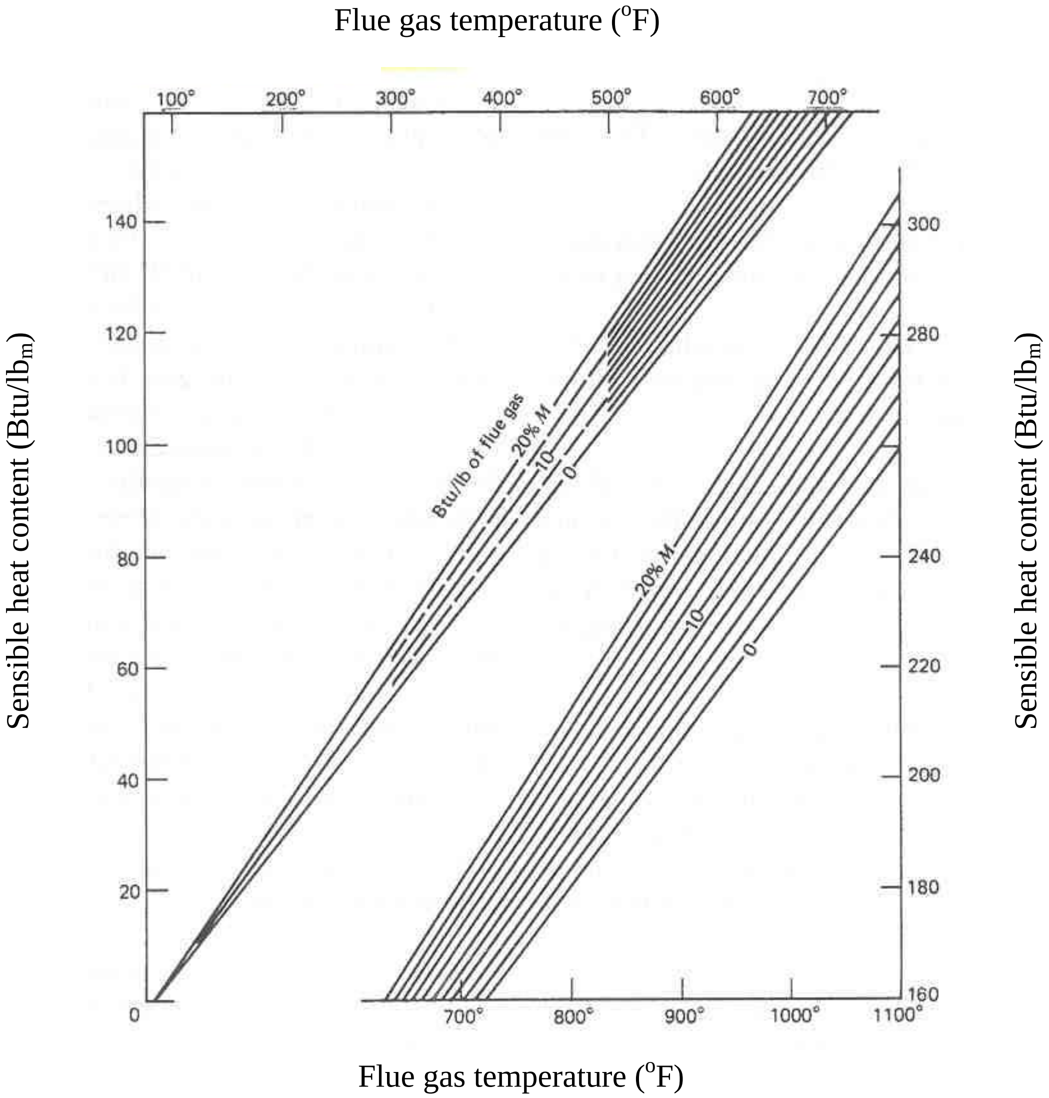


Figure G.1 Heat in flue gases vs percent moisture by weight. [39]. Derived from Keenan and Kayes 1948.

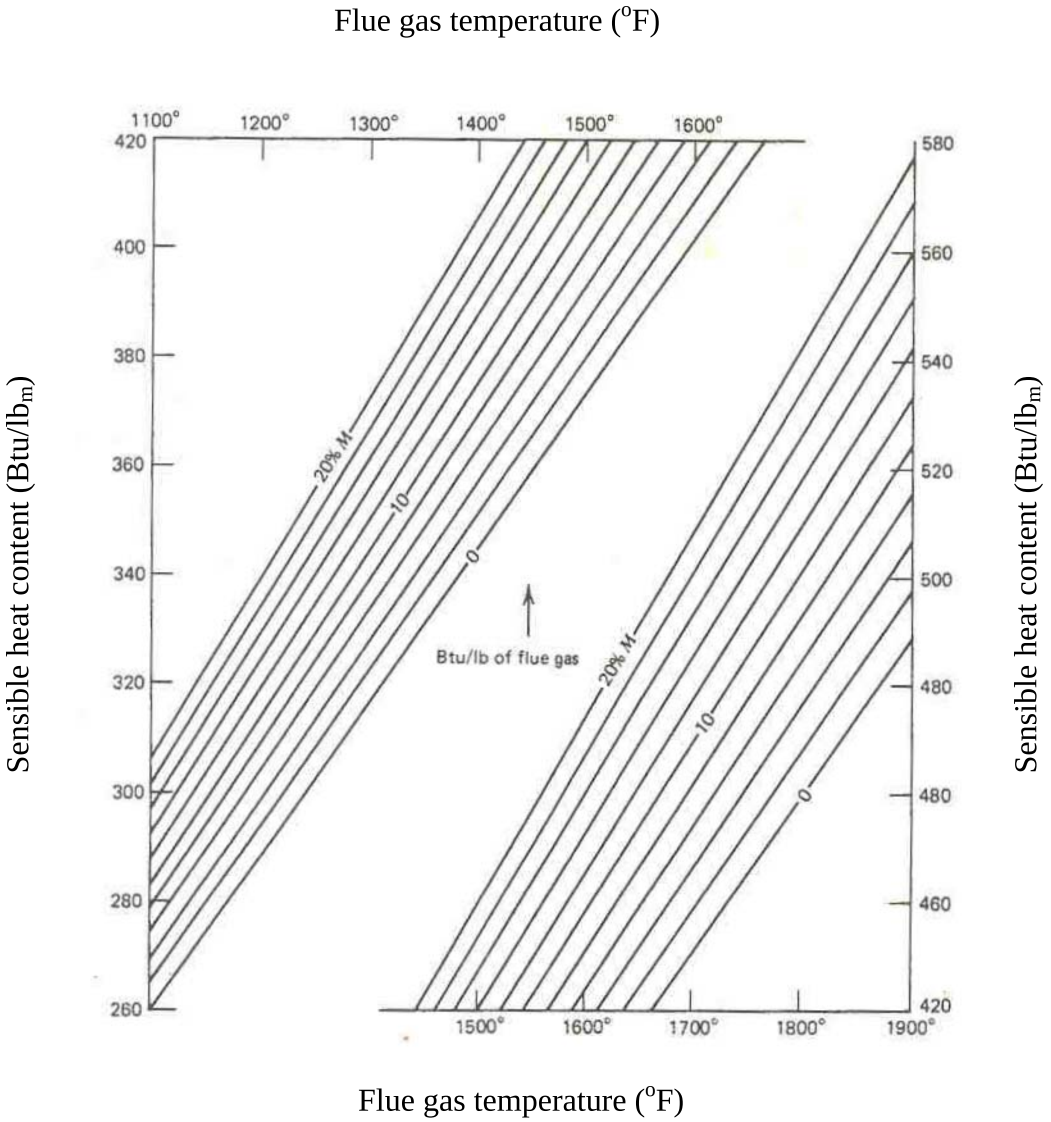


Figure G.1 Continued

APPENDIX H

STEAM TURBINE (RECOVERY ENERGY)

The steam turbine operating conditions follow:

@ inlet, superheated steam @ 752 °F and 615 psia

$S_1 = 1.6074 \text{ Btu/}\left(\text{lb}_\text{m} \cdot °\text{R}\right)$

$H_1 = 1377.47 \text{ Btu/lb}_\text{m}$

After an isentropic expansion in the turbine

@ outlet, saturated steam @ 1.47 psi and 115 °F

$S_{2l} = 0.1559 \text{ Btu/}\left(\text{lb}_\text{m} \cdot °\text{R}\right)$

$S_{2v} = 1.9456 \text{ Btu/}\left(\text{lb}_\text{m} \cdot °\text{R}\right)$

$H_{2l} = 82.97 \text{ Btu/lb}_\text{m}$

$H_{2v} = 1111.45 \text{ Btu/lb}_\text{m}$

$$x = \frac{S_1 - S_{2l}}{S_{2v} - S_{2l}} = \frac{1.6074 \dfrac{\text{Btu}}{\text{lb}_\text{m}\,°\text{R}} - 0.1559 \dfrac{\text{Btu}}{\text{lb}_\text{m}\,°\text{R}}}{1.9456 \dfrac{\text{Btu}}{\text{lb}_\text{m}\,°\text{R}} - 0.2313 \dfrac{\text{Btu}}{\text{lb}_\text{m}\,°\text{R}}} = 0.8462$$

$$H_2 = xH_{2v} + (1 - x)H_{2l}$$

$$= 0.8462 \times 1111.45 \text{ Btu/lb}_\text{m} + (1 - 0.8462) \times 82.97 \text{ Btu/lb}_\text{m} = 953.26 \text{ Btu/lb}_\text{m}$$

The energy recovered from the steam turbine = $W_{turb} = m(H_1 - H_2)\eta_{turb}$

$$W_{turb} = 47{,}615\,\frac{\mathrm{lb_m}}{\mathrm{h}}\left(1377.47\,\frac{\mathrm{Btu}}{\mathrm{lb_m}} - 953.26\,\frac{\mathrm{Btu}}{\mathrm{lb_m}}\right)\times 0.85 = 17{,}168{,}945.3\,\frac{\mathrm{Btu}}{\mathrm{h}}$$

$$= 5{,}032\ \mathrm{kW}$$

Therefore the total mechanical power available for the compressor is

[Gas turbine set] + [steam turbine recovery] = 11,190 kW + 5,032 kW = 16,222 kW

It should be noted that the power available from the gas and steam turbines surpasses the actual compressor needs; therefore, the power surplus will drive a generator that makes electricity for the grid, further reducing the desalinated water unitary cost. The total power requirement for pumping (746.3 kW) will be satisfied with a portion of this electrical power.

APPENDIX I

ENERGY EFFICIENCY OF THE COMBINED-CYCLE POWER SYSTEM

From the manufacturer’s chart, the gas turbine heat requirements are

2,719.5 GJ/day = 31,476 kW

therefore,

$$\eta_{energy} = \frac{\text{power output}}{\text{heat input}} = \frac{16{,}222\ \text{kW}}{31{,}476\ \text{kW}} = 0.5153$$

APPENDIX J

PUMPS

Feed Seawater Pumps

Table J-1 shows individual pressure and flow requirements at each vapor-compression-train stage. The shaft work of each the pumps is calculated from [42]

$$W_o = \frac{H m_v}{\eta_{pump} \rho_{seawater}}$$

where:

W_o = shaft work (kW)

H = total dynamic head (kPa)

m_v = mass flow rate (kg / s)

$\rho_{seawater} = 1{,}021.90 \text{ kg/m}^3$

η_{pump} = pump efficiency = 0.85

Results are shown in Table J.1. It is to be noted that distillate and brine discharges from the sensible heat exchanger do not need pumping. Brine and distillate flows lose most of the pressure in the heat exchanger, but still can flow out.

Table J-1. Individual shaft work required in each seawater feed pump

Pump	Volumetric flow rate (kg/s)	Discharge pressure required (kPa)	Shaft work (kW)
P-1	147.5	1480	251.3
P-2	146.03	1480	248.81
P-3	144.57	1480	246.32
TOTAL			746.43

APPENDIX K

CONDENSER

The required cooling water flow is estimated as follows

$$Q = mC_p\Delta T$$

where

$Q =$ heat load = 1,000 Btu/lb$_{\text{m, steam}}$

$m =$ cooling water flow (lb$_\text{m}$/h)

$C_p =$ cooling water specific heat = 1 Btu/lb$_\text{m}$ °F

$\Delta T =$ cooling water temperature increment

= (105 – 70) °F = 35 °F

Therefore

$$m = \frac{Q}{C_p\Delta T} = \frac{1{,}000\ \text{Btu/lb}_{\text{m, steam}}}{\left(1\ \text{Btu/}\left(\text{lb}_{\text{m, cooling water}}\cdot{}^\circ\text{F}\right)\right)\times\left(35\ {}^\circ\text{F}\right)} = 33\,\frac{\text{lb}_{\text{m, cooling water}}}{\text{lb}_{\text{m, steam}}}$$

Steam flow = 47,615 lb$_\text{m}$, therefore, the total cooling water flow required is

$$47{,}615\,\frac{\text{lb}_{\text{m, steam}}}{\text{h}}\times 33\,\frac{\text{lb}_{\text{m, cooling water}}}{\text{lb}_{\text{m, steam}}} = 1{,}571{,}295\,\frac{\text{b}_{\text{m, cooling water}}}{\text{h}} = 17{,}141\,\frac{\text{m}^3}{\text{day}} = 0.198\,\frac{\text{m}^3}{\text{s}}$$

From reference [34], the purchased cost of a multi-jet spray-type condenser is

cost = $19,000

APPENDIX L

DESALINATION PLANT COST ANALYSIS

Total Capital cost

Case A

Table 8.1 shows the calculated total capital investment = $23,484,724

Amortization[a]: @ 5% interest 30 years = $126,087.18/month = $4,146/day

[a] http://www.engineeringtoolbox.com/12_812qframed.html

The following formula is used to calculate the monthly loan payment [34]:

$$\widehat{R} = P_0 \frac{i}{1 - \left(1 + \frac{i}{m}\right)^{-mt}}$$

where:

$\widehat{R} =$ yearly loan payment ($/year)

$P_o =$ initial amount of the loan ($)

$i =$ nominal interest rate = 0.05

$t =$ repayment period = 30 years

$m =$ number of monthly payments per year = 12

$$\widehat{R} = 23{,}484{,}724\frac{0.05}{1-\left(1+\frac{0.05}{12}\right)^{-12(30)}} = \$1{,}513{,}046.16\,/\,\text{year} = \$126{,}087.18/\text{month}$$

Case B

Table 8.1 shows the calculated total capital investment = $26,887,517

Amortization[a]: @ 5% interest 30 years = $144,338.00/month = $4,745/day

[a]http://www.engineeringtoolbox.com/12_812qframed.html

The following formula is used to calculate the monthly loan payment [40]:

$$\widehat{R} = P_0\frac{i}{1-\left(1+\frac{i}{m}\right)^{-mt}}$$

where:

$\widehat{R} =$ yearly loan payment ($/year)

$P_o =$ initial amount of the loan ($)

$i =$ nominal interest rate = 0.05

$t =$ repayment period = 30 years

$m =$ number of monthly payments per year = 12

$$\widehat{R} = 26{,}887{,}517.\frac{0.05}{1-\left(1+\frac{0.05}{12}\right)^{-12(30)}} = \$1{,}732{,}056\,/\,\text{year} = \$144{,}338.00/\text{month}$$

Case C

Table 8.1 shows the calculated total capital investment = $33,453,796

Amortization[a]: @ 5% interest 30 years = $179,587.21/month = $5,904.23/day

[a] http://www.engineeringtoolbox.com/12_812qframed.html

The following formula is used to calculate the monthly loan payment [40]:

$$\widehat{R} = P_0 \frac{i}{1-\left(1+\frac{i}{m}\right)^{-mt}}$$

where:

$\widehat{R} =$ yearly loan payment ($/year)

$P_o =$ initial amount of the loan ($)

$i =$ nominal interest rate = 0.05

$t =$ repayment period = 30 years

$m =$ number of monthly payments per year = 12

$$\widehat{R} = 33{,}453{,}796\frac{0.05}{1-\left(1+\frac{0.05}{12}\right)^{-12(30)}} = \$2{,}155{,}046.52\,/\,\text{year} = \$179{,}624.07/\text{month}$$

APPENDIX M

COST OF TITANIUM SHEET-SHELL HEAT EXCHANGER CALCULATION

The following were developed in cooperation with Gooseneck^{TM} Manufacturing Co. They have experience with sheet metal fabrication.

The cost of the 0.007-in-thick titanium sheets is determined considering the following:

1. Cost per pound of material
2. Add cost of stamping the dimple patterns on the sheets
3. Manufacturing cost

Unitary cost of titanium sheet

Cost per pound [43]: \$14.00/lb$_m$, Grade 2 (rolled)

Sheet volume = (8 ft) (8 ft) (0.007 in) (ft / 12 in) = 0.0373 ft^3 = 1,056.219 cm^3

Sheet weight = (4.5 g/cm^3) (1,056.219 cm^3) = 4,52.98 g

= 4.75298 kg = 10.456 lb

Cost per sheet = (10.456 lb) (\$14/lb) = \$146.38 (rolled)

Cost per square foot = \$146.38/64 ft^2 = **\$2.28**

Manufacturing cost. Goosneck Manufacturing Company provided the following estimates of the cost to produce dimpled pattern on the sheets:

Cost of die tool = \$20,000

Machine cost index = \$70/h (includes labor and acess to stamping machine)

Working at 3 cycles per minute pressing 4 inches at a time, the rate of machine work is 1 ft/min, therefore, 1 sheet will take 8 minutes = 7.5 sheets/h or conservatively 7 sheets/h.

$$\text{Dimpling cost} = \left(\frac{\$70}{\text{h}}\right)\left(\frac{\text{h}}{7\,\text{sheets}}\right) = \frac{\$10}{\text{sheet}}$$

Because each sheet is 64 ft^2, the unitary cost is

Unitary cost = \$ 10.00 / 64 ft^2 = **\$0.1563 / $\mathbf{ft^2}$**

This cost does not include the amortized cost of the die; however, this cost is negligible in a large production run.

Cost of vessel:

Vessel Shell Thickness

The recommended design equation for a vessel under internal pressure is [41]

$$t = \frac{P \cdot r_i}{S \cdot E_j - 0.6P} + C_c \qquad \left\{ \begin{array}{l} t \le \frac{r_i}{2} \\ P \le 0.385 S \cdot E_j \end{array} \right\}$$

where

C_c = Allowance for corrosion = 0

P = Maximum allowable internal pressure

= 10% over the maximum operating pressure [41]

= $1.10 \times 120 \text{ psig} = 132 \text{ psig} = 910.1$ kPa

r_i = inside radius of shell before corrosion allowance is added (m) [41]

= 2 m

S = maximum allowable working stress (kPa)

= ¼ of the ultimate strength of the material [41]

= 88,350 kPa (carbon steel SEA J412)

$E_j =$ efficiency of joints, expressed as a fraction

= 0.85 (spot examined joints) [41]

Therefore, substituting

$$t = \frac{910.1\,\text{kPa} \times 2}{(88{,}350\,\text{kPa} \times 0.85) - (0.6 \times 910.1\,\text{kPa})} = 0.0296\,\text{m} = 29.6\,\text{mm}$$

Area per plate = 64 ft^2

Separation between plates 0.25 in = 0.0208 ft

Plate thickness = 0.007 in = 0.000583 ft

[plate thickness] + [separation] = unitary space per plate

= 0.0208 + 0.000583 = 0.02141 ft

vessel length (l) = (No. of plates) (0.02141 ft/plate)

vessel diameter = $\sqrt{8^2 + 8^2}$ = $\sqrt{128}$ = 11.31 ft approximate 4 m.

From Figure E-1 [34], consider a vessel 20 m = 65.6 ft long

Cost of purchase = $ 60,000.00

pressure adjustment factor = 1.6 for 109.7 psig operation

Number of plates = (65.6 ft)$\left(\frac{\text{plate}}{0.0214\,\text{ft}}\right) = 3{,}064\,\text{plates}$

Cost = $\left(\frac{1.6 \times \$60{,}000}{3{,}064\text{pplates}}\right)\left(\frac{\text{plates}}{64\,\text{ft}^2}\right) = \frac{\$0.4895}{\text{ft}^2}$

Unitary cost of vessel = **$0.4895/ft^2**

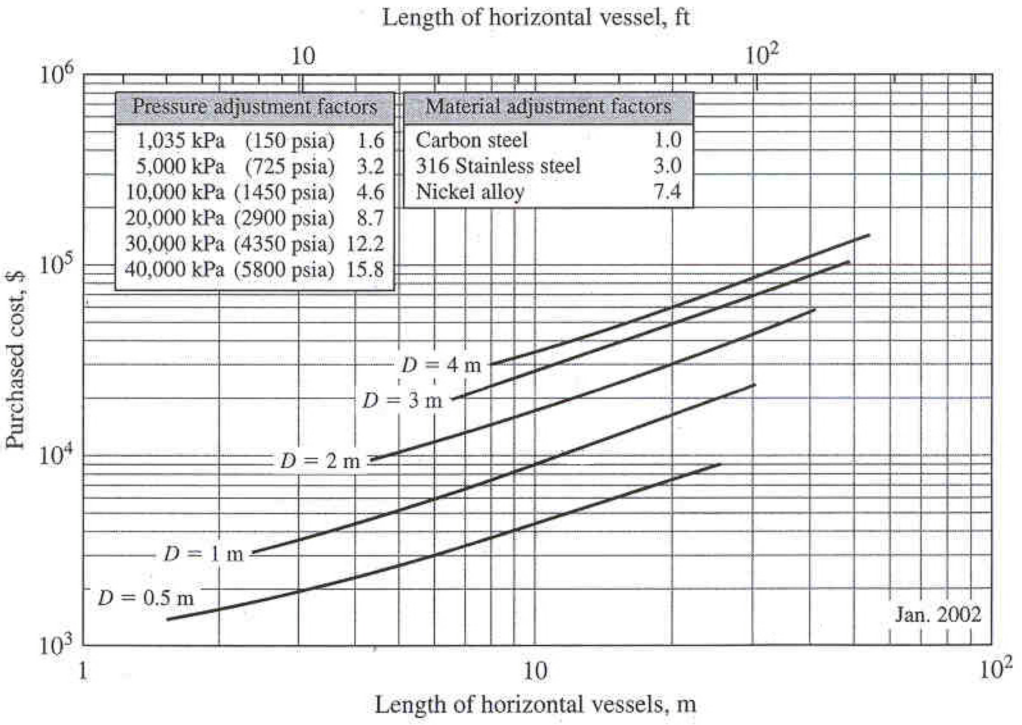


Figure M-1. Cost of purchase of vessels [41].

Table M-1 summarizes the cost of the heat exchanger components and provides a preliminary estimate total unitary cost.

Table M.1 Sheet-shell heat exchanger unitary cost

Sheet-shell heat exchanger cost per square foot	
	Cost ($)
Rolled titanium sheet (0.007 in)	2.28
Sheet manufacture	0.16
Vessel	0.49
Other (assembly, gaskets, vessel modification)	4.14
Total unitary cost	**7.07***

*Agrees with Rautenbach and Arzt [44] estimate.

APPENDIX N

THEORETICAL ENERGY CONSUMPTION FOR SEAWATER DESALINATION

The theoretical work required to desalinate seawater is obtained from the Gibbs free energy change

$$\Delta G = -RT\ln a$$

$$= -RT\ln\frac{p}{p_0}$$

where

$\Delta G =$ Gibbs free energy change (J/mol)

$R =$ Universal gas constant = 8.314 $\text{J/}(\text{mol}\cdot\text{K})$

$T =$ Absolute temperature = 300 K

$p =$ vapor pressure over 35 g/kg seawater solution (Pa)

$p_0 =$ vapor pressure over pure water (Pa)

$a =$ activity

The activity is obtained from Equation 1

$$\log_{10} a = \log_{10}\frac{p}{p_0} = hS + jS^2 \qquad \text{(N.1)}$$

$$= -2.1609\times10^{-4}\,\frac{\text{kg}}{\text{g}}\left(35\,\frac{\text{g}}{\text{kg}}\right) - 3.5012\times10^{-7}\,\frac{\text{kg}^2}{\text{g}^2}\left(35\,\frac{\text{g}}{\text{kg}}\right)^2$$

$$= -0.007992047$$

$$a = \frac{p}{p_0} = 10^{-0.007992047} = 0.981766$$

$$\Delta G = -\left(8.314\,\frac{\text{J}}{\text{mol}\cdot\text{K}}\right)(300\ \text{K})\ln 0.981766$$

$$= -45.9\,\frac{\text{J}}{\text{mol}} \times \frac{\text{mol}}{18\ \text{g}} \times \frac{1{,}000\ \text{g}}{\text{L}} \times \frac{1{,}000\ \text{L}}{\text{m}^3} \times \frac{\text{MJ}}{10^6\ \text{J}}$$

$$= 2.55\,\frac{\text{MJ}}{\text{m}^3}$$

VITA

Jorge Horacio Juan Lara Ruiz
9205 Shadowcrest
College Station, TX 77845

Education:

B.S. Mechanical Engineering, National Polytechnic Institute. Mexico 1975
M.S. Mechanical Engineering, Lamar University. 1979
M.Eng. Mechanical Engineering, Texas A&M University. 2003
Ph.D. Interdisciplinary Engineering, Texas A&M University. 2005

Professional Experience:

Production Manager. 1980-1985. Mexico City.
Professor. 1985- 2000. University of Puebla, Mexico
Director of 3 sponsored research projects. National Council of Science (CONACYT)
Research Assistant. 2001-2003. Energy Systems Laboratory. Texas A&M University

Publications:

1. J. Lara. Computer Aided Design of Centrifugal Pumps. Paper presented at Symposium for the Development of Latin American Countries, UNESCO, Puebla, Mexico, 1995

2. J. Lara. A Numerical Method for the Design of Radial Centrifugal Pumps, Paper presented at Turbo Machinery International Congress, Acapulco, Mexico, May 1996.

3. J. Lara. An Algorithm for the Aero Dynamical Design of Axial Flow Turbo Compressors, Paper presented at Turbo Machinery International Congress, Mexico City, November 1997.

4. J. Lara. Small Wind Tunnel Tests for the Characterization of Gas Turbine Blades, Paper presented at Turbo Machinery International Congress, Aberdeen, Scotland, October 1997.

5. J. Lara. A Report on the Design Manufacture and Tests of an Aero Dynamical Prototype of Axial Turbo Compressors, Paper presented at Turbo Machinery International Congress, Veracruz, Mexico, November 1998.