THEORETICAL AND EXPERIMENTAL ANALYSIS OF WATER DESALINATION SYSTEM USING LOW GRADE SOLAR HEAT

By

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To the loving memory of my grandfather and grandmother, whom I shall always remember.

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NOMENCLATURE

- A: Area (m^2)
- C: Solute concentration (%)
- C_A: Concentration of substance A in the solution (M)
- C_p: Specific heat (J/kg.°C)
- D: Diameter (m)
- D_c: Diameter of the riser tube in the collector (m)
- F_R: Collector heat removal factor (dimensionless)
- FT: Heat exchanger fouling factor (W/m².°C)
- g: Gravitational acceleration (m/s^2)
- Gr: Grashof number (dimensionless)
- h: Convection heat transfer coefficient (W/m².K), Head (m)
- h_{fg}: Latent heat of vaporization (J/kg)
- h(F): Saturated liquid enthalpy (kJ/kg)
- h(FCR): Saturated liquid enthalpy at critical temperature (kJ/kg)
- h(G): Saturated vapor enthalpy (kJ/kg)
- h(GCR): Saturated vapor enthalpy at critical temperature (kJ/kg)
- H_c: Vertical height of the bottom of the evaporator from collector's top (m)
- i: Interest rate (%)
- I: Incident solar radiation (W/m²), modified Bessel function of the first kind
- Ja: Jacob number (dimensionless)
- k: Thermal conductivity (W/m.K)
- K: Modified Bessel function of the second kind
- K_H: Henry's law constant
- l: Length (m)
- L₁: Length of the fluid channel in the collector (m)
- m: mass flow rate (kg/s)
- m_d : Daily distillate output per unit area (kg/m²)

- me: Instantaneous distillate output (kg/m².s)
- my: Annual average distillate output (kg/m².y)
- n: Useful life (year)
- Nc: Number of parallel risers in the collector
- N_{d,i}: Number of days in the I month
- NTU: Number of transfer units (dimensionless)
- Nu: Nusselt number (dimensionless)
- p: Perimeter (m)
- P: Pressure (Pa)
- PA: Vapor pressure of substance A (atm)
- Pr: Prandtl number (dimensionless)
- Q: Heat transfer rate (W)
- r: Radius (m)

 r_p : Ratio of pressure drop in the connecting pipes and heat exchanger to that of collector.

- Re: Reynolds number (dimensionless)
- s(F): Saturated liquid entropy (kJ/kg.K)
- s(FCR): Saturated liquid entropy at critical temperature (kJ/kg.K)
- s(G): Saturated vapor entropy (kJ/kg.K)
- s(GCR): Saturated vapor entropy at critical temperature (kJ/kg.K)
- t: Thickness (m), Time (hour)
- T: Temperature (°C)
- T(CR): Critical temperature (K)
- T(S): Saturation temperature (K)
- U: overall heat transfer coefficient $(W/m^2.K)$
- V: Volume (m³)
- \dot{V} : Volumetric flow rate (m³/s)
- v: Velocity (m/s)
- *v* : Evaporation rate per unit area (m/s)
- Z: Elevation (m)

Greek Letters

 ρ : Density (kg/m³)

 θ : Inclination angle

 μ : Viscosity (N.s/m²)

 γ : Kinematic viscosity (m²/s), Specific Weight (N/m³)

- η : Efficiency
- ε: Heat exchanger effectiveness (dimensionless)
- δ : Film thickness (m)
- β : Volumetric thermal expansion coefficient (K⁻¹), tilt angle
- β_T : Volumetric thermal expansion coefficient (= 5*10⁻⁴ °C⁻¹)
- $\beta_{\rm C}$: Solutal expansion coefficient (= 8*10⁻³ %⁻¹)
- β_2 : Empirical coefficient (= 4178.4 J/kg.°C)
- α : Thermal diffusivity (m²/s), absorptivity (dimensionless)
- α_1 : Empirical coefficient (= 0.0054, dimensionless)
- α_2 : Empirical coefficient (= -30.1 J/kg.°C)
- $\alpha_{\rm m}$: Empirical coefficient (= 10⁻⁷-10⁻⁶ kg/m².Pa.s.K^{0.5})
- τ : Transmisivity of the collector cover (dimensionless)

Subscripts

- a: Ambient
- b: Bottom
- c: Cold; condenser, collector
- D: Diameter
- e: Evaporation

env: Environment

- f: Fresh water, product, or fluid
- h: Hot; hydraulic, heat exchanger
- i: Of the injection pipe; inside
- in: Input
- ins: Insulation

L: Loss

- max: Maximum
- min: Minimum
- o: Outside, outlet
- s: Of sea water chamber or the evaporator; surface

- sat: Saturated conditions
- u: Useful
- v: Saturated vapor conditions
- w: Of the withdrawal pipe
- 0: Of the reference state

Superscripts

- 1: Present value
- 0: Previous value

Over bar

-: Average

Abstract of Dissertation Presented to the Graduate School of the University of Florida in Partial Fulfillment of the Requirements for the Degree of Doctor of Philosophy

THEORETICAL AND EXPERIMENTAL ANALYSIS OF WATER DESALINATION SYSTEM USING LOW GRADE SOLAR HEAT

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An innovative new concept of a solar distillation system which uses low grade heat was developed. The system utilizes natural forces of gravity and atmospheric pressure to create vacuum under which water can be evaporated at lower temperatures than conventional techniques, which would allow the use of low grade heat sources, such as inexpensive flat plate solar collectors and/or waste heat. The uniqueness of the system is in the way natural forces are used to create vacuum conditions, and its incorporation in a single system design where evaporation and condensation take place at appropriate locations without any external energy input other that low grade heat. The system consists of solar collectors or some other low grade heat source, an evaporator, and provisions to supply the saline water and withdraw the concentrated brine. The evaporator is connected to the condenser where the produced vapor is condensed and collected as the product. Both, the evaporator and condenser are placed at a height of about 10 m (the height required to have a water column that would balance the atmospheric pressure) from the ground level. The evaporator is connected to the saline water supply, and concentrated brine tanks, and the condenser is connected to the fresh water tank. All tanks are kept at the ground level.

In this research the concept was studied theoretically and experimentally, to investigate the effects of various operating conditions: depth of water body, withdrawal rate, condenser thermal resistance, and heat source temperature. A theoretical model was developed, to simulate the performance of the system which agreed with the experimental results. The effect of depth of the water body in the evaporator was found to be small. It mainly affects the time required by the system to reach steady state. As the depth of water body increases the system will need more time to reach the steady state conditions. Effect of withdrawal rate is significant if the withdrawal rate goes beyond a certain value, about 1 kg/hr for the present system. Effect of condenser thermal resistance is significant; the lower the thermal resistance, lower the fresh water temperature, the higher the system efficiency. Effect of heat source temperature was found to be significant, increasing the heat source temperature increases the system output significantly. Experimental and theoretical results show that the system is superior to a flat-basin solar still, and the output may be twice that of the still for the same input. Simulations show that if a system of 1 m^2 evaporator cross sectional area is connected to a flat plate solar collector of 1 m^2 absorber area, the system output can reach about 6.5 kg/day. The system energy and exergy efficiencies could reach 82 and 87 %, respectively.

Economic analysis for the system connected to a flat plate solar collector and operating under the climatic conditions of Gainesville, Florida, shows that the product cost will be around US\$0.135 per kg of distillate.

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CHAPTER 1 INTRODUCTION AND LITERATURE REVIEW

Introduction

The shortage of drinking water is expected to be the biggest problem of the world in this century due to unsustainable consumption rates and population growth. Pollution of fresh water resources (rivers, lakes and underground water) by industrial wastes has heightened the problem.

The total amount of global water reserves is about 1.4 billion cubic kilometers. Oceans constitute about 97.5% of the total amount, and the remaining 2.5% fresh water is present in the atmosphere, polar ice, and ground water. This means that only about 0.014% is directly available to human beings and other organisms [1]. So, development of new clean water sources is imperative. Desalination of sea and/or brackish water is an important alternative, since the only inexhaustible source of water is the ocean.

Besides the problem of water shortage, process energy constitutes another problem area. Desalination processes require significant amount of energy. It was estimated that the production of 1 million m³/day requires 10 million tons of oil per year [2]. Due to high cost of conventional energy sources, which are also environmentally harmful, renewable energy sources (particularly solar energy) have gained more attraction since their use in desalination plants will save conventional energy for other applications, reduce environmental pollution and provide free, continuous, and low maintenance energy source.

The use of solar distillation has been practiced for a long time, and it gained more attraction after the First World War. Solar desalination is suitable for remote, arid and semi-arid areas, where drinking water shortage is a major problem and solar radiation is high. The limitations of solar energy utilization for desalination are the high initial cost for renewable energy devices and intermittent nature of the sun. Due to these limitations the present capacity of solar desalination systems worldwide is about 0.01% of the existing large-scale conventional desalination plants [3]. So, efforts must be made to develop technologies, which will collect and use renewable energy more efficiently and cost effectively to provide clean drinking water besides developing technologies to store this energy to use it whenever it is unavailable.

The combination of solar energy with desalination processes can be classified into two main categories: 100% solar driven desalination plants or partial solar powered desalination plants. Solar plants could be designed to operate in a fully automatic fashion in the sense that when the sun rises, heat collection process is initiated automatically by a sensor measuring the solar radiation.

Desalination Processes

Different types of water desalination processes have been developed. Table 1.1 at the end of this section summarizes some of the main characteristics for some processes. These can be classified into the following two categories [4]:

Phase Change or Thermal Processes

Thermal energy sources, such as fossil fuels, nuclear energy or solar energy may be used to evaporate water, which is condensed to provide fresh water. The phase change desalination processes described here include multi-stage flash, multi-effect boiling, vapor compression, and freezing processes.

Multi-stage flash (MSF) process

The process consists of many stages; in each stage the steam produced in the previous stage condenses and simultaneously preheats the feed water. Thus, the temperature difference between the hot source and seawater is fractionated into a number of stages. Therefore, the system approaches ideal total latent heat recovery. The operation of such a system requires pressure gradients in different stages; i.e. stages should be at successively lower pressures. Figure 1.1 shows a schematic diagram of such a system. Seawater, preheated in various stages, enters the solar collector, where it is heated to nearly saturation temperature at the maximum system pressure. As the water enters the first stage through an orifice, its pressure is reduced, thus becomes superheated and flashes into steam. The steam produced passes through a demister to remove any suspended brine droplets, then to a heat exchanger where it condenses. This process is repeated through the various stages.

Multi-effect boiling (MEB) process

The process consists of a number of elements, called effects. The steam from one effect passes through the next one, where it condenses and causes evaporation of a part of the seawater. This process requires that the heated effect be kept at a pressure lower than that of the effect from which the steam originated. Figure 1.2 shows a schematic diagram of such a system. In this system the feed water passes through heat exchangers for preheat, then instead of entering solar collector or heater, it enters the top of the first effect, where the heating steam raises its temperature to the saturation temperature for the effect pressure, then another amount of steam from the solar collector is used to produce evaporation. The produced vapor is used in part to heat the incoming water and in part, to provide heat to the next effect. Also the sensible heat of condensation is used to preheat

the feed water. This process usually operates on a once through system without much of circulation of the brine through the system, thus reducing pumping power and scale formation.



Figure 1.1 Schematic diagram of multistage process, adapted from Kalogirou [4].



Figure 1.2. Schematic diagram of multi-effect boiling process, adapted from Kalogirou [4].

Vapor compression (VC) process

When vapor is compressed, its temperature and pressure increase. Vapor compression distillation process is based on this concept. In this process, the feed is heated, usually by steam, and a part of it is vaporized. The produced vapor, as shown in fig. 1.3, is compressed using a mechanical compressor or by mixing with small amounts

of high pressure steam (thermal compression) and returned through a tube to the chamber, where it condenses and gives its latent heat of condensation to the feed, thus causing a part of it to evaporate and the produced vapor is compressed again and the process continues. In mechanical compression, which is the most common one, a separate source of steam is required only for start up, and then the process converts mechanical energy to produce its own heat and thus eliminates the need for a large steam generator. The process usually consists of 1-3 stages.



Figure 1.3. Schematic diagram of a vapor compression process, adapted from Department of Water Resources [5].

Freezing process

The principle of operation is that as the salt water is cooled, ice crystals are formed which are salt free. Cooled seawater enters a freezing chamber (fig. 1.4) to form ice and a small amount of water vapor. Ice and the brine are then transported to a separation chamber, where the ice crystals are washed from the salts and moved to the melting chamber. The water vapor produced in the freezing chamber is compressed and supplied to the melting chamber, causing the ice to melt while the vapor itself condenses, forming a part of the product. The main advantage of this process is that it operates at very low temperatures that greatly reduce scale and corrosion problems.



Figure 1.4. Schematic diagram of freezing process, adapted from Department of Water Resources [5].

Single Phase or Membrane Processes

Processes that need electricity or mechanical power for desalination include:

Reverse osmosis (RO) process

When fresh water and seawater are separated into compartments by a

semipermeable membrane, as shown in fig. 1.5, fresh water will pass through the

membrane by osmosis. If pressure is exerted on the saline solution, the osmosis process

may be reversed. When the pressure on the saline water exceeds the natural osmotic

pressure, fresh water from the saline solution will pass through the membrane to the fresh

water side, leaving the salts in the concentrated brine. The higher the salt concentration in the feed water the higher the pressure required. As pressure increases, a stronger membrane will be required to prevent the passage of salts. Reverse osmosis is best for brackish water.



Figure 1.5. Schematic diagram of reverse osmosis process, adapted from Department of Water Resources [5].

Electrodialysis process

The principle of this process is reducing the salinity of seawater by transferring ions from the seawater, through a membrane, under the influence of electrical potential difference. This process combines the use of an electrically charged cell and an ionselective semipermeable membrane to desalt saline water. Figure 1.6 shows a schematic diagram of such a process. Salts are present in saline water in the form of ions; positive charged ions are called cations, and negative charged ions are called anions. As mineralized water passes through an electrodialysis cell, cations will be attracted to the negative electrode and anions to the positive electrode. Cation permeable membranes permit passage of cations only, such as sodium and calcium. Anion permeable membranes permit passage of anions only, such as chloride and sulfate. In practice, a large number of membranes are placed between the electrodes, forming a number of dilute (demineralized) and concentrate (brine) compartments.

Ion exchange

An ion exchanger is a porous bed of natural material or synthetic resins that have the ability to exchange ions in the resin with those in the mineralized waters that contact the bed. The beds, as shown in fig 1.7, are arranged in a series so that the mineralized water passes through the cation exchanger first then through the anion exchanger. In the cation exchanger, cations are taken from the mineralized water and hydrogen ion is put into the water. In the anion exchanger, anions are taken from the water and hydroxide ion is put into the water. Thus the compounds are removed from the saline water leaving fresh water, and hydrogen and hydroxide combine to give more fresh water. When the regenerated with an acid and a base to restore their ion-exchange properties.

Solar Desalination Processes

Solar energy may be used to supply the required energy for a desalination process either in the form of thermal energy or electricity. In either case, fresh water cost increases rapidly when the desalination plant does not consume the total energy delivered by the solar field; therefore for a given fresh water demand the desalination plant must be designed to consume the total production of the solar field.



Figure 1.6. Schematic diagram of electrodialysis process, adapted from Department of Water Resources [5].



Figure 1.7. Schematic diagram of ion exchange process, adapted from Department of Water Resources [5].

	Process	Size	Suitable for	Heat	Power Input	Output	Mean Features
			feed of initial	input	(kWh/m ³ of	Quality	
			concentration	(kJ/kg of	product)	(ppm of	
			(ppm)	product)	-	TDS)	
Phase Change or Thermal Processes	MSF	Medium- Large	10000-50000	294	3.7	50	 The most common and simple technique in use. Performance ratio, which is the ratio of tons of water produced per ton of initial steam, is about 10. The process is not sensitive to initial concentration. Part of the brine is circulated with the feed in order to increase water recovery.
	MEB	Small- Medium	10000-50000	123	2.2	50	-Performance ratio is about 8-16.-Operates usually on a once through system without large mass of brine circulating around the plant.
	VC	Small	10000-50000	-	16	50	-Part of water circulates to increase water recovery. -Ability to operate at low temperature makes it possible to use simple metal like aluminum.
Single Phase or Membrane Processes	RO	Small- Large	1000-10000	-	12	100-600	 -Pressure of 10-25 and 50-80 bars is required for brackish and seawater desalination, respectively. -Conversion of 90-95 and 35-50 % for brackish and seawater, respectively. -Fouling of membrane is a major problem. -Provides the cheapest product cost.
	ED	Small	1000-5000	-	12	400	 -Mainly used for brackish water desalination. -Due to low conductivity, which increases the energy requirements of very pure water, this process is not suitable for water of less than 400 ppm. -Membranes are insensitive to fouling.

Table 1.1 Characteristics of desalination processes [4,6]

Solar desalination processes may be classified into two categories:

Indirect Systems

The operating principle of such systems involves using two separate subsystems, a collector for solar energy conversion and a plant for using the collected energy to produce fresh water. The plant may be of any type described previously. Systems of this kind have been subjected to many analytical and experimental studies to improve their efficiency. Table 1.2 (at the end of this section) provides a brief summary of some of these studies. There are many factors that influence the selection of solar collectors for desalination processes [7]. Flat plate solar collectors are appropriate for low temperature processes; those processes involve the utilization of vacuum conditions, which are usually created with the help of vacuum pumps, steam ejectors or water jet ejectors. Evacuated tube collectors ensure some energy even on cloudy days, and their efficiency at high operating temperatures or low insolation is significantly better than flat plate collectors and hence give the highest operating times. Cylindrical tracking collectors can be more efficient than evacuated tube collectors, but have almost no output on cloudy days, besides they collect only a small fraction of the diffuse radiation. Parabolic concentrating collectors require very accurate two-axis tracking mechanisms and can produce temperatures more than 120 °C, which is higher than the temperature needed for solar desalination. For an optimum solar assisted desalination plant, the following parameters must be carefully considered for designing a system: maximum operating temperature of the system, type of solar collector, means of transferring heat to the process, and the type of plant to be used [7].

As some desalination processes can be operated by low-grade heat sources, conventional medium temperature solar collectors may be used to operate them. But due to large amounts of low-grade heat required for the processes, the units require enormous fields of conventional solar collectors. Construction and operation of such fields are impractical taking into account the associated interconnection plumbing, thermal insulation, flow balance, maintenance, heat storage and pumping. A solar system which is suitable for such purposes is a solar pond (SP) which involves adequate heat storage, passive heat collection, and heat transfer at low pumping costs and also can be built and operated at low cost. The operating temperature of a solar pond can be varied in the range of 30-95 °C. A salt gradient solar pond, consists of a thin upper convective layer, a mixed layer at the bottom to store the collected energy, and an insulation layer between the upper and bottom layers where convection is prevented by a stabilized density gradient. This allows the pond to act as a thermal trap. Desalination systems that can be driven by solar ponds include low temperature multi-effect distillation system, multistage flash distillation system, vapor compression distillation, and reverse osmosis. Since VC and RO desalting systems also need mechanical work, their operation with a solar pond requires that a part of the solar energy be converted to power (work), which can be very expensive. A solar pond is quite suitable to drive a low-temperature multi-effect desalination process for the following reasons: the temperature of the heat source supplied by the solar pond (60-75 °C) matches with that required for the low-temperature ME desalters operating at a top brine temperature of 50-60 °C and ME desalting system is very responsive to change in energy supply and operates stably under varying heat supply conditions. Coupling the solar pond into desalination systems was found to be

viable for desalination of seawater in an arid environment [8-12], and the performance of desalinator is satisfactory from a technical point of view, if the desalinator is made of materials that are resistive to corrosion caused by seawater [11,13,14].

Some researchers have proposed the coupling of desalination processes to parabolic trough solar collectors. Rodriguez et al. [15] proposed the use of parabolic trough collectors (with fresh water as the working fluid) as a direct steam generation (DSG) unit to drive a multi effect distillation system. They compared the cost of water produced from different seawater MED systems: solar powered MED plant (DSG parabolic troughs), solar-fossil fuel powered MED plant (oil based technology), and a MED plant driven by a conventional energy supply. The authors concluded that the solar MED technology could be competitive with conventional MED.

The use of evacuated tube collectors was studied by El-Nashar [16], who presented the results of optimization of a MED plant utilizing solar energy; the plant consists of an array of evacuated tube collectors, heat storage, and a seawater evaporator. The plant was designed to operate in a fully automatic fashion in the sense that heat collection process is initiated automatically by a sensor measuring the solar radiation. Also, the operation of the evaporator was initiated by temperature sensors measuring the level of thermal charge remaining in the storage tank. The author made the following conclusions: the cost of water produced is very sensitive to the cost of solar collector, and the minimum water cost is always achieved with the largest storage capacity.

Economic studies showed that solar desalination systems could be competitive under certain conditions, particularly in remote areas [2,17], and that it is better to use a partial solar driven desalination plant than a 100% solar powered one [2]. To find out the

effect of various configurations on the economics of a multi effect seawater distillation system, El-Nashar [18] compared three different configurations: (1) conventional system using a steam generator to supply the plant with steam and a diesel generator to supply electricity needed for pumping purposes; (2) solar assisted system, where the hot water was supplied to the evaporator using solar collectors (flat plate or evacuated tube collectors) and a diesel generator for pumping power; and (3) a solar system consisting of solar collectors (flat plate or evacuated tube collectors) to provide the evaporator with hot water and a photovoltaic (PV) array to provide electricity for pumping power. The author concluded that, at the present time solar energy can not compete favorably with fossil energy, but in remote areas where the real cost of fossil energy can be very high, the use of solar energy can be an attractive alternative. However, the cost of fresh water can be reduced with the increasing size of the units and through serial production [11]. In an effort to compare a solar assisted MSF distillation plant with a conventional one Rodriguez and Camacho [19] studied a solar assisted plant which is a conventional one coupled to a solar parabolic trough collector field, and uses thermal oil as heat transfer fluid and the hot oil is stored in an insulated tank. In this system the solar field could directly act as a brine heater, instead of using thermal oil as a heat transfer fluid. A little hot brine storage may be required to avoid the effect of solar irradiance transients. The authors concluded that the use of solar energy-under certain climatic conditions-could be competitive with conventional energy in MSF distillation plants.

Since the cost of converting solar energy to electricity is higher than that required to convert it into heat, desalination systems using electricity generated from solar energy, found less interest in terms of coupling those to solar devices to supply the required

energy. Many researchers studied reverse osmosis systems in which photovoltaic systems were used to provide the required electricity [1,20,21] and in some case hybrid systems (solar and wind) were used [22,23].

Direct Systems

Direct systems are those where the heat collection and distillation processes take place in the same equipment. Solar energy is used to produce the distillate directly in the solar still. Figure 1.8 shows a schematic diagram of a solar still. The still acts as a trap for the solar radiation (greenhouse effect). As solar radiation passes through the transparent cover, it gets absorbed by the absorber and partially by the saline water. As a result the water temperature, hence vapor pressure, increases and becomes higher than that of the cover. Water evaporates into the surrounding air, and the vapor rises to the cover by natural convection, where it condenses on the inner side of the cover. The condensate flows under the influence of gravity into the collection tank. The most common solar still is the greenhouse type. There are two types of greenhouse stills: symmetrical and asymmetrical. Greenhouse solar stills are simple to construct and operate. Main parts of greenhouse solar stills are as follows:

 Basin: It is where the saline water is kept. The basin must be insulated from the bottom to minimize heat loss to the ambient. It should be painted black or lined with black sheet to maximize its absorptivity. The basin may be 10-20 mm deep (shallow basin) or may be 100 mm or more (deep basin). It must be inclined slightly, around 1°, to allow easy drainage of the concentrated brine. The materials that can be used for the construction of the still must be resistant to saline water corrosion. It is also preferred that those materials be inexpensive, durable, and available locally. Aluminum, concrete, wood impregnated with resin to resist deterioration, iron painted to resist corrosion, and/or plastic may be used.

Type of solar	Author	Type of	Experiment	Additional comments
collector		desalination unit		
Solar Pond	Al-Hawaj and Drawish	ME	No	The system was solar assisted one
	[8]			
	Lu et al. [9]	MSF	Yes	The system has brine concentrated and recovery system to
				concentrate the rejected brine further, which then can be used
				to make a solar pond or produce salts
	Safi [10]	MSF	Yes	The plant was used for brackish water desalination
	Szacsvay et al. [11]	MSF	Yes	The unit has a deaerator to minimize the effect of non-
				condensable gases
	Tsilindiris [12]	ME	No	The water cost was estimated to be $2/m^3$
	Caruso and Naviglio [13]	VC	Yes	The system was manufactured completely from titanium to
	and Caruso et al. [14]			minimize corrosion
Parabolic	Rodriguez et al. [15]	ME	No	The collector was used as a direct steam generator
trough	Rodriguez and Camacho	MSF	No	Partially solar driven system
collector	[19]			
Evacuated	El-Nashar [16,18]	ME	No	A simulation model for fully automatic system with heat
tube collector				storage was developed
Photovoltaic	Bendfeld et al. [1]	RO	No	The system was assumed to supply a small village of 200
				persons with their water needs
	Joyce et al. [21]	RO	Yes	Small scale unit was built and tested
Hybrid system	Manolakos et al. [22]	RO	No	Micro hydropower plant was used for energy storage
(Solar and	Weiner et al. [23]	RO	Yes	Diesel generator was used for back up
wind)				- *

Table 1.2 Solar desalination systems

- 2. Transparent cover: The cover prevents the losses and keeps the wind away from cooling the hot water. Its inside surface also acts as a condenser where the vapor is condensed and collected as product. The slope of the cover must be such that the surface tension of water causes it to flow all the way down to reach the troughs, without falling back into the basin. The optimum tilt angle of the still cover is about 10 and 50° during summer and winter, respectively [24]. Cover is usually glass or plastic sheet. Glass is preferred because of its high transparency and rigidity. A disadvantage of the glass is that it breaks easily. Plastics are light in weight, low cost, do not break and are easy to handle. Their main disadvantage is short lifetime due to deterioration under ultraviolet (UV) light.
- 3. Collection troughs: They are placed at the lower edges of the cover to collect the distillate. The troughs should be constructed with enough pitch along their length to allow the distillate to flow to the lower end of the still, from which it is collected as a product.

Design of a solar still requires optimization of many factors: brine depth, tightness

to prevent vapor leakage, thermal insulation, cover slope, shape and material for the still.

A well designed still must also have adequate provision to collect the rainwater that falls on it, so the surrounding ground is not eroded nor flooded. Still components must be constructed such that they can be easily assembled. A large number of basin type designs have been used. They differ from each other in the types of materials used, geometry, the way of supporting the transparent cover and the provisions for supply and discharge water. A still requires frequent flushing of salty water to prevent precipitation, which reduces its absorptivity and hence the efficiency. Still efficiency, defined as the ratio of the energy used to evaporate the water to the solar energy incident on the still, is usually low and rarely exceeds 50%, with an average value of 30-40% [3]. The daily solar still production is about $3-4 1/m^2$ [4].



Figure 1.8. Schematic diagram of solar still

Simple solar stills have been studied extensively. Table 1.3 (at the end of this section) provides a brief summary of some of these studies. A theoretical analysis by Dunkle [25] in 1961, and the relations that he derived for the heat and mass transfer within the still, formed the basis for many research efforts since then. Dunkle found out that the mass transfer rate depends on the temperature difference between the water surface and the glass cover. In order to increase this temperature difference some researchers [26-32] studied the effect of coupling the solar still to a flat plate solar collector (fig. 1.9). The results showed that the still performance could improve significantly, but of course the system cost would increase. In some cases the daily productivity of the simple still increased from about 4 l/m^2 .day to about 8 l/m^2 .day for the coupled one [26]. Another way to increase the temperature difference is to reduce the temperature of the glass cover. This idea was investigated by Kumar and Tiwari [27] who considered the performance of a single and double effect solar still, with and without water flow over the still glass cover, and concluded that a still with water flows over the glass cover gives the best performance.


Figure 1.9. Schematic diagram of solar still connected to solar collector

Increasing the temperature difference between the saline water surface and the transparent cover could be increased by adding a condenser to the still (fig. 1.10), thus increasing the heat sink capacity, hence the still performance [33-36].

Evaporation at a low temperature, utilizing vacuum conditions, leads to a good improvement in the system efficiency as the evaporation rate increases with the reduction of pressure. System productivity higher than that from similar solar desalination systems operating under atmospheric pressure was reported by many researchers [37-41].

In all configurations described previously the latent heat of condensation was simply dissipated to the environment. However, the latent heat of condensation could be used to preheat the feed water, which would lead to an improvement in the still efficiency [42-44].

Latent heat of condensation could also be utilized for evaporating a part of the saline water in a multi-effect solar still (fig. 1.11). In such still, heat is usually supplied to the first effect from a solar collector, then the vapor produced in that effect ascends upward by natural convection, and condenses when it comes in contact with the bottom of the second effect, giving up its latent heat to that stage, thus evaporating a part of the water from it and the process continues till the last effect which may be covered by a

transparent cover, thus gaining an additional amount of heat directly from the sun. This modification in a still operating mechanism leads to a significant improvement in the still performance [45-50].



Figure 1.10. Schematic diagram of solar still with condenser



Figure 1.11. Schematic diagram of multieffect solar still

Another way to increase the efficiency of the still is to minimize the heat losses to the environment. A model to calculate the thermal losses (through the glass cover, bottom and side walls) from a solar still to the atmosphere was presented by Bobrovnikov et al. [51]. Tiwari [52] presented a thermal analysis of a tubular solar still, which consists of a tray of metal placed inside a glass tube, and validated the analysis experimentally for special case. The nocturnal performance of a solar still supplied with waste heat or with energy storage were also investigated. In some cases the heat is supplied during night from a waste heat source [53], or from stored thermal energy [54]. Adding a dye to the water inside the still improve its performance [55], since it enhances the absorptivity of water.

Fresh water may be obtained from saline water through a humidificationdehumidification cycle. In this process air is circulated by natural or forced convection over hot water supplied from a solar collector where it becomes humid. It is then passed through a condenser for dehumidification. This process was studied by many researchers [44,56-60]. The output can be two times that from a simple solar still [56].

As the driving force for evaporation is the vapor pressure difference between the saline water temperature and the cover temperature (condenser), desalination unit could be powered by salinity difference in an open cycle absorption heat pump [61], since the vapor pressure decreases as the salinity increases.

Concluding Remarks

To summarize, the literature review shows that the use of solar desalination is feasible under certain conditions, especially in remote areas. The output increases by decreasing the losses, lowering the boiling point of saline water by operating under vacuum conditions, making use of latent heat of condensation, and maximizing the driving force for evaporation by increasing the heat sink capacity. Those considerations usually increase the cost of the system, since they require using more material or using vacuum pumps, steam ejector or water jet ejector, to create the required vacuum.

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Table 1.3). V č	anous	types	01 SC	nai	sums

Main feature	Author	Still type	Experiment	Additional comments
Still connected to	Boukar and Harmim [25]	Single effect	Yes	Still output increases when connected to collector
collector	Kumar and Tiwari [27]	Single effect	Yes	Water flow over the cover to increase the temperature difference
	Lawrence and Tiwari [28]	Single effect	No	System was operated under thermosiphon mode
	Tiwari et al. [29]	Single effect	No	Complete thermal analysis. The increase in the collector area results in a decrease in the efficiency due to higher operating temperature which results in more losses
	Yadav and Jha [30]	Double effect	No	Two cases were studied: thermosiphon and forced convection modes
	Yadav [31]	Double effect	No	Two different arrangements were studied: first, heat was added via heat exchanger, second, water was heated in the collector then injected to the still, which gave higher efficiency
	Yadav and Prasad [32]	Single effect	No	Performance analysis was presented. The numerical results were compared with experimental results from other studies and a good agreement was found
Addition of	El-Bahi and Inan [33]	Single effect	Yes	The use of condenser increased the output by 70%
passive condenser	Fath and Elsherbiny [34]	Single effect	Yes	The use of condenser increased the output by 50%
	Khalifa [35]	Single effect	Yes	Internal condenser was used which also acted to preheat the feed
	Saatci [36]	Single effect	Yes	Heat pipe was used as a condenser
Vacuum units	Abu-Jabal et al. [37]	Three effects	Yes	Latent heat of condensation from one stage was used to evaporate water in the next stage
	Ibrahim and lowray [61]	Single effect	No	The system was powered by salinity difference
	Jubran et al. [38]	Three effects	No	Mathematical model which predicts the productivity and thermal characteristics was developed. An output of 9 kg/m^2 .day was reported
	Low and Tay [39] and Tay et al. [40]	Single	Yes	The system uses waste heat from steam turbine
	Uda et al. [41]	Multi effect	Yes	Electric water heater was used instead of solar collector

Main feature	Author	Still type	Experiment	Additional comments
Heat recovery (use	Kunze [42]	Single effect	Yes	Unit efficiency improved over the conventional one
of latent heat of condensation to	Mink et al. [43]	Single effect	Yes	Besides latent heat, sensible heat of condensation was used to preheat the feed
Heat recovery (use of latent heat of	Fernandez and Chargoy [46]	Multi effect	Yes	Each tray has a W-shape bottom that acts as a condenser for the pan below
condensation to evaporate part of the saline water in subsequent effect)	Franco and Saravia [47]	Multi effect	Yes	The effects were inclined metallic plates covered with cotton fabric at the top. At the bottom the vapor from the effect below condenses. Salt water is fed in the upper stage and flows along each surface falling from one stage to the next by gravity reaching finally a heated tray at the bottom
	Kumar and Tiwari [48]	Multi effect	Yes	Numerical model was developed and validated for a single effect still
	Toyama et al. [49] and Toyama et al. [50]	Multi effect	Yes	Heat and mass balance formulas were provided. The simulation results were compared with experimental ones, and a good agreement was obtained
	Schwarzer et al. [45]	Multi effect	Yes	The output could reach five or six times greater than that of a simple solar still
Heat loss minimization	Bobrovnikov et al. [51]	Single effect	No	Method to calculate the losses through the glass covers, bottom and side walls were presented
	Tiwari [52]	Single effect	No	The still was assumed to be a metal tray placed inside glass tube
Still designed for nocturnal	Malik and Tran [53]	Single effect	Yes	Heat was assumed to be supplied from a waste heat source
operation	Onyegegbu [54]	Single effect	Yes	Still with thermal energy storage
Addition of dye	Dutt et al. [55]	Double effect		The still performance was increased with the use of dye

Main feature	Author	Still type	Experiment	Additional comments
Humidification-	Saghafi [44] and Al-Hallaj	-	Yes	Latent heat recovery to heat the feed
dehumidification	et al. [56]			
process	Ben Bacha et al. [57]	-	Yes	The evaporation tower equipped with packed bed to
				increase the contact surface and improve humidification
				rate
	Chaibi et al. [58] and	-	Yes	Economic analysis was presented which showed that the
	Goosen et al. [24]			cost of fresh water produced is high
	Muller-Holst et al. [29]	-	Yes	Evaporator and condenser were place inside the same box
	Tchinda et al. [60]	-	Yes	Thermal analysis was presented. Numerical results
				agreed well with experiments

CHAPTER 2 THE RESEARCH

This chapter provides a description and the operating principle of the proposed system, the objectives, and the scope of work for conducting the research.

The System: Its Description and Operating Principle

This research involved the development and study of a solar distillation system utilizing an innovative new concept, which uses low-grade heat. The concept was proposed by Sharma and Goswami [62]. The concept utilizes natural means to create vacuum under which a liquid can be evaporated at much lower temperatures and with much less energy than the conventional techniques, which would allow the use of low grade heat sources, such as, flat plate solar collectors and/or waste heat. The uniqueness of this concept is in the way natural forces of gravity and the atmospheric pressure are used to create high vacuum conditions and its incorporation in a single system design where evaporation and condensation take place at the appropriate locations without any external energy input other than low grade heat. This also helps in a reduction of thermal energy input in the process. Conventional processes for the reduction of pressure require the use of vacuum pumps, steam ejectors or water jet ejectors.

The system, as shown in fig. 2.1, consists of an evaporation chamber in which there are provisions to feed the cold fluid directly to the chamber and provide solar or other low-grade thermal energy through a closed loop heat exchanger as well as provisions to withdraw the concentrated brine. The evaporation chamber is connected to a condenser, which dissipates the heat of condensation to the environment; the condenser is connected

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to a discharge pipe, about 10 m in height. The incoming cold fluid and the withdrawn brine pass through a tube-in-tube heat exchanger in order to extract the maximum possible energy from the hot brine.

To start up the unit, it is filled completely with water initially. The water is then be allowed to fall under the influence of gravity, creating a vacuum above the water level in the unit. Depending on the barometric pressure, the water will fall to a level of about 10 m above the ground level, leaving behind a vacuum. Vacuum is created by balancing the hydrostatic and the atmospheric pressures in the discharge and supply pipes, without any external mechanical energy.

It is known that the vapor pressure of seawater is about 1.84% less than that of fresh water over the temperature range of 0-100°C, this means that if the top of the two chambers; the saline water chamber (evaporator) and the fresh water chamber (condenser) are connected while maintained at the same temperature, water will distill from the fresh water side to saline water side. If the reverse process is to be maintained, i.e. distillation of water from saline water side to the fresh water side, the vapor pressure of saline water will have to be kept above that of fresh water. In the present system this can be done by increasing the temperature of the saline water utilizing solar energy. So, evaporation from the saline water side to the fresh water side is driven by the difference in the vapor pressure between the two sides. Vacuum enables the distillation of water at a low temperature level, requiring a smaller amount of thermal energy. This energy will be provided from solar collectors, which will operate at a higher efficiency because of lower collector temperatures, minimizing the heat loss to the environment. Simple flat plate

collectors may be used to heat the saline water in the evaporator. So, the unit is best suited for sites with access to sea and/or brackish water and high solar insolation.

As saline water in the evaporator starts evaporating, its salinity increases. As a result the molecular activity of water decreases because the dissolved salts interfere with the motion of the water molecules. Besides the average number of water molecules near the surface decreases and this tends to decrease the evaporation rate. Evaporation rate decreases by about 1% for each percentage increase in salinity [63]. Also, salinity increase moves the system towards the conditions of oversaturation, increasing the scale formation possibilities. So it becomes necessary to withdraw the concentrated brine at a certain flow rate, in such a way that balances the reduction in the evaporation rate and heat loss carried away with the brine as well as preventing the conditions of oversaturation. To make up for the withdrawn brine and evaporation, saline water must be injected at a rate equivalent to the withdrawal plus evaporation rates. The withdrawn water will be at a temperature equal to that of the evaporator, so it becomes necessary to recover the maximum possible energy from it. A tube-in-tube heat exchanger was used for this purpose, where injected water flow inside the inner tube and withdrawn water flow in the annulus in countercurrent directions. The heat exchanger area is such that a major part of the energy is recovered.

Under the influence of vacuum conditions at the saline water surface in the evaporator, water can be injected by the effect of atmospheric pressure; hence no pumping power is required.

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Figure 2.1. Schematic of the proposed system

The present distillation process is a relatively new, non-traditional, and innovative

heat efficient distillation process. First the process was studied and analyzed theoretically

and numerically, based on the numerical results, pilot scale unit was designed, built and tested.

Objectives of the Study

- 1. Develop a theoretical model for the concept to simulate the performance of the unit.
- 2. Use the model to design a pilot scale system based on the concept.
- 3. Conduct an experimental study of the water distillation system, and compare the experimental results with the theoretical ones.
- 4. Conduct an economic analysis of the system.

Scope of Work

Theoretical and experimental research was conducted to study the above-described

concept and develop a practical system based on this concept. The following tasks were

performed:

- 1. A thermodynamic analysis of each component and of the whole process was conducted. A mathematical model was developed that can be used in evaluating the performance of the system, by solving the coupled set of equations from the thermodynamic model.
- 2. Based on the findings of the theoretical study, experimental work was conducted, which included:
 - a. Design, construction, instrumentation and calibration of the experimental set up.
 - b. Data collection.
 - c. Data reduction, analysis and comparison of theoretical and experimental results.
- 3. Economic analysis.

CHAPTER 3 THEORETICAL ANALYSIS

Mathematical equations that describe the performance of each component of the system are presented in this chapter. The method of solving these sets of equations, to predict the system performance, is presented in the appendix A. The computer program, written in MATLAB is presented in appendix B. Three versions of the program were produced, the first one to simulate the case of constant heat source temperature, the second one assumes that a solar collector is the heat source, and the third one to simulate the experimental conditions. A flow chart for the computer program is given in appendix C.

Mass, Solute Concentration, and Energy Balance

A complete mathematical description of the proposed distillation unit requires equations for mass, energy and salt balances.

In applying those balances, it was assumed that no temperature stratification occurs in the system and the heat capacity of the evaporator material is neglected.

Referring to fig. 3.1, application of conservation of mass gives

$$\frac{d}{dt}(\rho V) = \rho_i \dot{V}_i - \rho_w \dot{V}_w - \rho_e \dot{V}_e$$
(3.1)



Figure 3.1. Mass balance of the water body in the evaporator

Initial condition:

$$(\rho V) (0) = \rho_0 V$$
 (3.2)

where the subscripts *i*, *w*, and *e* refer to the entering saline water, withdrawn concentrated brine, and the mass evaporating, respectively, and the subscript 0 denote the reference state for the saline water at 25 °C.

If we assume that the vapor produced will be totally condensed in the condenser side to form the product, that is, no particles will be collapsed to the evaporator, then the amount of fresh water produced is equivalent to the amount evaporated,

$$\rho_f \dot{V}_f = \rho_e \dot{V}_e \tag{3.3}$$

Application of solute conservation (fig. 3.2.) results in

$$\frac{d}{dt}(\rho C V) = (\rho C)_i \dot{V}_i - (\rho C)_s \dot{V}_w$$

$$(3.4)$$

 $(\rho \dot{V} C)_i \quad (\rho \dot{V} C)_w$

Figure 3.2. Salt balance of the water body in the evaporator

Initial condition:

$$(\rho CV) \ (0) = \rho_0 C_0 V \tag{3.5}$$

where *C* is the solute concentration.

At the water surface the temperatures of the vapor and liquid water are assumed equal, so radiation effects are negligible. Since the chamber is evacuated, convection effects are also assumed to be negligible. Application of the conservation of energy (fig. 3.3.) gives

$$\frac{d}{dt}(\rho C_p VT) + \frac{d}{dt}(\rho C_p VT)_{evaporator} = \dot{Q}_{input} + (\rho C_p T)_i \dot{V}_i - (\rho C_p T)_w \dot{V}_w - \dot{Q}_e - \dot{Q}_{loss}$$
...(3.6)



Figure 3.3. Energy balance of the water body in the evaporator

Initial condition:

$$(\rho C_p VT) \quad (0) = \rho_0 C_{p,0} VT_0, \text{ and}$$

$$(\rho C_p VT)_{evaporator} (0) = (\rho C_p V)_{evaporator} T_0$$
(3.7)

Mass Transfer

Methods of vacuum distillation can be classified according to the way in which the

vapor is formed, into three categories [64]:

- 1. Evaporative distillation: Which occurs when the saturation vapor pressure is less than that of the surrounding gas, so that no bubbles are formed. Evaporation rate is controlled by diffusion or convection of the vapor to the condenser.
- 2. Ebullition: Which occurs when the saturated vapor pressure exceeds the pressure of the surrounding gas. It is accompanied by the formation of bubbles, whose formation and movement create turbulence, which enhances the mass transfer. Ebullition is always accompanied by a certain amount of evaporative distillation (surface evaporation) that increases with the reduction of pressure.
- 3. Molecular distillation: Which is the process of evaporation into a gas free space (vacuum). The controlling factor is the rate at which molecules escape from the heated surface of the liquid and are received at the condenser. Therefore, for best results the condenser should be in the vicinity of the evaporating surface. In this process, the equilibrium between the evaporated molecules and the liquid is continually disturbed by condensation so that, in accordance with the physical laws, equilibrium has to be re-established. This means that more molecules will

evaporate from the liquid surface, and there will be no definite temperature at which distillation starts. It will take place whenever there is a temperature gradient between the evaporating surface and the condenser.

The maximum rate of distillate production in molecular distillation, is given by the Langmuir-Knudsen interpretation of the Maxwell-Boltzmann law [65]:

$$\dot{m} = \frac{P_s A}{\sqrt{2\pi MRT_s}} \tag{3.8}$$

where A, P, M, R, and T^1 are the cross sectional area of the evaporator, vapor pressure, molecular weight, gas constant, and the temperature of the fluid inside the evaporator, respectively.

This relation is valid in its present form if the device used for molecular distillation complies with the following requirements:

- Components under vacuum are wide, so pressure differentials are avoided.
- The liquid is evenly distributed as a thin film.
- Distance between the evaporating and condensing surfaces is not greater that the mean free path of the residual gas.
- Condensation temperature is about 50-100°C below the evaporation temperature.
- Substance to be distilled undergoes preliminary degassing to minimize the amount of noncondensable gas.

The mass transfer in the proposed unit is similar to category 3, molecular distillation, described in the previous page. Since it would not comply with the abovementioned requirements, the above relation for evaporation does not apply directly. The net rate of mass transfer at the phase interface is the sum of the rates of vaporization and condensation which can be calculated using the following equations, based on the analysis by Schrage [66]:

¹ Nomenclature in this section is as defined in the text

Vaporization Rate:

If dn is the number density of molecules which lie in a volume element dc of velocity space at a velocity c, then

$$dn = f.dc = f.dUdVdW \tag{3.9}$$

Where, U, V, W are the components of the molecular velocity c in x, y, z directions, and f is a velocity distribution function.

If the vapor is uniform, statistical mechanics predicts the form of the velocity distribution function, known as Maxwell velocity distribution function, and is given as,

$$f = A \exp[-\beta^2 (c - \overline{c})^2]$$

$$f = A \exp[-\beta^2 ((U - \overline{U})^2 + (V - \overline{V})^2 + (W - \overline{W})^2)$$
(3.10)

where A and β are constants which depend upon the thermodynamic state of the

gas, \overline{c} is linear velocity, and $\overline{U}, \overline{V}, \overline{W}$ are the components of the linear velocity.

Substituting eq. 3.10 in eq. 3.9 and integrating overall velocity space imposes a necessary condition upon the two constants *A* and β [66],

$$A = n \frac{\beta^3}{\pi^{\frac{3}{2}}}$$
(3.11)

$$\beta = \sqrt{\frac{1}{2RT}} \tag{3.12}$$

As shown in fig. 3.4, the origin of coordinate axes is assumed to be at the phase interface and the x-axis is directed into the gas. Water surface is in the yz plane.



Figure 3.4. Coordinate system at the water surface

Then, the mass flow rate per unit area from the phase interface is given as

$$\overset{\bullet}{m} + = \int_{-\infty-\infty0}^{\infty} \int_{-\infty-\infty0}^{\infty} m U \overline{f}_s dU dV dW$$
(3.13)

where, \dot{m} + is the vaporization rate per unit area,

m is the mass of molecule, and

 $\overline{f_s}$ is a distribution function for the gas in equilibrium with the surface.

Assuming the gas to be uniform,

$$\overline{f_s} = n \left(\frac{\beta^3}{\pi^{\frac{3}{2}}}\right) \exp[-\beta^2 \left((U - \overline{U})^2 + (V - \overline{V})^2 + (W - \overline{W})^2\right)]$$
(3.14)

$$\beta = \sqrt{\frac{1}{2RT}} \tag{3.15}$$

$$\overline{U} = \overline{V} = \overline{W} = 0 \tag{3.16}$$

Eq. 3.13 becomes

$$\dot{m} + = \rho \left(\frac{\beta_s^3}{\pi^{\frac{3}{2}}}\right) \int_{-\infty-\infty}^{\infty} \int_{0}^{\infty} U \exp\left[-\beta_s^2 (U^2 + V^2 + W^2)\right] dU dV dW$$
$$\dot{m} + = \rho \left(\frac{\beta_s^3}{\pi^{\frac{3}{2}}}\right) \left(\frac{1}{2\beta_s^2}\right) \left(\frac{\pi^{\frac{1}{2}}}{\beta_s}\right) \left(\frac{\pi^{\frac{1}{2}}}{\beta_s}\right)$$

$$\mathbf{\dot{m}} + = \frac{\rho}{2\pi^{\frac{1}{2}}\beta_s} \tag{3.17}$$

where the subscript *s* is used to denote the properties corresponding to the temperature of the saline water in the evaporator.

Substituting for β_s and assuming ideal gas relationship,

$$\overset{\bullet}{m} + = \rho \sqrt{\frac{RT_s}{2\pi}} = p_s \sqrt{\frac{1}{2\pi RT_s}}$$

$$(3.18)$$

Condensation Rate:

Following the same procedure as above, it can be shown that, the condensation rate per unit area at the phase interface is given as [66]

$$\overset{\bullet}{m} = -\left(\frac{\rho_{gi}}{\rho_{gs}}\right) \left(\frac{T_{gi}}{T_s}\right)^{\frac{1}{2}} \Gamma \overset{\bullet}{m} +$$
(3.19)

$$\Gamma = \exp(-\beta_{gi}^2 U_{gi}^2) - \beta_{gi} U_{gi} \pi^{\frac{1}{2}} (1 - \Phi \beta_{gi} U_{gi})$$
(3.20)

where m- is the condensation rate per unit area, Γ is a correction factor, Φ is a probability function, and the subscript g denotes that the properties of the gas, and i refers to the interface.

The net rate of mass transfer per unit area i.e. the difference between vaporization and condensation rates, is given as

$$\dot{m} = \sigma \left[\dot{m} + - \left(\frac{\rho_{gi}}{\rho_{gs}} \right) \left(\frac{T_{gi}}{T_s} \right)^{1/2} \Gamma \dot{m} + \right]$$

$$\dot{m} = \frac{\sigma}{\left(2\pi R\right)^{1/2}} \left(\frac{p_s}{T_s^{1/2}} - \Gamma \frac{p_{gi}}{T_{gi}^{1/2}} \right)$$
(3.21)

where σ is the accommodation coefficient for evaporation or condensation, which is equal to the ratio of the vapor molecules sticking to the phase interface to those impinging on it.

Based on the above result, Bemporad [67] found a correlation for evaporation rate between two chambers, one containing seawater and the other containing fresh water, which are connected via a vacuum chamber. The evaporation rate per unit area, \dot{v}_e , from seawater chamber to fresh water chamber is given as (assuming that evaporation is not impeded by foreign gas molecules)

$$\mathbf{v}_{e} = \frac{\alpha_{m}}{\rho_{f}} \left[f(C) \frac{p(T_{s})}{(T_{s} + 273)^{0.5}} - \frac{p(T_{f})}{(T_{f} + 273)^{0.5}} \right]$$
(3.22)

where f(c) is a correction factor, which accounts for the solute concentration is given as [67]

$$f(C) = 1 - \alpha_1 C \tag{3.23}$$

 α_m and α_l are empirical coefficients, ρ_f is the density of fresh water, and T_s and T_f are the temperatures of saline and fresh water in °C, respectively.

In the above equation the pressure drop between the evaporator and condenser is neglected, however, this should be calculated (discussed in detail in a later section) and added to that of the fresh water, then the above equation becomes

$$v_e = \frac{\alpha_m}{\rho_f} \left[f(C) \frac{p(T_s)}{(T_s + 273)^{0.5}} - \frac{p(T_f) + \Delta p}{(T_f + 273)^{0.5}} \right]$$
(3.24)

Evaporation energy is given as

$$\dot{Q}_e = \rho_f h_{fg}(T_s) \dot{V}_e \tag{3.25}$$

Evaporator Heat Exchanger

The energy supplied to the system from the heat source, is assumed to be in the form of hot water, via heat exchanger. Heat transfer rate in the evaporator is a function of the available temperature difference. It is difficult to determine the temperature of the liquid at all locations along the heating surface, but an average value, hence the amount of heat transferred, can be calculated as shown below.

For an infinitesimal element of width Δx , fig. 3.5, the energy balance gives

$${}^{\bullet} m_c C_{p,cf} \frac{dT_{cf}(x)}{dx} \Delta x = -\pi D_h U_h (T_{cf} - T_s) \Delta x$$

$$(3.26)$$

where D_h and U_h are the diameter and overall heat transfer coefficient of the evaporator heat exchanger, and the subscript *cf* refers to the collector fluid flowing through the evaporator heat exchanger.

The boundary conditions: $T_{cf}(x) = T_{co}$ at x = 0 (neglecting heat loss in the connecting pipes). The subscript *co* refers to the temperature of the fluid at the collector outlet.



Figure 3.5. Control volume for the flow through the evaporator heat exchanger

Solving eq. 3.26 we get

$$T_{cf}(x) = T_{s} + (T_{co} - T_{s}) \exp\left[\frac{-\pi D_{h} U_{h} x}{m_{c} C_{p,cf}}\right]$$
(3.27)

Net amount of heat transferred to the water can be written as,

$$\dot{Q}_u = \pi D_h U_h l_h (\overline{T}_{cf} - T_s)$$
(3.28)

where l_h is the length of the evaporator heat exchanger and \overline{T}_{cf} is the fluid average temperature, given as

$$\overline{T}_{cf} = \frac{1}{l_h} \int_{0}^{l_h} T_{cf}(x) dx$$
(3.29)

Substitution of eq. 3.27 in eq. 3.29 and carrying out the integration, and then substitution of the result in eq. 3.28, gives

as

$$\dot{Q}_{u} = \overset{\bullet}{m}C_{pcf}(T_{co} - T_{s})\left[1 - \exp\left(\frac{-\pi D_{h}U_{h}l_{h}}{\overset{\bullet}{m_{c}}C_{p,cf}}\right)\right]$$
(3.30)

From eq. 3.27, fluid temperature at the outlet of the heat exchanger, may be written

$$T_{cf,outlet} = T_s + (T_{co} - T_s) \exp\left[\frac{-\pi D_h U_h l_h}{\frac{1}{m_c} C_{p,cf}}\right]$$
(3.31)

Neglecting heat loss in the connecting pipes, the temperature of the fluid entering the collector, T_{ci} , is given by

$$T_{ci} = T_{cf,outlet} \tag{3.32}$$

where $T_{cf,outlet}$ is the temperature of the collector fluid flowing through the evaporator heat exchanger at the exit.

The overall heat transfer coefficient, U_h , between the working fluid and saline water may be calculated as

$$U_{h} = \frac{1}{\frac{1}{h_{f}} + FT}$$
(3.33)

where *FT* is the heat exchanger fouling factor, which may vary in practice from 0.0005 for clean tubes to 0.001 W/m².°C for adverse scale conditions [68], and h_f is the heat transfer coefficient between the collector fluid and the evaporator heat exchanger, which can be calculated as follows [69];

If the flow is laminar, i.e.

$$\operatorname{Re}_{D} = \frac{4\rho_{0}V_{h}}{\pi D_{h}\mu} \leq 2300 \tag{3.34}$$

then,

$$Nu_D = 3.66$$
 (3.35)

If the flow is turbulent, i.e.

 ${\rm Re}_{D} > 2300$

then,

$$Nu_D = 0.023 \,\mathrm{Re}_D^{0.8} \,\mathrm{Pr}^{0.4} \tag{3.36}$$

Tube-in-Tube Heat Exchanger

The injection pipe, which carries the seawater to the evaporator, is coaxial and internal to the withdrawal pipe (fig. 3.6), so, a major part of the energy of withdrawn water can be recovered.

The temperature of the water entering to the evaporator, T_i , can be calculated as follows [69]:

$$T_{i} = \frac{\dot{Q}_{h}}{\rho_{0} V_{i} C_{p0}} + T_{0}$$
(3.37)



Figure 3.6 Cross section of the tube-in-tube heat exchanger

where, \dot{Q}_h is the actual heat transfer rate, given as

$$\dot{Q}_h = \varepsilon \dot{Q}_{\max} \tag{3.38}$$

 $\overset{\bullet}{Q}_{\mathrm{max}}$ is the maximum possible heat transfer rate, given by

•
$$Q_{\max} = C_{\min}(T_s - T_0)$$
 (3.39)

 ε is the heat exchanger effectiveness, given by

$$\varepsilon = \frac{1 - \exp[-NTU(1 - C_r)]}{1 - C_r \exp[-NTU(1 - C_r)]}$$
(3.40)

NTU is the number of transfer units, given as

$$NTU = \frac{UA}{C_{\min}}$$
(3.41)

$$C_r = \frac{C_{\min}}{C_{\max}}$$
(3.42)

 C_{min} and C_{max} are the minimum and maximum of C_c and C_h , respectively, where

$$C_{c} = \rho_{0} \dot{V}_{i} C_{p0}$$
(3.43)

$$C_h = \rho_s \dot{V}_w C_{ps} \tag{3.44}$$

The product UA is given as

$$UA = \frac{1}{\frac{1}{2\pi r_i lh_i} + \frac{\ln(r_o/r_i)}{2\pi kl} + \frac{1}{2\pi r_o lh_o}}$$
(3.45)

The heat transfer coefficient between the injected water and the injection pipe, h_i , can be calculated from eq. 3.35 if the flow is laminar and from eq. 3.36 if the flow is turbulent.

The heat transfer coefficient between the withdrawn water and the withdrawal pipe,

 h_o , can be calculated as follows

If the flow is laminar, i.e.

$$\operatorname{Re}_{D} = \frac{\rho(D_{o} - D_{i})}{\mu} \frac{\rho_{s} \dot{V}_{w}}{0.25\rho\pi(D_{o}^{2} - D_{i}^{2})} \le 2300$$
(3.46)

Then the value of Nusselt number can be selected from the following table 3.1.

D_i/D_o	Nu _i
0	-
0.05	17.46
0.1	11.56
0.25	7.37
0.5	5.74
1	4.86

Table 3.1 Nusselt number for laminar flow in annulus [69]

If the flow is turbulent, eq. 3.36 can be used, with the diameter replaced by the hydraulic diameter [69].

Condenser Analysis

Evaporation of water from saline water chamber tends to cool it, while

condensation of vapor tends to heat fresh water, and for the process of distillation to be

continuous, heat is to be added continuously to the evaporator and rejected from fresh water chamber. The condensation heat transfer will mainly be dissipated to the environment via the condenser. So the condenser should be able to dissipate the amount of energy given by

$$\dot{Q}_{c} = \dot{m}(h_{fg} + 0.68C_{p}(T_{s} - T_{if}))$$
(3.47)

where, h_{fg} is the latent heat of condensation and the second term in right hand side of the above equation accounts for the sensible heat transfer from the condensate.

This amount of heat is transferred through the condensate film, then conducted through the condenser wall, and eventually transferred to the environment by convection (if we neglect radiation).

The condensation heat transfer coefficient is expected to be very large compared to the free convection heat transfer coefficient from the outside surface of the condenser. Since the velocity of vapor is small, and the condensation rate is low, the condensate will flow as a thin annular film inside the tube. Then it flows in a longitudinal direction along the bottom side of the tube. For flow with Re < 30,000, the average film heat transfer coefficient is given as [70]

$$\bar{h}_{c} = 0.725 \left[1 - \frac{\theta}{\pi} \right] \left[\frac{h_{fg}^{*} g k_{c}^{3} (\rho_{l} - \rho_{v})}{\gamma_{l} D (T_{s} - T_{if})} \right]^{\frac{1}{4}}$$
(3.48)

where, θ is the half angle made by joining the center of the tube to the two edges of the flowing liquid at the bottom, as shown in fig. 3.7.



Figure 3.7. Cross section through the condenser

Film properties involved are to be evaluated at an intermediate temperature between the interface, T_i , and the inside surface of the condenser, T_{ci} , so that the temperature jump across the film is accounted for [71]

$$T_{film} = T_{ci} + 0.25(T_i - T_{ci}) \tag{3.49}$$

where T_i is the interface temperature calculated by assuming that all of the heat transferred from the vapor to the interface, Q_{s-i} , is conducted through the liquid film to the condenser surface, Q_{i-ci} . Therefore, the interface temperature does not change with time, i.e.

$$\frac{dT_i}{dt} = 0 \tag{3.50}$$

and

$$\dot{Q}_{s-i} = \dot{Q}_{i-ci} \tag{3.51}$$

or

$$\dot{Q}_{c} = \bar{h}_{c} A_{i} (T_{i} - T_{ci})$$
(3.52)

where A_i is the surface area of the liquid film at the interface

For heat conduction through the condenser wall,

$$\dot{Q}_{c} = \frac{2\pi l_{c} k_{c} (T_{ci} - T_{co})}{\ln(r_{co} / r_{ci})} = \frac{T_{i} - T_{co}}{\frac{1}{2\pi r_{ci} l_{c} \bar{h}_{c}} + \frac{\ln(r_{co} / r_{ci})}{2\pi l_{c} k_{c}}}$$
(3.53)

where T_{co} , r_{ci} , r_{co} , l_c , and k_c are the condenser outside surface temperature, inside radius, outside radius, length, and thermal conductivity, respectively.

The condenser is assumed to be a horizontal tube with circular fins. The average heat transfer from the tips of the fins is given by [72]

$$Nu_s = cRa_s^b \tag{3.54}$$

This equation is valid for

$$2 \le Ra_s \le 10^4$$

and

$$1.36 < 1/\xi < 3.73$$

where

$$\xi = \frac{D_{co}}{D_{fin}}$$

Rayleigh number and other constants are given by

$$Ra_{s} = \frac{g\beta(T_{co} - T_{a})S^{3}}{\alpha\gamma} \frac{S}{D_{fin}}$$
(3.55)

b=0.29

$$c = 0.44 + 0.12 / \xi$$
, and

S is the distance between two successive fins.

Heat transfer from the cylinder and lateral fin surfaces is given by [72]

$$Nu_{s} = \frac{Ra_{s}}{12\pi} \left\{ 2 - \exp\left[-\left(\frac{c1}{Ra_{s}}\right)^{\frac{3}{4}} \right] - \exp\left[-\beta\left(\frac{c1}{Ra_{s}}\right)^{\frac{3}{4}} \right] \right\}$$
(3.56)

where

$$1.67 < 1/\xi < \infty$$

$$\beta = 0.17\xi + \exp(-4.8\xi)$$
$$c1 = \left[\frac{23.7 - 1.1(1 + 152\xi^2)^{\frac{1}{2}}}{1 + \beta}\right]^{\frac{4}{3}}$$

The rate of heat transferred from the condenser (fins and prime surface) can be calculated as

$$Q_c = [h_{co,tip} NA_{f,tip} \eta_f + h_{co} NA_{f,sides} \eta_f + h_{co} A_b](T_{co} - T_a)$$
(3.57)

where N is the number of fins, and η_f is the fin efficiency calculated as follows [69],

$$\eta_f = C \frac{K_1(mr_{co})I_1(mr_{2c}) - I_1(mr_{co})K_1(mr_{2c})}{I_0(mr_{co})K_1(mr_{2c}) + K_0(mr_{co})I_1(mr_{2c})}$$

where

$$C = \frac{2r_{co}/m}{r_{2c}^2 - r_{co}^2}$$
$$r_{2c} = r_{fin} + t/2$$
$$m = \left(\frac{2h_{co}}{k_c t_{fin}}\right)^{\frac{1}{2}}$$

and I_0 , I_1 and K_0 , K_1 are the modified Bessel functions of the first and second kind, respectively.

Heat Loss

For simulations with constant temperature heat source, solar collector, and for indoor tests, heat loss from the evaporator was assumed to be by free convection, whereas for outdoor tests, heat loss to the environment is controlled by the wind speed. If the wind speed is less than 0.5 mph, heat loss was assumed to be by free convection, and if the wind speed is greater than 0.5 mph, heat loss to the environment was assumed to be by

forced convection [73]. In both cases radiative heat transfer was neglected. Heat was assumed to be transferred from the bottom, side, and top. Each part has a different heat transfer coefficient. Heat loss by conduction from the evaporator to condenser is prevented with the help of an insulating section made of polycarbonate.

Heat loss from the bottom to the environment can be calculated as

$$Q_{bottom} = \frac{T_s - T_{bottom}}{t_{steel} A_s + \frac{t_{ins}}{k_{ins} A_s} + \frac{t_{wood}}{k_{wood} A_s}} = \frac{T_{bottom} - T_a}{\frac{1}{h_{bottom} A_s}}$$
(3.58)

The bottom natural convection heat transfer coefficient is given by the following formula [69],

$$\overline{Nu} = 0.27 R a_L^{0.25}, \qquad (10^5 \le R a_L \le 10^{10}) \qquad (3.59)$$

where Ra_L is the Rayleigh number given by

$$Ra_{L} = \frac{g\beta(T_{bottom} - T_{a})l^{3}}{\alpha\gamma}$$

l is characteristic length, given by

$$l = \frac{A_s}{p}$$

where A_s , and p are evaporator cross sectional area and perimeter, respectively.

Heat loss from the side is given by

$$Q_{side} = \frac{T_{s} - T_{side}}{\ln\left(\frac{r_{s,o}}{r_{s}}\right) / 2\pi k_{steel} l_{s}} + \frac{\ln\left(\frac{r_{ins,o}}{r_{s,o}}\right) / 2\pi k_{ins} l_{s}}{2\pi k_{ins} l_{s}} = \frac{T_{side} - T_{a}}{\frac{1}{h_{s}} 2\pi r_{ins,o} l_{s}}$$
(3.60)

where $r_{s,o}$, r_s , $r_{ins,o}$, and l_s are the outside radius, inside radius of the evaporator, the radius from the center of the evaporator to the outside surface of the insulation, and the evaporator height.

The side natural convection heat transfer coefficient is given by [69]

$$\overline{Nu} = 0.68 + \frac{0.67Ra_{L}^{0.25}}{\left[1 + \left(0.492/Pr\right)^{9/16}\right]^{4/9}} \qquad (Ra_{L} \le 10^{9})$$
(3.61)

The top part of the evaporator is treated as an inclined plate, with an inclination angle θ . Heat loss from that part is calculated as

$$Q_{top} = \frac{T_s - T_{top}}{t_{steel} / k_{steel} A_{top}} + \frac{t_{ins}}{k_{ins} A_{top}} = \frac{T_{top} - T_a}{1 / h_{top} A_{top}}$$
(3.62)

The top natural convection heat transfer coefficient is calculated as [74]

$$\overline{Nu} = 0.56 (Ra_L \cos\theta)^{0.25} \qquad (10^5 \le Ra_L \cos\theta \le 10^{11}) \qquad (3.63)$$

In case of natural convection, iteration is required to solve for the above heat transfer coefficients, since they as well as air properties are functions of temperature, which is also not known initially.

For the case of forced convection, heat transfer coefficient can be calculated as [75]

$$\overline{Nu} = 0.42 \,\mathrm{Re}^{0.6}$$
 (3.64)

where the characteristics length is the cube root of the evaporator volume.

Operating Pressure

Any liquid surface inside a vacuum system is a source of vapor, and as long as any liquid remains in the system, the minimum pressure attainable is the vapor pressure of that liquid at the existing temperature, and reduction in the temperature of any part of the system will result in a reduction of the vapor pressure of any vapor present.

The pressure in the evaporator of proposed unit can be taken as the sum of the pressure in the vapor space at the point of condensation, which is the saturation vapor

pressure at the interface temperature and the pressure due to non-condensable gases, and the pressure drop occurring in the column (the parts that connect the evaporator to the condenser).

The four most abundant gases in seawater are nitrogen, oxygen, argon, and carbon dioxide. The solubility of CO_2 in seawater is greater than the other gases. This is due mainly to the reactivity of CO_2 in seawater leading to the carbonate and bicarbonate equilibrium:

$$CO_2 + H_2O \Leftrightarrow H_2CO_3 \Leftrightarrow H^+ + HCO_3^- \Leftrightarrow 2H^+ + CO_3^{-2}$$

Concentration of the above gases in seawater is given the table 3.2 [76].

Table 3.2 Concentration of gases in seawater

Element	Concentration (ppm)	Some probable dissolved species
Carbon	28	$HCO_{3}^{-}, CO_{3}^{-2}, CO_{2}^{-2}$
Nitrogen	11.5	N_2 gas, NO_3^- , NH_4^-
Oxygen	6	O_2 gas
Argon	0.43	Ar gas

The gases dissolve in the seawater in proportion to their atmospheric partial pressure. The solubility of gases decreases with the increase of temperature and salinity and increases with increasing pressure.

The mixture of water vapor and non-condensable gases can be treated as an ideal gas mixture, for which the total pressure is given as

$$P = P_{gases} + P_{H_2O} \tag{3.65}$$

The partial pressure of water is equal to the vapor pressure at T_i ,

$$P_{H_2O} = P_{\nu}(T_i)$$
(3.66)

The partial pressure of non-condensable gases can be found using the Dalton model,

$$P = \sum_{i=1}^{j} P_i$$
 (3.67)

where

$$P_i = \frac{n_i \,\bar{R} T}{V} \tag{3.68}$$

Pressure drop can be calculated by applying the continuity and the energy equations between the inlet and the outlet of the column connecting the evaporator to the condenser:

$$(\rho A v)_{in} = (\rho A v)_{out} \tag{3.69}$$

$$\frac{P_{out}}{\gamma} + \frac{v_{out}^2}{2g} + z_{out} = \frac{P_{in}}{\gamma} + \frac{v_{in}^2}{2g} + z_{in} - h_L$$
(3.70)

where, h_L is the head loss between the inlet and outlet of the column, which can be calculated as

$$h_L = \sum k_L \frac{v^2}{2g} \tag{3.71}$$

 k_L is the loss coefficient for the various parts of the column, taken as 0.2 for the truncated cone and 0.3 for the elbow [77].

In vacuum distillation, there is a larger volume of vapor to be handled. In order to avoid any significant impedance to vapor flow in the connecting pipes, the tubing should be as short and as wide as possible. It is often the practice to use connecting tubes of diameter larger than (1/10) of the evaporator diameter, at least in the upper part of the

evaporator, where the lowest pressure prevails and flooding is most likely to take place [65].

Heat Source

To start and maintain distillation, continuous supply of heat is required, which goes to preheat the feed, evaporate the water, and to compensate for the heat losses. This heat can be supplied by flat plate solar collectors. Useful energy collected by a solar collector, is given by [78]

$$Q_{u} = m_{c} C_{p,cf} (T_{co} - T_{ci})$$
(3.72)

or

$$Q_{u} = F_{R}A_{c}[I_{c}(\tau\alpha) - U_{L}(T_{ci} - T_{a})]$$
(3.73)

From eq. 3.72 and eq. 3.73, collector outlet temperature can be calculated as

$$T_{co} = T_{ci} + \frac{ZF_R A_c}{m_c C_{p,cf}}$$

$$(3.74)$$

where

$$Z = I_c(\tau \alpha) - U_L(T_{ci} - T_a)$$

If the collector operates on a thermosiphon mode, the mass flow rate in the collector can be evaluated from the following expression [79],

$$\mathbf{\dot{m}}_{c} = \left[A_{c} Z \frac{\pi N_{c} D_{c}^{4} g \beta_{f} \rho_{f} f'}{128 L_{1} \gamma C_{p,cf} (1+r_{p})} \left(\frac{L_{c}}{2} \sin \beta + H_{c} \right) \right]^{0.5}$$
(3.75)

The vapor ascending from the evaporator should reach the condenser without the introduction or removal of heat. This requirement can be satisfied sufficiently with the use of usual insulating materials up to a temperature of 100 °C. So the evaporator and all

parts before the condenser should be insulated. In such a case heat losses are minimized and may be neglected.

Thermal Efficiency

The analysis describes the simultaneous heat and mass transfers in the unit. The performance of the system can be described in terms of the extent to which the heat added, \dot{Q}_u is used to evaporate the saline water to produce fresh water, $\dot{m}_f h_{fg}$. The extent of energy conversion can be expressed by the following ratio, which is known as thermal efficiency,

$$\eta_t = \frac{\stackrel{\bullet}{m_f \cdot h_{fg}}}{\stackrel{\bullet}{Q}_u} \tag{3.76}$$

The above expression is for the instantaneous efficiency, while a daily efficiency may be calculated as

$$\eta_{daily} = \frac{\int \dot{m} h_{fg} dt}{\int \dot{Q}_u dt} \cong \frac{m_{day} h_{fg} (av.)}{\Delta t \Sigma \dot{Q}_u}$$
(3.77)

Exergy Analysis

The maximum available work per unit mass of a given heat transfer fluid at a constant ambient temperature is a function of the temperature of that fluid, and is given as

$$w_{\max} = (h - h_0) - T_0(s - s_0)$$
(3.78)

where, h and s are the specific enthalpy and specific entropy of the heat transfer fluid, respectively, and h_0 and s_0 are the specific enthalpy and specific entropy of the heat transfer fluid at the ambient temperature, respectively. However, actual processes will have irreversabilities, which will result in a reduction in the amount of the maximum work. These irreversabilities can be calculated for each component of the system, this enables us to identify where the large losses take place and try to minimize them.

Second law of thermodynamics allows the estimation of the irreversibility of the various components of the system. The second law efficiency is a measure of utilization of the available energy. The fraction of the available energy converted into useful work is known as the second law efficiency and can be calculated as

$$\eta_2 = \frac{w_{net}}{availability} = \frac{w_{net}}{w_{max}}$$
(3.79)

For the present system, this efficiency is given as

$$\eta_{2} = \frac{m_{f} h_{fg}}{m_{c} (e_{c,out} - e_{c,in})}$$
(3.80)

The exergy balance for the present system is given as

$$\frac{dE_s}{dt} = \dot{m}\left(e_{c,out} - e_{c,in}\right) + \dot{m}_i e_i - \dot{m}_w e_w - \dot{m}_f e_s + \sum_j \left(1 - \frac{T_0}{T_j}\right) \dot{\mathcal{Q}}_{loss,j} - \dot{E}_d$$
(3.81)

The exergy destruction is then calculated as

$$\dot{E}_{d} = \dot{m} \left(e_{c,out} - e_{c,in} \right) + \dot{m}_{i} e_{i} - \dot{m}_{w} e_{w} - \dot{m}_{f} e_{s} + \sum_{j} \left(1 - \frac{T_{0}}{T_{j}} \right) \dot{\mathcal{Q}}_{loss,j} - \left(\frac{E^{1} - E^{0}}{\Delta t} \right) (3.82)$$

The specific exergy, e, is given as (neglecting kinetic and potential energy),

$$e = (h - h_0) - T_0(s - s_0)$$
(3.83)

The exergy balance for individual components are given as

Tube-in-tube heat exchanger

$$\overset{\bullet}{E}_{d} = \overset{\bullet}{m}_{w} [(h_{in} - h_{out}) - T_{0}(s_{in} - s_{out})]_{w} + \overset{\bullet}{m}_{i} [(h_{in} - h_{out}) - T_{0}(s_{in} - s_{out})]_{i}$$
(3.84)

Evaporator heat exchanger

$$\overset{\bullet}{E}_{d} = \overset{\bullet}{m} \Big[\Big(h_{c,out} - h_{c,inlet} \Big) - T_0 \Big(s_{c,out} - s_{c,inlet} \Big) \Big] + \left(1 - \frac{T_0}{T_{he,o}} \right) \overset{\bullet}{Q}_{u}$$
(3.85)

Condenser

$$\overset{\bullet}{E}_{d} = \overset{\bullet}{m}_{f} \left[\left(h_{in} - h_{out} \right) - T_{0} \left(s_{in} - s_{out} \right) \right]_{c} + \left(1 - \frac{T_{0}}{T_{co}} \right) \overset{\bullet}{\mathcal{Q}}_{c}$$
(3.86)

Scaleup of the System

The principle of similarity is applied here to study the effect of scaleup the system. Geometric, kinematic, and dynamic similarities should exist between the model and prototype. These similarities can be interpreted in terms of dimensionless groups and variables; the numerical values of all of the dimensionless groups should remain constant during scaleup. One way to obtain the dimensionless groups and variables is to transform the various balances: mass, salt, and energy, into dimensionless form. The following nondimensional parameters are introduced,

$$\dot{V}_{in}^{*} = \frac{\dot{V}_{in}}{\left(\frac{V}{t_{R}}\right)}, \\ \dot{V}_{i}^{*} = \frac{\dot{V}_{i}}{\left(\frac{V}{t_{R}}\right)}, \\ \dot{V}_{w}^{*} = \frac{\dot{V}_{w}}{\left(\frac{V}{t_{R}}\right)}, \\ \dot{V}_{e}^{*} = \frac{\dot{V}_{e}}{\left(\frac{V}{t_{R}}\right)}$$

where t_R is a reference time to be defined later.

$$\rho^{*} = \frac{\rho_{s}}{\rho_{0}}, \rho_{in}^{*} = \frac{\rho_{in}}{\rho_{0}}, \rho_{i}^{*} = \frac{\rho_{i}}{\rho_{0}}, \rho_{e}^{*} = \frac{\rho_{e}}{\rho_{0}}, \rho_{evaporator}^{*} = \frac{\rho_{evaporator}}{\rho_{0}}$$
$$C_{p}^{*} = \frac{C_{p,s}}{C_{p,0}}, C_{p,in}^{*} = \frac{C_{p,in}}{C_{p,0}}, C_{p,i}^{*} = \frac{C_{p,i}}{C_{p,0}}, C_{p,evaporator}^{*} = \frac{C_{p,evaporator}}{C_{p,0}}$$
$$T^* = \frac{T_s}{T_0}, T_{in}^* = \frac{T_{cout} - T_{cinlet}}{T_0}, T_i^* = \frac{T_i}{T_0}, T_{evaporator}^* = \frac{T_{evaporator}}{T_0}, T_{top}^* = \frac{T_{top} - T_a}{T_0}, T_{side}^* = \frac{T_{side} - T_a}{T_0},$$

$$T_{bottom}^* = \frac{T_{bottom} - T_a}{T_0}$$

$$C^* = \frac{C_s}{C_0}$$

$$t^* = \frac{t}{t_R}$$

$$V^* = \frac{V_{evaporator}}{V}$$

Substituting in mass conservation equation (eq. 3.1) gives

$$\frac{d\rho^{*}}{dt^{*}} = \rho_{i}^{*} \overset{\bullet}{V}_{i}^{*} - \rho^{*} \overset{\bullet}{V}_{w}^{*} - \rho_{e}^{*} \overset{\bullet}{V}_{e}^{*}$$
(3.87)

Initial condition:

$$\rho^*(0) = 1 \tag{3.88}$$

Substituting in solute conservation equation (eq. 3.4) gives

$$\frac{d}{dt^*} \left(\rho^* C^* \right) = \overset{\bullet^*}{V_i} \rho_i^* - \overset{\bullet^*}{V_w} \rho^* C^*$$
(3.89)

Initial condition:

$$\rho^* C^*(0) = 1 \tag{3.90}$$

Substituting in energy conservation equation (eq. 3.6) gives

$$\frac{d}{dt^{*}} \left(\rho^{*} C_{p}^{*} T^{*} \right) + \frac{d}{dt^{*}} \left(\rho^{*} V^{*} C_{p}^{*} T^{*} \right)_{evaporator} = \overset{\bullet}{V_{in}} \rho_{in}^{*} C_{p,in}^{*} T_{in}^{*} + \overset{\bullet}{V_{i}} \rho_{i}^{*} C_{p,i}^{*} T_{i}^{*} - \overset{\bullet}{V_{w}} \rho^{*} C_{p}^{*} T^{*} - \overset{\bullet}{V_$$

The quantities $\frac{hAt_R}{\rho_0 C_{p,0}V}$ must be dimensionless. We can define the reference time,

 t_R , as

$$t_R = \frac{\rho_0 C_{p,0} V}{h_{top} A_{top}}$$

So that the dimensionless quantity,

$$\frac{h_{top}A_{top}t_R}{\rho_0 C_{p,0}V} = 1$$

Also, we can define the following dimensionless quantity as

$$\frac{h_{fg}}{C_{p,0}T_0} = \frac{1}{Ja}$$

where, Ja is the Jakob number, defined here as

$$Ja = \frac{C_{p,0}T_0}{h_{fg}}$$

Assume that the density, specific heat, and volume of the evaporator material

remain constant over the operating temperature range. Substituting in eq. 3.91 gives

$$\frac{d}{dt^{*}}(\rho^{*}C_{p}^{*}T^{*}) + (\rho^{*}V^{*}C_{p}^{*})_{evaporator} \frac{dT_{evaporator}^{*}}{dt^{*}} = V_{in}^{*}\rho_{in}^{*}C_{p,in}^{*}T_{in}^{*} + V_{i}^{*}\rho_{i}^{*}C_{p,i}^{*}T_{i}^{*} - V_{w}^{*}\rho^{*}C_{p}^{*}T^{*} - \frac{V_{e}^{*}\rho_{e}^{*}}{Ja} - \frac{V_{e}^{*}\rho_{e}^{*}}{Ja} - T_{top}^{*} - \frac{h_{side}A_{side}}{h_{top}A_{top}}T_{side}^{*} - \frac{h_{bottom}A_{bottom}}{h_{top}A_{top}}T_{bottom}^{*} - \frac{(3.92)}{(3.92)}$$

Initial condition:

$$\rho^* C_p^* T^*(0) = 1, \text{ and}$$

 $T_{evaporator}^*(0) = 1$
(3.93)

CHAPTER 4 EXPERIMENTAL STUDY

The main objectives of the experimental study are to investigate the feasibility of

the proposed concept and to compare the theoretical results with the experimental ones.

Experimental unit has been designed, built and tested.

Experimental Set Up

The experimental set up, schematic of which is shown in fig. 4.1, a photo of the

outdoor set up is shown in fig. 4.2 and a photo of the indoor set up is shown in fig. 4.3,

consists of the following main components:

- 1. Hot water side: A schematic of the hot water side is shown in fig. 4.4 and a photo is shown in fig. 4.5. A storage type of electric water heater is used to supply the hot water instead of a solar collector, in order to maintain steady temperature at the inlet of the evaporator. A circulating pump is used to circulate the hot water through the heat exchanger. The heat exchanger is a copper tube of 2.4 m length and an outside diameter of 1.27 cm. The heat exchanger was designed to give a temperature drop between the inlet and the outlet of about 10°C, which represents the optimum temperature rise across the flat plate solar collectors.
- 2. Evaporator side: Evaporators may be operated batch wise or continuously. Batch operation is sometime used when small amounts are needed. Its operation generally requires more energy than continuous operation. Those evaporators are usually operated such that filling, evaporation, and concentrate removal are consecutive steps. Under such operating conditions, the evaporator body must be large enough to hold the charge and the heating coil remains submerged even when the volume is reduced due to evaporation. Batch process is best for small systems, for certain industrial products that require large residence times, or for products that are difficult to handle. Semibatch method may be used, where the feed is continuously added to maintain a constant liquid level until the charge reaches final concentration. Continuous evaporators have continuous feed and discharge. The evaporator of the proposed unit could be operated in any of these modes. In building such a unit, it is to be taken into consideration that seawater is very corrosive. Therefore, the materials should be carefully selected. Titanium is generally suitable as are some of the nickel and copper/nickel alloys. Stainless steel is less suitable since it suffers from corrosion especially at high temperature.

For the purpose of this experiment, and to minimize the cost of the experimental set up, carbon steel was used for the evaporator after applying certain coating to minimize corrosion. The evaporator is a cylinder of 0.2 m^2 cross sectional area, and 0.2 m height and fixed with a truncated cone at the top. At a height of 0.16 m from its bottom a 1 cm lip is fixed to collect any condensate which might result from vapor condensation in the inside of the evaporator before reaching the condenser. The lip has a pitch sufficient enough for the water to flow to a common point, which is connected to the condensate receiver tank. Since the operating pressure is usually vacuum, the system components should be such that they can withstand vacuum conditions. The shell of the evaporator is 0.254 cm thick carbon steel and the bottom is 0.635 cm thick. The method of calculating these dimensions is given in appendix E. The evaporator has a provision for feed water, through a 1.27 cm copper tube, and a provision for withdrawing the concentrated brine, which is a 2.54 cm PVC tube. The injection and withdrawal pipes form a tube-in-tube heat exchanger. The evaporator has a provision to provide the required energy through a closed loop heat exchanger. A photo that shows the evaporator and condenser is shown in fig. 4.6.

- 3. Condenser side: The condenser is a 4 inches copper tube of 0.5 m length, 0.25 cm thickness. On its lateral surface 10 copper fins of 25.4 cm diameter and 0.0635 cm thickness are soldered 4 cm apart. The condenser was designed to dissipate the required amount of energy. The condenser is connected to the evaporator by flanges; a piece of polycarbonate is fixed between them to act as a thermal block to prevent heat transfer by conduction from the evaporator to condenser. At the other end, the condenser is connected to a condensate receiver via 1.27 cm PVC pipe.
- 4. Auxiliary components: These include tanks for supply, concentrate withdrawal and distillate receiver along with connecting pipes. Those are shown in fig. 4.7.
- 5. Supporting structure: A 32 ft high scaffold.

Measurements

Temperature, pressure and wind speed (for some experiments) were measured at

the locations shown in fig. 4.1. Temperatures were measured using T-type

thermocouples, pressure was measured using pressure transducer (model 68075), and

wind speed was measured using three-cup anemometer (model 010C wind speed sensor).

Those were connected to an Iotech data acquisition system, which recorded averaged

values with the help of DaqView software. The mass flow rate through the heat

exchanger was measured at the location shown in fig. 4.4. The mass flow rate was kept constant during each experiment.

The thermocouples, pressure transducer, and flow meter were calibrated. The method of calibration is provided in appendix F.

Uncertainty analysis of measurements was conducted to establish a confidence in the measurements. The method and results of the analysis are given in appendix F.

Experiments

The system was tested under various operating conditions, to study the effect of those conditions on the system performance. These conditions are as shown in table 4.1.

	Heat source	Depth of water	Withdrawal rate	Number of tests
	temperature (°C)	body (m)	(kg/hr)	
Outdoor tests	60	0.08	0	6
	60	0.08	0.05	1
	60	0.08	0.1	6
	60	0.08	0.5	1
	60	0.06	0.1	1
	60	0.1	0.1	1
	60	0.1	0	1
	50	0.08	0.1	1
	50	0.08	0	6
	50	0.1	0	1
	40	0.08	0.1	1
Indoor tests	60	0.06	0	1
	60	0.08	0	6
	60	0.1	0	1
	50	0.08	0	1
	40	0.08	0	1

Table 4.1 Tests conditions



Figure 4.1. Schematic of the experimental set up



Figure 4.2. Photo of the experimental set up (outdoor tests)



Figure 4.3. Photo of the experimental set up (indoor tests)



Figure 4.4. Schematic of the hot water side



Figure 4.5. Photo of the hot water side



Figure 4.6. Photo of the evaporator-condenser



Figure 4.7. Photo of the auxiliary components

CHAPTER 5 RESULTS AND DISCUSSION

This chapter presents the theoretical and experimental results and a comparison between them. The theoretical results were obtained by simulating the performance of the system with a constant temperature heat source and with a solar collector. The mathematical relations presented in chapter 3 were employed to determine the performance of the proposed system. Scaleup effect is also presented. Finally, Quality of the product is presented.

System Specifications and Reference Conditions

For all simulations the system specifications and dimensions were assumed to be the same as the one in the actual experimental system. The heat exchanger, through which the required thermal input is supplied to the saline water, is assumed to be a copper tube of 2.4 m length and 1.27 cm outside diameter. The evaporator is a cylinder of 0.2 m² cross sectional area, 0.2 m height, with a truncated cone fixed on top of it. The evaporator has a provision for feed water, through a 1.27 cm copper tube, enclosed partially by 2.54 cm CPVC pipe that is used for withdrawing the concentrated brine. The two pipes form a tube-in-tube heat exchanger. The condenser is a 10.16 cm copper tube of 0.5 m length, 0.25 cm thickness. On its lateral surface, 10 copper fins of 25.4 cm diameter and 0.0635 cm thickness are soldered 4 cm apart. The other end of the condenser is connected to a condensate receiver via 1.27 cm PVC pipe. A reference state for the saline water in the evaporator was fixed as 25 °C temperature, 3.5% solute concentration and 1021 kg/m³ density.

Results and Discussion

Constant Heat Source Temperature

An electric water heater was assumed to supply the hot water at a constant temperature during each test. For all calculations the ambient temperature was taken as 25 °C. The heat transfer fluid through the evaporator heat exchanger was water with a mass flow rate of 10 kg/hr, this mass flow rate is equivalent to an optimum mass flow rate through solar collectors which is in the range of 50-75 kg/hr.m² collector area [76], assuming that in real life the system will be supplied with its energy requirements from a solar collector of 1 m² area for each 1 m² of evaporator area.

Operating conditions were varied to study the effect of those changes on the system performance. Effect of the depth of the water body was investigated with the withdrawal rate taken as 0.1 kg/hr and the heat source temperature as 60 °C. The results are shown in figs. 5.1 and 5.2 for steady state conditions. It is clear from fig. 5.1 that the depth of water body affects mainly the time period required to get to the steady state. Once steady state is reached, the effect on the system output and saline water temperature is small may be neglected. For example, as the depth of water body varied from 0.05 to 0.1 m the system output rate at steady state conditions varied only from 0.133 to 0.129 kg/hr, and the temperature of saline water at steady state conditions varied for subsequent calculations. At this depth the system output rate at steady state was 0.131 kg/hr, the water temperature was 47.3 °C, and energy and exergy efficiencies were 80 % and 86 %, respectively. Figure 5.2 also shows that the effect of depth of water body on the system energy and exergy efficiencies, and the exergy destruction is small. As the depth of water body increased



Figure 5.1. Effect of depth of water body inside the evaporator on the system performance at steady state conditions (T_{c,out}=60 °C, Q_w=0.1 kg/hr)



Figure 5.2. Variation of energy and exergy efficiencies and exergy destruction with the depth of water body at steady state conditions (T_{c,out}=60 °C, Q_w=0.1 kg/hr)

from 0.05 to 0.1 m, the energy and exergy efficiencies decreased from 83.3 % and 89.6 % to 77.7 % and 83.6 %, respectively. The exergy destruction in the heat exchanger increased from 47.7 to 50 W and that in the condenser decreased from 66.9 to 65.1 W. The exergy destruction in the tube-in-tube heat exchanger was very small (<1 W) and is neglected.

The second parameter considered was the effect of withdrawal rate whose effect on the performance of the system is shown in fig. 5.3. The heat source temperature was assumed to be 60 °C. The energy and exergy efficiencies decreased from 80 and 86 % for a withdrawal rate of 0.05 kg/hr to about 44.7 and 48 % at a withdrawal rate of 5 kg/hr, respectively. For the same variation in the withdrawal rate, system output rate decreased from 0.131 to 0.096 kg/hr. The efficiency and output rate were almost constant in the withdrawal range of 0 to 1 kg/hr, but looking at the concentration curve -based on steady



Figure 5.3. Effect of withdrawal rate on the system performance at steady state conditions (T_{c,out}=60 °C, h_s=0.08 m)

state conditions in terms of salt concentration- which was assumed to be achieved if the solute concentration does not vary by more than 0.0001 % in 60 seconds, we see that the concentration starts increasing rapidly as the withdrawal rate is decreased from 0.5 kg/hr. At a withdrawal rate of 0.1 kg/hr, the salt concentration at the steady state conditions will be about 7.1 %, and if the withdrawal rate is reduced further, there will be a danger of scale formation. As the withdrawal rate increases above 1 kg/hr, more energy will be carried away by the concentrated brine, hence more losses and less output. A withdrawal rate of 0.1 kg/hr will be used in subsequent calculations.

Another parameter that affects the system performance is condenser thermal resistance. Its effect on the system performance is shown in figs. 5.4 and 5.5.

Values for the condenser thermal resistance were calculated by varying the condenser area, while keeping the heat source temperature constant at 60 °C, depth of water body at 0.08 m, and withdrawal rate at 0.1 kg/hr. Figure 5.4 shows that at higher thermal resistance, which corresponds to a smaller condenser area, fresh and saline water temperatures will be higher, hence more losses, as a result less output and low efficiency as clear from fig. 5.5. Also at higher thermal resistance, which will result in higher saline water temperature, the temperature difference between the saline water and heat transfer fluid is reduced, thus the heat input will decrease as shown in fig. 5.5. However, reducing that resistance below a certain value has a small effect in the system output, but requires larger condenser. For example, to reduce the thermal resistance from 0.0667 °C/W where the system output rate was about 0.131 kg/hr to 0.0208 °C/W where the system output rate was about 0.142 kg/hr, the condenser area has to be almost doubled. Figure 5.4 also shows that the pressure inside the system increases as the condenser thermal resistance

increases. A vacuum equivalent to 3.7 kPa (abs) or less can be created depending on the condenser area and the ambient temperature at which condensation will take place.



Figure 5.4. Variation of saline and fresh water temperatures and pressure inside the system with the condenser thermal resistance at steady state conditions $(T_{c.out}=60 \text{ }^{\circ}\text{C}, Q_w=0.1 \text{ kg/hr}, h_s=0.08 \text{ m})$



Figure 5.5. Effect of the condenser thermal resistance on the system performance at steady state conditions (T_{c,out}=60 °C, Q_w=0.1 kg/hr, h_s=0.08 m)

The last parameter considered was the temperature of the heat source, which was investigated over a temperature range of 40-100 °C. As expected, increasing the temperature of the heat source increases the saline water and fresh water temperatures, as can be seen from fig. 5.6. When the heat source temperature increased from 40 to 100 °C, the steady state temperature of the saline water increased from 34.8 to 66.8 °C, and that of fresh water increased from 27.2 to 40 °C. Not only the temperature in both chambers increased, but also the temperature difference between them, from 7.6 °C to 26.8 °C in this case, and as the temperature difference; hence the system performance will be improved.



Figure 5.6. Effect of heat source temperature on the system performance at steady state conditions (Q_w=0.1 kg/hr, h_s=0.08 m)

For the same increase in the temperature as above, the output rate increases from 0.041 to 0.396 kg/hr (fig. 5.7). The energy and exergy efficiencies were defined in eqs. 3.76 and 3.80, respectively. For convenience these equations are reproduced here,

$$\eta_t = \frac{m_f \cdot h_{fg}}{\dot{Q}_u} \tag{5.1}$$

$$\eta_{2} = \frac{m_{f} h_{fg}}{m_{c} (e_{c,out} - e_{c,in})}$$
(5.2)

The specific exergy, *e*, is given as (neglecting kinetic and potential energy),

$$e = (h - h_0) - T_0(s - s_0)$$
(5.3)

As the heat source temperature increases from 40 to 100 °C, the energy and exergy efficiencies increase from 61.8 and 66.7 % to 90.7 and 97.2 %, respectively.

Figure 5.7 shows the effect of heat source temperature on the input and output rate of the system, where both show the same trend, that is, increase with the increase of the heat source temperature, i.e. as the temperature of the heat source increases the amount of energy extracted from the heat transfer fluid also increases.



Figure 5.7. Effect of the heat source temperature on the system output at steady state conditions (Q_w=0.1 kg/hr, h_s=0.08 m)

The previous results were for the system under steady state conditions. The behavior of the system during transient operation is shown in figs. 5.8 and 5.9. The variation of saline and fresh water temperatures, operating pressure and energy and exergy efficiencies with time during transient operation is shown in fig. 5.8. Steady state conditions were assumed to be achieved if the saline water temperature does not vary more that 0.01 °C over a time period of 60 seconds. With a heat source temperature of 60 °C, steady state conditions were achieved in about 5 hours, when the saline water temperature water temperature was about 47.3 °C, fresh water temperature about 31 °C, the system operating pressure about 4.7 kPa (abs), and energy and exergy efficiencies were 80 and 86 %, respectively.



Figure 5.8. Variation of saline and fresh water temperatures, system pressure, and system efficiencies with time (T_{c,out}=60 °C, Q_w=0.1 kg/hr, h_s=0.08 m)

Variation of heat input, fresh water output, and solute concentration with time during the transient operation is shown in fig. 5.9. The heat input started from a high value at the beginning and decreased with time until steady state conditions were reached. The high amount of heat input at the beginning of the operation was used mainly to raise the temperature of the saline water in the evaporator, and a small part is used to evaporate the water. As the system reached steady state conditions the heat supplied was used to evaporate the water, compensate for the heat losses, and preheat the feed. The output rate increased until it reached a steady state value of about 0.131 kg/hr. The accumulated output during the transient period was about 0.416 kg. As shown in the figure, solute concentration would still be increasing by the time steady state conditions, based on saline water temperature, were reached.



Figure 5.9. Variation of energy input, fresh water output, and solute concentration with time (T_{c,out}=60 °C, Q_w=0.1 kg/hr, h_s=0.08 m)

Figure 5.10 shows the variation of exergy efficiency and destruction with time. The heat exchanger exergy destruction started from high value and decreased as the system headed towards steady state, when that value was 49 W. This was due to the fact that the water temperature inside the evaporator was low at the beginning, thus the heat transfer fluid experienced high temperature drop between the inlet and the outlet, as a result high

value of exergy destruction. The condenser exergy destruction behaved just the opposite, started from low value and increased as the system headed towards steady state, when that value was 65.6 W, this was due to the fact that the output rate was maximum under steady state conditions, and the latent heat of condensation of the produced vapor was dissipated to the environment via the condenser.



Figure 5.10. Variation of exergy efficiency and exergy destruction with time (T_{c,out}=60 °C, Q_w=0.1 kg/hr, h_s=0.08 m)

System Simulation with a Solar Collector

A solar collector with an efficiency given by

$$\eta = 0.72 - \frac{5.6\Delta T}{I_c}$$
(5.4)

operating in a thermosyphon mode was used. A collector area equal to the evaporator cross sectional area of 0.2 m^2 was assumed. For all calculations the ambient temperature was taken as 25 °C. The solar radiation values were taken for a clear day in June for

Gainesville, Florida (latitude 29.68 °N, longitude 82.27 °W) for a south-facing collector tilted at an angle equal to the latitude.

The amount of incident solar radiation on the collector and the amount of useful heat supplied by the collector to the system is shown in fig. 5.11. This is shown for different depths of water body (h_s) inside the evaporator. The smaller the water depth, the lower the amount of useful energy gain, this is because the heat capacity of the water inside the evaporator will be less at smaller depth.





Effect of some parameters studied with constant heat source temperature simulations were also studied here. Those parameters are: depth of water body, withdrawal rate, and fresh water temperature. Figure 5.12 shows the variation of the saline water temperature inside the evaporator and the collector outlet temperature for different depths of water body with time. The withdrawal rate was taken as 0.1 kg/hr. The smaller the depths of water body the higher the temperature during the peak solar radiation, as it is clear from the figure. At a water depth of 0.04 m, the maximum collector outlet temperature reached about 59.5 °C, and the water temperature was about 50.7 °C. The corresponding values at a water depth of 0.1 m were 55 and 47.1 °C, respectively. The peak temperature is reached faster at a smaller depth of water.



Figure 5.12. Variation of saline water and collector outlet temperatures with time $(Q_w=0.1 \text{ kg/hr})$

The variation of system output with time for different amounts of water inside the evaporator is shown in fig. 5.13. The highest output is for the minimum amount of water. The maximum output rate at a water depth of 0.04 m was about 0.158 kg/hr, which reduced to about 0.123 kg/hr as water depth increased to 0.1 m. The accumulated output (during day time only) for a water depth of 0.04 m was about 1.19 kg, which decreased to

about 0.89 kg at a water depth of 0.1 m. However, the total daily output decreased from 1.37 to 1.19 kg, as shown in fig. 5.14, which gives the total daily output as a function of



Figure 5.13. System output at different depths of water with time (Q_w=0.1 kg/hr)



Figure 5.14. System daily output as a function of depth of water body (Q_w=0.1 kg/hr)

the depth of the water body inside the evaporator. The output split between day and night is also shown in fig. 5.14. The night time output is the result of the energy being stored in the system during the initial hours of operation. As the night time output is function of the energy stored, the higher the depth of water body, the higher the night time output, as clear from the figure. For example, as the water depth increases from 0.04 to 0.1 m, the night output increases from 0.18 to 0.3 kg, respectively, but the day time output decreases from 1.19 to 0.89 kg, respectively.

The collector and system daily efficiencies as a function of the depth of the water body are shown in fig. 5.15. The collector energy and exergy efficiencies increases slightly with the increase of the depth of the water body, whereas the system efficiencies deceases more rapidly. As the depth of water increases from 0.04 to 0.1 m, the collector energy and exergy efficiencies increase from 56.5 and 51.5 to 58.7 and 53.4 %,



Figure 5.15. Energy and exergy efficiencies for the collector and the system as a function of the depth of the water body (Q_w=0.1 kg/hr)

respectively, whereas the system energy and exergy efficiencies decreased from 85.5 and 93.7 % to 71.7 and 78.7 %.

The second parameter considered is the withdrawal rate. Figure 5.16 shows the variation of the temperature inside the evaporator and the collector outlet temperature with time for different withdrawal rates. Depth of water body was 0.08 m. The highest temperature corresponds to the lowest withdrawal rate. However, this difference is small if the withdrawal rate is less than 1 kg/hr.



Figure 5.16. Variation of saline water and collector outlet temperatures with time (h_s=0.1 m)

Effect of withdrawal rate on the system output is shown in fig. 5.17. As the withdrawal rate increased, the system output decreased, because the withdrawn water carried an amount of heat with it as it left the evaporator. The maximum output rate at a

withdrawal rate of 0.05 kg/hr was about 0.134 kg/hr; this amount remained almost constant as the withdrawal rate increased to 0.1 kg/hr. The accumulated output at this withdrawal rate was about 0.983 kg. As the withdrawal rate increased to 2 kg/hr, the output rate deceased to 0.119 kg/hr and the accumulated output decreased to about 0.879 kg.





Figure 5.18 shows the total daily system output as a function of withdrawal rate. The output is shown split between day and night time. Both day and night time outputs decreased as the withdrawal rate increased. The day and night time output decreased from 0.983 and 0.273 kg at withdrawal rate of 0.1 kg/hr to 0.879 and 0.238 kg at a withdrawal rate 2 kg/hr, respectively. This is because more withdrawal rate means more heat carried away from the system, which is a loss, hence reduces the output.





The collector and system daily efficiencies as a function of withdrawal rate are shown in fig. 5.19. The collector energy and exergy efficiency increased very slightly with the increase of the withdrawal rate. As the withdrawal rate increased from 0.05 to 5 kg/hr, collector energy and exergy efficiencies increased from 58 and 52.8 to 58.7 and 53.5 %, respectively. However, the system energy and exergy efficiencies decreased from 76.6 and 84.1 to 67.3 and 73.8 %, respectively, for the same increase in withdrawal rate. It is clear from the figure that the system efficiencies decreased very slightly as the withdrawal rate increased to about 1 kg/hr.



Figure 5.19. Energy and exergy efficiencies for the collector and the system as a function of withdrawal rate (h_s=0.1 m)

The last parameter considered here is the effect of condenser thermal resistance, which is shown in fig. 5.20. The thermal resistance varies with the condenser area. As the area increases, thermal resistance decreases, heat can be dissipated rapidly and efficiently, which reduces the fresh water temperature, which in turn increases the vapor pressure difference between saline and fresh water sides. Since the vapor pressure difference is the driving force for mass transfer, the output increases as this driving force increases. For example, as the number of condenser fins, N, were increased from 10 to 20, the maximum output rate and the daily accumulated output increased from 0.134 kg/hr and 0.983 kg to 0.142 kg/hr and 1.052 kg, respectively. However, a compromise must be reached between the increase in the output and the increase in the condenser cost.



Time (min) Figure 5.20. Effect of condenser thermal resistance on the system performance (h_s =0.1 m, Q_w =0.1 kg/hr)

Figure 5.21 shows the exergy destruction in the evaporator heat exchanger and



Figure 5.21. Variation of exergy destruction with time (h_s=0.1 m, Q_w=0.1 kg/hr)

condenser. The heat exchanger exergy destruction values increase as the collector outlet temperature increases. The condenser exhibits the same behavior, as the collector outlet temperature increases, the system output increases, and as the vapor condenses inside the condenser releasing its latent heat of condensation to the environment, the exergy destruction in the condenser increases.

Figure 5.22 shows the variation of pressure and concentration of saline water with time. The pressure reached a maximum value of about 5 kPa (abs). The concentration continued to increase till the end of the test day. Considering the value at the end of a day as the initial concentration for the next day, a steady state value of 6.75 % would be reached after 46 days, as shown in fig. 5.23. The steady state condition was assumed to be achieved if the increase in the concentration for a day was less than 0.01 %. Also shown in the figure is the total daily output, which decreased as the concentration



Figure 5.22. Variation of concentration and pressure with time ($h_s=0.1$ m, $Q_w=0.1$ kg/hr)

increased. This result agrees with the result of Keren et al. [63], who found that the evaporation rate decreases by about 1 % for each 1 % increase in the salinity.



Figure 5.23. Variation of concentration and accumulated output with time (h_s =0.1 m, Q_w =0.1 kg/hr)

Experimental Results

A small scale system with specifications as described in chapter 4 was tested.

Outdoor and indoor tests were performed. Some tests were repeated six times to ensure

reproducibility of the results and to establish the error bounds experimentally.

Uncertainty analysis was conducted; the obtained uncertainty limits are given in table 5.1.

Details of the uncertainty analysis are given in appendix G.

Quantity	Uncertainty
Flow rate	5.3 %
Output	6.8 %
Heat input	5.25 %
Efficiency	8.4 %

Table 5.1. Uncertainty limits

Outdoor tests results

An experimental set up was constructed, where the system was placed at the required height, about 10 m, to create a vacuum naturally, see figs. 4.1 and 4.2. A number of tests were performed, with various combinations of operating conditions shown in table 4.1. For convenience the table is reproduced here.

	Heat source	Depth of water	Withdrawal rate	Number of tests
	temperature (°C)	body (m)	(kg/hr)	
Outdoor tests	60	0.08	0	6
	60	0.08	0.05	1
	60	0.08	0.1	6
	60	0.08	0.5	1
	60	0.06	0.1	1
	60	0.1	0.1	1
	60	0.1	0	1
	50	0.08	0.1	1
	50	0.08	0	6
	50	0.1	0	1
	40	0.08	0.1	1
Indoor tests	60	0.06	0	1
	60	0.08	0	6
	60	0.1	0	1
	50	0.08	0	1
	40	0.08	0	1

The results obtained are presented below. The first set of tests were performed under the following operating conditions: heat source temperature was 60 °C, the withdrawal rate 0.1 kg/hr, and depth of water body 0.08 m. Figure 5.24 shows the temperature profiles of the saline water as a function of time. Also included in the figure are the ambient temperature profiles during those tests. As can be seen from the figure, all temperature profiles have the same trend, and at steady state conditions the difference between the maximum and the minimum saline water temperatures is 2.1 °C, or about 4 %. This difference can be attributed to the uncertainty of measurements and the variation in the ambient temperature during those tests.



Figure 5.24. Saline water and ambient temperatures for six tests (Tc,out=60 °C, Q_w=0.1 kg/hr, and h_s=0.08 m), (each color represents one test day)

The outputs during those six tests are shown in fig. 5.25. If the first day, which has



Figure 5.25. Theoretical and experimental output for six tests (Tc,out=60 °C, Q_w=0.1 kg/hr, and h_s=0.08 m)

significantly higher output than the other days due to low ambient temperature, is excluded, the difference between the maximum output (0.823 kg) and the minimum output (0.739 kg) is about 0.084 kg, or about 10 %, which is higher than the uncertainty of measurements. Those six tests were conducted under the same experimental conditions except for the ambient temperature, which was different for each day. This explains the difference between the results from various tests. The ambient temperature affects the system output significantly, the lower the ambient temperature, the lower the fresh water temperature, and the higher the output, and vice versa. Also shown in the figure are the outputs obtained from theoretical simulation, where the measured temperature profiles of the heat source fluid and the ambient were used with the computer program to predict the system performance. The experimental and theoretical results agree very well. The maximum difference was about 0.049 kg, or about 5 %, which is within the uncertainty of the measurements. It is to be mentioned here that the experimental values were always less than the theoretical ones. This may be attributed partly to the fact that the theoretical model assumes that all molecules evaporated from the saline water inside the evaporator will reach the condenser and condense as liquid water, whereas in real life a number of those molecules might fall back to the pool. Also the model assumes that the fins are an integral part of the condenser, whereas those were soldered to the condenser surface, this might reduce the rate of heat transfer from the condenser, hence the system output will be reduced. Another reason might be that the model assumes that the heat loss from the system is by natural convection, whereas wind may increase that. However, wind speed measurements were made and the obtained wind speed profiles were used in the computer program (after modifying to account for forced convection) and that effect was

found to be small, as will be shown later. Both experimental and theoretical outputs are broken down into day time output (time during which heat is supplied to the system) and night time output (time after the heat is no longer supplied to the system), as shown in fig. 5.25. The night time output is a result of the heat being stored in the system during the initial hours of operation, and varies slightly for different tests, depending on the saline water temperature at the end of the test and the ambient temperature. The maximum and minimum experimental night time outputs (excluding the first day) were 0.319 and 0.290 kg, respectively. The saline water temperatures at the end of these two days were 49.66 and 49.4 °C, respectively, but the average ambient temperatures were 27 and 30 °C, respectively.

Additional results from the previous six tests are presented in figs. 5.26-5.28. Figure 5.26 shows how the saline water temperature and the heat input vary with time



Figure 5.26. Experimental and theoretical saline water temperature and heat input with time (Tc,out=60 °C, Q_w=0.1 kg/hr, and h_s=0.08 m)

both experimentally and theoretically. Also shown in the figure are the heat source and ambient temperatures. The saline water temperature increased with time and reached a steady state value of about 50 and 48 °C, for experimental and theoretical results, respectively. The higher value obtained from the experiments may be due to the fact that the temperature was measured at a distance of 10 cm from the evaporator wall, where the temperature may be slightly higher than that near the wall, where the heat loss takes place. In the theoretical model, the saline water temperature was assumed to be uniform throughout the evaporator. The heat input to the system started from a high value, where the system began to heat up and a major part of the energy was used to raise the temperature of the saline water and the evaporator material, i.e. stored as a sensible heat, which would be utilized during the night time, as was discussed earlier. As the system reached steady state, the experimental and theoretical values of the energy input were about 109 and 103 W, respectively. The difference is about 5 %, which is within the uncertainty of measurements.

Figure 5.27 shows the variation of the theoretical output, output rate, and the energy and exergy efficiencies with time, based on measured values of the temperature profiles of the heat source fluid and the ambient. The accumulated daily output during six hours test reached a value of 0.495 kg compared to and 0.462 kg obtained experimentally. The difference is about 6.6 %, which is within the uncertainty of measurements. At steady state the output rate was 0.115 kg/hr compared to 0.108 kg/hr obtained experimentally, the difference is about 6 %, which is within the uncertainty of measurements. The energy and exergy efficiencies started from very low values at the beginning of the test day and increased with time. When the system reached steady state
conditions the energy and exergy efficiencies reached values of 74 and 79 %, respectively.



Figure 5.27. Variation of the output, output rate, and energy and exergy efficiencies with time (Tc,out=60 °C, Q_w=0.1 kg/hr, and h_s=0.08 m)

Exergy destruction in the system components, evaporator heat exchanger, condenser, and tube-in-tube heat exchanger was analyzed. Exergy destruction in the evaporator heat exchanger and the condenser is shown in fig. 5.28. The exergy destruction in the evaporator heat exchanger started from a high value, about 150 W, and decreased with time, as the system reached steady state, with the lowest value being about 40 W. The trend for exergy destruction in the condenser was the opposite of that of the heat exchanger. It started from very low value and increased with time, reaching a maximum value of about 44 W at steady state, since the output at the beginning was very small and increased with time. The exergy destruction in the condenser could be minimized if the latent heat of condensation of the produced vapor was utilized, as in a multistage evaporator. However, in the present system it was dissipated to the environment. Exergy destruction in the tube-in-tube heat exchanger was very small (<<1 W) and may be neglected.



Figure 5.28. Variation of exergy destruction with time (Tc,out=60 °C, Q_w =0.1 kg/hr, and h_s =0.08 m)

The following five figures (5.29-5.33) present results similar to those in the previous five figures (5.24-5.28), however, the system was operated under batch process conditions, i.e. no withdrawal or injection during the test. All trends are the same for the two sets of operating conditions. Figure 5.29 shows the temperature profiles obtained for the saline water and ambient as a function of time. The maximum and minimum saline water temperatures were 50.7 and 46.9 °C, respectively. A difference of 3.8 °C, was mainly due to uncertainty of measurements and the variation in the ambient conditions.



Figure 5.29. Saline water and ambient temperatures for six tests (Tc,out=60 °C, Q_w=0 kg/hr, and h_s=0.08 m), (each color represents one test day)

The experimental and theoretical outputs and their breakdown between day and night is shown in fig. 5.30. A good agreement between the theoretical and experimental



Figure 5.30. Theoretical and experimental output for six tests (Tc,out=60 °C, Q_w=0 kg/hr, and h_s=0.08 m)

results can be seen. The maximum difference for a single day was 0.057 kg, as shown in the figure. The figure also shows that the output was the highest for that day. This was due to the low ambient temperature for that day. The average total daily output for the other five days was about 0.755 kg.

The results for a single day show a good agreement between the experimental and theoretical results. Under steady state conditions, the difference between the experimental and theoretical values of saline water temperature was about 2.5 °C, and the difference in the heat input was about 4 W or about 3.5 %, as shown in fig. 5.31.



Figure 5.31. Variation of the experimental and theoretical saline water temperature and heat input with time (Tc,out=60 °C, Q_w=0 kg/hr, and h_s=0.08 m)

Figure 5.32 shows the theoretical results for the output, output rate, and energy and exergy efficiencies. The accumulated output during the day was 0.494 kg compared to 0.457 kg obtained experimentally. The output rate was 0.124 kg/hr compared to 0.119 kg/hr obtained experimentally, the difference is about 4 %. The energy and exergy

efficiencies increased with time, at steady state conditions their values were of 75 and 80 %, respectively.



Figure 5.32. Variation of output, output rate, and energy and exergy efficiencies with time (Tc,out=60 °C, Q_w=0 kg/hr, and h_s=0.08 m)

The exergy destruction for the evaporator heat exchanger and condenser started from 200 and 1.6 W and ended at 44.7 and 52.2 W, respectively, which is shown in fig. 5.33. It is to be mentioned here that there were no significant differences between the results of the previous two sets of tests. This agrees well with the simulation results, which showed that the effect of withdrawal rate on the system performance is small, if the withdrawal rate is below 1 kg/hr.

Another set of operating conditions were selected and the test was repeated six times, those conditions were: heat source temperature 50 °C, batch process, and depth of water body 0.08 m. Figure 5.34 shows the variation of saline water temperature with time for those tests. All temperature profiles obtained have the same trend. The maximum

temperature difference was about 3 °C. The average saline water temperature at steady state was about 44 °C, compared to 49 °C when the heat source temperature was 60 °C.



Figure 5.33. Variation of exergy destruction with time (Tc,out=60 °C, Q_w =0 kg/hr, and h_s =0.08 m)



Figure 5.34. Saline water and ambient temperatures for six tests (Tc,out=50 °C, Q_w=0 kg/hr, and h_s=0.08 m), (each color represents one test day)

The outputs from these tests are shown in fig. 5.35. Again a good agreement between the experimental and theoretical results is observed. Differences were within the uncertainty of measurements. It is to be noted that reducing the temperature of the heat source fluid by about 10 °C, reduced the system output significantly, from about 0.750 to about 0.5 kg for the heat source temperature of about 60 and 50 °C, respectively. This is due to the reduction in the vapor pressure (which varies exponentially with temperature) difference between the evaporator and condenser.



Figure 5.35. Theoretical and experimental output for six tests (Tc,out=50 °C, Q_w =0 kg/hr, and h_s =0.08 m)

The experimental and theoretical saline water temperature and heat input for a day are shown in fig. 5.36. As the system reached steady state conditions, the saline water

temperature reached a value of about 44 °C from the experimental measurements. The theoretical model predicted a value that is 1 °C lower. The corresponding values when the heat source temperature was 60 °C were 50.6 and 48 °C. It is to be mentioned that these two test days (one with the heat source temperature of 60 °C and the other with 50 °C) had a comparable ambient conditions. The heat input reached values of about 79 and 75 W, from experimental and theoretical results, respectively. The corresponding values when the heat source temperature was 60 °C were 113 and 109 W, respectively.



Figure 5.36. Variation of the experimental and theoretical saline water temperature and heat input with time (Tc,out=50 °C, Q_w=0 kg/hr, and h_s=0.08 m)

Figure 5.37 shows the output, output rate, and energy and exergy efficiencies. The accumulated daily outputs (for six hours test) were 0.303 and 0.311 kg and the output rates were 0.078 and 0.08 kg/hr, for the experimental and theoretical results, respectively. By the end of the test the energy and exergy efficiencies were about 72 and 75 %, compared to about 75 and 80 % when the heat source temperature was about 60 °C.



Figure 5.37. Variation of output, output rate, and energy and exergy efficiencies with time (Tc,out=50 °C, Q_w=0 kg/hr, and h_s=0.08 m)

Exergy destruction, shown in fig. 5.38, in the evaporator heat exchanger started from a high value and decreased to about 29 W by the end of the test day, as compared to



Figure 5.38. Variation of exergy destruction with time (Tc,out=50 °C, Q_w =0 kg/hr, and h_s =0.08 m)

a value of about 45 W when the heat source temperature was about 60 °C. Exergy destruction in the condenser increased till it reached a value of about 30 W, compared to 52 W when the heat source temperature was about 60 °C.

As can be seen, for the case with a heat source temperature of 50 °C, the input, output , energy and exergy efficiencies, and exergy destruction were all less than those for the heat source temperature of 60 °C.

The effect of depth of water body on the system performance is shown in fig. 5.39. Three different depths were considered: 0.06, 0.08, and 0.1 m. The heat source temperature was set to 60 °C and the withdrawal rate was 0.1 kg/hr. This effect should not be significant, but due to the variation in the ambient conditions, there are some differences at various depths. Although the saline water temperature for the test day



Figure 5.39. Effect of depth of water body on the system performance (outdoor tests)

corresponding to a water depth of 0.1 m was the lowest, about 47 °C, compared to about 50 °C for the other two days. That day had the highest output of about 0.908 kg, and the output rate of about 0.124 kg/hr under steady state conditions, compared to an output of about 0.750 kg and an output rate of about 0.118 kg/hr for the other two test days. This is because the average ambient temperature for that day was 21 °C compared to about 27 °C for the other two days. However, the agreement between the experimental and theoretical results for each depth is a very good. Since the theoretical model simulates the system performance well, if we consider various depths under the same ambient conditions, a small difference in the system output will be observed, as was shown in fig. 5.1, and as seen from indoor tests (discussed later).

The effect of withdrawal rate is shown in fig. 5.40. Three different withdrawal rates



Withdrawal Rate (kg/hr)

Figure 5.40. Effect of withdrawal rate on the system performance (outdoor tests)

were considered: 0, 0.1, and 0.5 kg/hr. The heat source temperature was 60 °C and the depth of water body was 0.08 m. The ambient conditions during those tests were comparable, that is why the expected trends were obtained. The effect in the range of the withdrawal rates considered was very small. The output and output rates decreased slightly with the increase in the withdrawal rate, from 0.763 kg and 0.119 kg/hr, respectively, under batch process operation to 0.755 kg and 0.116 kg/hr at a withdrawal rate of 0.5 kg/hr.

The effect of heat source temperature is shown in fig. 5.41. Three different temperatures were considered, 40, 50, and 60 °C. The withdrawal rate was 0.1 kg/hr, and



Figure 5.41. Effect of heat source temperature on the system performance (outdoor tests) the depth of water body was 0.08 m. Although the ambient conditions were different for the different tests, the effect of the heat source temperature is significant. The output, output rate, and saline water temperature all increased as the heat source temperature

increased. As the heat source temperature increased from 40 to 50 to 60 °C, the output increased from 0.275 to 0.517 to 0.752 kg, respectively, the output rate increased from 0.04 to 0.069 to 0.108 kg/hr, respectively, and the saline water temperature at steady state increased from 39.3 to 45.4 to 50.3 °C, respectively. All numbers in the above discussion were experimental results. The theoretical results agree very well with the experimental results, as seen from the figure.

As mentioned earlier, wind speed measurements were made, and the heat loss due to forced convection was considered. Forced convection means more losses, hence lower saline water temperature and output. However, this effect appeared to be very small, since the system was very well insulated. Figure 5.42 shows the experimental and



Figure 5.42. Experimental and theoretical saline water temperature profile

theoretical saline water temperature profiles for a single test. The theoretical profiles were obtained for two cases: in the first one, heat loss from the system was assumed to be due to natural convection and in the second case forced convection was assumed. As can be seen from the figure the agreement between the experimental and theoretical results was good, and the difference between the two theoretical cases is small (<1 °C).

Additional tests were conducted under different operating conditions, to investigate the effect of wind speed on the simulation results. The outputs obtained from those tests are shown in fig. 5.43. As clear from the figure, the output obtained from theoretical



Figure 5.43. Experimental and theoretical outputs for different tests.

simulations was lower in the forced convection case than in the natural convection one. However, this difference was small and a comparison of either case with the experimental results shows a good agreement.

The variation of the system pressure and fresh water temperature with time is shown in fig. 5.44. The ambient temperature has a strong impact on those values. For experimental results the system pressure increased from about 2.3 to 5.6 kPa (abs) by the end of the test. The theoretical values were lower than the experimental ones, and at steady state conditions, the system pressure was about 5 kPa (abs). This difference appears to be large, but knowing that the pressure varies exponentially with temperature, if the predicted temperature is 2 °C less than the actual one, this would result in such a



Figure 5.44. Variation of pressure and fresh water temperature with time (Tc,out=40 °C, Q_w =0.1 kg/hr, and h_s=0.08 m)

difference in the pressure values. Here the experimental and theoretical fresh water temperatures were 34.4 and 32.3 °C, respectively.

The pressure inside the system is due to the vapor pressure of water and noncondensable gases. To study how the noncondensable gases evolve during the tests, the amount of dissolved oxygen (DO) in saline water was measured using dissolved oxygen meter (model 51A). A test was conducted to see how the dissolved oxygen content in the water changes in the evaporator with time under vacuum conditions. Water samples were taken from the evaporator under vacuum conditions each hour and analyzed for DO. The results are shown in fig. 5.45.





The above figure shows that about 75% of the oxygen evolved during the six hours, when there was no heating. The temperature also affects this amount. The measurements made before injecting the water into the evaporator for a specific day gave DO as 7.7

ppm, while at the end of the test day (after six hours) that value was 0.95 ppm, i.e. around 88% of the DO was evolved with heating.

Indoor tests results

In the results presented previously for the outdoor tests, significant differences were sometimes observed, due to the fact that the ambient conditions were different for those tests. In order to ensure the reproducibility of results with minimum variation, indoor tests were conducted under controlled ambient conditions. The experimental set up for these tests was shown in fig.4.3. Unlike the outdoor tests where vacuum was created by natural means, a vacuum pump was used to create vacuum for the indoor tests. All tests were performed under batch process operation. A set of six tests was performed under the following operating conditions: heat source temperature of 60 °C and depth of water body as 0.08 m. The saline water temperature profiles for these tests are shown in fig. 5.46. The agreement among the results of all the tests was very good. Under steady state conditions the maximum and minimum saline water temperatures obtained were 51.1 and 50.6 °C, respectively, a difference of only 0.5 °C.

The outputs obtained during those tests are shown in fig. 5.47. Again a very close agreement was obtained between the theoretical and experimental results, and among the experimental results themselves. The difference between the maximum and minimum total daily outputs obtained experimentally was 0.0014 kg, which is about 1.5 %. The maximum difference between theoretical and experimental values for a day was about 0.0024 kg, which is about 2.5 %. The average total daily output was about 0.936 kg of which 0.591 kg was obtained during the day time and 0.345 kg was obtained during the night time. The corresponding theoretical values were: average total daily output 0.954



Figure 5.46. Saline water temperature and surrounding temperatures for six tests (indoor tests, Tc,out=60 °C, Q_w=0 kg/hr, and h_s=0.08 m), (each color represents one test day)



Figure 5.47. Theoretical and experimental output for six tests (indoor tests, Tc,out=60 °C, Q_w =0 kg/hr, and h_s=0.08 m)

kg, average day time output 0.601 kg, and night time output 0.353 kg. The difference between the experimental and the corresponding theoretical values are about 1.9, 1.7, and 2.3 %, respectively, all are within the uncertainty of measurements.

A single day was selected to discuss the results. The experimental and theoretical saline water temperatures and heat input are shown in fig. 5.48. By the end of the test the experimental and theoretical saline water temperatures and heat input were 50.5 and 48.6



Figure 5.48. Variation of experimental and theoretical saline water temperature and heat input with time (indoor tests, Tc,out=60 °C, Q_w=0 kg/hr, and h_s=0.08 m)

°C, and 113 and 112 W, respectively. The energy and exergy efficiencies, output, and output rate are shown in fig. 5.49. The accumulated output during the day was 0.607 kg, and at steady state the output rate was 0.138 kg/hr. Energy and exergy efficiencies were 81.8 and 87.7 %, respectively.



Figure 5.49. Variation of the output, output rate, and energy and exergy efficiencies with time (indoor tests, Tc,out=60 °C, Q_w=0 kg/hr, and h_s=0.08 m)

Variation of exergy destruction with time is shown in fig. 5.50. The evaporator heat exchanger exergy destruction started from a high value of about 155 W, and decreased



Figure 5.50. Variation of exergy destruction with time (indoor tests, Tc,out=60 °C, Q_w=0 kg/hr, and h_s=0.08 m)

with time till it reached a value of 47 W at the end of the test day. Condenser exergy destruction increased with time till it reached a value of about 64 W by the end of the test day.

Effect of the depth of water body on the system performance is shown in fig. 5.51. Three different depths were considered: 0.06, 0.08, and 0.1 m. The total daily output decreased from 0.970 to 0.907 kg as the depth of water increased from 0.06 to 0.1 m. The output rate also decreased from 0.142 to 0.135 kg for the same increase in depth. Finally, shown in the figure, the saline water temperature decreased from 50.8 to 49.6 °C for the same increase in depth. The numbers in the above discussion are from experimental observations for indoor tests for controlled ambient conditions. However, the agreement between theoretical and experimental values was very good, the output rates almost coincided.



Figure 5.51. Effect of depth of water body on the system performance (indoor tests)

Figure 5.52 shows the effect of heat source temperature for three different temperatures: 40, 50, and 60 °C. As the heat source temperature increased from about 40 to 60 °C, the total daily output increased from 0.451 to 0.936 kg, the output rate increased from 0.053 to 0.138 kg/hr, and saline water temperature increased from 39 to 50.6 °C. All numbers above are from experimental observations, which agree very well with the theoretical ones.



Figure 5.52. Effect of heat source temperature on the system performance (indoor tests)

Scaling Effect

To generalize the analysis, the basic equations were written in dimensionless form. This allows us to predict the performance of a system having dimensions other than those of the model. The various dimensionless parameters for the model were obtained; some of those are shown in the following two figures. Figure 5.53 shows the saline water temperature, power input, and fresh water output in dimensionless form. Also fig. 5.54 shows the reference time of the system as a function of dimensionless time.



Figure 5.53. Saline water temperature, power input, and fresh water output as a function of time, in dimensionless form





If we consider a case where the desired output twice that from the present system, i.e. the system size must be increased by scaleup ratio (the ratio of prototype production

rate to model production rate) of two. For the larger system all dimensionless variables have to remain constant. The reference conditions, reference time, and depth of water body are assumed to be the same for the model and the prototype. For the model operating under the following conditions: heat source temperature 60 °C, withdrawal rate 0.1 kg/hr, and depth of water body 0.08 m, when the model reaches steady state, the following dimensionless variables (defined in chapter 3) are obtained:

$$\dot{V}_{in}^{*} = 12.2243, \\ \dot{V}_{i}^{*} = 0.2787, \\ \dot{V}_{w}^{*} = 0.1223, \\ \dot{V}_{e}^{*} = 0.161 \\ \rho^{*} = 0.9898, \\ \rho_{in}^{*} = 0.9628, \\ \rho_{i}^{*} = 0.9551, \\ \rho_{e}^{*} = 0.9707, \\ \rho_{evaporator}^{*} = 7.6925 \\ C_{p}^{*} = 0.9991, \\ C_{p,in}^{*} = 1.0277, \\ C_{p,i}^{*} = 1, \\ C_{p,evaporator}^{*} = 0.1066 \\ T^{*} = 1.8927, \\ T_{in}^{*} = 0.3722, \\ T_{i}^{*} = 1.3892, \\ T_{evaporator}^{*} = 1.8927, \\ T_{iop}^{*} = 0.236, \\ T_{side}^{*} = 0.196, \\ T_{bottom}^{*} = 0.284 \\ C^{*} = 1.035, \\ V^{*} = 3.725, \\ t^{*} = 0.2413 \\ \end{array}$$

The dimensionless evaporation rate is given as,

$$\mathbf{\dot{V}}_{e}^{*} = \frac{\mathbf{\dot{V}}_{e}}{\left(\frac{V}{/t_{R}}\right)}$$
(5.5)

Since the dimensionless evaporation rate, depth of water body, and reference time are to remain constant, eq. 5.5 above can be satisfied if the evaporator cross sectional area is increased by a factor of two. Since the cross sectional area is to be increased two times, to satisfy the other dimensionless parameters: the injection flow rate, withdrawal flow rate, and flow rate through the evaporator heat exchanger are to be increased by a factor of two accordingly. From the values of dimensionless temperatures, various temperature values can be found; those values place restrictions on the size of the evaporator heat exchanger, condenser size and the tube-in-tube heat exchanger.

The simulation program can be used to calculate the output for the above mentioned conditions and other similar cases and to study the effect of scaleup ratio. When scaling up the evaporator cross sectional area, the evaporator diameter has to be increased, which affects the condenser diameter. Increasing the area of the condenser requires keeping the new diameter/length ratio the same as the model. Another restriction here relates to the ratio of the condenser diameter to evaporator diameter. As mentioned previously in chapter 3, it is often the practice to maintain this ratio larger than 0.1. In the model this ratio is 0.2 and the same ratio is to be maintained for the larger system. Condenser outside surface area is sized accordingly, by selecting the number of fins and their diameter, keeping the ratio of the fin diameter to the condenser diameter the same in each case. Heat transfer area of the evaporator heat exchanger is scaled up by the same scaleup ratio. Here two cases were considered; first increasing the area while maintaining the diameter/length ratio, and the second just increasing the area by using a heat exchanger of the same diameter but different length. For example, increasing the heat exchanger area by a factor of two, the new heat exchanger will consist of two coils each of 0.0127 m diameter and 2.4 m long connected in parallel, such that the mass flow rate through each of them remains the same as the one in the model. For the heat transfer area of the tube-in-tube heat exchanger, again two different cases were considered; first, increasing the area by increasing the diameters and length of the inside and the outside tubes, while keeping all ratios the same, the second case, increasing the area by simply increasing the length of the tubes. Table 5.1 shows dimensions of the system for a number of scaleup ratios. Using the dimensions given in table 5.1 above with the simulation program, gives the results shown in figs. 5.55 and 5.56. Figure 5.55 shows the

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energy end exergy efficiencies for various scaleup ratios, and for the two cases discussed

previously, when all dimensions ratios were maintained and when area ratios only were

Dimensions	Scaleup ratio			
	1	2	3	4
Evaporator cross sectional area (m ²)	0.2	0.4	0.6	0.8
Evaporator diameter (m)	0.5	0.72	0.88	1.02
Condenser area (m ²)	0.16	0.32	0.48	0.64
Condenser inside diameter (m)	0.102	0.144	0.176	0.203
Condenser length (m)	0.5	0.71	0.87	1
Number of fins	10	10	10	10
Fin diameter (m)	0.254	0.355	0.432	0.497
Evaporator heat exchanger area (m ²)	0.096	0.192	0.287	0.383
Evaporator heat exchanger diameter (m)	0.013	0.018	0.022	0.025
Evaporator heat exchanger length (m)	2.4	3.4	4.16	4.8
Tube-in-tube heat exchanger area (m^2)	0.04	0.08	0.12	0.16
Internal tube diameter (m)	0.013	0.018	0.022	0.025
Annulus diameter (m)	0.025	0.036	0.044	0.051
Tube-in-tube heat exchanger length (m)	1	1.41	1.73	2
Heat source fluid mass flow rate (kg/hr)	10	20	30	40
Withdrawal rate (kg/hr)	0.1	0.2	0.3	0.4

Table 5.1. System dimensions for various scaleup ratios



Figure 5.55. Effect of scaleup ratio on the system efficiencies

maintained. The figure shows that both energy and exergy efficiencies remain almost constant when scaling up was made in terms of area, but when scaling up was made while maintaining all dimensions ratios, the efficiencies decreased slightly as scaleup ratio increased. This is due to the fact that increasing the dimensions of the system, particularly the diameter of the heat exchangers, will reduce the heat transfer coefficient, hence the efficiencies will decrease.

Although the efficiencies remained almost constant as the system was scaled up, the output from the system would differ slightly. The larger system would take longer time to get to steady state, especially when all dimensions ratios were maintained. For example, if the system is to be scaled up by scaleup ratio of two, the accumulated output by the end of transient operation period, 286 minutes, should be 0.834 kg and the output rate under the steady state conditions should be 0.263 kg/hr. The results obtained, fig. 5.56, show that when area scaling was maintained, the system will reach steady state after 290 minutes, the accumulated output will be 0.823 kg, and the output rate under steady state after 311 minute, the accumulated output would be 0.8 kg and the output rate under steady state after 311 minute, the accumulated output would be 0.8 kg and the output rate under steady state conditions would be 0.234 kg/hr.

Quality of Product

The developed distillation process involves evaporating the saline water and the produced vapor, which is usually salt free, condenses to form the product. Four samples from the produced water were analyzed for the total dissolved solids (TDS), the analysis showed that the quality of water produced is very high. Saline water with TDS of about 35,000 ppm was used which was reduced to about 90 ppm in the fresh water produced.

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Figure 5.56. Effect of scaling ratio on the system performance

CHAPTER 6 SALINE WATER AND SCALE FORMATION

This chapter presents the classification of water resources. Definition of saline water, and the major constituents and physical and chemical properties of sea water. It also presents various types of scale and corrosion associated with saline water and their prevention methods.

Saline Water Composition

Water resources can be classified into five different categories, according to the total solids dissolved in it, as shown in table 6.1.

Water sample	Total dissolved solids, mg/l		
Well water	300-500		
Typical river water	200-750		
Typical brackish water	1500-6000		
Typical seawater	36000		
Water for irrigation	1000		

Table 6.1 Classification of water resources [80]

Saline water, which includes both sea and brackish water, contains large amounts of dissolved salts, which makes it unsuitable for direct use in industry, household or agriculture. The total salt concentration in saline water is expressed in terms of either salinity (S), which is defined as the total amount of solids (grams) contained in 1 kg of saline water after all carbonate has been converted to oxide, and all bromide and iodide have been replaced by chloride and all organic matter has been completely oxidized, or Chlorinity (Cl), which is defined as the mass of silver necessary to precipitate the halogens (Cl⁻ and Br⁻) in 328.5233 g of seawater [81]. The two are related by the following relation [80],

S(,...) = 1.80655(Cl,...)

where **f** indicates per thousand

Seawater contains about 3.5% of dissolved salts, with major constituents given in table 6.2. For the purpose of the present work, seawater samples used for different tests were prepared using sea salt. This salt is produced from the Red Sea water using solar evaporation technique. An amount of 0.0346 kg of this salt was dissolved in 0.9654 kg of tap water to produce 1 kg of water with the properties of seawater with salinity of 3.5 %.

Seawater samples do not have the same composition, but all ions are present in the same ratios and the only variation is in the amount of pure water present. So if the percentage composition of any one ion was measured then the amount of all other ions could be accurately found by calculation.

Constituent	Seawater at S=3.5 % (g/kg)	(g/kg) ÷ Chlorinity
Sodium (Na ⁺)	10.77	0.556
Magnesium (Mg ²⁺)	1.29	0.068
Calcium (Ca ^{$2+$})	0.4121	0.02125
Potassium (K^+)	0.399	0.0206
Sr^{2+}	0.0079	0.00041
Chloride (Cl ⁻)	19.354	0.9989
Sulfate (SO_4^{2-})	2.712	0.1400
Bicarbonate (HCO ₃ ⁻)	0.1424	0.00735
Bromide (Br)	0.0673	0.00348
F	0.0013	0.000067
В	0.0045	0.000232
	$\Sigma = 35$	$\Sigma = 1.82$

 Table 6.2 Major composition of seawater [81]

Seawater contains a wide variety of dissolved organic compounds. The total amount of the dissolved organics is very low, about 2 ppm, but their composition is very complex [82]. The Organic matter (OM) is represented by living (Autotrophic and Heterotrophic) organisms and their excretory products and after death remains, but may also be inert or non-living. The latter is found in large accumulations (fuel deposition, soils), as well as in dispersed state in most mountain rocks and ocean waters. The inert OM is believed to be of biogenic origin and in reservoirs it dominates "living" OM. In seas and oceans the nonliving OM is mainly of autochthonous origin, while the intake from the land is comparatively small. Its primary source is phytoplankton [83]. A synopsis of the organic compounds already identified in seawater is given in table 6.3.

Component	Concentration as mg C/m ³ (ppb)
Vitamins	0.0065
Total fatty acids	5
Urea	5
Total free sugar	10
Total carbohydrates	200
Total free amino acid	10
Total combined amino acids	50
Dissolved organic carbon (including all of the	500-2000
above components)	
Dissolved organic nitrogen	$75-230 \text{ (as mg N/m}^3)$

Table 6.3 Presently known average organic composition of seawater [84]

Volatile component of organic matter constitutes a small fraction (<10 %) of the total organic matter in most marine systems. This small fraction includes these organic compounds of high vapor pressure, low molecular weight, and low water solubility, which can be purged or vaporized from water systems under natural conditions of wind and turbulence. Volatile matter is sometimes considered as the fraction of the total organic matter, which is lost during acidification and purging steps required for the determination of dissolved organic carbon in the seawater [85].

In some desalination plants filtration of feed water by activated carbon is made to remove dissolved organic materials. Biocides are also used as a pretreatment to prevent any microbiological activity. If we consider the present system, the degree to which organic liquids or solids present in seawater will tend to move to the gas phase in the space above the pool, depends on the vapor pressure of these substances. The higher the vapor pressure the higher the tendency of the substances to accumulate in the gas phase. Partitioning of these substances between the gas and liquid phase in seawater can be described by Henry's law [81].

$$\frac{C_A}{P_A} = k_B$$

However, due to their very low concentration and the fact that the unit will be flushed at regular intervals to remove any noncondensable gases, these gases will not be allowed to accumulate to a degree where they impose any serious problems.

Properties of Saline Water

Physical properties of saline water depend on the salt concentration present and to a lesser extent on the relative proportions of salts. In desalination the relative proportion becomes very important, since it is necessary to treat the feed water to prevent scale formation. Seawater properties can be simulated as 3.45% by weight NaCl solution [86]. The latent heat of vaporization of seawater is almost identical to that of fresh water, but the specific heat of normal seawater is smaller by several percent than that of pure water. Pure seawater freezes at about -1.9 °C and doubly concentrated seawater at -3.8 °C [87]. When it freezes, it crystallizes as a fine slush still holding up to 50 weight percent brine in the space between the crystals.

Scale formation and corrosion are the main problems of desalination. The dominant chemical and physical characteristics of seawater are as follows [88]:

1. Abundant supply of dissolved oxygen, which is the most important environmental factor affecting corrosion of structural steels, copper alloys, and stainless steels. The oxygen content of seawater varies between 0-12 ppm depending upon temperature, salinity, and biological activity. The solubility of oxygen is largely determined by the temperature, decreases with the increase of temperature. Concentrating seawater reduces the solubility of oxygen, so for steel, corrosion

rate decreases as concentration increases. Brackish underground waters have lower dissolved oxygen.

- 2. High chloride ions concentration, which penetrate protective films and enhances corrosion reactions. Sulfate ion acts in the same manner but to a lesser extent than chloride ion. Seawater contains about 19000 ppm chloride, which is about 55 times the chloride concentration needed to ensure 100% probability of corrosion mild steel.
- 3. Excellent electrolytic conductivity, which is higher than that of surface waters.
- 4. Naturally occurring corrosion inhibitors (anodic inhibitors), like phosphate, silicate, carbonate, fluoride, and hydroxyl ions occur in very low concentrations.
- 5. Presence of heavy metal ions (Cu, Zn, Cd, Pb).

The above-mentioned characteristics enhance corrosion. The following two

characteristics are corrosion retarding factors:

- 6. Abundant supply of calcareous scale formers (cathodic inhibitors), like, calcium, strontium and magnesium ions. The alkalinity developed at the cathodic places sets up reactions, those reactions result in deposition of tight and adherent films of lime salts (CaCO₃, SrCO₃, MgCO₃, and Mg(OH)₂). This film resists oxygen supply, thus low corrosion rate.
- 7. Growth of slime and marine, their growth act as a mechanical barrier to the diffusion of oxygen and act to lower corrosion rate.

Scale Formation and Prevention Methods

Scale is defined as the deposition of minerals on a solid surface. In distillation processes it might be found on the evaporator surface, its presence leads to operating difficulties and/or loss of efficiency. It reduces heat transfer through the surfaces affected; therefore, scale prevention is an integral part of the design, operation, and cost of the process. Sometimes pretreatment is necessary where the objective may be the removal or at least reduction of the concentration of ions which can be components of scale. The components of scale are derived from raw water except when corrosion takes place simultaneously with the deposition of scale. It usually contains the following chemical compounds: magnesium oxide, calcium carbonate and calcium sulfate. These compounds are found in different solid forms, either pure or in mixtures. The formation of scale can occur where the solubility limits (the maximum amount of solid that can be dissolved in a liquid at a specified temperature) of these compounds are exceeded. Magnesium chloride and sodium chloride are highly soluble, highly soluble salts are generally not components of the scale, which forms in distillation plants. Seawater can be concentrated to about one-tenth of its volume before the concentration of sodium chloride reaches saturation [87]. When pure water is separated from saline water the concentration increases and eventually reaches the saturation limit with respect to one or more salts. Also, when raw water saturated with a salt of inverted solubility (like calcium sulfate, where its solubility decreases if the temperature is increased beyond 38 °C) is heated, the solubility limit is eventually exceeded, even without evaporating water, because the solubility of such salts at high temperatures is less than at room temperature. In both cases, supersaturated solutions are obtained (solutions containing more salt than the solubility limit).

Two types of scale can be formed in distillation plants:

1. Alkaline scales: which results from the decomposition of bicarbonate content of seawater. When bicarbonate ion breaks down on heating, alkaline scales occur. The following reaction takes place:

$$2HCO_3^- \Leftrightarrow CO_2^+ + CO_3^{--} + H_2O \tag{6.1}$$

The carbonate ion can react in either of the following ways:

$$Ca^{++} + CO_3^{--} \Leftrightarrow CaCO_3^{\downarrow} \tag{6.2}$$

or,
$$CO_3^{--} + H_2O \Leftrightarrow OH^- + HCO_3^-$$
 (6.3)

Then the hydroxyl ion can react with the magnesium ion present to form magnesium hydroxyl scale as follows:

$$Mg^{++} + 2OH^{-} \Leftrightarrow Mg(OH)_{2}$$
(6.4)

When the temperature is below 82 °C, eq. 6.2 predominates and $CaCO_3$ scale deposition might occur. As the temperature increases above 82 °C eq. 6.3 predominates and hydroxyl ion formation is more favorable, which leads to the formation of $Mg(OH)_2$. $CaCO_3$ and $Mg(OH)_2$ are both present in ordinary seawater. Those components have inverted solubility.

2. Calcium sulfate scaling: deposition of calcium sulfate takes place because of its inverted solubility. No economic control method is available other than ensuring that the concentration-temperature path is within the solubility confines of the various crystalline modification of calcium sulfate; its deposition must be prevented.

Although supersaturated solutions are unstable, solid salts do not necessarily deposit immediately from these solutions. When these solutions come into contact with a crystal of a solid surface all dissolved solids they contain, above saturation, precipitate out. Crystals of the salt in supersaturated solution and many other solids can act as centers of crystallization. At the surface of the heating tubes the solubility of these materials is lower than that in the bulk of the solution (due to inverted solubility), hence supersaturation is reached there first and is immediately followed by deposition of solid. So it can be stated that supersaturated, centers of crystallization must also be present. Scale will deposit on those centers even if they are provided in another region (other than heating surface) of the equipment.

In applying stability data for calcium sulfate, magnesium hydroxide, and calcium carbonate to seawater, Spiegler [87] made the following conclusions:

1. If seawater is concentrated to two-thirds of its volume (1.5 degree of concentration) there will be a chance of calcium sulfate scale formation provided that seeds are present to induce crystallization. However, when the degree of concentration reaches three, the danger of scale deposition become serious, so it is necessary to discard the brine when it reaches this concentration limit.

- 2. Seawater is saturated with calcium carbonate; hence, scale may be deposited even at low temperature. However, the solubility of calcium carbonate can be greatly increased by adding acid, even weak acids such as carbonic are effective.
- 3. Magnesium hydroxide scale forms at higher temperature and/or when seawater has been concentrated to a considerable extent. Precipitation of magnesium hydroxide liberates acid that inhibits the precipitation of calcium carbonate.
- 4. In seawater stills working at low temperature (up to 60 °C), the scale will mainly consists of calcium carbonate.

To prevent scale formation it is necessary to know under what conditions of temperature and concentration the raw water becomes unstable, so that we may design the process such that these conditions are never reached or by controlling the degree of super saturation so that deposition of solids occurs only after the water has left the unit. Figure 6.1 shows how the solute concentration varies with time. The steady state, which is assumed to be reached if the solute concentration does not vary by more than 0.0001 % in 60 second period, will be reached after about 10 days of continuous operation, and the solute concentration (total dissolved solids) will be about 7.1 %. However, the most important factor in scale prevention is the choice of the lowest possible operating temperature, since super saturation increases with temperature. There are different methods to prevent scale formation:

- 1. Mechanical or chemical means of loosening deposited scale: This is done when the system is cold, a charge of steam passes through the heating surfaces, this produces a thermal shock, which is always sufficient to loosen the deposited scale. This is usually done along with the addition of some chemical agents, like derivatives of sulphonic acid, which weaken the bond between scale and heating surface.
- 2. Seeding: by establishing a preferential nucleation sites, such that these sites can be removed, processed and returned to the system economically.
- 3. Ion exchange: by passing the raw feed through a resin bed which replaces calcium and magnesium ions with sodium ions, as in the following equation:
$$R - Na + \frac{1}{2}Ca^{++} \rightarrow R - \frac{1}{2}Ca + Na^{+}$$

$$(6.5)$$

After sometime, the resin becomes spent and requires regeneration with a concentrated brine stream as follows:

$$R + \frac{1}{2}Ca + Na^{+} \rightarrow R - Na + \frac{1}{2}Ca^{++}$$
 (6.6)

This method is expensive and is usually used to treat boiler feed with low concentration of scale forming salts, unlike seawater with high concentration.



Figure 6.1 Variation of solute concentration with time

4. Proprietary components: adding compounds to seawater, which lead to delay in the onset of precipitation and a deformation of the crystal lattice, so that the scale does not adhere. The principal proprietary compound is Hagevap, which is a mixture of sodium polyphosphate, lignin sulphonic acid derivatives and various asters of polalkylene glycols.

5. pH control: acid treatment is very good way to prevent scale formation and consists of supplying hydrogen ions to break down the bicarbonate ions. The following reaction takes place:

$$HCO_3^- + H^+ \Leftrightarrow CO_2^\uparrow + H_2O \tag{6.7}$$

Sulphuric acid is usually the cheapest source of hydrogen ion followed by hydrochloric acid and ferric chloride.

Corrosion and Its Prevention

All types of corrosion that take place in desalination plants are electrochemical in nature. Electrical current flows through the seawater and at the place where it leaves the metal and enters the water (anode), corrosion takes place. The other end, i.e. where the current returns to the metal, is called cathode. A difference in electric potential must exist between anode and cathode for the corrosion mechanism to proceed, besides an electrical path is needed to complete the circuit, this path may be provided by metal structure itself or by physical contact between the metals. Corrosion may be minimized by:

- 1. Deaeration of dissolved oxygen: complete removal of dissolved gases is necessary to minimize corrosion of evaporator internals, prevent carbonate scale, and minimize condenser fouling by noncondensable gases.
- 2. Brine pH should be controlled to minimize corrosion yet remain below the magnesium hydroxide scaling point. Control within the range 7-7.7 is desirable.

Corrosion can be controlled through the selection of proper materials for the

desalination plants. The following materials can be used:

- 1. Metals and alloys: Stainless steel, copper alloys, aluminum alloys, and titanium are good materials. Carbon steel can be used but the corrosion rates are high. This material in heavy thickness behaves well when the water chemistry and physical conditions are suitably controlled.
- 2. Plastics: a number of plastic materials and coatings with a satisfactory performance are available to the desalination plants. Temperature must be carefully considered when selecting thermoplastics.
- 3. Corrosion resistant cements.

Environmental Aspects

Effluents from desalination processes contain highly concentrated water that depends on the water recovery from the feed. In desalination plants seawater is concentrated to about twice of the original sea solution. The concentrate also contains chemicals used for pretreatment of the feed water and washing solutions. If the desalination plant is close to open sea, small-scale operations do not cause a serious damage to the marine life, but in large-scale operations the problem become more serious. However, the effluent can be diluted and spread to overcome the problem.

The issue of effluents becomes more serious for inland plants, where there is no access to open sea for purging them. The concentrate may increase the salinity of the underground water if it penetrates the earth. A possible solution to this problem is a zero discharge treatment, i.e. evaporative separation between solids and water, so solids may be stored properly inland, or deep well injection.

CHAPTER 7 ECONOMIC ANALYSIS

Introduction

Solar energy may prove to be economical for saline water desalination due to one

or more of the following reasons:

- 1. Location: Many arid and semi-arid areas are coastal and have high insolation rates.
- 2. Seasonal changes: In some areas, the demand for fresh water increases during tourism season, at this time the insolation rates are high.
- 3. Lack of conventional energy sources in many remote areas.
- 4. Environmental impact: It is known that saline water desalination processes are energy intensive ones. It is estimated that for the production of 1 m³ of fresh water from saline water, a minimum of 0.7 kWh of energy is required, whereas in reality much more energy is required (see table 1.1). Besides the energy cost, there are environmental concerns regarding the effect of using conventional energy sources. Such as ozone depletion and global warming.
- 5. Economics: For some areas, like the Mediterranean islands, fresh water is transported by ships, which makes it very costly.

The cost of the produced fresh water depends on the cost of the desalination

system. Solar energy systems, in general, are capital intensive but require low operational

and maintenance costs. The actual cost of a desalination system depends on the materials

used for construction. Prices may differ considerably from one location to another, if

local materials and local personnel are used.

Cost Analysis

To be able to calculate the cost of a unit of fresh water produced, we need to calculate the total amount of fresh water and the total cost of the system, for a certain period of time, say annually.

The daily distillate output can be calculated from the following equation:

$$m_d = \int_{t=1}^{24} m_e dt$$

The annual average distillate output per day is given as

$$m_y = \left(\frac{1}{365}\right) \sum_{i=1}^{365} m_{d,i}$$

The total annual cost of the system is given as

Annual Cost (AC) = First Annual Cost (FAC) + Annual Maintenance (AM) – Annual Salvage Value (ASV) – Tax Savings (TS)

If CC is the capital cost of the system (table 7.1 provides a breakdown of the cost of various system components, these figures were obtained from the products catalogs of different manufacturers) and CRF is the capital recovery factor, the first annual cost of the system (FAC) can be calculated as,

$$FAC = CC * CRF$$

where

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1}$$

n is the lifetime of the system, solar distillation plants have a mean lifetime of about 20 years [24].

Maintenance is required to ensure a continuous supply of water, replacement of broken glass, cleaning the system and corrosion and scale control costs. The Annual Maintenance Cost (AMC) can be taken as a percent of the first annual cost. However, this cost is not expected to be high. The brine disposal cost is not included here, which might be a big problem especially if the system is not close to open sea.

If S is the salvage value of the system, the first annual salvage value (ASV) can be calculated as

$$ASV = S * SFF$$

where

SFF is the sinking fund factor, given by

$$SFF = \frac{i}{\left(1+i\right)^n - 1}$$

The salvage value of the system, S, can be taken as a percent of the initial cost of the system.

If the income tax law determines that any expenses incurred for the production of a business income are tax exempted. Then costs such as maintenance, and money paid for interest are subtracted from the income for tax purposes. Then,

Tax savings = net profit tax rate*(maintenance cost + interest paid + depreciation) Income tax savings for non-income producing system, can be calculated as, Income tax savings = effective tax rate * interest payment

Income tax savings were not included in the numerical analysis to follow; they were presented only to complete the theoretical analysis.

The product cost per kg, PC, is given by

$$PC = \frac{AC}{\sum_{i=1}^{365} m_{d,i}}$$

Table 7.1 Cost breakdown of	the system components (base year	2002)
Item	Dimensions	Estimated cost (\$US)
Solar collector	1 m^2	200
Evaporator	1 m^2 cross sectional area	500
Condenser	10.16 cm copper tube with fins	50
Evaporator heat exchanger	1.27 cm copper tube, 5 m long	30
Tube-in-tube heat exchanger	1.27 cm inside copper tube, 2.54	25
	cm outside PVC tube	
PVC pipes	PVC tube of 1.27 cm diameter	30
	and 30 m long	
Pipe fittings		20
Storage tanks	Four, 20 liters capacity	60
Supporting structure	32 ft high structure	*
Labor		240**
Total cost		1155
Miscellaneous		110
Total system cost		1265
Salvage value (25% of the cost		250
of usable items)		
* The supporting structure is assume	ed already existing	

* The supporting structure is assumed already existing
** Assuming the erection of the system will require 8-man hour, and the labor cost is US\$30/man hour

Effect of different parameters such as interest rate, system lifetime, and annual

maintenance cost on the cost of the produced distilled water are shown in table 7.2.

1 4010 /	.2 111		nous pu			or mut	or produ	leeu			
		Capital	First	Annual		Annual				Annual	Annual
Interest	Useful	recovery	annual	maintenance	Sinking	salvage	Annual	Annual	Annual	useful	cost
rate	life	factor	cost	(10 % of	fund	value	cost	output	cost	energy	per
(%)	(years)	(CRF)	(FAC)	FAC)	factor	(ASV)	(AC)	(kg)	per kg	(kWh)	kwh
5	10	0.130	165	16	0.080	20	160	1000	0.160	650.0	0.247
5	15	0.096	120	12	0.046	12	120	1000	0.120	650.0	0.189
5	20	0.080	100	10	0.030	8	110	1000	0.110	650.0	0.161
8	10	0.149	190	19	0.069	18	190	1000	0.190	650.0	0.293
8	15	0.117	150	15	0.037	10	155	1000	0.155	650.0	0.237
8	20	0.102	130	13	0.022	6	135	1000	0.135	650.0	0.210
10	10	0.163	205	20	0.063	16	210	1000	0.210	650.0	0.325
10	15	0.131	165	16	0.031	8	175	1000	0.175	650.0	0.270
10	20	0.117	150	15	0.017	4	160	1000	0.160	650.0	0.246

Table 7.2 Effect of various parameters on the cost of water produced

Based on the results obtained, a decision regarding the feasibility of the system for use in a certain location can be made by calculating the solar savings, defined as

Solar savings = *cost of conventional energy* – *costs of solar energy*

If the savings are negative, then they are losses instead of gains. For example, if the interest rate is 5 % and the expected life of the system is 10 years, the annual cost of the system is about US\$160, and the annual production is about 1000 kg, which gives a product cost of US\$0.16. Assuming the average latent heat of evaporation to be 0.65 kWh/kg, we will need 650 kWh to produce the 1000 kg of distilled water. Since the annual cost is US\$160, the cost of one kWh will be US\$0.247.

As clear from the above figures, the cost of fresh water produced is high. The cost is expected to drop significantly if the economic analysis is conducted for another place, where the costs are lower than those in the United States and the intensity of incident solar radiation is higher than that for Gainesville.

The total production cost of fresh water for brackish water systems (RO) is about \$0.25 to \$0.6 per m³ (for systems having capacities of 4000 to 40000 m³ per day) and for seawater desalting systems (MSF, MEB) is about \$1 to \$4 per m³ (for systems having capacities of 4000 to 20000 m³ per day) [17]. Under these conditions the use of solar energy for distillation is not economically feasible at this time. However, solar energy systems get will be less expensive over time, and are expected to become competitive to conventional energy sources at least for some geographic locations.

CHAPTER 8 CONCLUSIONS

A water distillation system that uses low grade heat was studied theoretically and

experimentally. The system makes use of natural forces of gravity and atmospheric

pressure to create a vacuum under which water can be evaporated more rapidly at low

temperatures.

Experimental tests were performed outdoor and indoor. The effect of various

parameters: depth of water body inside the evaporator, withdrawal rate, condenser

thermal resistance, and heat source temperature on the system performance were studied.

The results showed that

- 1. The depth of water body has small effect on the system performance, but as the depth of water body increases, the system will need more time to reach the steady state conditions.
- 2. The withdrawal rate has a significant effect on the system performance. When it goes beyond 1 kg/hr, the system output starts decreasing rapidly. Reducing the withdrawal rate below 0.1 kg/hr will result in a high solute concentration inside the system, increasing the possibility of scale formation, besides reducing the system output, since the system output and solute concentration were found to be inversely proportional.
- 3. Condenser thermal resistance affects the system performance significantly. The lower the thermal resistance, the lower the fresh water temperature, the higher the vapor pressure difference between the evaporator and condenser, i.e. the higher the driving force for mass transfer, thus improving the system performance.
- 4. The temperature of the heat source is the most important factor that influences the performance of the system. As this increases the system output improves significantly.

Experimental results showed that the output from a system of 0.2 m^2 evaporator

cross sectional area could reach 0.95 kg for a six hours test. A theoretical simulation

model was developed, which predicted the system performance very well. Based on a simulation for Gainesville, Florida, the daily output from a system of 1 m^2 evaporator cross sectional area connected to a solar collector of 1 m^2 absorber area could reach 6.5 kg, which is almost double the amount from flat basin solar still of about 3-4 kg/day.m²

[4].

The experimental study shows that the proposed concept for solar desalination is

feasible. This might result in the following:

- 1. Self-sustainable solar desalination appropriate for supplying rural areas with potable water.
- 2. Use of solar energy will save conventional energy sources for other applications and reduce pollution to the environment.
- 3. An energy efficient technology, where vacuum will be utilized to evaporate saline water at low temperatures. This technique offers the potential to reduce energy consumption substantially, so that simple flat plate solar collectors can be used to supply the system with its energy requirements.

The proposed system has the following additional advantage:

- It has low operating temperatures, which reduces the heat loss to the environment, and allows the system to operate more efficiently. Besides at low temperatures the scale formation problem is minimized.
- The system can be operated in either a continuous or a batch process mode. When operating in a continuous process mode, the withdrawal rate can be adjusted such that the solute concentration remains well below the oversaturation limits where scale formation becomes a serious problem.
- The quality of water produced is very high. Saline water with TDS of about 35,000 ppm was used which was reduced to about 90 ppm in the fresh water produced.

The disadvantages of the system are as follows:

- It operates under a vacuum condition, which requires careful design, manufacturing, and operation.
- It requires a special supporting structure, a tower of about 10 m height, if that is not something available, like a roof of an existing building.

Economic analysis showed that if the system is used in Gainesville, FL, the cost of 1 kg of distilled water produced will be about US\$0.135, based on 20 years lifecycle at 8 % interest rate.

System performance can be improved significantly if it is made to operate as a multistage system, i.e. the vapor produced in one stage is used to evaporate a part of the water in the next stage. In such an arrangement, heat is to be added to the lowest stage.

APPENDIX A CALCULATIONS PROCEDURE

To be able to solve the system of equations obtained from the theoretical analysis, first the differential equations are to be written in a form suitable for numerical calculations. For this purpose a backward finite difference scheme is used.

Solute conservation eq. 3.4 is rewritten as

$$\frac{d}{dt}(Y_sV) = Y_i \overset{\bullet}{V}_i - Y_s \overset{\bullet}{V}_w$$

where *Y* is a compounded variable $=\rho C$

$$\frac{(Y_{s}^{1} - Y_{s}^{0})V}{\Delta t} = Y_{i}^{0} \overset{\bullet}{V}_{i}^{0} - Y_{s}^{0} \overset{\bullet}{V}_{w}^{0}$$

$$let, \frac{V}{\Delta t} = a$$

$$aY_{s}^{1} = aY_{s}^{0} + Y_{i}^{0} \overset{\bullet}{V}_{i}^{0} - Y_{s}^{0} \overset{\bullet}{V}_{w}^{0}$$

$$Y_{s}^{1} = \frac{1}{a} \left[Y_{s}^{0} \left(a - \overset{\bullet}{V}_{w}^{0} \right) + Y_{i}^{0} \overset{\bullet}{V}_{i}^{0} \right]$$
(A.1)

The constraint imposed by the stability criteria of the backward finite difference

method, requires that $(a - V_w)$ be positive.

Energy conservation eq. 3.6 is rewritten as

$$\frac{d}{dt}(X_{s}V) + \frac{d}{dt}(W_{2}T)_{s} = \stackrel{\bullet}{m}C_{pf}(T_{co} - T_{s})Z + X_{i}\stackrel{\bullet}{V}_{i} - X_{s}\stackrel{\bullet}{V}_{w} - Q_{e} - Q_{loss}$$

where $X = (\rho C_p T)_s$

$$W_{2} = (\rho C_{p} V)_{steel}, \text{ and}$$

$$Z = 1 - \exp\left[-\frac{\pi D_{h} U_{h} l_{h}}{\dot{m} C_{pf}}\right]$$

$$\frac{(X_{s}^{1} - X_{s}^{0}) W}{\Delta t} + \frac{W_{2} (T_{s}^{1} - T_{s}^{0})}{\Delta t} = \dot{m} C_{pf} T_{co} Z - \dot{m} C_{pf} T_{s}^{0} Z + X_{i}^{0} \dot{V}_{i}^{0} - X_{s}^{0} \dot{V}_{w}^{0} - Q_{e}^{0} - Q_{loss}^{0}$$

$$let, \frac{V}{\Delta t} = a, \text{ and}$$

$$\frac{W_{2}}{\Delta t} = W_{1}$$

$$aX_{s}^{1} + W_{1}T_{s}^{1} = aX_{s}^{0} + W_{1}T_{s}^{0} + \dot{m} C_{pf} T_{co} Z - \dot{m} C_{pf} T_{s}^{0} Z + X_{i}^{0} \dot{V}_{i}^{0} - X_{s}^{0} \dot{V}_{w}^{0} - Q_{e}^{0} - Q_{loss}^{0}$$

$$G = \frac{1}{a} \left[X_{s}^{0} \left(a - \dot{V}_{w}^{0} \right) - \dot{m} C_{pf} T_{s}^{0} Z + W_{1} T_{s}^{0} + \dot{m} C_{pf} T_{co} Z + X_{i}^{0} \dot{V}_{i}^{0} - Q_{e}^{0} - Q_{loss}^{0} \right] \quad (A.2)$$
where $G = X_{s}^{1} + \frac{W_{1}}{a} T_{s}^{1}$

$$let, \frac{W_{1}}{a} = W$$

$$G = X_{s}^{1} + WT_{s}^{1} = (\rho C_{p})_{s}^{1} T_{s}^{1} + WT_{s}^{1} = ((\rho C_{p})_{s}^{1} + W)T_{s}^{1}$$

Since, the density and specific heat varies as the temperature and/or salinity varies; saline water temperature, concentration, density and specific heat that appear in the compounded variables G and Y, are obtained by solving for each time increment the following system of four coupled equations,

$$G = ((\rho C_p)_s^1 + W)T_s^1$$
$$Y_s^1 = (\rho C)_s^1$$

$$C_{ps}^{1} = \alpha_{2}C_{s}^{1}$$
$$\rho_{s}^{1} = \rho_{0} \Big[1 - \beta_{T}(T_{s}^{1} - T_{0}) + \beta_{c}(C_{s}^{1} - C_{0}) \Big]$$

Those can be rearranged to give,

$$C_{s}^{1} = \frac{Y_{s}^{1}}{\rho_{0} \left[1 - \beta_{T} \left(\frac{GC_{s}^{1}}{Y_{s}^{1} \alpha_{2} C_{s}^{1} + Y_{s}^{1} \beta_{2} + WC_{s}^{1}} - T_{0} \right) + \beta_{c} \left(C_{s}^{1} - C_{0} \right) \right]}$$
(A.3)

$$C_{ps}^{1} = \alpha_2 C_s^{1} + \beta_2 \tag{A.4}$$

$$\rho_s^1 = \frac{Y_s^1}{C_s^1}$$
(A.5)

$$T_{s}^{1} = \frac{G}{\rho_{s}^{1}C_{ps}^{1} + W}$$
(A.6)

Since the system of equations requires an iterative solution, the Gauss-Seidel method was used. System specifications were fixed. Reference state was identified; initial conditions on concentration and temperatures were specified. Then based on those initial conditions: density (eq. D.11), specific heat (eq. D.9), latent heat of vaporization (eq. D.2), vapor pressure in the evaporator and condenser (eq. D.1), hence evaporation rate (eq. 3.23) were calculated. The amount of heat input to the system via the heat exchanger placed inside the evaporator was also calculated (eq. 3.30), which depends on both the temperature of saline water and heat source temperature (if the mass flow rate and heat exchanger specifications are held constant). Heat loss is calculated using eqs. 3.58-3.64. The values of the compounded variables Y and G are calculated (eq. A.1) and (eq. A.2), respectively. This will be done for each time step (60 seconds) and at the end of each step the new values of the solute concentration (eq. A.3), as clear from the above equations fixed point iteration will be performed to solve for the solute concentration, specific heat

(eq. A.4), density (eq. A.5), and temperature (eq. A.6) will be calculated. The injection water temperature (eq. 3.37) is also calculated after knowing the saline water temperature, the withdrawal flow rate and injection flow rate (eq. 3.1), assuming that the mass of water inside the evaporator remains constant. Fresh water temperature is then calculated. This is done by first assuming the outside surface temperature of the condenser, then calculate the temperature at the interface (eq. 3.53) based on the amount of heat to be dissipated (eq. 3.47), then calculate the outside surface temperature by equating the amount of heat transferred across the film and condenser material (eq. 3.53) and that convected from its outside finned surface (eq. 3.57) and iterating till convergence (temperature difference between two successive iteration less than 0.01°C) is achieved. Equation 3.53 is used again to calculate the condenser inside surface temperature. Then, eq. 3.50 is used to calculate the film temperature. Steady state conditions for the unit are achieved when the difference in saline water temperature at the beginning and the end of a time step becomes less than 0.01°C. If this difference is greater than 0.01°C, the above processes are repeated till convergence is achieved.

APPENDIX B COMPUTER PROGRAM

%***Computer program written in MATLAB (version 5.1)***
S*************************************
%**Calculations of areas, perimeter, volume and length*
%**required for subsequent calculations*************
S*************************************
Af=pi*rci^2;
Aci=pi*dci*lc;
Aco=pi*dco*lc;
As=pi*rs^2;
Ahe=SF*(pi*dhe*lh);
Pside=2*pi*rs;
lbottom=As/Pside;
Aside=Pside*lside;
Abottom=As;
vbottom=As*tbottom;
vside=Aside*tside;
vtop=Atop*ttop;
vtotal=vbottom+vside+vtop;
v=As*hs;

vt=As*ht+0.5*(As+Af)*0.15+Af*lc;

```
vnet=vt-v;
  %***************************Gas pressure**********
  massAr=pl*ppmAr*v*D0/1e6;
  massC=pl*ppmC*v*D0/1e6;
  massN2=pl*ppmN2*v*D0/1e6;
  massO2=pl*ppmO2*v*D0/le6;
  PAr=massAr*(R/MAr)*Ts/vnet;
  PC=massC*(R/MC)*Ts/vnet;
  PN2=massN2*(R/MN2)*Ts/vnet;
  PO2=massO2*(R/MO2)*Ts/vnet;
  Pgas=PAr+PC+PN2+PO2;
  Cp=a2*C+b;
  Cpi=a2*Ci+b;
  %Cp=4186*(1.0049-0.01621*C+3.5261e-4*C^2-((3.2506-
1.4795*C+0.07765*C^2)*10^-4*Ts)+((3.8013-
1.2084*C+0.06212*C^2)*10^-6*Ts^2));
```

hfg=2.3246e3*(1.0727e3-1.0167*Ts+1.4087e-4*Ts^2-

5.1462e-6*Ts^3);

```
/9);
```

hside=Nuside*ka/lside;

```
X1side=1/(hside*2*pi*rs3*lside);
```

```
X2side=log(rs2/rs)/(2*pi*ksteel*lside)+log(rs3/rs2)/(2*
```

```
pi*kinsulation*lside);
```

```
Y1side=(Twside-Ta)/X1side;
```

```
Y2side=(Ts-Twside)/X2side;
```

while abs (Y1side-Y2side)>0.1

if Y1side>Y2side

Twside=Twside-0.05;

else

```
Twside=Twside+0.1;
```

end

Taverageside=(Twside+Ta)/2;

```
Bside=1/(Taverageside+273);
```

```
Raside=g*Bside*(Twside-Ta)*lside^3/(alphaa*nua);
```

```
Nuside=0.68+(0.67*Raside^0.25)/(1+(0.492/Pr)^(9/16))^(
```

4/9);

```
hside=Nuside*ka/lside;
```

X1side=1/(hside*2*pi*rs3*lside);

X2side=log(rs2/rs)/(2*pi*ksteel*lside)+log(rs3/rs2)/(2

*pi*kinsulation*lside);

Y1side=(Twside-Ta)/X1side;

Y2side=(Ts-Twside)/X2side;

end

Qlossside=Aside*hside*(Twside-Ta);

Taveragebottom=(Twbottom+Ta)/2;

Bbottom=1/(Taveragebottom+273);

Rabottom=g*Bbottom* (Twbottom-

Ta) *lbottom^3/ (alphaa*nua);

Nubottom=0.27*Rabottom^0.25;

hbottom=Nubottom*ka/lbottom;

X1bottom=1/(hbottom*As);

X2bottom=tbottom/(ksteel*As)+tinsulation/(kinsulation*A

s);

Y1bottom=(Twbottom-Ta)/X1bottom;

Y2bottom=(Ts-Twbottom)/X2bottom;

while abs (Y1bottom-Y2bottom) >0.1

if Y1bottom>Y2bottom

Twbottom=Twbottom-0.05;

else

Twbottom=Twbottom+0.1;

end

Taveragebottom=(Twbottom+Ta)/2;

Bbottom=1/(Taveragebottom+273);

Rabottom=g*Bbottom*(Twbottom-

Ta) *lbottom^3/(alphaa*nua);

Nubottom=0.27*Rabottom^0.25;

hbottom=Nubottom*ka/lbottom;

X1bottom=1/(hbottom*As);

X2bottom=tbottom/(ksteel*As)+tinsulation/(kinsulation*

As);

Y1bottom=(Twbottom-Ta)/X1bottom;

Y2bottom=(Ts-Twbottom)/X2bottom;

end

Qlossbottom=As*hbottom*(Twbottom-Ta);

Taveragetop=(Twtop+Ta)/2;

Btop=1/(Taveragetop+273);

Ratop=g*Btop*(Twtop-Ta)*ltop^3/(alphaa*nua);

Nutop=0.56* (Ratop*cos(theta))^0.25;

htop=Nutop*ka/ltop;

X1top=1/(htop*Atop);

X2top=ttop/(ksteel*Atop)+tinsulation/(kinsulation*Atop)

;

```
Y1top=(Twtop-Ta)/X1top;
```

```
Y2top=(Ts-Twtop)/X2top;
```

while abs (Y1top-Y2top)>0.1

if Y1top>Y2top

Twtop=Twtop-0.05;

else

Twtop=Twtop+0.1;

end

Taveragetop=(Twtop+Ta)/2;

Btop=1/(Taveragetop+273);

Ratop=g*Btop*(Twtop-Ta)*ltop^3/(alphaa*nua);

Nutop=0.56* (Ratop*cos(theta))^0.25;

htop=Nutop*ka/ltop;

X1top=1/(htop*Atop);

X2top=ttop/(ksteel*Atop)+tinsulation/(kinsulation*Atop

);

Y1top=(Twtop-Ta)/X1top;

Y2top=(Ts-Twtop)/X2top;

end

Qlosstop=Atop*htop*(Twtop-Ta);

Qloss=Qlossside+Qlossbottom+Qlosstop;

%values of temperature, density, specific heat and solute

```
Xi=Di*Cpi*Ti;
```

hhe=3.66*kcf/dhe;

Uh=1/((1/hhe)+FT);

Z=1-exp(-Ahe*Uh/(m*Cpc));

X=(1/a) * (X0* (a-Qw) -m*Cpc*Ts*Z+w1*Ts+m*Cpc*Tcout*Z-

he+Xi*Qi-Qloss);

%*********surface temperature of the heat exchanger**

Uhe=1/(1/hhe);

Zhe=1-exp(-Ahe*Uhe/(m*Cpc));

```
Qcollectoruseful=m*Cpc*(Tcout-Tcinlet);
    The=Tcout-(Qcollectoruseful/(Zhe*m*Cpc));
    Y0=Ds*C;
    Yi=Di*Ci;
    Y = (1/a) * (Y0* (a-Qw) + Yi*Qi);
    C1=Y/(D0*(1-bt*(X*C/(Y*a2*C+Y*b+w*C)-T0)+bc*(C-C0)));
    while abs(C1-C) > 0.01
      C=C1;
       C1=Y/(D0*(1-bt*(X*C/(Y*a2*C+Y*b+w*C)-T0)+bc*(C-
CO)));
    end
    C=C1;
    Cp=a2*C+b;
    %Cp=4186*(1.0049-0.01621*C+3.5261e-4*C^2-((3.2506-
1.4795*C+0.07765*C^2)*10^-4*Ts)+((3.8013-
1.2084 \times C + 0.06212 \times C^{2} \times 10^{-6} \times Ts^{2});
    Ds=Y/C;
    Ts1=X/(Ds*Cp+w);
    %*****calculate the amount of gases librated*******
    massAr=pl*ppmAr*Qi*Di*dt/1e6;
```

massC=pl*ppmC*Qi*Di*dt/1e6;

massN2=pl*ppmN2*Qi*Di*dt/1e6;

mass02=pl*ppm02*Qi*Di*dt/1e6;

```
150
```

```
PAr=massAr*(R/MAr)*Ts/vnet;
PC=massC*(R/MC)*Ts/vnet;
PN2=massN2*(R/MN2)*Ts/vnet;
PO2=massO2*(R/MO2)*Ts/vnet;
dPgas=PAr+PC+PN2+PO2;
Pgas=Pgas+dPgas;
Cc=Di*Qi*Cpi;
Ch=Ds*Qw*Cp;
if Cc-Ch<0
 Cmin=Cc;
 Cmax=Ch;
else
 Cmin=Ch;
 Cmax=Cc;
end
Re=(4*Di*Qi) / (pi*di*us);
hi=(Nu*ks)/di;
```

```
ho=(Nui*ks)/dhy;
```

```
UA=1/(1/(pi*di*l*hi)+log((ri+ti)/ri)/(2*pi*kc*l)+1/(pi*
```

(di+2*ti)*l*ho));

NTU=UA/Cmin;

```
Cr=Cmin/Cmax;
E = (1 - \exp(-NTU^{*}(1 - Cr))) / (1 - Cr^{*}exp(-NTU^{*}(1 - Cr)));
qmax=Cmin*(Ts-T0);
q=E*qmax;
Ti=q/Cc+T0;
Tcinlet=Ts+(Tcout-Ts)*exp(-Ahe*Uh/(m*Cpc));
Re=(4*mf) / (pi*dci*uv);
Tfa=(Tco+Ta)/2;
ka=1.163*(0.021+(0.66e-4)*Tfa);
ua=(1/3600) * (6.1848e-2+(1.6632e-4) *Tfa);
Da=353.44/(Tfa+273.15);
Cpa=4186*(0.2317+1.6213e-5*Tfa+3.9593e-8*Tfa^2-1.6213e-
```

11*Tfa^3);

```
Pr=ua*Cpa/ka;
```

mf=mf;

hfgm=hfg+(3/8)*Cpf*(Ts-Tif);

hc=0.555*((g*Df*(Df-Dv)*kf^3*hfgm)/(uf*dco*(Ts-

Tif)))^0.25;

coeff=1/(2*pi*rci*lc*hc)+log(rco/rci)/(2*pi*lc*kc);

Tif=(1/(1+0.68*mf*Cpf*coeff))*(mf*coeff*hfg+0.68*mf*Cpf

*Ts*coeff+Tco);

```
Tci=mf*(hfg+0.68*Cpf*(Ts-
```

Tif))*log(rco/rci)/(2*pi*lc*kc)+Tco;

```
%Tci=(mf*hfg*log(rco/rci)+0.68*Cpf*mf*log(rco/rci)*Ts+2
```

```
*pi*lc*kc*Tco)/(2*pi*lc*kc+0.68*Cpf*mf*log(rco/rci));
```

xi=dco/Dfin;

B=1/(Tfa+273);

Tco=Tco;

RaS=(g*B*(Tco-Ta)*Pr*S^3/(nua)^2)*(S/Dfin);

Beta=(0.17*xi)+exp(-4.8*xi);

CC=((23.7-1.1*(1+(152*xi^2))^0.5)/(1+Beta))^(4/3);

```
hco=(ka/S)*(RaS/(12*pi))*(2-exp(-(CC/RaS)^0.75)-exp(-
```

Beta*(CC/RaS)^0.75));

CCC=0.44+0.12/xi; hcot=(ka/S)*(CCC*RaS^0.29); rfin=Dfin/2; r2c=(Dfin/2)+(tfin/2); Aftip=N*pi*Dfin*tfin; Afside=N*2*pi*(rfin^2-rco^2); Ab=2*pi*rco*(lc-N*tfin); mm=sqrt((2*hco)/(kc*tfin));

C2=(2*rco/mm)/(r2c^2-rco^2);

fineff=C2*(BESSELK(1, (mm*rco))*BESSELI(1, (mm*r2c))-

BESSELI(1,(mm*rco))*BESSELK(1,(mm*r2c)))/(BESSELI(0,(mm*rco

;

```
Tcol=(1/((log(rco/rci)*constant)+2*pi*lc*kc))*(2*pi*lc*
kc*Tci+log(rco/rci)*Ta*constant);
```

```
while abs(Tco1-Tco)>0.0001
```

Tco=Tco1;

Tfa=(Tco+Ta)/2;

ka=1.163*(0.021+(0.66e-4)*Tfa);

ua=(1/3600)*(6.1848e-2+(1.6632e-4)*Tfa);

Da=353.44/(Tfa+273.15);

Cpa=4186*(0.2317+1.6213e-5*Tfa+3.9593e-8*Tfa^2-

1.6213e-11*Tfa^3);

Pr=ua*Cpa/ka;

hfgm=hfg+(3/8) *Cpf*(Ts-Tif);

hc=0.555*((g*Df*(Df-Dv)*kf^3*hfgm)/(uf*dco*(Ts-

Tif)))^0.25;

coeff=1/(2*pi*rci*lc*hc)+log(rco/rci)/(2*pi*lc*kc);

Tif=(1/(1+0.68*mf*Cpf*coeff))*(mf*coeff*hfg+0.68*mf*Cp

f*Ts*coeff+Tco);

Tci=mf*(hfg+0.68*Cpf*(Ts-

Tif))*log(rco/rci)/(2*pi*lc*kc)+Tco;

```
%Tci=(mf*hfg*log(rco/rci)+0.68*Cpf*mf*log(rco/rci)*Ts+
```

```
2*pi*lc*kc*Tco)/(2*pi*lc*kc+0.68*Cpf*mf*log(rco/rci));
```

xi=dco/Dfin;

Tfa=(Tco+Ta)/2;

B=1/(Tfa+273);

RaS=(g*B*(Tco-Ta)*Pr*S^3/(nua)^2)*(S/Dfin);

Beta=(0.17*xi)+exp(-4.8*xi);

CC=((23.7-1.1*(1+(152*xi^2))^0.5)/(1+Beta))^(4/3);

```
hco=(ka/S)*(RaS/(12*pi))*(2-exp(-(CC/RaS)^0.75)-
```

 $\exp(-\text{Beta}*(CC/RaS)^{0.75});$

CCC=0.44+0.12/xi;

hcot=(ka/S)*(CCC*RaS^0.29);

rfin=Dfin/2;

r2c=(Dfin/2)+(tfin/2);

Aftip=N*pi*Dfin*tfin;

Afside=N*2*pi*(rfin^2-rco^2);

Ab=2*pi*rco*(lc-N*tfin);

mm=sqrt((2*hco)/(kc*tfin));

C2=(2*rco/mm)/(r2c^2-rco^2);

fineff=C2*(BESSELK(1, (mm*rco))*BESSELI(1, (mm*r2c))-

BESSELI(1, (mm*rco))*BESSELK(1, (mm*r2c)))/(BESSELI(0, (mm*rco))*BESSELK(1, (mm*r2c))+BESSELK(0, (mm*rco))*BESSELI(1, (mm*r2c)));

constant=N*fineff*hcot*Aftip+N*fineff*hco*Afside+hco*Ab;

```
Tco1=(1/((log(rco/rci)*constant)+2*pi*lc*kc))*(2*pi*lc*kc*T
ci+log(rco/rci)*Ta*constant);
```

end

Tco=Tco1;

Tci=(mf*hfg*log(rco/rci)+0.68*Cpf*mf*log(rco/rci)*Ts+2*

pi*lc*kc*Tco)/(2*pi*lc*kc+0.68*Cpf*mf*log(rco/rci));

q2=2*pi*lc*kc*(Tci-Tco)/log(rco/rci);

Tf=Tif;

if (Tcout+273.15)>=273.15&(Tcout+273.15)<300

ah=0;bh=0;ch=0;dh=0;e1h=6.24698837e2;e2h=-

2.34385369e3;

e3h=-9.50812101e3;e4h=7.16287928e4;e5h=-

1.63535221e5;

e6h=1.66531093e5;e7h=-6.47854585e4;hfcr=2.0993e3;

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as=0;bs=0;cs=0;ds=0;e1s=-

1.83692956e3;e2s=1.47066352e4;

e3s=-4.31466046e4;e4s=4.86066733e4;e5s=7.9975096e3;

e6s=-5.83339887e4;e7s=3.31400718e4;sfcr=4.4289;

else

(Tcout+273.15)>=300&(Tcout+273.15)<600;

ah=8.839230108e-1;bh=0;ch=0;dh=0;e1h=-

2.67172935;e2h=6.22640035;

e3h=-1.31789573e1;e4h=-1.91322436;e5h=6.87937653e1;

e6h=-1.24819906e2;e7h=7.21435404e1;hfcr=2.0993e3;

as=9.12762917e-1;bs=0;cs=0;ds=0;e1s=-

1.75702956;e2s=1.68754095;

e3s=5.82215341;e4s=-6.33354786e1;e5s=1.88076546e2;

e6s=-2.52344531e2;e7s=1.28058531e2;sfcr=4.4289;

end

Tcr=647.3;

Tccout=(Tcr-(Tcout+273.15))/Tcr;

hcout=1000*hfcr*(ah+bh*Tccout^(1/3)+ch*Tccout^(5/6)+dh*

Tccout^(7/8)+e1h*Tccout+e2h*Tccout^2+e3h*Tccout^3+e4h*Tccou

t^4+e5h*Tccout^5+e6h*Tccout^6+e7h*Tccout^7);

scout=1000*sfcr*(as+bs*Tccout^(1/3)+cs*Tccout^(5/6)+ds*

Tccout^(7/8)+e1s*Tccout+e2s*Tccout^2+e3s*Tccout^3+e4s*Tccou

t^4+e5s*Tccout^5+e6s*Tccout^6+e7s*Tccout^7);

ah=0;bh=0;ch=0;dh=0;e1h=6.24698837e2;e2h=-

2.34385369e3;

e3h=-9.50812101e3;e4h=7.16287928e4;e5h=-

1.63535221e5;

e6h=1.66531093e5;e7h=-6.47854585e4;hfcr=2.0993e3; as=0;bs=0;cs=0;ds=0;e1s=-

1.83692956e3;e2s=1.47066352e4;

e3s=-4.31466046e4;e4s=4.86066733e4;e5s=7.9975096e3;

e6s=-5.83339887e4;e7s=3.31400718e4;sfcr=4.4289;

else

(Tcinlet+273.15)>=300&(Tcinlet+273.15)<600;

ah=8.839230108e-1;bh=0;ch=0;dh=0;e1h=-

2.67172935;e2h=6.22640035;

e3h=-1.31789573e1;e4h=-1.91322436;e5h=6.87937653e1;

e6h=-1.24819906e2;e7h=7.21435404e1;hfcr=2.0993e3;

as=9.12762917e-1;bs=0;cs=0;ds=0;e1s=-

1.75702956;e2s=1.68754095;

e3s=5.82215341;e4s=-6.33354786e1;e5s=1.88076546e2;

e6s=-2.52344531e2;e7s=1.28058531e2;sfcr=4.4289;

end

Tccinlet=(Tcr-(Tcinlet+273.15))/Tcr;

hcinlet=1000*hfcr*(ah+bh*Tccinlet^(1/3)+ch*Tccinlet^(5/

6) + dh * Tccinlet ^ (7/8) + e1h * Tccinlet + e2h * Tccinlet ^ 2 + e3h * Tccinl

et^3+e4h*Tccinlet^4+e5h*Tccinlet^5+e6h*Tccinlet^6+e7h*Tccin
let^7);

scinlet=1000*sfcr*(as+bs*Tccinlet^(1/3)+cs*Tccinlet^(5/ 6)+ds*Tccinlet^(7/8)+e1s*Tccinlet+e2s*Tccinlet^2+e3s*Tccinl et^3+e4s*Tccinlet^4+e5s*Tccinlet^5+e6s*Tccinlet^6+e7s*Tccin let^7);

if (Ts+273.15)>=273.15&(Ts+273.15)<300

as=0;bs=0;cs=0;ds=0;e1s=-

1.83692956e3;e2s=1.47066352e4;

e3s=-4.31466046e4;e4s=4.86066733e4;e5s=7.9975096e3;

e6s=-5.83339887e4;e7s=3.31400718e4;sfcr=4.4289;

else

(Ts+273.15)>=300&(Ts+273.15)<600;

as=9.12762917e-1;bs=0;cs=0;ds=0;e1s=-

1.75702956;e2s=1.68754095;

e3s=5.82215341;e4s=-6.33354786e1;e5s=1.88076546e2;

e6s=-2.52344531e2;e7s=1.28058531e2;sfcr=4.4289;

end

Tcs=(Tcr-(Ts+273.15))/Tcr;

hs=1000*(4.2045016*Ts-0.0678226*C*Ts+1.47532e-3*C^2*Ts-

6.8002552e-4*Ts^2+3.095114e-4*C*Ts^2-0.1624438e-

4*C^2*Ts^2+5.3015464e-6*Ts^3-1.6853152e-

6*C*Ts^3+0.0853674e-6*C^2*Ts^3-0.04881);

ssw=1000*sfcr*(as+bs*Tcs^(1/3)+cs*Tcs^(5/6)+ds*Tcs^(7/8
)+e1s*Tcs+e2s*Tcs^2+e3s*Tcs^3+e4s*Tcs^4+e5s*Tcs^5+e6s*Tcs^6
+e7s*Tcs^7);

Xs=1.42185e-3-3.1337e-7*Ts+4.2446e-9*Ts^2; Ys=-2.1762e-4+4.1426e-7*Ts-1.6285e-9*Ts^2; Zs=1.0201e-5+1.5903e-8*Ts-2.3525e-10*Ts^2; sss=(Xs*(C*10)+Ys*(C*10)^1.5+Zs*(C*10)^2)/1000;

massofsalt=mw*C/100;

m water=mw-massofsalt;

ss=(m water/mw) *ssw+(massofsalt/mw) *sss;

%******Saline water-old values****************

if (Tsold+273.15) >=273.15& (Tsold+273.15) <300

as=0;bs=0;cs=0;ds=0;e1s=-

1.83692956e3;e2s=1.47066352e4;

e3s=-4.31466046e4;e4s=4.86066733e4;e5s=7.9975096e3;

e6s=-5.83339887e4;e7s=3.31400718e4;sfcr=4.4289;

else

(Tsold+273.15)>=300&(Tsold+273.15)<600;

as=9.12762917e-1;bs=0;cs=0;ds=0;e1s=-

1.75702956;e2s=1.68754095;

e3s=5.82215341;e4s=-6.33354786e1;e5s=1.88076546e2;

e6s=-2.52344531e2;e7s=1.28058531e2;sfcr=4.4289;

end

Tcsold=(Tcr-(Tsold+273.15))/Tcr;

hsold=1000*(4.2045016*Tsold-0.0678226*C*Tsold+1.47532e-3*C^2*Tsold-6.8002552e-4*Tsold^2+3.095114e-4*C*Tsold^2-0.1624438e-4*C^2*Tsold^2+5.3015464e-6*Tsold^3-1.6853152e-6*C*Tsold^3+0.0853674e-6*C^2*Tsold^3-0.04881);

ssw=1000*sfcr*(as+bs*Tcsold^(1/3)+cs*Tcsold^(5/6)+ds*Tc sold^(7/8)+e1s*Tcsold+e2s*Tcsold^2+e3s*Tcsold^3+e4s*Tcsold^ 4+e5s*Tcsold^5+e6s*Tcsold^6+e7s*Tcsold^7);

Xs=1.42185e-3-3.1337e-7*Tsold+4.2446e-9*Tsold^2;

Ys=-2.1762e-4+4.1426e-7*Tsold-1.6285e-9*Tsold^2;

Zs=1.0201e-5+1.5903e-8*Tsold-2.3525e-10*Tsold^2;

sss=(Xs*(C*10)+Ys*(C*10)^1.5+Zs*(C*10)^2)/1000;

massofsalt=mwater*C/100;

m water=mwater-massofsalt;

ssold=(m water/mwater)*ssw+(massofsalt/mwater)*sss;

Td=Ts-(q/(Qw*Ds*Cp));

if (Td+273.15)>=273.15&(Td+273.15)<300

as=0;bs=0;cs=0;ds=0;e1s=-

1.83692956e3;e2s=1.47066352e4;

e3s=-4.31466046e4;e4s=4.86066733e4;e5s=7.9975096e3;

e6s=-5.83339887e4;e7s=3.31400718e4;sfcr=4.4289;

else

(Td+273.15)>=300&(Td+273.15)<600;

1.75702956;e2s=1.68754095;

e3s=5.82215341;e4s=-6.33354786e1;e5s=1.88076546e2;

e6s=-2.52344531e2;e7s=1.28058531e2;sfcr=4.4289;

end

Tcd=(Tcr-(Td+273.15))/Tcr;

hd=1000*(4.2045016*Td-0.0678226*C*Td+1.47532e-3*C^2*Td-

6.8002552e-4*Td^2+3.095114e-4*C*Td^2-0.1624438e-

4*C^2*Td^2+5.3015464e-6*Td^3-1.6853152e-

6*C*Td^3+0.0853674e-6*C^2*Td^3-0.04881);

sdw=1000*sfcr*(as+bs*Tcd^(1/3)+cs*Tcd^(5/6)+ds*Tcd^(7/8

)+e1s*Tcd+e2s*Tcd^2+e3s*Tcd^3+e4s*Tcd^4+e5s*Tcd^5+e6s*Tcd^6 +e7s*Tcd^7);

Xs=1.42185e-3-3.1337e-7*Td+4.2446e-9*Td^2;

Ys=-2.1762e-4+4.1426e-7*Td-1.6285e-9*Td^2;

Zs=1.0201e-5+1.5903e-8*Td-2.3525e-10*Td^2;

sss=(Xs*(C*10)+Ys*(C*10)^1.5+Zs*(C*10)^2)/1000;

massofsalt=mw*C/100;

m water=mw-massofsalt;

sd=(m water/mw) *sdw+(massofsalt/mw) *sss;

%*******inlet to tube-in-tube heat exchanger******

if (T0+273.15)>=273.15&(T0+273.15)<300

as=0;bs=0;cs=0;ds=0;e1s=-

1.83692956e3;e2s=1.47066352e4;
e3s=-4.31466046e4;e4s=4.86066733e4;e5s=7.9975096e3;

e6s=-5.83339887e4;e7s=3.31400718e4;sfcr=4.4289;

else

(T0+273.15) >= 300 & (T0+273.15) < 600;

as=9.12762917e-1;bs=0;cs=0;ds=0;e1s=-

1.75702956;e2s=1.68754095;

e3s=5.82215341;e4s=-6.33354786e1;e5s=1.88076546e2;

```
e6s=-2.52344531e2;e7s=1.28058531e2;sfcr=4.4289;
```

end

TcO=(Tcr-(T0+273.15))/Tcr;

hinlet=1000*(4.2045016*T0-0.0678226*C0*T0+1.47532e-

3*C0^2*T0-6.8002552e-4*T0^2+3.095114e-4*C0*T0^2-0.1624438e-4*C0^2*T0^2+5.3015464e-6*T0^3-1.6853152e-

6*C0*T0^3+0.0853674e-6*C0^2*T0^3-0.04881);

sinletw=1000*sfcr*(as+bs*Tc0^(1/3)+cs*Tc0^(5/6)+ds*Tc0^
(7/8)+e1s*Tc0+e2s*Tc0^2+e3s*Tc0^3+e4s*Tc0^4+e5s*Tc0^5+e6s*T
c0^6+e7s*Tc0^7);

Xs=1.42185e-3-3.1337e-7*T0+4.2446e-9*T0^2; Ys=-2.1762e-4+4.1426e-7*T0-1.6285e-9*T0^2; Zs=1.0201e-5+1.5903e-8*T0-2.3525e-10*T0^2; sss=(Xs*(C0*10)+Ys*(C0*10)^1.5+Zs*(C0*10)^2)/1000; massofsalt=mi*C/100; m_water=mi-massofsalt; sinlet=(m_water/mi)*sinletw+(massofsalt/mi)*sss; if (Ti+273.15)>=273.15&(Ti+273.15)<300

as=0;bs=0;cs=0;ds=0;e1s=-

1.83692956e3;e2s=1.47066352e4;

e3s=-4.31466046e4;e4s=4.86066733e4;e5s=7.9975096e3;

e6s=-5.83339887e4;e7s=3.31400718e4;sfcr=4.4289;

else

(Ti+273.15)>=300&(Ti+273.15)<600;

as=9.12762917e-1;bs=0;cs=0;ds=0;e1s=-

1.75702956;e2s=1.68754095;

e3s=5.82215341;e4s=-6.33354786e1;e5s=1.88076546e2;

e6s=-2.52344531e2;e7s=1.28058531e2;sfcr=4.4289;

end

Tci=(Tcr-(Ti+273.15))/Tcr;

hi=1000*(4.2045016*Ti-0.0678226*C0*Ti+1.47532e-

3*C0^2*Ti-6.8002552e-4*Ti^2+3.095114e-4*C0*Ti^2-0.1624438e-4*C0^2*Ti^2+5.3015464e-6*Ti^3-1.6853152e-

6*C0*Ti^3+0.0853674e-6*C0^2*Ti^3-0.04881);

siw=1000*sfcr*(as+bs*Tci^(1/3)+cs*Tci^(5/6)+ds*Tci^(7/8
)+e1s*Tci+e2s*Tci^2+e3s*Tci^3+e4s*Tci^4+e5s*Tci^5+e6s*Tci^6
+e7s*Tci^7);

Xs=1.42185e-3-3.1337e-7*Ti+4.2446e-9*Ti^2; Ys=-2.1762e-4+4.1426e-7*Ti-1.6285e-9*Ti^2; Zs=1.0201e-5+1.5903e-8*Ti-2.3525e-10*Ti^2; sss=(Xs*(C0*10)+Ys*(C0*10)^1.5+Zs*(C0*10)^2)/1000;

massofsalt=mi*C/100;

m water=mi-massofsalt;

si=(m water/mi) *siw+(massofsalt/mi) *sss;

if (Tif+273.15)>=273.15&(Tif+273.15)<300

ah=0;bh=0;ch=0;dh=0;e1h=6.24698837e2;e2h=-

2.34385369e3;

e3h=-9.50812101e3;e4h=7.16287928e4;e5h=-

1.63535221e5;

e6h=1.66531093e5;e7h=-6.47854585e4;hfcr=2.0993e3;

as=0;bs=0;cs=0;ds=0;e1s=-

1.83692956e3;e2s=1.47066352e4;

e3s=-4.31466046e4;e4s=4.86066733e4;e5s=7.9975096e3;

e6s=-5.83339887e4;e7s=3.31400718e4;sfcr=4.4289;

else

```
(Tif+273.15)>=300&(Tif+273.15)<600;
```

ah=8.839230108e-1;bh=0;ch=0;dh=0;e1h=-

2.67172935;e2h=6.22640035;

e3h=-1.31789573e1;e4h=-1.91322436;e5h=6.87937653e1;

e6h=-1.24819906e2;e7h=7.21435404e1;hfcr=2.0993e3;

as=9.12762917e-1;bs=0;cs=0;ds=0;e1s=-

1.75702956;e2s=1.68754095;

e3s=5.82215341;e4s=-6.33354786e1;e5s=1.88076546e2;

e6s=-2.52344531e2;e7s=1.28058531e2;sfcr=4.4289;

end

Tcif=(Tcr-(Tif+273.15))/Tcr;

hif=1000*hfcr*(ah+bh*Tcif^(1/3)+ch*Tcif^(5/6)+dh*Tcif^(
7/8)+e1h*Tcif+e2h*Tcif^2+e3h*Tcif^3+e4h*Tcif^4+e5h*Tcif^5+e
6h*Tcif^6+e7h*Tcif^7);

sif=1000*sfcr*(as+bs*Tcif^(1/3)+cs*Tcif^(5/6)+ds*Tcif^(
7/8)+e1s*Tcif+e2s*Tcif^2+e3s*Tcif^3+e4s*Tcif^4+e5s*Tcif^5+e
6s*Tcif^6+e7s*Tcif^7);

ahv=1;bhv=4.57874342e-1;chv=5.08441288;dhv=-1.48513244;

e1hv=-4.81351884;e2hv=2.69411792;e3hv=-7.39064542;

e4hv=1.04961689e1;e5hv=-

5.46840036;e6hv=0;e7hv=0;hgcr=2.0993e3;

asv=1;bsv=3.77391e-1;csv=-2.78368;dsv=6.93135;

elsv=-4.34839;e2sv=1.34672;e3sv=1.75261;e4sv=-6.22295;

e5sv=9.99004;e6sv=0;e7sv=0;sgcr=4.4289;

Tcv=(Tcr-(Ts+273.15))/Tcr;

hv=1000*hgcr*(ahv+bhv*Tcv^(1/3)+chv*Tcv^(5/6)+dhv*Tcv^(

7/8)+e1hv*Tcv+e2hv*Tcv^2+e3hv*Tcv^3+e4hv*Tcv^4+e5hv*Tcv^5+e 6hv*Tcv^6+e7hv*Tcv^7);

sv=1000*sgcr*(asv+bsv*Tcv^(1/3)+csv*Tcv^(5/6)+dsv*Tcv^(7/8)+e1sv*Tcv+e2sv*Tcv^2+e3sv*Tcv^3+e4sv*Tcv^4+e5sv*Tcv^5+e 6sv*Tcv^6+e7sv*Tcv^7); ahr=0;bhr=0;chr=0;dhr=0;e1hr=6.24698837e2;e2hr=-

2.34385369e3;

e3hr=-9.50812101e3;e4hr=7.16287928e4;e5hr=-

1.63535221e5;

e6hr=1.66531093e5;e7hr=-6.47854585e4;hfcr=2.0993e3;

asr=0;bsr=0;csr=0;dsr=0;e1sr=-

1.83692956e3;e2sr=1.47066352e4;

e3sr=-4.31466046e4;e4sr=4.86066733e4;e5sr=7.9975096e3;

e6sr=-5.83339887e4;e7sr=3.31400718e4;sfcr=4.4289;

Tcr=647.3;

Tcr=(Tcr-(T0+273.15))/Tcr;

hr=1000*hfcr*(ahr+bhr*Tcr^(1/3)+chr*Tcr^(5/6)+dhr*Tcr^(

7/8) +e1hr*Tcr+e2hr*Tcr^2+e3hr*Tcr^3+e4hr*Tcr^4+e5hr*Tcr^5+e
6hr*Tcr^6+e7hr*Tcr^7);

sr=1000*sfcr*(asr+bsr*Tcr^(1/3)+csr*Tcr^(5/6)+dsr*Tcr^(

7/8) +e1sr*Tcr+e2sr*Tcr^2+e3sr*Tcr^3+e4sr*Tcr^4+e5sr*Tcr^5+e
6sr*Tcr^6+e7sr*Tcr^7);

ecout=(hcout-hr)-T0*(scout-sr);

ecinlet=(hcinlet-hr)-T0*(scinlet-sr);

es=(hs-hr)-T0*(ss-sr);

ed=(hd-hr)-T0*(sd-sr);

einlet=(hinlet-hr)-T0*(sinlet-sr);

```
ei=(hi-hr)-T0*(si-sr);
  eif=(hif-hr)-T0*(sif-sr);
  ev=(hv-hr)-T0*(sv-sr);
  E1=mwater*((hs-hr)-T0*(ss-sr));
  E0=mwater*((hsold-hr)-T0*(ssold-sr));
  Qcollectoruseful=Qcollectoruseful;
  aaa=m*(ecout-ecinlet);
  Ed=m*(ecout-ecinlet)+mi*ei-mw*es-mf*ev-(1-
TO/Twside) *Qlossside-(1-TO/Twbottom) *Qlossbottom-(1-
T0/Twtop) *Qlosstop-(E1-E0)/dt;
  Ed tube=mw*(es-ed)+mi*(einlet-ei);
  Ed he=m*(ecout-ecinlet)+(1-T0/The)*(-Qcollectoruseful);
  Ed cond=mf*(ev-eif)+(1-T0/Tco)*(-mf*hfgm);
  eff2=(mf*hfg)/(m*(ecout-ecinlet));
  eff1=mf*hfq/Qcollectoruseful;
  end
  ۹٫۰۰۰ ۲٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰ ۹٫۰۰۰
```

Cp=a2*C+b;

Ps=exp(63.042-7139.6/(Ts+273)-6.2558*log(Ts+273))*100; Pf=exp(63.042-7139.6/(Tf+273)-6.2558*log(Tf+273))*100; dp=0.5*Dv*((mf/(Dv*Af))^2-

(mf/(Dv*As))^2)+Dv*((klc+kle)/2)*(mf/(Dv*As))^2;

```
Ds=D0*(1-bt*(Ts-T0)+bc*(C-C0));
```

hfg=2.3246e3*(1.0727e3-1.0167*Ts+1.4087e-4*Ts^2-

```
5.1462e-6*Ts^3);
```

fc=1-a1*C;

qe=(am/Df)*((fc*Ps)/(Ts+273)^0.5-(Pf+dp)/(Tf+273)^0.5);

Qe=qe*As;

mf=Qe*Df;

he=mf*hfg;

ex***

```
Taverageside=(Twside+Ta)/2;
```

Bside=1/(Taverageside+273);

Raside=g*Bside*(Twside-Ta)*lside^3/(alphaa*nua);

```
Nuside=0.68+(0.67*Raside^0.25)/(1+(0.492/Pr)^(9/16))^(4
```

/9);

```
hside=Nuside*ka/lside;
X1side=1/(hside*2*pi*rs3*lside);
```

```
X2side=log(rs2/rs)/(2*pi*ksteel*lside)+log(rs3/rs2)/(2*
pi*kinsulation*lside);
```

```
Y1side=(Twside-Ta)/X1side;
```

```
Y2side=(Ts-Twside)/X2side;
```

while abs (Y1side-Y2side)>0.1

if Y1side>Y2side

Twside=Twside-0.05;

else

Twside=Twside+0.1;

end

Taverageside=(Twside+Ta)/2;

Bside=1/(Taverageside+273);

Raside=g*Bside*(Twside-Ta)*lside^3/(alphaa*nua);

```
Nuside=0.68+(0.67*Raside^0.25)/(1+(0.492/Pr)^(9/16))^(
```

4/9);

```
hside=Nuside*ka/lside;
```

```
X1side=1/(hside*2*pi*rs3*lside);
```

```
X2side=log(rs2/rs)/(2*pi*ksteel*lside)+log(rs3/rs2)/(2
```

*pi*kinsulation*lside);

Y1side=(Twside-Ta)/X1side;

Y2side=(Ts-Twside)/X2side;

end

Qlossside=Aside*hside*(Twside-Ta);

Taveragebottom=(Twbottom+Ta)/2;

Bbottom=1/(Taveragebottom+273);

Rabottom=g*Bbottom*(Twbottom-

Ta) *lbottom^3/(alphaa*nua);

Nubottom=0.27*Rabottom^0.25;

hbottom=Nubottom*ka/lbottom;

X1bottom=1/(hbottom*As);

X2bottom=tbottom/(ksteel*As)+tinsulation/(kinsulation*A

s);

Y1bottom=(Twbottom-Ta)/X1bottom;

Y2bottom=(Ts-Twbottom)/X2bottom;

while abs (Y1bottom-Y2bottom)>0.1

if Y1bottom>Y2bottom

Twbottom=Twbottom-0.05;

else

Twbottom=Twbottom+0.1;

end

Taveragebottom=(Twbottom+Ta)/2;

Bbottom=1/(Taveragebottom+273);

Rabottom=g*Bbottom* (Twbottom-

Ta) *lbottom^3/ (alphaa*nua);

Nubottom=0.27*Rabottom^0.25;

hbottom=Nubottom*ka/lbottom;

X1bottom=1/(hbottom*As);

```
X2bottom=tbottom/(ksteel*As)+tinsulation/(kinsulation*
```

As);

```
Y1bottom=(Twbottom-Ta)/X1bottom;
```

```
Y2bottom=(Ts-Twbottom)/X2bottom;
```

end

```
Qlossbottom=As*hbottom*(Twbottom-Ta);
```

Taveragetop=(Twtop+Ta)/2;

Btop=1/(Taveragetop+273);

Ratop=g*Btop*(Twtop-Ta)*ltop^3/(alphaa*nua);

```
Nutop=0.56*(Ratop*cos(theta))^0.25;
```

htop=Nutop*ka/ltop;

X1top=1/(htop*Atop);

X2top=ttop/(ksteel*Atop)+tinsulation/(kinsulation*Atop)

;

```
Y1top=(Twtop-Ta)/X1top;
```

```
Y2top=(Ts-Twtop)/X2top;
```

while abs (Y1top-Y2top)>0.1

if Y1top>Y2top

Twtop=Twtop-0.05;

else

Twtop=Twtop+0.1;

end

Taveragetop=(Twtop+Ta)/2;

Btop=1/(Taveragetop+273); Ratop=g*Btop*(Twtop-Ta)*ltop^3/(alphaa*nua); Nutop=0.56*(Ratop*cos(theta))^0.25; htop=Nutop*ka/ltop; Xltop=1/(htop*Atop); X2top=ttop/(ksteel*Atop)+tinsulation/(kinsulation*Atop)

);

Y1top=(Twtop-Ta)/X1top;

Y2top=(Ts-Twtop)/X2top;

end

Qlosstop=Atop*htop*(Twtop-Ta);

Qloss=Qlossside+Qlossbottom+Qlosstop;

%****Solution of the set of coupled equations to find the new****

%****values of temperature, density, specific heat and solute****

laminar or not

```
Tfa=(Tco+Ta)/2;
ka=1.163*(0.021+(0.66e-4)*Tfa);
ua=(1/3600)*(6.1848e-2+(1.6632e-4)*Tfa);
Da=353.44/(Tfa+273.15);
```

Cpa=4186*(0.2317+1.6213e-5*Tfa+3.9593e-8*Tfa^2-1.6213e-11*Tfa^3);

```
Pr=ua*Cpa/ka;
```

mf=mf;

hfgm=hfg+(3/8)*Cpf*(Ts-Tif);

hc=0.555*((g*Df*(Df-Dv)*kf^3*hfgm)/(uf*dco*(Ts-

Tif)))^0.25;

```
coeff=1/(2*pi*rci*lc*hc)+log(rco/rci)/(2*pi*lc*kc);
```

```
Tif=(1/(1+0.68*mf*Cpf*coeff))*(mf*coeff*hfg+0.68*mf*Cpf
```

*Ts*coeff+Tco);

```
Tci=mf*(hfg+0.68*Cpf*(Ts-
Tif))*log(rco/rci)/(2*pi*lc*kc)+Tco;
xi=dco/Dfin;
B=1/(Tfa+273);
Tco=Tco;
RaS=(g*B*(Tco-Ta)*Pr*S^3/(nua)^2)*(S/Dfin);
Beta=(0.17*xi)+exp(-4.8*xi);
CC=((23.7-1.1*(1+(152*xi^2))^0.5)/(1+Beta))^(4/3);
hco=(ka/S)*(RaS/(12*pi))*(2-exp(-(CC/RaS)^0.75)-exp(-
```

```
Beta*(CC/RaS)^0.75));
```

```
CCC=0.44+0.12/xi;
hcot=(ka/S)*(CCC*RaS^0.29);
rfin=Dfin/2;
r2c=(Dfin/2)+(tfin/2);
Aftip=N*pi*Dfin*tfin;
Afside=N*2*pi*(rfin^2-rco^2);
Ab=2*pi*rco*(lc-N*tfin);
mm=sqrt((2*hco)/(kc*tfin));
C2=(2*rco/mm)/(r2c^2-rco^2);
```

fineff=C2*(BESSELK(1, (mm*rco))*BESSELI(1, (mm*r2c))BESSELI(1, (mm*rco))*BESSELK(1, (mm*r2c)))/(BESSELI(0, (mm*rco))
*BESSELK(1, (mm*r2c))+BESSELK(0, (mm*rco))*BESSELI(1, (mm*r2
c)));

constant=N*fineff*hcot*Aftip+N*fineff*hco*Afside+hco*Ab
;
Tcol=(1/((log(rco/rci)*constant)+2*pi*lc*kc))*(2*pi*lc*

kc*Tci+log(rco/rci)*Ta*constant);

while abs(Tco1-Tco)>0.0001

Tco=Tco1;

Tfa=(Tco+Ta)/2;

ka=1.163*(0.021+(0.66e-4)*Tfa);

ua=(1/3600) * (6.1848e-2+(1.6632e-4) *Tfa);

Da=353.44/(Tfa+273.15);

Cpa=4186*(0.2317+1.6213e-5*Tfa+3.9593e-8*Tfa^2-

1.6213e-11*Tfa^3);

Pr=ua*Cpa/ka;

hfgm=hfg+(3/8)*Cpf*(Ts-Tif);

hc=0.555*((g*Df*(Df-Dv)*kf^3*hfgm)/(uf*dco*(Ts-

Tif)))^0.25;

```
coeff=1/(2*pi*rci*lc*hc)+log(rco/rci)/(2*pi*lc*kc);
```

```
Tif=(1/(1+0.68*mf*Cpf*coeff))*(mf*coeff*hfg+0.68*mf*Cp
```

f*Ts*coeff+Tco);

Tci=mf*(hfg+0.68*Cpf*(Ts-

Tif))*log(rco/rci)/(2*pi*lc*kc)+Tco;

xi=dco/Dfin;

Tfa=(Tco+Ta)/2;

B=1/(Tfa+273);

```
RaS=(g*B*(Tco-Ta)*Pr*S^3/(nua)^2)*(S/Dfin);
Beta=(0.17*xi)+exp(-4.8*xi);
CC=((23.7-1.1*(1+(152*xi^2))^0.5)/(1+Beta))^(4/3);
hco=(ka/S)*(RaS/(12*pi))*(2-exp(-(CC/RaS)^0.75)-
exp(-Beta*(CC/RaS)^0.75));
```

CCC=0.44+0.12/xi;

 $hcot=(ka/S)*(CCC*RaS^{0.29});$

rfin=Dfin/2;

r2c=(Dfin/2)+(tfin/2);

Aftip=N*pi*Dfin*tfin;

Afside=N*2*pi*(rfin^2-rco^2);

Ab=2*pi*rco*(lc-N*tfin);

mm=sqrt((2*hco)/(kc*tfin));

C2=(2*rco/mm)/(r2c^2-rco^2);

fineff=C2*(BESSELK(1, (mm*rco))*BESSELI(1, (mm*r2c))-

```
BESSELI(1, (mm*rco))*BESSELK(1, (mm*r2c)))/(BESSELI(0, (mm*rco))*BESSELK(1, (mm*r2c))+BESSELK(0, (mm*rco))*BESSELI(1, (mm*r2c)));
```

constant=N*fineff*hcot*Aftip+N*fineff*hco*Afside+hco*Ab

```
;
```

```
Tcol=(1/((log(rco/rci)*constant)+2*pi*lc*kc))*(2*pi*lc*
kc*Tci+log(rco/rci)*Ta*constant);
```

end

Tco=Tco1;

APPENDIX C PROGRAM FLOW CHART



APPENDIX D PHYSICAL PROPERTIES

This appendix provides the relations used to estimate the physical properties of fresh water, vapor, saline water, and air as a function of temperature and salinity (in case of saline water).

Water:

Vapor pressure [67]:

$$P(T) = \exp\left(63.042 - \frac{7139.6}{(T+273.15)} - 6.2558\ln(T+273.15)\right) * 100$$
(D.1)

Latent heat of water [67]:

$$h_{fg}(T) = 1000 * (3146 - 2.36(T + 273))$$
 (D.2)

Specific heat [89]:

$$C_{p} = 4.2174 - 3.720283 * 10^{-3}T + 1.412855 * 10^{-4}T^{2}$$

$$- 2.654387 * 10^{-6}T^{3} + 2.093236 * 10^{-8}T^{4}$$
(D.3)

Density [89]:

$$\rho = (999.83952 + 16.945176T - 7.9870401 * 10^{-3}T^2 - 46.170461 * 10^{-6}T^3 + 105.56302 * 10^{-9}T^4 - 280.5423 * 10^{-12}T^5)/(1 + 16.879850 * 10^{-3}T)$$

(D.4)

Enthalpy [90]:

$$Y(S) = A + BT(C)^{\frac{1}{3}} + CT(C)^{\frac{5}{6}} + DT(C)^{\frac{7}{8}} + \sum_{N=1}^{7} E(N)T(C)^{N}$$
(D.5)

$$T(C) = \frac{\left[T(CR) - T(S)\right]}{T(CR)}$$
$$T(CR) = 647.3K$$
$$Y(S) = \frac{h(F)}{h(FCR)}$$

Range I

$$273.15 \le T(S) < 300K$$

$$A = 0, B = 0, C = 0, D = 0$$

- E(1) = 6.24698837E2
- E(2) = -2.34385369E3
- E(3) = -9.50812101E3
- E(4) = 7.16287928E4
- E(5) = -1.63535221E5
- E(6) = 1.66531093E5
- E(7) = -6.47854585E4
- h(FCR) = 2.0993E3

Range II

$$300 \le T(S) < 600K$$

$$A = 8.839230108E - 1$$

- B=0, C=0, D=0
- E(1) = -2.67172935
- E(2) = 6.22640035
- E(3) = -1.31789573E1

$$E(4) = -1.91322436$$

 $E(5) = 6.87937653E1$
 $E(6) = -1.24819906E2$
 $E(7) = 7.21435404E1$
 $h(FCR) = 2.0993E3$

Entropy [90]:

$$Y(S) = A + BT(C)^{\frac{1}{3}} + CT(C)^{\frac{5}{6}} + DT(C)^{\frac{7}{8}} + \sum_{N=1}^{7} E(N)T(C)^{N}$$
(D.6)

$$T(C) = \frac{\left[T(CR) - T(S)\right]}{T(CR)}$$

T(CR) = 647.3K

$$Y(S) = \frac{s(F)}{s(FCR)}$$

Range I

$$273.15 \le T(S) < 300K$$

$$A = 0, B = 0, C = 0, D = 0$$

E(1) = -1.83692956E3

$$E(2) = 1.47066352E4$$

- E(3) = -4.31466046E4
- E(4) = 4.86066733E4
- E(5) = 7.9975096E3
- E(6) = -5.83339887E4
- E(7) = 3.31400718E4

$$s(FCR) = 4.4289$$

Range II

 $300 \le T(S) < 600K$

A = 9.12762917E - 1

$$B=0, C=0, D=0$$

E(1) = -1.75702956

$$E(2) = 1.68754095$$

E(3) = 5.82215341

E(4) = -6.33354786E1

E(5) = 1.88076546E2

E(6) = -2.52344531E2

$$E(7) = 1.28058531E2$$

s(FCR) = 4.4289

Vapor:

Enthalpy [90]:

$$Y(S) = A + BT(C)^{\frac{1}{3}} + CT(C)^{\frac{5}{6}} + DT(C)^{\frac{7}{8}} + \sum_{N=1}^{7} E(N)T(C)^{N}$$
(D.7)

$$T(C) = \frac{\left[T(CR) - T(S)\right]}{T(CR)}$$

T(CR) = 647.3K

$$Y(S) = \frac{h(G)}{h(GCR)}$$

 $273.15 \le T(S) < 647.3K$

$$A = 1$$

$$B = 4.57874342E - 1$$

$$C = 5.08441288$$

$$D = -1.48513244$$

$$E(1) = -4.81351884$$

$$E(2) = 2.69411792$$

$$E(3) = -7.39064542$$

$$E(4) = 1.04961689E1$$

$$E(5) = -5.46840036$$

E(6) = 0

E(7) = 0

$$h(GCR) = 2.0993E3$$

Entropy [90]:

$$Y(S) = A + BT(C)^{\frac{1}{3}} + CT(C)^{\frac{5}{6}} + DT(C)^{\frac{7}{8}} + \sum_{N=1}^{7} E(N)T(C)^{N}$$
(D.8)

$$T(C) = \frac{[T(CR) - T(S)]}{T(CR)}$$

$$T(CR) = 647.3K$$

$$Y(S) = \frac{s(G)}{s(GCR)}$$

$$273.15 \le T(S) < 647.3K$$

$$A = 1$$

$$B = 3.77391E - 1$$

$$C = -2.78368$$

D = 6.93135 E(1) = -4.34839 E(2) = 1.34672 E(3) = 1.75261 E(4) = -6.22295 E(5) = 9.99004 E(6) = 0 E(7) = 0s(GCR) = 4.4289

Seawater:

The vapor pressure and latent heat of evaporation of seawater is almost identical to that of fresh water [87].

Specific heat [91]:

$$C_{p} = 4.2045016 - 0.0678226C + 1.47532 * 10^{-3} C^{2} - \left[(13.60051 - 6.190228C + 0.3248876C^{2}) * 10^{-4} T \right]$$
(D.9)
+ $\left[(15.904639 - 5.0559456C + 0.2561026C^{2}) * 10^{-6} T^{2} \right]$

If the temperature effect is neglected, then the specific heat is given as [66]

$$C_p = \alpha_2 C + \beta_2 \tag{D.10}$$

Density [67]:

$$\rho = \rho_0 (1 - \beta_T \Delta T_0 + \beta_C \Delta C_0) \tag{D.11}$$

Enthalpy:

The enthalpy of seawater is a function of temperature, pressure, and composition. The effect of pressure is given by [89]

$$\left(\frac{\partial h}{\partial P}\right)_{T} = V - T \left(\frac{\partial V}{\partial T}\right)_{P} \tag{D.12}$$

Numerical values for $\left(\frac{\partial h}{\partial P}\right)_T$ show that the enthalpy is not very dependent on pressure [89]. From Maxwell's relations,

$$\left(\frac{\partial h}{\partial T}\right)_p = C_p \tag{D.13}$$

Since the specific heat is given as a function of temperature and salinity, eq. D.9, substituting in eq. D.13 and carrying out the integration we get the following expression for the enthalpy,

$$h = \begin{cases} 4.2045016T - 0.0678226CT + 1.47532 * 10^{-3} C^{2}T \\ - \left[(6.8002552 - 3.095114C + 0.1624438C^{2}) * 10^{-4} T^{2} \right] \\ + \left[(5.3015464 - 1.6853152C + 0.0853674C^{2}) * 10^{-6} T^{3} \right] - 0.04881 \end{cases} * 1000$$
(D.14)

Entropy:

The entropy of seawater is a function of temperature, pressure, and composition [89],

$$S = f(T, P, n_i) \tag{D.15}$$

The change in entropy can be expressed as

$$dS = \left(\frac{\partial S}{\partial T}\right)_{P} dT + \left(\frac{\partial S}{\partial P}\right)_{T} dP + \sum_{i} \left(\frac{\partial S}{\partial n_{i}}\right)_{T,P} dn_{i}$$
(D.16)

Temperature effect is related to heat capacity by

$$\left(\frac{\partial S}{\partial T}\right)_{p} = \frac{C_{p}}{T} \tag{D.17}$$

Pressure effect is related to expansibility by

$$\left(\frac{\partial S}{\partial P}\right)_T = -\left(\frac{\partial V}{\partial T}\right)_P \tag{D.18}$$

Since the water can be treated as an incompressible substance in the operating temperature range (upto about 50 °C), the pressure effect may be neglected.

At a constant pressure and temperature, the changes in entropy is a function of composition,

$$dS = \bar{S}_1 \, dn_1 + \bar{S}_2 \, dn_2 \tag{D.19}$$

where, $\bar{S_1}$ is the partial molal entropy of water in seawater and $\bar{S_2}$ is the partial molal entropy of sea salt.

Since both temperature and salinity will vary, the entropy of seawater was taken as a function of these two variables. Its numerical value was taken as the summation of the water and salt entropies. The entropy of water in the seawater can be calculated from eq. D.6. The specific relative entropy of sea salt in seawater can be calculated from the following equation [89],

$$s = XC + YC^{1.5} + ZC^{2}$$
(D.20)

$$X = 1.42185 * 10^{-3} - 3.1337 * 10^{-7}T + 4.2446 * 10^{-9}T^{2}$$

$$Y = -2.1762 * 10^{-4} + 4.1426 * 10^{-7}T - 1.6285 * 10^{-9}T^{2}$$

$$Z = 1.0201 * 10^{-5} + 1.5903 * 10^{-8}T - 2.3525 * 10^{-10}T^{2}$$

$$C \text{ is in ppm.}$$
Air:

Specific heat [50]:

$$C_p = 0.9992 * 10^3 + 0.14319T_f + 1.1010 * 10^{-4}T_f^2 - 6.7851 * 10^{-8}T_f^3$$
(D.21)

Density [49]:

$$\rho = \frac{353.44}{\left(T_f + 273.15\right)} \tag{D.22}$$

Viscosity [50]:

$$\mu = 1.718 * 10^{-5} + 4.62 * 10^{-8} T_f$$
(D.23)

Conductivity [50]:

$$k = 0.0244 + 0.7673 * 10^{-4} T_f$$
 (D.24)

In the above equation T_f is given as

$$T_f = \frac{T_a + T_{co}}{2} \tag{D.25}$$

APPENDIX E EVAPORATOR DESIGN

The evaporator was designed according to ASME standards. The evaporator crosssection and height were selected. The maximum allowable external pressure for a given shell thickness of a certain material is given by

$$P = \frac{4B}{3(d/t)} \tag{E.1}$$

The design process is as follows:

Select the material.

Assume a material thickness, *t*.

Find the ratios: h/d and d/t.

From appropriate figures given in ASME code [92], find the value of constant B.

Calculate the pressure from eq. E.1 above. If the value of calculated pressure is

greater than the design pressure, the assumed thickness should be satisfactory.

For the bottom of the evaporator, the minimum required thickness could be calculated from the following formula:

$$t = d\sqrt{\frac{CP}{SE}}$$
(E.2)

where C is a factor depends on the method of attachment (the bottom to the shell), and shell thickness (dimensionless). Its value for the present case is 0.5.

d is the diameter, mm.

E is the joint efficiency; its value is 0.8 for the present case.

P is the design pressure, psi.

S is the maximum allowable stress value, psi.

APPENDIX F CALIBRATION

Thermocouples

The thermocouples were calibrated with two temperature points; the boiling ($T_{H,c} = 100 \text{ °C}$) and freezing points ($T_{L,c} = 0 \text{ °C}$) of water. Total of 250 readings were recorded during 10 seconds period. Those were averaged for each thermocouple at the high and low points, to get the measured high temperature ($T_{H,m}$) and the measured low temperature ($T_{L,m}$). Assuming that the response of thermocouples varies linearly with temperature change. Then if a thermocouple measure a temperature value T_m , the actual temperature, T, can be calculated as

$$T = \frac{T_{H,c} - T_{L,c}}{T_{H,m} - T_{L,m}} (T_m - T_{L,m}) + T_{L,c}$$
(F.1)

Pressure Transducer

The pressure transducer used has the range of -14.7 - 15 psig and its output is 4 – 20 mA. This output is converted into voltage response through the data acquisition system by using 250 Ω resistance. The corresponding output will then be 1 – 5 V.

The transducer was calibrated with two pressure points. The atmospheric pressure $(P_{H,c})$ and the vacuum created when the unit is started $(P_{L,c})$, where a vacuum gauge of ± 2 % accuracy was used to measure the pressure.

Since the output from the transducer (V_m) is voltage, this can be converted to pressure (P_m) by

$$P_{m} = \frac{P_{\max} - P_{\min}}{V_{\max} - V_{\min}} (V_{m} - V_{\min}) + P_{\min}$$
(F.2)

Total of 200 readings were recorded during 10 seconds period. Those were averaged at the high pressure points to get the measured high pressure ($P_{H,m}$), and at low pressure point to get the measured low pressure ($P_{L,m}$). As for the case of thermocouples, assuming that the response varies linearly with the pressure change, the actual pressure can be calculated as

$$P = \frac{P_{H,c} - P_{L,c}}{P_{H,m} - P_{L,m}} (P_m - P_{L,m}) + P_{L,c}$$
(F.3)

Flow Meter

The flow meter was calibrated at the operating temperature range. The actual volume of flow through the flow meter was collected and measured, and then the actual mass flow rate can be calculated. This process was repeated several times to insure accuracy.

APPENDIX G UNCERTAINTY ANALYSIS

Information obtained through measurements are to be used to produce physically correctly interpreted data along with their estimated error or uncertainty. Error or uncertainty estimation will allow us to estimate the reliability of experimental data which is a very important task, because without knowing that the data is useless.

Error and Uncertainty

Error, ε , is the difference between the value of measurement and the true value.

Value of measurement here means the experimental results after we have applied any correction we may think necessary, like calibration factor. The true value can never be known, and then we have to deal with mean value, which usually contains error. We can write

$$\varepsilon = r - r \tag{G.1}$$

where r is the measured value, and

r is the mean value.

For a given number of measurements, N, the mean value, \overline{r} , is the arithmetic mean of the measured values and is given as

$$\bar{r} = \frac{1}{N} \sum_{i=1}^{N} r_i \tag{G.2}$$

The error as defined above is known as absolute error. Another definition may be used, relative error, which is the ratio of the absolute error to the mean value,

$$\varepsilon_r = \frac{r - \bar{r}}{\bar{r}} = \frac{\varepsilon}{\bar{r}}$$
(G.3)

Errors are made whenever measurements are made; no measurement is free from them. Errors may be classified into [93]

Systematic errors: those are mainly attributed to the equipments used, in case they have a certain defects.

Personal errors: those errors arise whenever the experimenter judgment is used.

Mistakes in collecting and recording the data.

Assignable causes: neglecting control of certain variables, which may have certain effect on the results.

Random errors: those are attributed to the working effect of some variables, which are left without control.

Uncertainty of measurement is defined as the range within which the true value should lie, since it is defined as a range, if the measured value is r, and the uncertainty is U, this means that the true value should lie between r+U and r-U, therefore the numerical value of the uncertainty is the width of this range, 2U. The uncertainty of the mean can be taken as $\frac{1}{\sqrt{N}}$ of the certainty of individual value. So, for an infinite number of readings the uncertainty of the mean is zero.

If similar measurements are repeated many times, under the same conditions, the normal "Gaussian" distribution usually applies [94]. If a single measurement is made, it will be unlikely equal to the mean value, but it should not differ from the mean value significantly, if so, the measurement is to be rejected.

The standard deviation provides a measure of error of experiment and is defined as the positive square root of the mean of the square of the deviation from the expected value, calculated for (*N-1*) members of the series. For a given set of individual measurements, their standard deviation from the mean value, σ , is given as [94],

$$\sigma = +\sqrt{\frac{\sum_{i=1}^{N} (r_i - \bar{r})^2}{N - 1}}$$
(G.4)

 $\sigma = +\sqrt{\frac{1}{N} \sum_{i=1}^{N} (r_i - \bar{r})^2}$ (G.5)

If, N >> 1

Since the true value, μ , cannot be known, it follows that the true error can never be known for any measurement, and we shall strive to get the best value within a level of confidence. The level of confidence of the measurement is to be chosen according to an acceptable tolerance. Table G.1 gives the different confidence levels along with confidence intervals.

Confidence level	Confidence interval
0.5	<i>r</i> -0.674 <i>σ</i> < <i>µ</i> < <i>r</i> +0.674 <i>σ</i>
0.8	<i>r</i> -1.282 <i>σ</i> < <i>μ</i> < <i>r</i> +1.282 <i>σ</i>
0.9	<i>r-1.645σ</i> < <i>μ</i> < <i>r</i> +1.645σ
0.95	<i>r</i> -1.960σ<μ< <i>r</i> +1.960σ
0.999	<i>r-3.291σ</i> < <i>μ</i> < <i>r</i> + <i>3.291σ</i>

Table G.1: Confidence intervals for various confidence levels, adopted from [93]

So, if we decide to operate with a level of confidence of 90%. Then the uncertainty of measurement repeated N times is given as,

$$\delta = \frac{1}{\sqrt{N}} 1.645\sigma \tag{G.6}$$

Propagation of Errors

If a final result is calculated based on a number of measured quantities, the total uncertainty is the combination of uncertainties of individual components, each one of those will have certain influence on the final result. Let us assume that a final result, R, depends on many individual measurements, r_i , by [93],

$$R = F(r_1, r_2, ..., r_N) \tag{G.7}$$

where F is a known functional form.

The variation of r_i by an amount dr_i , will affect *R* by an amount given by (making use of the chain rule of differential calculus),

$$dR = \frac{\partial R}{\partial r_1} dr_1 + \frac{\partial R}{\partial r_2} dr_2 + \dots + \frac{\partial R}{\partial r_N} dr_N$$
$$= \sum_{i=1}^N \frac{\partial R}{\partial r_i} dr_i$$
(G.8)

The uncertainty of R is given as,

$$\delta = \sqrt{\sum_{i=1}^{N} \left(\frac{\partial R}{\partial r_i}\right)^2} \delta_i^2 \tag{G.9}$$

Uncertainty of Measured Values

Temperature

The thermocouples were calibrated at two temperatures, the boiling and freezing points of water, 100 and 0 °C, respectively. The actual temperature is given as

$$T_{actual} = \frac{T_{h,c} - T_{l,c}}{T_{h,m} - T_{l,m}} (T_m - T_{l,m}) + T_{l,c}$$
(G.10)

Applying eq. G.9 to find the uncertainty of temperature measurements, we can write

$$\delta_{temp.} = \left\{ \left(\frac{\partial T_{actual}}{\partial T_{h,c}} \delta T_{h,c} \right)^2 + \left(\frac{\partial T_{actual}}{\partial T_{l,c}} \delta T_{l,c} \right)^2 + \left(\frac{\partial T_{actual}}{\partial T_{h,m}} \delta T_{h,m} \right)^2 + \left(\frac{\partial T_{actual}}{\partial T_{l,m}} \delta T_{l,m} \right)^2 + \left(\frac{\partial T_{actual}}{\partial T_m} \delta T_m \right)^2 \right\}^{\frac{1}{2}}$$

Assuming that the boiling and freezing points of water used are 100 and 0 °C and those are accurate, we get

$$\delta_{temp.} = \left\{ \left(\frac{\partial T_{actual}}{\partial T_{h,m}} \delta T_{h,m} \right)^2 + \left(\frac{\partial T_{actual}}{\partial T_{l,m}} \delta T_{l,m} \right)^2 + \left(\frac{\partial T_{actual}}{\partial T_m} \delta T_m \right)^2 \right\}^{\frac{1}{2}}$$
(G.11)

Note that $\delta T_{h,m}$, $\delta T_{l,m}$, and δT_m should be equal for each thermocouple.

Differentiating eq. G.10 with respect to $T_{h,m}$, $T_{l,m}$, and T_m , and substitute in eq. G.11, upon arrangement we get

$$\delta_{temp.} = \delta T_m \left\{ 2 \frac{(T_{h,c} - T_{l,c})^2}{(T_{h,m} - T_{l,m})^2} \left[\frac{(T_m - T_{l,m})^2}{(T_{h,m} - T_{l,m})^2} - \frac{(T_m - T_{l,m})}{(T_{h,m} - T_{l,m})} + 1 \right] \right\}^{\frac{1}{2}}$$
(G.12)

A population of 1200 measurements was recorded for each thermocouple. Then if a sample of 200 measurements is taken and eq. G.6 is used to find the uncertainty for each thermocouple. The uncertainty will be less than 0.02 °C for all thermocouples.

Substitution in eq. G.12 gives the uncertainty in temperature measurements, which is less than 0.07 °C for all temperature measurements.

Pressure

As for the thermocouples, pressure transducer was calibrated at two points; atmospheric pressure and the vacuum attained inside the unit at the time of its starting,
where a vacuum gauge was used to measure the vacuum. The actual pressure inside the unit is given as

$$P_{actual} = \frac{P_{h,c} - P_{l,c}}{P_{h,m} - P_{l,m}} (P_m - P_{l,m}) + P_{l,c}$$
(G.13)

Applying eq. G.9 to find the uncertainty of the pressure measurement, we can write

$$\delta_{pressure} = \left\{ \left(\frac{\partial P_{actual}}{\partial P_{h,c}} \, \delta P_{h,c} \right)^2 + \left(\frac{\partial P_{actual}}{\partial P_{l,c}} \, \delta P_{l,c} \right)^2 + \left(\frac{\partial P_{actual}}{\partial P_{h,m}} \, \delta P_{h,m} \right)^2 + \left(\frac{\partial P_{actual}}{\partial P_{l,m}} \, \delta P_{l,m} \right)^2 + \left(\frac{\partial P_{actual}}{\partial P_m} \, \delta P_m \right)^2 \right\}^{1/2}$$

Assuming that the atmospheric pressure is accurate and does not vary. Also, note that $\delta P_{h,m}$, $\delta P_{l,m}$, and δP_m should be equal. Then,

$$\delta_{pressure} = \left\{ \left(\frac{\partial P_{actual}}{\partial P_{l,c}} \delta P_{l,c} \right)^2 + \left(\frac{\partial P_{actual}}{\partial P_{h,m}} \delta P_m \right)^2 + \left(\frac{\partial P_{actual}}{\partial P_{l,m}} \delta P_m \right)^2 + \left(\frac{\partial P_{actual}}{\partial P_m} \delta P_m \right)^2 \right\}^{\frac{1}{2}}$$
(G.14)

Differentiating eq. G.13 with respect to $P_{l,c}$, $P_{h,m}$, $P_{l,m}$, and P_m , and substitute the result in eq. G.14, upon arrangement we get

$$\delta_{pressure} = \left\{ \left[1 - \frac{(P_m - P_{l,m})}{(P_{h,m} - P_{l,m})} \right]^2 \delta P_{l,c}^2 + 2\delta P_m^2 \frac{(P_{h,c} - P_{l,c})^2}{(P_{h,m} - P_{l,m})^2} \left[\frac{(P_m - P_{l,m})^2}{(P_{h,m} - P_{l,m})^2} - \frac{(P_m - P_{l,m})}{(P_{h,m} - P_{l,m})} + 1 \right] \right\}^{\frac{1}{2}}$$
(G.15)

As in the case of thermocouples, a population of 1200 measurements was recorded for the transducer. Then if a sample of 200 measurements is taken and eq. G.6 is used to find the uncertainty of the transducer, the result will be 0.006 psi. The accuracy of the vacuum gauge used is $\pm 2\%$, which is 0.294 psi over the full scale. Substitution in eq. G.15 gives the uncertainty of pressure measurement, which is about 0.256 psi.

Flow Rate

The flow meters are usually calibrated for a certain fluid at a certain density. The flow meter used for measuring the flow rate through the heat exchanger was calibrated by the manufacturer for a certain temperature. To find the actual flow rate for the operating temperature, the following relation can be used,

$$Q_{actual} = Q_m \left[\frac{\rho_c}{\rho}\right]^{\frac{1}{2}}$$
(G.16)

However, the flow meter was calibrated for the actual flow rate at the operating temperature range. That process was repeated many times to assure a good accuracy, each time the flow rate was measured for 1 minute interval. A graduated tube with 5 ml graduation was used to measure the flow rate, tube graduation give the liquid volume at a temperature of 20 °C. Assuming that the volume of liquid does not vary significantly in the operating temperature range, the actual flow rate can be calculated. Five measurements for a typical flow rate were taken, the standard deviation was calculated, and then if one measurement is taken and eq. G.6, which gives the uncertainty within 90 % level of confidence, is used to calculate the uncertainty, it will be about 8.8 ml.

Distillate Output

Distillate receiver tank is 7 cm in width and 32 cm in length and 5 liter capacity. The water level inside the tank was observed at the beginning of each test. At the end of the test the distillate accumulated was discharged and measured. To find the accuracy of that process, a similar tank was used to perform measurement tests. The tank was filled with water to a certain level and weighted. A known amount of water (which was weighted as well) was added to the tank, then that amount was recovered. The tank and the recovered water were weighted again to find the difference between the initial and final weights. The test was repeated ten times. The standard deviation was found from eq. G.4, then, if eq. G.6, which gives the uncertainty within 90 % level of confidence, is used to calculate the uncertainty on carrying out one measurement, the result will be about 7.4 ml. However, the above arrangement was used to measure the output for a number of tests, then another arrangement were used, water flow arrangement. A glass bottle having an opening at a certain height received the water from the condenser. The bottle was initially filled with water up to that height. Then whatever amount of water was received from the condenser would escape through the opening, where it was received by another bottle and measured. This arrangement gives a better accuracy. The uncertainty was estimated to be less than 2 ml.

The flow rate through the evaporator heat exchanger affects the amount of distillate output directly. The uncertainty of the flow rate measurement is 8.8 ml, which represents 5.3 % of the flow rate. As the distillate output varies with the flow rate, we would expect the output to be 5.3 % off the correct value, which represents about 6.4 ml of the hourly output. Adding this to the uncertainty in measuring the output, the total uncertainty in the hourly distillate output is about 13.8 and 8.4 ml for the old and new arrangements, respectively. The uncertainty of the daily output will be 5.3 % of the total daily output plus 7.4 and 2 ml for the old and new arrangements, respectively.

Uncertainty of Calculated Values

Heat Input

The heat input to the unit depends on the temperature of the water at the inlet and exit of the heat exchanger and the flow rate. It is given as,

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$$Q_{input} = m_c C_p (T_{co} - T_{ci})$$
(G.17)

Making use of eq. G.9, the uncertainty can be written as

$$\delta_{input} = \left[\left(\frac{\partial Q_{input}}{\partial T_{co}} \delta T_{co} \right)^2 + \left(\frac{\partial Q_{input}}{\partial T_{ci}} \delta T_{ci} \right)^2 + \left(\frac{\partial Q_{input}}{\bullet} \delta m_c \right)^2 \right]^{\frac{1}{2}}$$
(G.18)

. .

Differentiating eq. G.17 with respect to T_{co} , T_{ci} , and m_c and substitute the result in eq. G.18, we get

$$\delta_{input} = \left[\left(\stackrel{\bullet}{m}_{c} C_{p} \delta T_{co} \right)^{2} + \left(\stackrel{\bullet}{-m}_{c} C_{p} \delta T_{ci} \right)^{2} + \left(C_{p} (T_{co} - T_{ci}) \delta \stackrel{\bullet}{m}_{c} \right)^{2} \right]^{\frac{1}{2}}$$
(G.19)

Substituting values for a typical test in eq. G.19, the uncertainty will be about 4.5 W. The total error will be about 5.25 %, with the largest contribution due to error in flow rate measurements.

Efficiency

The system efficiency depends on the heat input, which depends on the temperatures of the water at the inlet and outlet of the heat exchanger and the mass flow rate, and the output, which depends on the amount of distillate produced and its latent heat of evaporation. The expression for efficiency is given as

$$\eta = \frac{\dot{m}h_{fg}}{\dot{m}_c C_p (T_{co} - T_{ci})} = \frac{\dot{m} [3146 * 10^3 - 2360(T_s + 273)]}{\dot{m}_c C_p (T_{co} - T_{ci})}$$
(G.20)

Making use of eq. G.9, the uncertainty can be written as

$$\delta\eta = \left[\left(\frac{\partial\eta}{\partial m} \delta \overset{\bullet}{m} \right)^2 + \left(\frac{\partial\eta}{\partial T_s} \delta T_s \right)^2 + \left(\frac{\partial\eta}{\partial m_c} \delta \overset{\bullet}{m_c} \right)^2 + \left(\frac{\partial\eta}{\partial T_{co}} \delta T_{co} \right)^2 + \left(\frac{\partial\eta}{\partial T_{ci}} \delta T_{ci} \right)^2 \right]^{\frac{1}{2}}$$
(G.21)

Differentiating eq. G.19, and substitute in eq. G.20, we get

$$\delta\eta = \frac{1}{\overset{n}{m_c} C_p (T_{co} - T_{ci})} \begin{cases} \left[3146 * 10^3 - 2360(T_s + 273) \right]^2 (\delta \overset{\bullet}{m})^2 + (2360 \overset{\bullet}{m} \delta T_s)^2 + \left[3146 * 10^3 - 2360(T_s + 273) \right]^2 \overset{\bullet}{m}^2 \\ \left[\frac{\left[3146 * 10^3 - 2360(T_s + 273) \right]^2 \overset{\bullet}{m}^2}{\left[m_c C_p (T_{co} - T_{ci}) \right]^2} \\ \left[\frac{\left[(C_p (T_{co} - T_{ci}) \delta \overset{\bullet}{m_c})^2 + (\overset{\bullet}{m_c} C_p \delta T_{co})^2 + (\overset{\bullet}{m_c} C_p \delta T_{ci})^2 \right] \\ \right] \end{cases}$$
(G.22)

Substituting values for a typical test in eq. G.21, gives the total error of 8.4 %, with the largest contribution due to error in flow rate and distillate measurements.

LIST OF REFERENCES

- Bendfeld, J., Broker, Ch., Menne, K., Ortjohann, E., Temme, L., Vob, J., and Carvallo, P.C.M., 1998, "Design of a PV-powered reverse osmosis plant for desalination of brackish water," Proceedings of 2nd World Conference and Exhibition on Photovoltaic Solar Energy Conversion, Vienna, Austria, pp. 3075-3077.
- 2. Kalogirou, S., 1997, "Economic analysis of a solar assisted desalination system," Renewable Energy, **12**(4), pp. 351-367.
- Delyannis, E. and Belessiotis, V., 2001, "Solar energy and desalination," in *Advances in solar energy, An annual review of research and development,* Goswami, D.Y. (Ed), 14, American Solar Energy Society, Inc, Boulder, Colorado, pp. 287-330.
- 4. Kalogirou, S., 1997, "Survey of solar desalination systems and system selection," Energy, **22**(1), pp. 69-81.
- 5. Department of Water Resources, Resources Agency, State of California, 1974, *Desalting alternatives in ten California communities*. Department of Water Resources, Sacramento.
- 6. Semiat, R., 2000, "Desalination: present and future," Water International, **25**(1), pp. 54-65.
- 7. Hamed, O.A., Eissa, E.I., and Abdalla, W.E., 1993, "Overview of solar desalination," Desalination, **93**, pp. 563-579.
- 8. AlHawaj, O., and Darwish, M.A., 1994, "A solar pond assisted multi-effect desalting system," Desalination, **99**, pp. 119-135.
- 9. Lu, H., Walton, J.C., and Swift, A.H.P., 2001, "Desalination coupled with salinitygradient solar ponds," Desalination, **136**, pp. 13-23.
- 10. Safi, M.J., 1998, "Performance of a flash desalination unit intended to be coupled to a solar pond," Renewable Energy, **14**(1-4), pp. 339-343.
- 11. Szacsvay, T., Noser, P.H., and Posnansky, M., 1999, "Technical and economic aspects of small-scale solar-pond-powered seawater desalination systems," Desalination, **122**, pp. 185-193.

- 12. Tsilingiris, P.T., 1995, "The analysis and performance of large-scale stand-alone solar desalination plants," Desalination, **103**, pp. 249-255.
- 13. Caruso, G., and Naviglio, A., 1999, "A desalination plant using solar heat as a heat supply, not affecting the environment with chemicals," Desalination, **122**, pp. 225-234.
- 14. Caruso, G., Naviglio, A., Principi, P., and Ruffini, E., 2001, "High-energy efficiency desalination project using a full titanium desalination unit and a solar pond as the heat supply," Desalination, **136**, pp. 199-212.
- 15. Rodriguez, L.G., Marrero, A.I.P., and Camacho, C.G., 1999, "Application of direct steam generation into a solar parabolic trough collector to multieffect distillation," Desalination, **125**, pp. 139-145.
- 16. El-Nashar, A.M., 1993, "An optimal design of a solar desalination plant," Desalination, **93**, pp. 579-614.
- 17. Abdel Rassoul, R.A., 1998, "Potential for economic solar desalination in the Middle East," Renewable Energy, **14**(1-4), pp. 345-349.
- 18. El-Nashar, A.M., 2001, "The economic feasibility of small solar MED seawater desalination plants for remote areas," Desalination, **134**, pp. 173-186.
- 19. Rodriguez, L.G., and Camacho, C.G., 1999, "Conditions for economical benefits of the use of solar energy in multi-stage flash distillation," Desalination, **125**, pp. 133-138.
- 20. Hasnain, S.M., and Alajlan, S., 1998, "Coupling of a PV-powered R.O. brackish water desalination plant with solar stills," Renewable Energy, **14**(1-4), pp. 281-286.
- Manolakos, D., Papadakis, G., Papantonis, D., and Kyritsis, S., 2001, "A simulation-optimization programme for designing hybrid energy systems for supplying electricity and fresh water through desalination to remote areas. Case study: the Merssini village, Donoussa island, Aegean Sea, Greece," Energy, 26, pp. 679-704.
- 22. Joyce, A., Loureiro, D., Rodrigues, C., and Castro, S., 2001, "Small reverse osmosis units using PV systems for water purification in rural places," Desalination, **137**, pp. 39-44.
- 23. Weiner, D., Fisher, D., Moses, E.J., Katz, B., and Meron, G., 2001, "Operation experience of a solar-and wind-powered desalination demonstration plant," Desalination, **137**, pp. 7-13.
- Goosen, M.F.A., Sablani, S.S., Shayya W.H., Paton, C., and Al-Hinai, H., 2000, "Thermodynamic and economic considerations in solar desalination," Desalination, 129, pp. 63-89.

- 25. Dunkle, R.V., 1961, "Solar water distillation: The roof type still and a multiple effect diffusion still," International Conference of Heat Transfer, University of Colorado, Publication No. 108, pp. 895-902.
- 26. Boukar, M. and Harmim, A., 2001, "Effect of climatic conditions on the performance of a simple solar still: a comparative study," Desalination, **137**, pp. 15-22.
- 27. Kumar, S. and Tiwari, G.N., 1996, "Performance evaluation of an active solar distillation system," Energy, **21**(9), pp. 805-808.
- 28. Lawrence, S.A. and Tiwari, G.N., 1990, "Theoretical evaluation of solar distillation under natural circulation with heat exchanger," Energy Conversion and Management, **30**(3), pp. 205-213.
- 29. Tiwari, G.N., Kumar, S., Sharma, P.B., and Khan, M.E., 1996, "Instantaneous thermal efficiency of an active solar still," Applied Thermal Energy, **16**(2), pp. 189-192.
- 30. Yadav, Y.P. and Jha, L.K., 1989, "A double-basin solar still coupled to collector and operating in the thermosiphon mode," Solar Energy, **14**(10), pp. 653-659.
- 31. Yadav, Y.P., 1993, "Performance analysis of a solar still coupled to a heat exchanger," Desalination, **91**, pp. 135-144.
- 32. Yadav, Y.P., and Prasad, A.S., 1995, "Performance analysis of a high temperature solar distillation system," Energy Conversion and Management, **36**(5), pp. 365-374.
- 33. El-Bahi, A. and Inan, D., 1999, "A solar still with minimum inclination, coupled to an outside condenser," Desalination, **123**, pp. 79-83.
- 34. Fath, H.E.S. and Elsherbiny, S.M., 1993, "Effect of adding a passive condenser on solar still performance," Energy Conversion and Management, **34**(1), pp. 63-72.
- 35. Khalifa, A.N., 1985, "Evaluation and energy balance study of a solar still with an internal condenser," Journal of Solar Energy Research, **3**(1), pp. 1-11.
- 36. Saatci, A.M., 1984, "Heat-pipe assisted solar still," Proceedings of the 1984 Annual Meeting American Solar Energy Society, Inc, pp. 249-253.
- 37. Abu-Jabal, M.S., Kamiya, I., and Narasaki, Y., 2001, "Proving test for a solarpowered desalination system in Gaza-Palestine," Desalination, **137**, pp. 1-6.
- 38. Jubran, B.A., Ahmed, M.I., Ismail, A.F., and Abakar, Y.A., 2000, "Numerical modeling of a multi-stage solar still," Energy Conversion & Management, **41**, pp. 1107-1121.

- 39. Low, S.C. and Tay, J.H., 1991, "Vacuum desalination using waste heat from a steam turbine," Desalination, **81**, pp. 321-331.
- 40. Tay, J.H., Low, S.C., and Jeyaseelan, S., 1996, "Vacuum desalination for water purification using waste heat," Desalination, **106**, pp. 131-135.
- 41. Uda, K., Sato, H., and Watanabe, K., 1994, "Development of advanced evacuated solar still," Joint Engineering Conference, ASME 1994, pp. 513-519.
- 42. Kunze, H., 2001, "A new approach to solar desalination for small-and mediumsize use in remote areas," Desalination, **139**, pp. 35-41.
- 43. Mink, G., Aboabboud, M.M., and Karmazsin, E., 1998, "Air-blown solar still with heat recycling," Solar Energy, **62**(4), pp. 309-317.
- 44. Saghafi, M., 1994, "Development of a solar desalination plant," Renewable Energy, **5**, Part I, pp. 529-531.
- 45. Schwarzer, K., Vieira, M.E., Farber, C., and Muller, C., 2001, "Solar thermal desalination system with heat recovery," Desalination, **137**, pp. 23-29.
- 46. Fernandez, J.L. and Chargoy, N., 1990, "Multi-stage, indirectly heated solar still," Solar Energy, **44**(4), pp. 215-223.
- 47. Franco, J. and Saravia, L., 1994, "A new design for a passive atmospheric multistage still," Renewable Energy, **4**(1), pp. 119-122.
- 48. Kumar, S. and Tiwari, G.N., 1999, "Optimization of design parameters for multieffect active distillation systems using Runge-Kutta method," Desalination, **121**, pp. 87-96.
- 49. Toyama, S., Aragaki, T., and Murase, K., 1983, "Simulation of a multi-effect solar distillator," Desalination, **45**, pp. 101-108.
- 50. Toyama, S., Aragaki, T., Salah, H.M., Murase, K., and Sando, M., 1987, "Simulation of a multi-effect solar still and the static characteristics," Journal of Chemical Engineering of Japan, **20**(5), pp. 473-478.
- 51. Bobrovnikov, G.N., Achilov, B.M., Kamilov, O.S., Kakharov, S., and Odinaev, A., 1979, "Determining the thermal losses to the atmosphere from a solar still," Geliotekhnika, **15**(4), pp. 70-71.
- 52. Tiwari, G.N., 1988, "Nocturnal water production by tubular solar stills using waste heat to preheat brine," Desalination, **69**, pp. 309-318.
- 53. Malik, M.A.S. and Tran, V.V., 1973, "A simplified mathematical model for predicting the nocturnal output of a solar still," Solar Energy, **14**(4), pp. 371-385.

- 54. Onyegegbu, S.O., 1986, "Nocturnal distillation in basin-type solar stills," Applied Energy, **24**, pp. 29-42.
- 55. Dutt, D.K., Kumar, A., Anand, J.D. and Tiwari, G.N., 1989, "Performance of a double-basin solar still in the presence of dye," Applied Energy, **32**, pp. 207-223.
- 56. Al-Hallaj, S., Farid, M.M., and Tamimi, A.R., 1998, "Solar desalination with a humidification-dehumidification cycle: performance of the unit," Desalination, **120**, pp. 273-280.
- 57. Ben Bacha, H., Maalej, A.Y., Ben Dhia, H., Ulber, I., Uchtmann, H., Engelhardt, M., and Krelle, J., 1999, "Perspective of solar-powered desalination with the SMCEC technique," Desalination, **122**, pp. 177-183.
- 58. Chaibi, M.T., Safi, M.J., and Hsairi, M., 1991, "Performance analysis of a solar desalination unit in south Tunisia," Desalination, **82**, pp. 197-205.
- 59. Muller-Holst, H., Engelhardt, M., and Scholkopf, W., 1999, "Small-scale thermal seawater desalination simulation and optimization of system design," Desalination, **122**, pp. 255-262.
- 60. Tchinda, R., Kaptouom, E., and Njomo, D., 1999, "Heat and mass transfer processes in a solar still with an indirect evaporator-condenser," Energy Conversion & Management, **41**, pp. 93-107.
- 61. Ibrahim, A. and Lowrey, P., 1992, "A preliminary analysis of distillation powered by a salinity difference," General Papers in Heat Transfer, ASME 1992, HTD-**204**, pp. 79-88.
- 62. Sharma, S.K., and Goswami, D.Y, 1994, "Low temperature energy conversion system," unpublished document, University of Florida.
- 63. Keren, Y., Rubin, H., Atkinson, J., Priven, M., and Bemporad, A., 1993, "Theoretical and experimental comparison of conventional and advanced solar pond performance," Solar Energy, **51**(4), pp. 255-270.
- 64. Pirani, M. and Yarwood, J., 1961, *Principles of vacuum engineering*, Chapman and Hall, London.
- 65. Krell, E., 1982, *Handbook of laboratory distillation*, Elsevier Scientific Publishing Company, Amsterdam, pp. 159.
- 66. Schrage, R.W., 1953, *A theoretical study of interphase mass transfer*, Columbia University Press, New York.
- 67. Bemporad, G.A., 1995, "Basic hydrodynamic aspects of a solar energy based desalination process," Desalination, **54**(2), pp. 125-134.

- 68. Porteous, A., 1975, *Saline water distillation processes*, Longman Group Limited, London.
- 69. Incropera, F.P. and DeWitt, D.P., 1996, *Fundamentals of heat and mass transfer*, fourth edition, John Wiley & Sons, New York.
- 70. Lock, G.S.H., 1994, *Latent heat transfer: An introduction to fundamentals*, Oxford Science Publications, New York
- 71. Herranz, L.E., Anderson, M.H., and Corradini, M.L., 1998, "A diffusion layer model for steam condensation within the AP600 containment," Nuclear Engineering and Design, **183**, pp. 133-150.
- 72. Rohsenow, W.M., Hartnett, J.P., and Cho, Y. I., 1998, *Handbook of heat transfer*, third edition, McGraw-Hill Book Company, New York.
- 73. Duffie, J.A., and Beckmann, W.A., 1980, *Solar engineering of thermal processes*, second edition, John wiley & sons, New York.
- 74. Fujii, T., and Imura, H., 1972, "Natural convection heat transfer from a plate with arbitrary inclination," International Journal of Heat and Mass Transfer, **15**, pp. 755-767.
- 75. Mitchell, J.W., 1976, "Heat transfer from spheres and other animal forms," Biophysical Journal, **16**, p. 561.
- 76. Open university course team, 1989, *Seawater: Its composition, properties, and behavior*, first edition, Pergamon press, New York.
- 77. Munson, B.R., Young, D.F., and Okiishi, T.H., 1998, *Fundamentals of fluid mechanics*, third edition, John Wiley & Sons, New York.
- 78. Goswami, D.Y., Kreith, F., and Kreider, J.F., 2000, *Principles of solar engineering*, second edition, Taylor & Francis, Philadelphia.
- 79. Adhikari, R.S., and Kumar, A., 1993, "Transient simulation on a multi-stage stacked tray solar still," Desalination, **91**, pp. 1-20.
- 80. Lior, N., 1986, *Desalination and water purification, 2. Measurements and control in water desalination*, Elsevier scientific publishing company, Amsterdam.
- 81. Stumm, W., and Morgan, J.J., 1996, *Aquatic chemistry*, third edition, John Wiley and Sons, New York.
- 82. Roberge, P.R., 1999, *Handbook of corrosion engineering*, McGraw-Hill, New York.

- 83. Skopintsev, B.A., 1981, "Decomposition of organic matter of plankton humification and hydrolysis," in *Marine organic chemistry*, Dunrsma, E.K., and Davson, R. (Editors), Elsevier Scientific Publishing Company, Amsterdam.
- 84. Hunter, K.A., and Liss, P.S., 1981, "Organic sea surface films," in *Marine organic chemistry*, Dunrsma, E,K., and Davson, R. (Editors), Elsevier Scientific Publishing Company, Amsterdam.
- 85. Mackinnon, M.D., 1981, "The measurement of organic carbon in seawater," in *Marine organic chemistry*, Dunrsma, E,K., and Davson, R. (Editors), Elsevier Scientific Publishing Company, Amsterdam.
- 86. Bromley, L.A., Singh, D., and Ray, P., 1974, "Thermodynamic properties of sea salt-solutions," AICHE Journal, **20** (2), pp. 326-335.
- 87. Spiegler, K.S., 1977, *Salt-water purification*, second edition, Plenum Press, New York.
- 88. Schrieber, C.F., 1986, "Measurements and control related to corrosion and scale in water desalination installations," in *Desalination and water purification 2, Measurements and control in water desalination*, Lior, N. (Ed), Elsevier, New York.
- 89. Millero, F.J., 2001, *Physical chemistry of natural waters*, John Wiley and Sons, New York.
- 90. Irvine, T.F. (Jr), and Liley, P.E., 1984, *Steam and gas tables with computer equations*, Academic Press, New York.
- 91. Mamayev, O.I., 1975, *Temperature-salinity analysis of world ocean waters*, Elsevier Scientific Publishing Company, Amsterdam.
- 92. ASME Boiler and pressure vessel code, An American national standard. Section VIII: Rules for construction of pressure vessels, Division 1, 1977 edition, ASME, New York.
- 93. Wilson, E.B., Jr., 1952, *An introduction to scientific research*, McGraw-Hill Book Company, Inc., New York.
- 94. Boros, A., 1989, *Fundamental studies in engineering: Measurement evaluation*, Elsevier Scientific Publishing Company, Amsterdam.

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Saleh received his bachelor's degree in mechanical engineering in 1991 from the Regional Engineering College-Durgapur (India). In the same year he joined the University of Jordan where he received his M.Sc. degree in mechanical engineering in 1994. He worked in different places before joining the Ph.D. program at the University of Tennessee in 2000; a year later he transferred to the University of Florida.