A COMPREHENSIVE STUDY OF SOLAR DESALINATION WITH A HUMIDIFICATION - DEHUMIDIFICATION CYCLE

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## TABLE OF CONTENTS

List of Tables..................................................................................................................................................... vii  
List of Figures....................................................................................................................................................... ix  
Nomenclature....................................................................................................................................................... x  
Abstract............................................................................................................................................................... xii  
Acknowledgements............................................................................................................................................ xii  
Executive Summary.................................................................................................................................................. xiii  

### 1. Comprehensive Literature Review.............................................................................................................. 1  
1.1. Introduction.................................................................................................................................................. 1  
1.2. History and Background: Solar Stills........................................................................................................ 1  
1.3. Humidification-Dehumidification Technique............................................................................................ 4  
  1.3.1. Principle of the Humidification-Dehumidification Process................................................................. 4  
  1.3.2. Methods of Extracting Water from Humid Air under Atmospheric Condition............................... 4  
    1.3.2.1. Atmospheric Water Vapour Processing......................................................................................... 5  
    1.3.2.2. Dew Collection.......................................................................................................................... 5  
    1.3.2.3. Adsorption Method................................................................................................................. 5  
    1.3.2.4. Adsorption-Refrigeration Method.............................................................................................. 6  
    1.3.2.5. Vapour-Compression Refrigeration Method............................................................................. 6  
    1.3.2.6. Absorption Method.................................................................................................................. 7  
    1.3.2.7. Humidification by Pervaporation.............................................................................................. 7  
    1.3.2.8. HD Using Hydrophobic Capillary Contactors........................................................................... 7  
1.4. Solar Humidification-Dehumidification (HD).......................................................................................... 8  
  1.4.1. Early Development............................................................................................................................ 8  
  1.4.2. Forced Convection in Solar HD......................................................................................................... 8  
1.5. Multi-Effect Humidity (or HD) Process (MEH)..................................................................................... 9  
  1.5.1. Principle of MEH............................................................................................................................. 9  
  1.5.2. MEH Units based on Open-Water / Closed Air Cycle................................................................. 10  
  1.5.3. MEH Units based on Open-Air / Closed Water Cycle.................................................................. 13  
  1.5.4. Solar Multiple Condensation Evaporation Process..................................................................... 15  
  1.5.5. Aero-Evapo-Condensation Process................................................................................................. 16  
  1.5.6. Carrier Gas Process..................................................................................................................... 17
1.6. Summary of Studies Conducted on Humidification-Dehumidification Process.................................18

1.7. Efficiency and Production Improvement Studies on Solar Desalination...........................................18
  1.7.1. Improvements in Evaporator Performance.........................21
  1.7.2. Improvements in Condenser Performance...........................23
  1.7.3. Studies on Convection Systems........................................24
  1.7.4. Heat Recovery Systems....................................................24
  1.7.5. Heat Storage Systems......................................................25
  1.7.6. Recent Research on Solar Collectors..................................26

1.8. Conclusion .................................................................................26

1.9. Outlook .....................................................................................26

2. Mathematical modelling of solar desalination unit with HD cycle ..............27
  2.1. Background..............................................................................27
  2.2. Mathematical Modelling..........................................................27
  2.3. Energy and Mass Balances......................................................28
  2.4. Desalination Units Constructed in Iraq, Jordan and Malaysia.............30
    2.4.1. Humidifier Design.............................................................31
    2.4.2. Condenser Design.............................................................31
  2.5. Mass Transfer Coefficient in the Humidifier..................................32
  2.6. Heat Transfer in the Condenser.................................................35
  2.7. Simulation Results for the Variations in Collector, Condenser and Humidifier Areas on the Productivity of an MEH Unit.................................37
  2.8. Conclusions..............................................................................46

3. Economic Evaluation...........................................................................47
  3.1. Introduction..............................................................................47
  3.2. Solar Distillation.......................................................................48
    3.2.1. Solar Stills..........................................................................48
    3.2.2. Cost Analysis Method for Solar Distillation Units...............49
    3.2.3. Solar-thermal Processes.........................................................49
  3.3. Combination of Different Solar Powered Systems..........................50
    3.3.1. Solar-MED Units.................................................................50
    3.3.2. Solar-MSF and Solar-RO Systems.......................................53
    3.3.3. RO Units Powered by Renewable Energy............................55
3.4. Economics of Solar Humidification-Dehumidification and MEH Processes…………………………………………………………………………………56
  3.4.1. Recent Cost Studies for MEH Units………………………………………59
  3.4.2. Investment and Operational Costs………………………………………59

3.5. Cost of Other Humidification-Dehumidification Processes………………..60
  3.5.1. Dewvaporation.......................................................................60
  3.5.2. Carrier-Gas Process.................................................................61
  3.5.3. Aero-Evapo-Condensation Process.........................................62

3.6. Case for Cost Reduction...................................................................62
  3.6.1. Solar Collectors and Thermal Storage........................................62
    3.6.1.1. Types and Cost of Collectors...............................................63
    3.6.1.2. Thermal Storage..................................................................63

3.7. Conclusions.....................................................................................64

References................................................................................................66
LIST OF TABLES

Table 1.1. Studies on closed - Air/Open - water cycle systems.........................19
Table 1.2. Studies on open - Air/Closed - water cycle systems.........................21
Table 1.3. Operating data for a Pilot Solar - Powered Multi - Effect Distiller……...22
Table 2.1. Specifications of the condenser, humidifier, and solar collector used in the three desalination units.........................35
Table 2.2. The daily production of the desalination unit with various condenser and collector surfaces areas, humidifier area = 11.9 m^2 ……38
Table 2.3. The daily production of the desalination unit with various condenser and humidifier surface areas, collector area = 20 m^2……..40
Table 2.4. The daily production of the desalination unit with various collector and humidifier surface areas, condenser area = 9 m^2……..42
Table 2.5. The daily production of the desalination unit with water flow-rate…..45
Table 3.1. Distribution of costs for conventional (RO and MSF) plants and for plants driven with renewable energy.................................47
Table 3.2. Price of desalting by conventional solar stills.................................48
Table 3.3. Summary of water cost for Simple and Multi - Effect solar stills………50
Table 3.4. Cost comparison of Solar Pond powered desalination with conventional SWRO ..............................................................51
Table 3.5. Cost of water production using conventional and Solar-powered MSF Systems, Plant Capacity: 1m^3. d^-1.................................54
Table 3.6. Cost of water production using conventional and Solar-powered RO systems, plant capacity: 1m^3. d^-1.................................54
Table 3.7. Effect of different parameters on the cost of distilled water obtained from a proposed Solar Still Plant coupled to a PV-Powered RO Desalination Plant..............................................…....55
Table 3.8. Cost of RO-Plants powered on the basis of renewable energy.........56
Table 3.9. Summary of water cost from different Solar – Powered Desalination units.................................................................57
Table 3.10. Comparison of HD process (with waste heat) with other processes.....60
Table 3.11.  Cost summary of desalination plant with different capacities ...........61

Table 3.12.  Capital cost comparison for large systems...............................61

Table 3.13.  Operating cost comparison for large systems.............................62

Table 3.14.  Summary of estimated capital cost for different types of HD systems.................................................................64

Table 3.15.  Summary of cost of water from HD and MEH processes................64

Table 3.16.  Operational cost for other HD processes with solar energy usage.................................................................65
LIST OF FIGURES

Figure 1.1. Multi-Effect Still: (Rheinlander and Grater\textsuperscript{12}) ..............................................3

Figure 1.2. Schematic Diagram of an Experimental MEH Desalination Unit Operated With Forced or Natural Air Convection .................11

Figure 1.3. Illustration of a Multi-Effect Distillation Unit without Storage Implementation .................................................................13

Figure 1.4. Schematic Diagram of an Experimental MEH Desalination Unit Operated With Forced or Natural Air Convection .................14

Figure 1.5. MEH Unit with Open-Air/Closed-Water Cycle .........................18

Figure 2.1. Sketch of a Natural Draft Air Circulation MEH Desalination Unit .......28

Figure 2.2. Details of the Humidifier’s Packing Used in the Desalination Units ....33

Figure 2.3. Details of the Condenser Used in the Desalination Units ...............34

Figure 2.4a. Effects of the Condenser and Collector Surface Areas on the Daily Production of the Desalination Unit, Humidifier Area = 11.9 m\textsuperscript{2} .......37

Figure 2.4b. Effects of the Condenser and Collector Surface Areas on the Daily Production of the Desalination Unit, Humidifier Area = 11.9 m\textsuperscript{2} ......38

Figure 2.5a. Effects of the Condenser and Humidifier Surface Areas on the Daily Production of the Desalination Unit, Collector Area = 20 m\textsuperscript{2} ........39

Figure 2.5b. Effects of the Condenser and Humidifier Surface Areas on the Daily Production of the Desalination Unit, Collector area = 20 m\textsuperscript{2} ..........40

Figure 2.6a. Effects of the Collector and Humidifier Surface Areas on the Daily Production of the Desalination Unit, Condenser Area = 9 m\textsuperscript{2} ..........41

Figure 2.6b. Effects of the Collector and Humidifier Surface Areas on the Daily Production of the Desalination Unit, Condenser Area = 9 m\textsuperscript{2} .......42

Figure 2.7. Effect of the Water Flow-rate on the Daily Production of the Desalination Unit .................................................................44

Figure 2.8. Solar intensity Measurements at Arab Gulf Countries, and the Daily production of the Desalination Unit ...........................................44

Figure 2.9. Temperature Distribution and Desalination Production of the Desalination Unit .................................................................45
NOMENCLATURE

Acronyms and Abbreviations

CO₂ Carbon Dioxide
CGP Carrier Gas Process
ETC Evacuated Thermal Collector
FPC Flat Plate Collector
HD Humidification-Dehumidification
HDD Humidification-Dehumidification Desalination
IA Average annual interest and amortization rate, % of investment
GOR Gain Output Ratio
L/G Liquid to Gas Ratio
ME Multiple Effect
MED Multiple Effect Desalination
MEH Multiple Effect Humidification
MES Multiple Effect Stack
MR Average annual maintenance and repair, labour and materials, % of investment
MSF Multi Stage Flash
MVC Multi Vapour Compression
N Number of effects
PR Performance Ratio
RO Reverse Osmosis
SME Solar Multiple Condensation Evaporation
SMCEC Solar Multiple Condensation Evaporation Cycle
SP Solar Pond
SWRO Sea Water Reverse Osmosis
TI Average annual taxes and insurance charges, % of investment
TVC Thermal Vapour Compression
VC Vapour Compression

A Area of still on which yields are based [m²]
Acond surface area of the condenser [m²]
Aunit surface area of the desalination unit [m²]
Acol surface area of solar collector [m²]
a humidifier surface area per unit volume [m².m⁻³]
C Cost of water [$m⁻³]
Cc Surface cost of collectors [$m⁻²]
Ce Cost of evaporator heat transfer area [$m⁻²]
Cs Total cost (fixed and operating) of salt water supply [$m⁻³]
Cpₜw specific heat capacity of water [J.kg⁻¹.K⁻¹]
Cunit specific heat capacity of the unit [J.kg⁻¹.K⁻¹]
c Cost (wages) of operating labour [$man·hr⁻¹]
G air mass flow rate [kg·s⁻¹]
H enthalpy of air [J.kg⁻¹]
h individual heat transfer coefficient [W.m⁻².K⁻¹]
hOverall heat transfer coefficient of the evaporator [kW.m⁻²°C]
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>i</td>
<td>time counter in the stepwise computation</td>
<td></td>
</tr>
<tr>
<td>I_T</td>
<td>solar radiation intensity received by the collector</td>
<td>W.m⁻²</td>
</tr>
<tr>
<td>I</td>
<td>Total capital investment</td>
<td>$</td>
</tr>
<tr>
<td>I_s</td>
<td>Insulation</td>
<td>MJ.m⁻².d⁻¹</td>
</tr>
<tr>
<td>K</td>
<td>mass transfer coefficient in the humidifier</td>
<td>kg.m⁻².s⁻¹</td>
</tr>
<tr>
<td>L</td>
<td>water mass flow rate</td>
<td>kg.s⁻¹</td>
</tr>
<tr>
<td>m_w</td>
<td>productivity of the unit (kg distilled water.s⁻¹)</td>
<td></td>
</tr>
<tr>
<td>M_unit</td>
<td>mass of the unit</td>
<td>kg</td>
</tr>
<tr>
<td>n</td>
<td>payment period, years</td>
<td></td>
</tr>
<tr>
<td>O</td>
<td>Annual operating labour (man-hours)</td>
<td></td>
</tr>
<tr>
<td>Q_heater</td>
<td>heating power supplied by the heater</td>
<td>watt</td>
</tr>
<tr>
<td>r</td>
<td>Annual interest rate</td>
<td>%</td>
</tr>
<tr>
<td>T</td>
<td>temperature</td>
<td>°C</td>
</tr>
<tr>
<td>T_amb</td>
<td>ambient temperature</td>
<td>°C</td>
</tr>
<tr>
<td>U_cond</td>
<td>overall heat transfer coefficient</td>
<td>W.m⁻².K⁻¹</td>
</tr>
<tr>
<td>U_loss</td>
<td>heat loss coefficient</td>
<td>W.m⁻².K⁻¹</td>
</tr>
<tr>
<td>V</td>
<td>volume of the humidifier</td>
<td>m³</td>
</tr>
<tr>
<td>Y_c</td>
<td>Annual unit yield of collected rain water</td>
<td>m³.m⁻²</td>
</tr>
<tr>
<td>Y_d</td>
<td>Annual unit yield of distilled water</td>
<td>m³.m⁻²</td>
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**Greek Letters**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
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<tbody>
<tr>
<td>α</td>
<td>Boiling point elevation of brine,</td>
<td>°C</td>
</tr>
<tr>
<td>∆t</td>
<td>time increment</td>
<td>s</td>
</tr>
<tr>
<td>∆T</td>
<td>Temperature difference between any two consecutive effects</td>
<td>°C</td>
</tr>
<tr>
<td>η_col</td>
<td>collector efficiency</td>
<td>-</td>
</tr>
</tbody>
</table>

Symbols not listed above are defined in the text.
ABSTRACT

Major desalination processes consume a large amount of energy derived from oil and natural gas as heat and electricity, while emitting harmful CO2. Solar desalination has recently emerged as a promising renewable energy powered technology for producing fresh water. Combining the principle of humidification - dehumidification with solar desalination, results in an increase in the overall efficiency of the desalination plant and therefore appears to be the best method of water desalination with solar energy. A detailed study of the mechanism of this process is presented in this report, as well as a comprehensive mathematical modelling of this technology applied to an existing MEH (Multi Effect Humidification) unit, and an economical evaluation of the process. Comparison with the cost of currently available solar desalination processes presented in this report leads to the conclusion that a better understanding of this method of solar desalination is highly desirable. Simulation verification and design optimisation of the performance of a unit based on varying the three major components (humidifier, condenser and collector surface areas) of the unit is perhaps the first such optimisation to date that could be a critical step in the commercialisation of solar desalination based on the humidification - dehumidification principle.

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EXECUTIVE SUMMARY

A solar desalination unit based on the principle of humidification – dehumidification is investigated in this project. Various units based on the principle of solar powered humidification – dehumidification has been constructed in different parts of the world. The basic principle of all these units is that evaporation of seawater and the condensation of water vapour from the humid air takes place in the unit at ambient pressure and at temperatures between 40°C and 85°C. However, the economics of solar desalination have thus far, not been evaluated on a comprehensive basis i.e. comparing with alternative techniques of desalination. This requires a rationally based design optimisation of the solar-powered humidification - dehumidification process.

The main tasks of the present project are:

- Comprehensive survey and summary of work previously conducted on solar desalination utilising the humidification - dehumidification process
- A comprehensive review of the calculation methods applied in solar desalination using humidification - dehumidification principles with a strong emphasis on the heat and mass balance equations
- Economic evaluation and cost analysis of the process in comparison with the economics of all the other solar - based desalination techniques

Due to the large energy consumption in the major commercial desalination processes, along with the growing concern about CO₂ emission, there is a strong interest in alternate sources of energy to run desalination units, and in particular, renewable energy sources. In countries with abundant solar energy, solar desalination could be one of the most successful applications of solar energy. Solar stills developed as desalination units have inherently a major problem of energy loss in the form of latent heat of condensation of water. Solar desalination based on the humidification - dehumidification principle leads to a major improvement in the efficiency of solar desalination units.

Two types of MEH (Multi Effect Humidification) units are studied, open-air closed-water cycle and open-water closed-air cycle. Three units based on both forced and natural convection of air are comprehensively reviewed. A rigorous mathematical model is developed to study the performance of one of the MEH units. A computer simulation program was used to study the effects of the collector, condenser and humidifier size. This is considered the only reliable method to study these effects and is possibly the first of its kind thus far.

Simulation of the performance of a typical MEH unit was carried out to account completely for the effect of different parameters. The complicated effect of water feed flow-rate on the desalination production was also studied. This showed the existence of a maximum desalination production. An economic evaluation and cost analysis of solar desalination with humidification - dehumidification principles was conducted in comparison with other solar-based desalination techniques. The cost components of the desalination process are presented in detail in this study.
1. COMPREHENSIVE LITERATURE REVIEW

1.1. Introduction

Water is available in abundance on the earth; however, there is a shortage of potable water in many countries. In the Gulf countries and many others, the non-renewable energy from oil and natural gas is used to desalinate water from seawater in multi-effect evaporators. It is also common in some places to use electric power to run Reverse Osmosis units for water desalination. In the first method, a large quantity of heat is required to vapourise the water, while the second method requires electric power to generate high pressure to force the water component of seawater through a membrane. Both methods consume large amounts of energy and require high skill for their operation. Nevertheless, these two methods until recently were considered as the most practical way of desalinating seawater. The Gulf countries, known for their shortage of drinking water, are also known for the availability of oil as a source of cheap energy. Due to the fossil fuel based energy consumption in both methods, CO$_2$ emission will always be an issue of environmental concern. However, there are many places where energy is too expensive to run such types of desalination processes. Sometimes, fresh water is required at locations far from the energy grid-lines and requires a local source of energy. Hence, such countries with rich resources of energy i.e. the Gulf countries, have shown strong interest in the desalination processes that often utilise renewable energy sources. Generally, water shortages occur at places with high solar radiation and usually peak during the hot summer months at maximum solar radiation. Hence, solar desalination could be one of the most successful applications of solar energy in most of the hot climate countries that have limited resources of fresh water.

1.2. History and Background: Solar Stills

Interest in seawater desalination goes back to the fourth century BC, when the Greek sailors obtained drinking water from seawater. The first work published on solar desalination was by Arab alchemists in 1551. However, the first solar still was designed and constructed in Chile by the Swedish engineer, Carlos Wilson in 1872, as described by Malik et al.$^1$ Since then, no work had been carried out on solar desalination until the end of the First World War. During World War II, Maria Telkes$^2$, developed a plastic still inflated with air, which was used by the US Navy and Air Force in emergency life rafts and published a series of more than twenty papers on this type of solar still, up to 1964. It is not the objective of this report to comprehensively review the work done on solar stills, however, certain features of its design led to new solar desalination processes such as the multi effect humidification-dehumidification (MEH), which is the objective of this report.

Even with very good insulation, the single-basin-still was found to distil water at low efficiency (usually below 45%), depending on the operating conditions, reported by Malik et al.$^1$, Cooper$^3$, Kudish et al.$^4$ and Farid and Hamad$^5$. The low efficiency of the still is mainly due to the high heat loss from its glass cover. A double glass cover reduces heat losses but also reduces the transmitted portion of the solar radiation and increases the cost significantly. The productivity increases when the solar still is
operated under reduced pressure (Yeh et al.\(^6\)), however, this was found impracticable because of the difficulties associated with reduced pressure operation.

Solar radiation impinging on a horizontal surface is not at its maximum except near the equator. A large number of investigators have modified the horizontal single-basin-still, usually fixed on a horizontal surface, to an inclined type to receive maximum solar radiation. Later, the tilted-tray and the wick-type solar still were developed. However, the construction costs of these complicated stills added significant cost penalties while the increase in the production of the stills was very limited.

The loss of energy in the form of latent heat of condensation of water at the glass cover is the major problem of the single-basin-still. Some investigators (Tiwari et al.\(^7\)) arranged the still in such a way as to have the water flow over the glass cover. Preheating of the feed water by passing it over the glass cover allowed only partial recovery of the latent heat with an increase in the still’s production up to 6 L.m\(^{-2}\) day. The flow of water over the glass cover reduced the amount of solar radiation received by the water in the still. Accumulation of salts and vapour leaks frequently caused defects in these units.

Further work to improve the efficiency of solar stills was carried out by El-Bahi and Inan\(^8\). The effect of adding an outside passive condenser to a single-basin-type solar still with minimum inclination (4\(^o\)), was investigated experimentally. This solar still yielded a daily output of up to 7 L.m\(^{-2}\). It is reported, that, in their study the solar still operated during the summer months with an efficiency of up to 75%. When the solar still was operated without a condenser, the yield decreased to 70% of that with the condenser.

A major improvement in solar still design is possible through the multiple use of the latent heat of condensation in the still. In such a unit, consisting of multi-cells, heat is supplied only to the first cell from a flat plate solar collector. Water is evaporated in the second effect as it trickles over a metallic surface heated by the condensation of the vapour from the first effect which allows the utilisation of the latent heat at different levels. A comprehensive review and technical assessment of various single and multi-effect solar stills is presented by Fath\(^9\), who highlights the impact of utilising latent heat of condensation via multi-effect solar stills. Mink et al\(^10\), have conducted an in-depth study on heat recycling, using a laboratory-scale solar still of 1.0 m\(^2\) area, designed to recycle the condensation heat of the distillate. The exposed wick surface area was 1.0 m\(^2\), with thermal incident energy of 650 W.m\(^{-2}\) being supplied by a solar simulator at a tilt angle of 20\(^o\). The forced circulation of ambient air was achieved using a low pressure (200 Pa) variable speed ventilator. Preliminary results showed an increase in productivity per unit area by a factor of two to three, compared with tilted-wick or basin-type solar stills respectively.

One of the most recent designs of this type of still is that described by Grater et al.\(^11\) and Rheinlander and Grater\(^12\) of a four-effect still. The evaporation process in a four-effect still for the desalination of sea and brackish water was experimentally investigated in a test facility under different modes and configurations of heat recovery, and natural or forced convection in the four distillation chambers (“effects”). The theoretical distillate output from a 4-effect distillation unit is 8.7
kg.m$^{-2}$h$^{-1}$ with an energy input of 2.0 kW.m$^{-2}$, for an active cross section of 1 m$^2$ of the apparatus, representing 4 m$^2$ of evaporator and 4 m$^2$ of condenser surface. The experimental unit consisted of the base module of the four-effect distillation unit with an active evaporation cross-section of 1.7 m$^2$ and the heating and cooling cycles.

The four-effect distillation unit consisted of four stages comprising of heating and cooling plates and three intermediate plates. The multi-effect still unit was tilted by 3° - 5° from the vertical. The heating cycle was connected via a heat exchanger to the heat transfer fluid test facility to simulate a solar collector cycle. The energy of the last condensation surface was absorbed by the cooling cycle, connected to a wet cooling tower via a heat exchanger. All operating conditions that were made possible by this configuration were examined by the use of a model. It was observed that the main advantage of the heat recovery system was the substantial improvement of the Gain Output Ratio (GOR) (defined as the ratio of the energy consumed in the production of the condensate to the energy input), by decreasing the demand for primary heating of the first effect. The GOR increases to 80% due to heat recovery from the distillate latent heat and higher concentration mass flows. Intermediate screens and forced convection in the colder effects improved the distillate output and therefore, GOR of the investigated four-effect still, considerably. For heating inlet temperatures greater than 90°C, the model predicted a distillate output in the first effect that was 50% higher than the measured values. The theoretically determined distillate output of 8.7 kg.m$^{-2}$h$^{-1}$, with an energy input of 2.0 kW.m$^{-2}$ and a GOR of 2.9, was not achieved in the experiments.

It should be noted, that the unit of Grater et al.$^{11}$ and Rheinlander and Grater$^{12}$, was operated with hot water at a constant temperature of 96°C. Under such high operating conditions, the evaporation and condensation would be very efficient but long-term operation was not practical due to scale formation. Furthermore, if a solar collector was used to drive the desalination unit then its collection efficiency would drop to very low values at such a high temperature.
Thus, multi-effect solar stills may carry out a more efficient desalination of seawater, but for only small capacities, since the condenser and the evaporator are integral parts of the still. The low heat and mass transfer coefficients in this type of still require operation at relatively high temperature and the use of large and expensive metallic surfaces for the evaporation and condensation.

In the following sections, a new class of solar desalination system is discussed; this design is based on a more efficient utilisation of the latent heat of condensation.

**1.3. Humidification-Dehumidification Technique**

**1.3.1. Principle of the Humidification-Dehumidification Process**

The most promising recent development in solar desalination is the use of the humidification-dehumidification (HD) process. The principle of operation of the HD process has been reviewed by Bourouni et al. The HD process is based on the fact that air can be mixed with significant quantities of vapour. The vapour carrying capability of air increases with temperature, i.e. 1 kg of dry air can carry 0.5 kg of vapour and about 670 kcal when its temperature increases from 30°C to 80°C. When flowing air is in contact with salt water, a certain quantity of vapour is extracted by air at the expense of the sensitive heat of salt water, which provokes cooling. The distilled water, on the other hand, may be recovered by bringing the humid air in contact with a cooling surface, which causes the condensation of a part of the vapour mixed with air. Generally, the condensation occurs in another exchanger in which salt water is preheated by the latent heat of recovery. An external heat contribution is therefore necessary to compensate for the sensitive heat loss.

The humidification-dehumidification technique is especially suited for seawater desalination when the demand for water is decentralised. Several advantages of this technique can be presented which includes, flexibility in capacity, moderate installation and operating costs, simplicity and the possibility of using low temperature energy (solar, geothermal, recovered energy or cogeneration). In this process, air is heated and humidified by the hot water received from a solar collector and it is dehumidified in a large surface condenser using the saline feed. Most of the latent heat of condensation is used for preheating the feed.

**1.3.2. Methods of Extracting Water from Humid Air under Atmospheric Conditions**

Prior to focusing on the details of the different types of humidification-dehumidification processes and their analysis, we briefly review work aimed to utilise the humidity in the atmosphere as a source of fresh water, and methods for extracting the water from humid air.
Methods for extraction of water from humid air include mechanical methods, refrigeration methods (absorption and vapour compression), adsorption methods and absorption methods.

1.3.2.1. Atmospheric Water Vapour Processing

Recently, Wahlgren\textsuperscript{14}, conducted a comprehensive review of atmospheric water vapour processing (AWVP). Three classes of “processor machines” for potable water production were distinguished. The machines served to cool a surface below the dew point of the humid air to concentrate water vapour through the use of solid or liquid desiccants to induce and control convection in a tower structure. Desalination, in theory, requires less than 1 kWh to remove salt ions from seawater to produce 1 m\textsuperscript{3} of fresh water. In practice, about 23 kWh are needed. In theory, AWVP consumes 681 kWh to condense 1 m\textsuperscript{3} of liquid water. By using natural heat sinks such as the atmosphere and deep-sea cold seawater, AWVP performance can be improved so that in the case of seawater coolant it is more productive than high-temperature evaporative desalination in terms of energy input per unit volume of product water. The natural range of absolute humidity is from 4 g to 22 g of water.m\textsuperscript{-3} of moist air with many population centres having values between 5 g.m\textsuperscript{-3} and 10 g.m\textsuperscript{-3}. A 40\% efficient machine in the rural surroundings of Antofagasta, Chile (absolute humidity 10.9 g.m\textsuperscript{-3}) with airflow of 10 m\textsuperscript{3}.s\textsuperscript{-1}, would produce 3767 L.day\textsuperscript{-1}. At a modest consumption of 50 L per person per day, 75 people could have their domestic water requirements satisfied. In Garissa, Kenya, with normal absolute humidity of 17.0 g.m\textsuperscript{-3}, an average family would need 900 L per day for drinking and household consumption requiring a 60\% efficient AWVP machine capable of 1 m\textsuperscript{3}.s\textsuperscript{-1} airflow. No costs or capacities for these machines (referred here) are mentioned, but energy requirements stated for the three classes of machines mentioned, are relatively higher than for the solar-based HD process.

1.3.2.2. Dew Collection

In 1981, Rajvanshi\textsuperscript{15} reported a scheme for large-scale dew collection as a source of fresh water supply. In the desert environment, dew collection takes place due to the night sky radiation cooling. This however, results in an insufficient quantity of water production. A more efficient method proposed would be to pass deep sea cold water through suitable heat exchangers for dew condensation. A heat exchanger field of 129,000 m\textsuperscript{2} can condense 643 m\textsuperscript{3} of dew over a period of 24-hours. Cold water for dew condensation may be obtained from a depth of 600 m. Three 200 kW wind machines power the pumping of 8,320,000 kg.hr\textsuperscript{-1} of this cold water. This method has also been studied by Wahlgren\textsuperscript{14} in his design of AWVP machines (mentioned above).

1.3.2.3. Adsorption Method

An adsorption method considered for obtaining fresh water from the humid air involves passing humid air over a hot adsorbing material when the water is adsorbed and cools the adsorbing material. In the regeneration process, the adsorbing material is heated. Later, the humid air flows over a colder condenser surface to condense the carried water and absorbs its water content. Alayli et al\textsuperscript{16}, in 1987, reported a study to
extract water from wet air based on the adsorption principle. A two phase cycle comprising of a nocturnal phase and a diurnal phase was proposed. In the nocturnal phase, an adsorbent composite material type ‘A,’ is exposed to the surrounding atmosphere in which the temperature and relative humidity rate can vary. The material ‘A’, is humidified by physico-chemical adsorption. In the diurnal phase, solar radiation heats up the wet composite material ‘A,’ to about 100°C. The water contained in the material is drawn up and it condenses on a cold plate. Experimental investigations using a certain type of composite material (‘A’) operating with conditions of 20°C temperature and relative humidity 50%, yielded 1 L.m\(^{-2}\) of drinking water per day. The quantity of water increased to 2 - 4 L.m\(^{-2}\) of material surface when the relative humidity was 80% and when another composite material (type ‘B’), was used.

\textbf{1.3.2.4. Absorption-Refrigeration Method}

Among the absorption-refrigeration methods to extract fresh water from humid air, Aly\textsuperscript{17}, presented a non-conventional method suitable for collecting the humidity from air in hot regions. In this process, water is a by-product of the cycle, originally used in air-conditioning. A solar driven LiBr-H\(_2\)O absorption-cooling machine is used with an open absorber where the ventilation air is dehumidified by direct contact with a concentrated LiBr-H\(_2\)O solution. The diluted solution is regenerated in a generator (concentrator), where the collected water is recovered to allow the concentrated solution to repeat the cycle. The recovered vapour is condensed and the condensation heat is used on a second generator, at a lower pressure, to promote the required cooling effect for the air-conditioning evaporator. The condensed vapour represents the fresh water by-product of the process. Further, its efficient air-conditioning function process contributes to decentralised fresh water production in hot regions. The by-product water production amounts to 3.11 L.m\(^{-2}\).day of the collector area, which is higher than that of basin-solar-stills.

\textbf{1.3.2.5. Vapour Compression-Refrigeration Method}

Vlachogiannis et al.\textsuperscript{18}, have proposed a novel desalination concept combining the principles of humidification-dehumidification and mechanical vapour compression refrigeration. They constructed a laboratory prototype to analyse and study the concept. Their process combines the principles of intensive evaporation and vapour compression refrigeration with a heat pump (mechanically intensified evaporation MIE). Air is injected into the evaporation chamber through a porous bottom wall and is dispersed as small bubbles. The emerging saturated stream is then compressed by a blower to a slightly higher pressure and directed to the adjacent condensation chamber. Due to the increased pressure, water condensation occurs at a slightly higher temperature than evaporation and the latent heat is transferred to the evaporation chamber through the thermally conducting sidewall. Experiments indicated that the prototype was successful. The concept could be promising for future developments.

A study to investigate the combination of desalination with cooling and dehumidification air conditioning was conducted by Khalil\textsuperscript{19}, for the climatic conditions of UAE coastal regions. The quantity of fresh water obtained depends on parameters i.e. properties of the humid air, air velocity, cooling coils, surface area and
heat exchange arrangement. In his study for maximum condensate yield, the heat and mass transfer mechanisms were analysed and the coil conditions optimised.

### 1.3.2.6. Absorption Method

Among the absorption-based techniques, Abualhamayel and Gandhidasan\(^{20}\), proposed the use of a suitable liquid desiccant to extract fresh water from humid air. The night-time moisture absorption and the day-time moisture desorption takes place in the same unit. The performance of the unit was predicted analytically for typical summer climatic data in Dhahran, Saudi Arabia, by solving the energy balance equations. For given operating conditions, it was shown that it was possible to obtain 1.92 kg.m\(^{-2}\) with the unit. The influence of absorbent concentration and flow rate on the performance of the system was analysed and it was found that the increase in absorbent solution flow rate increases the rate of absorption of water from the atmosphere but decreases the desorption rate of water during daytime operation. Further study is required to determine the economic feasibility of the system.

### 1.3.2.7. Humidification by Pervaporation

Novel methods of desalination based on the humidification-dehumidification principle have been proposed and studied by other investigators. Korngold et al.\(^{21}\), studied a new desalination process consisting of air humidification by pervaporation through hydrophilic or microporous hydrophobic hollow fibres followed by dehumidification using cooling water. Hot saline water is passed through hollow fibres in a recycled air-sweep prevaporation process, the saline water heated by waste heat, solar energy or any other sources of energy. The flux of water through the hollow fibres is in the range of 1.5 L.m\(^{-2}\).h\(^{-1}\) - 3.0 L.m\(^{-2}\).h\(^{-1}\), when water temperatures were between 45\(\degree\)C - 65\(\degree\)C. The calculated energy requirement for pumping air and water in a pilot plant unit of capacity 6.3 L.h\(^{-1}\) with 4 m\(^2\) of anion-exchange hollow fibres was approximately 2 kWh.m\(^{-3}\), when the hot water temperature was 60\(\degree\)C.

### 1.3.2.8. HD Using Hydrophobic Capillary Contactors

Bergero and Chiari\(^{22}\), presented an experimental and theoretical analysis of air humidification-dehumidification processes using hydrophobic capillary contactors. The HD processes were carried out in a hollow-fibre membrane contactor. During the humidification process, experiments were carried out using three different mass flow rates of water (19, 35, 54 kg.h\(^{-1}\)), while two different mass flow rates of LiCl saturated solution (25, 41 kg.h\(^{-1}\)), were used for air dehumidification. Air flow rates were also varied, ranging between 30 m\(^3\).h\(^{-1}\) - 80 m\(^3\).h\(^{-1}\) and the variations in the relative humidity of air and the temperatures of air and liquid were considered. Experiments indicated high mass transfer efficiency for both humidification and dehumidification.
1.4. Solar Humidification-Dehumidification (HD)

1.4.1. Early Development

In 1968, Garg et al.\textsuperscript{23}, reported a study with the aim of developing the HD technique for water desalination in the arid zones of India. A solar still developed in the Central Salt and Marine Chemicals Research Institute, Gujarat, India, had a productivity of 2.94 L.m\textsuperscript{-2} - 3.91 L.m\textsuperscript{-2} of still area, depending upon the variations in the intensity of solar radiation. Certain drawbacks of the solar-still technique were overcome in the solar powered HD technique. In the first stage of development of the HD technique, a 3 L.day\textsuperscript{-1} (24-hours) capacity experimental unit was fabricated with a tower of packing height 30 cm using Raschig rings as the packing material. The total height of the humidifier was 60 cm, with 15 cm top and bottom heads. The humidification unit was coupled with a surface condenser (dehumidifier). The distillate collected from the unit had a concentration of less than 50 ppm of salt. An electric heater was used to heat the brine. From the experimental runs, it was determined that for lower temperatures of approximately 55°C, a liquid-gas ratio (L/G) of the order of 3 was suitable. This unit had a production of 3.4 L.day\textsuperscript{-1} for a brine temperature of 60°C. A second unit was fabricated which had a capacity of 136 L.day\textsuperscript{-1} (24-hours). Experimental runs determined that the rate of production of fresh water, increased with an increase in the temperature of brine, providing other conditions e.g. liquid and gas rates were kept constant. The Reynolds number (airflow) calculated for the 3 litre capacity unit was Re = 285 and the 136 litre capacity was Re = 300. From the experiments on a 136 L.day\textsuperscript{-1} capacity unit, the rate of production of fresh water obtained was about 61.25 L.day\textsuperscript{-1} for a brine temperature of 59°C with an L/G ratio of 2.03. The production was lower than the designed capacity because the heat transfer area in the condenser unit was not sufficient, even though the heat transfer coefficient was relatively high, so that complete condensation of water vapour was not achieved. Tabulated results show an L/G of 3 was favourable and resulted in the production of 72 L.day\textsuperscript{-1}. A pilot plant with a capacity of 4540 litres of fresh water per day was designed for further study.

1.4.2. Forced convection in Solar HD

The scarcity of water and fossil fuel in the Canary Islands has led to the study of solar assisted desalination on the Islands. A forced convection humidification-dehumidification process has been analysed by Veza and Ruiz\textsuperscript{24}. The process of forced convection solar distillation differs from the conventional still, in that, vapour from salt water is absorbed by flowing air and sucked out to an external cooler where it is collected as condensate. The effect of convection was studied during water evaporation and vapour condensation at an external condenser. The humidification-dehumidification process is not adiabatic due to the external contribution of energy from solar radiation.

Two simulation models for the proposed process were developed and its output in terms of temperature and humidity was predicted. A simplified model using mass and energy balance relationships is presented as well as a general model which predicts
the system behaviour with greater accuracy with features such as recycling air or using new external air. A methodology for determining energy and exergy efficiency has been included, which is also applicable to other solar collection and conversion processes. An experimental design was adapted to validate the model by simulating various experimental pilot plant conditions. The design consisted of two stills with similar characteristics, except for the lapilli on the bottom, to assess differences. Still dimensions were 10 m long and 1 m wide for each still, with a cover of standard glass. At the exit, a seawater-cooled condenser was present and fans induced the forced flow of air. Feed water, air flow rates, water temperature (at three different points along the still), air temperature and humidity were measured. Ambient temperature and humidity, wind velocity and solar irradiance were also measured at the experimental site. The model results compare favourably with experimental data obtained from the pilot plant. The relationship between different parameters has been determined and optimum operating values have been selected in the study.

1.5. Multi-Effect Humidity (or HD) Process (MEH)

1.5.1. Principle of MEH

The principle of MEH plants is the distillation under atmospheric conditions by an air loop saturated with water vapour. The air is circulated by natural or forced convection (fans). The evaporator-condenser combination is termed “humidification cycle”, because the airflow is humidified in the evaporator and dehumidified in the condenser. The term “multiple effect” is not in reference to the number of constructed stages, but to the ratio of heat input to heat utilised for distillate production (GOR>1).

In studies by Grater et al.\textsuperscript{11}, efficient evaporation and condensation could be achieved at high temperature (near to 100°C); however, the thermal efficiency even for the highest quality flat plate collector drops significantly at such elevated temperatures. At moderate operating temperatures, intensive heat and mass transfer must be maintained in the evaporator and condenser. This necessitates the development of a new generation of solar desalination units.

In recognising this fact, extensive research has been carried out at different research institutes in Germany, reported by Heschl\textsuperscript{25}, to develop a more efficient utilisation of solar energy for water desalination. Grune\textsuperscript{26}, introduced the multi-effect humidity (or HD) process in the early 1960’s and his 1970 study gives an interesting insight into the process. The multiple-effect process promises higher specific productivity due to the separation of the three basic processes of energy collection, evaporation and condensation.

The MEH process extends the concept of the forced convection solar still further, by separation of heat collection and evaporation units. At the University of Arizona, pilot plant work from 1956 to 1963, initiated the construction of an experimental pilot solar energy multiple-effect humidification plant in 1963. The plant was constructed to test the feasibility of using solar energy as a heat source in a humidification system. Further work was initiated in 1964 by the University of Arizona in co-operation with
the University of Sonora, Mexico, when a larger pilot scale solar desalting plant at Puerto Penasco, Sonora, Mexico, was constructed. The Multi-Effect Humidification (MEH) process was developed over the years and a few units constructed and tested in different countries. There are two types: the open-water/ closed air cycle and open-air/ closed water cycle, described below.

1.5.2. MEH Units Based on Open-Water/ Closed-Air Cycle

In these plants, heat is recovered by air circulation between a humidifier and a condenser using natural or forced draft circulation. As shown in Figure 1.2, (Farid), the saline water feed to the condenser is preheated by the evolved latent heat of condensation of water. This heat is usually lost in the single-basin still. The saline water leaving the condenser is further heated in a flat-plate solar collector and then sprayed over the packing in the humidifier. Some of the MEH units used an integrated collector, evaporator, and condenser (Heschl and Sizman). The reported efficiency of these desalination units was significantly higher than that of a single-basin-still. These types of desalination units are very suitable for small production capacities in remote areas. The process is easy to operate and maintain and does not require skilled operators.

Chaibi, carried out a performance study of a solar MEH unit, installed in the south of Tunisia for potable water and irrigation. Several tests on storage, evaporation and condensation were carried out and an estimation of the cost of fresh water production was also given. The study showed that the plant, which was intended to produce 12 L.m$^{-2}$.day of fresh water, did not achieve this result. The highest production was approximately 6 L.m$^{-2}$.day.

Heschl, has constructed an MEH plant, which uses natural-draft air circulation. Textile heat exchangers were used for efficient evaporation and condensation with the minimum pressure drop. A GOR value higher than 3 was reported.

Khedr, performed a techno-economic investigation of an air HD desalination process. The results showed that 76% of the energy consumed in the humidifier was recovered by condensation. The cost calculations showed that the HD process has significant potential as an alternative for small capacity desalination plants and allows operation of systems with a small output, 10 m$^3$.day$^{-1}$. 
An increase in desalination productivity was achieved by increasing the water temperature at the inlet to the humidifier of the MEH unit, also, air circulation was found essential for raising the performance. Madani and Zaki\textsuperscript{31}, constructed a test rig of an MEH solar desalination plant working on the humidification principle. The unit yielded 0.63 L.m\textsuperscript{-2} hr\textsuperscript{-1} - 1.25 L.m\textsuperscript{-2} hr\textsuperscript{-1} of fresh water on a typical summer day at noontime (2.5 L.m\textsuperscript{-2} d\textsuperscript{-1} - 5 L.m\textsuperscript{-2} d\textsuperscript{-1} for a 4-hr.d\textsuperscript{-1} peak time operation), which is as low as some efficient single-basin-stills.

From 1990 to 1996, Farid and co-workers built three MEH desalination units in Iraq, Jordan and Malaysia. The unit constructed in Iraq was operated with forced air circulation (Farid and Al-Hajaj\textsuperscript{32}), while the unit constructed in Jordan was operated with both forced and natural draft air circulation (Al-Hallaj\textsuperscript{33}; Al-Hallaj et al\textsuperscript{34}; Nawayseh et al.\textsuperscript{35, 36, 37}). Based on the experience of operating these units, a third unit operated with natural draft air circulation was constructed in Malaysia (Nawayseh et al.\textsuperscript{36, 37}). These units were built with a single stage for the purpose of generating sufficient information to construct a rigorous mathematical model that could be used in the design and simulation of such units and also to optimise the performance of existing MEH units. The design and performance simulation of these units will be discussed (in Part 2.0) in detail.

**Figure 1.2. Schematic diagram of an experimental MEH desalination unit operated with forced or natural air circulation (Farid\textsuperscript{27})**
The Bavarian Center of Applied Energy Research and T.A.S. GmbH & Co. KG at Munich, Germany have addressed the performance optimisation of a multi-effect humidification unit (MEH) built in the Canary Islands. The unit was based on a patent design developed at the University of Munich, described earlier. The design of the unit was similar to that used by Farid et al.\textsuperscript{27}, except, that the humidifier and condenser were kept in the same unit and the unit was designed for higher capacity. The design of these two units was based on natural convection and not forced convection. After installation, their long-term performance was measured from 1992 to 1997. These distillation units illustrate the energy saving procedure of multi-effect humidification. Water is evaporated at ambient pressure and condensed where more than 70% of the heat of evaporation can be recovered. The performance of the units have been improved over the years and an average daily production of 100 litres out of 8.5 m\textsuperscript{2} collector area (11.8 L.m\textsuperscript{-2}.d\textsuperscript{-1}), has been obtained from the systems without thermal storage.

Muller-Holst et al.\textsuperscript{38}, studied a MEH unit shown in Figure 1.3. The desalination plant consists of a solar unit which provides the thermal energy and a desalination module that uses multi-effect distillation to treat the water. Seawater was fed to the unit and evaporated under ambient pressure; the saturated air is transported by free convection to the condenser area where it condenses on the surface of the plastic heat exchanger. The evaporator consists of vertically suspended tissues or fleece, made of Polypropylene and over which, the hot seawater is normally distributed. In the evaporator, partial evaporation cools the brine that leaves the evaporation unit concentrated at a temperature of approximately 45\degree C. The condenser is a Polypropylene bridged double plate heat exchanger through which the cool brine is pumped upwards. The condensate runs down the plates and trickles into a collecting basin. The heat of condensation is mainly transferred to the cold brine, as it flows upwards inside the heat exchanger. The temperature of the brine rises from 40\degree C to approximately 75\degree C. The brine is then heated up to the evaporator inlet temperature, which is between 80\degree C and 90\degree C by a heat source such as the highly efficient solar collectors or by heat from the thermal storage tank or by waste heat. The salt content of the brine and the condenser inlet temperature can be increased by a partial reflux from the evaporator outlet to the brine storage tank. If re-circulated brine needs to be cooled, the feed water is sent through a cooler before it reaches the condenser.

Based on this concept, a pilot plant with direct flow through the collectors has been working almost without any maintenance or repair for a period of more than seven years on the island of Fuerteventura. The results from Fuerteventura for a distillation unit without thermal storage showed that the daily average heat recovery factor (GOR) was between 3 and 4.5. A similar distillation unit in the laboratory at ZAE Bayern yielded a GOR of more than 8 at steady state conditions. The optimised module produced 40 L.h\textsuperscript{-1} of fresh water, but it was shown that a production of 1000 kg.day\textsuperscript{-1} is possible when the unit was operated continuously for 24-hours. Based on a collector area of 8.5 m\textsuperscript{2}, the productivity of the optimised module worked out to be 113 L.m\textsuperscript{-2}.day\textsuperscript{-1} for a 24-hour run. It was realised that an improvement of the overall system efficiency could be reached by adding thermal storage as an alternate heat source to enable 24-hour operation of the distillation module. It was achieved by using extra collectors and hot water storage tanks. In a related study, Ulber et al.\textsuperscript{39}, investigated the concept of a solar thermal desalination plant with a heat storage tank installed. A unit was constructed in 1997 in Tunisia with the financial support of the German Ministry of Economic Co-operation and Development (BMZ). In addition, a
concept was developed and implemented in Sfax, Tunisia, which included a conventional heat storage tank and heat exchanger between the collector circuit (desalted water) and the distillation circuit. This enabled continuous (24 h.d$^{-1}$) distillate production. A major factor prompting a 24 h.d$^{-1}$ operation of these units was the realisation that the major capital cost of these units was due to the condenser and humidifier. It had been suggested that to improve the performance of the MEH unit constructed in Tunisia using 38 m$^2$ collectors by including a 2 m$^3$ of storage tank. A similar suggestion had been made to extend the operation of the unit constructed in Jordan and Malaysia to be operated on 24 h.d$^{-1}$ period (Al-Hallaj et al. 34; Nawayseh et al. 37).

1.5.3. MEH Units Based on Open-Air/ Closed-Water Cycle

In the study of multi-effect processes by Böhner 40, a description of a closed water circulation system is presented, see Figure 1.4. The closed water circulation is in contact with a continuous flow of cold air in the evaporation chamber. The air is heated and loaded with moisture as it passes upwards through the falling hot water in the evaporation chamber. After passing through a condenser cooled with cold seawater, the partially dehumidified air leaves the unit while the condensate (distillate) is collected. Water is recycled or re-circulated. Incoming cold air provides a cooling source for the circulating water before it re-enters the condenser. This system with a closed salt water cycle ensures a high utilisation of the salt water for fresh water production. In the closed water cycle, the salt water is evaporated continuously in the evaporation chamber $e.g.$ 1 m$^3$ saltwater with 1% salt, results in 330 L distillate water. The brine has a rest salt concentration of approximately 1.5%.
Some investigators (Assouad and Lavan\textsuperscript{41}), applied an open-air cycle for obtaining good productivity. The air was vented to the atmosphere after its partial dehumidification in the condenser while the water was circulated in a closed cycle. The productivity of the units working on this principle was high, but the power required for air circulation was also very high. The system consisted of a humidifier, a solar still in the form of a flow channel, a condenser and a pond. The solar still was a long glass-covered channel about 200 m long. Sensitivity studies carried out on the channel continued to explore parameters such as wind velocity, air flow rate, inlet water temperature and flow rate. The channel dimensions selected, length of 177 m and width of 1.69 m. Vertical dimensions chosen were 0.31 m x 0.76 m. It was observed that the performance increased with increasing air flow rate but practically levelled off at about 2500 kg.h\textsuperscript{1}.

Recently, a MEH unit based on this principle was built at Kuwait University (Delyannis and Belessiotis\textsuperscript{42}). It received energy from a salt gradient solar pond of 1700 m\textsuperscript{2} producing 5.8 L.m\textsuperscript{-2}.day, used to load the air with humidity. Fresh water is collected by cooling the air in a dehumidifying column, producing 9.8 m\textsuperscript{3}.day\textsuperscript{-1} of distillate. In a similar study, Younis et al\textsuperscript{43}, describes a solar operated HD desalination system, which consists of a solar pond, humidifying column, dehumidifying stack and the necessary fans and pumps. For an average solar intensity of 21,000 kJ.m\textsuperscript{-2}, and 22\% efficiency, a heat rate of 90.9 kW thermal energy can be obtained from the 1700 m\textsuperscript{2} solar pond built at Kuwait. A performance ratio of 3 (800 kJ.kg\textsuperscript{-1} of distillate) is obtainable, with an output of 9.8 m\textsuperscript{3}.d\textsuperscript{-1}, as mentioned in the study.
The process described by Khalil\textsuperscript{19}, as discussed earlier (Section 1.3.2) where the moist air is passed over the cooling coil of an air conditioner falls under the open-air/closed-water cycle MEH category. It is noted that the method may be economical only if the produced fresh water was considered as an air-conditioning by-product.

Dai and Zhang\textsuperscript{44}, have also built an MEH unit operated in an open-air, closed water cycle. The unit was 1 m x 1 m x 1.5 m and capable of producing up to 100 L.hr\textsuperscript{-1} of fresh water. They replaced the collector with a boiler to provide the hot water, which was sprayed at the surface of the honeycomb packing of the humidifier. A fan was used to force the process air to flow through the humidifier in a cross flow arrangement. The hot humid air was then passed through a condenser, cooled by cold seawater prior to feeding the boiler. The seawater captures some of the latent heat of condensation thereby improving the efficiency of the unit. The water from the humidifier was recycled to the storage tank since it was warm and its salinity, not very high. However, some bleeding of this water was required to prevent the accumulation of salt in the unit. An efficiency of 0.8 was obtained using hot water feed from a boiler. This corresponded to an efficiency of about 0.68 when a solar collector was included in the calculation. The unit production did not exceed 6.2 L.m\textsuperscript{-2}.day. The authors showed a strong effect of the humidifier feed water temperature, which had been reported previously in all types of MEH units. The effect of air flow-rate on the production efficiency showed a maximum value. Increasing the airflow rate first increased the heat and mass transfer coefficients in the humidifier and condenser but eventually lowered the operating temperature. This was the reason for the maximum efficiency observed.

Another MEH unit based on open-air cycle and referred to as "Dewvaporation", was built at Arizona State University, for the production of 45.4 kg.day\textsuperscript{-1} of condensate, with GOR values in excess of 7.5. The evaporator unit was constructed out of strips of thin water wettable plastics and operated at a low-pressure drop. This system was studied and experimentally operated by Beckman\textsuperscript{45}, and could emerge as an economically feasible for small capacity plant applications. As mentioned in his study, RO technology faces competition from other seawater desalination techniques such as MVC, MSF, and MED with and without Thermal Vapour Compression. The electrically driven MVC plants consume more electricity than RO plants. The thermally driven plants attempt to recycle the applied heat continually to minimise the operating costs. The energy reuse factor; economically varies from 6 to 12. The optimum GOR value depends on factors such as plant capacity, cost of energy and materials, interest and tax rates.

Other studies carried out on desalination systems based on the HD principle are described in the following sections. Although all these processes are based on the HD principle, the respective researchers have presented them under different process titles and descriptions.

### 1.5.4. Solar Multiple Condensation Evaporation Process

In 1991, Graef\textsuperscript{46}, studied a desalination process based on a Solar Multiple Condensation Evaporation (SME) cycle, and in further related studies Ben Bacha et al.\textsuperscript{47}, presented a study of a water desalination installation using the Solar Multiple Condensation Evaporation Cycle (SMCEC). In the study by Graef\textsuperscript{46}, it is mentioned
that the number of heat recovery cycles depended on the condenser surface area and temperature of the cooling water. A collector efficiency of 58% and a water temperature of 65°C - 75°C could produce 6 L.m\(^{-2}\).day\(^{-1}\) of condensate based on 1 m\(^2\) collector and condenser surface areas. Tests in Sfax, Tunisia, produced condensate of 4 L.m\(^{-2}\).day with a collector efficiency of 46% (theoretical 14.3 L). Both types of desalination units, namely, SME 3.6 and SME 200, were fabricated by Aquasolar GmbH & Co., and Aquasolar (SARL), Tunisia. The SME 3.6 is most suitable for a single family, producing up to 50 L.d\(^{-1}\) and has been in series production since 1991 in Tunisia.

The SMCEC based desalination unit presented by Ben Bacha et al.\(^{47}\), belongs to a new generation of decentralised installations for water desalination using solar energy with heat recuperation. Similar to the solar HD and MEH units, the SMCEC based units are well suited for developing countries with extended rural areas because of its simplified design, low maintenance, extended life time (over 20 years), almost zero energy consumption and low capital cost. A detailed modelling, simulation and experimental validation for this type of installation permits the optimisation of size of the solar collectors, evaporation tower and condensation tower similar to the modelling and simulation for a MEH unit studied and presented in Part 2.0 of this report. The SMCEC based desalination unit consists of three main parts: solar collector, condensation tower and evaporation tower. The flat-plate collector is equipped with an absorber of a Polypropylene material covered by a Hostoflan membrane or glass. The absorber is made up of very thin and tightly spaced capillary tubes where the salt water circulates. The evaporation tower produced the water vapour. Thorn trees are utilized to increase the water spray and improve evaporation. At the beginning, the brackish or seawater was heated by the solar collector. Then, the hot water was injected to the top of the evaporation tower. An atomizer with a special shape was used to insure the formation of uniform drops of the hot water in all the sections of the tower. Air circulation in the evaporation is possible either by natural or forced convection.

To examine the validity of the model proposed by Ben Bacha et al\(^{47}\), experimental measurements were taken using the pilot desalination unit located at the National School of Engineering of Sfax, Tunisia. The specifications of the pilot unit are: solar collector area of 7.2 m\(^2\) (effective transmission absorption of 0.83 and loss coefficient 3.73 W.m\(^{-2}\).K\(^{-1}\)), evaporation tower of 1.2 m x 0.5 m x 2.55 m, with solid packing of thorn trees and a condensation tower of 1.2 m x 0.36 m x 3 m. Based on model simulation and experimental validation, the optimum operation and production for the SMCEC unit requires perfect insulation of the unit, high water temperature and flow rate at the entrance of the evaporation tower, low water temperature at the entrance of the condenser, hot water recycling by injection at the top of the evaporation chamber and a storage tank to store the excess hot water that would extend water desalination beyond sunset.

1.5.5. Aero-Evapo-Condensation Process

Bourouni et al\(^{48}\), conducted an experimental investigation with a desalination plant using the “aero-evapo-condensation” process. The unit consisted of a falling film evaporator and condenser made of Polypropylene, and designed to work at low temperatures (70°C - 90°C), specifically using geothermal energy. The prototype was patented by the firm Caldor-Marseille, France in 1994. This prototype included two
cross flow heat exchangers, a horizontal falling film evaporator and horizontal falling film condenser. The two exchangers are made of Polypropylene and affect the humidification-dehumidification of air. The influence of the various thermal and hydrodynamic parameters on the unit performance was investigated. Results showed that the performance of the unit increased with inlet hot water and air temperatures. On the other hand, it was observed that the performance of the unit decreased when the air velocity and hot liquid flow rate was increased. A critical film flow rate corresponding to the film breakdown was determined. At this value, a maximum amount of evaporated water was obtained. Horizontal-tube falling film evaporators have an advantage over vertical-tube evaporators in dealing with problems such as; liquid distribution, levelling, non-condensable gases on the tube side, fouling and liquid entertainment. Another parameter affecting the heat transfer coefficients is the water distribution system at the top of the horizontal tube. Instead of the common “perforated-plate” water distribution system; the more accurately controlled “thin-slot” water distribution system was shown to be preferable.

1.5.6. Carrier Gas Process

Larson et al\textsuperscript{49} and Hamieh et al\textsuperscript{50}, in their studies, reported in 1989, presented results of the “Carrier-Gas Process” (CGP) of EvCon Corp., which demonstrated potential for desalination of seawater and brackish water and for the concentration of various process streams and industrial wastewaters. It operated at temperatures below the normal boiling point and at ambient pressure. A novel feature in this process was that the two chambers, one for evaporation and the other for condensation, as shown in Figure 1.5, are physically separated by a common heat-transfer wall. The CGP process provided results over a wide range of performance ratios and production densities simply by varying the temperatures and air flows. This type of flexibility is not possible in conventional Multi Stage Flash (MSF) installations. The system can also be operated using renewable heat supplies including solar and ambient air. Thus, this system is also a convenient choice for remote and arid regions of the world where conventional technology is too expensive. Although the future looks encouraging for this relatively new technology, further experiments and field testing is required before the CGP system would be commercially available and could replace the conventional Reverse Osmosis and MSF installations in many areas of the world.

ECOTERM\textsuperscript{51}, an organization formed by engineers and architects based in Barcelona, claim to be developing a pilot plant for desalination using the Carrier Gas Process. This principle is similar to the humidification-dehumidification technique and has been studied separately as an alternative desalination process. Expected results from the proposed pilot unit include a product water flow rate of 1 m\textsuperscript{3}.h\textsuperscript{-1} for an air flow rate of 7.55 kg.h\textsuperscript{-1} and a heat exchange surface area of 500 m\textsuperscript{2}. The possible plant sizes suggested are for productivity between 100 L.h\textsuperscript{-1} and 1000 m\textsuperscript{3}.h\textsuperscript{-1} or higher. Capability to work with low temperatures of approximately 40°C and using forms of energy such as residual heat of cogeneration, solar energy and geothermal energy are the possibilities of this proposed unit.
1.6. Summary of Studies Conducted on HD Desalination Process

A summary of the main technical results reported by various researchers is grouped as shown in Tables 1.1, and 1.2. Almost all the investigators state that the effect of water flow rate on the performance of the unit is important. The effect of air flow rate on productivity is termed insignificant by all authors except Younis et al. Also, all researchers express preference for natural convection since air flow rate has no significant effect on unit productivity. However, forced circulation could be feasible with another cost-effective source of energy such as wind energy. The effect of air flow rate is only noticeable at temperatures of approximately 50°C, as reported by Al-Hallaj et al.

![Diagram](image)

Figure 1.5. Carrier-Gas Process System Schematic Diagram (Larson et al.)

Another variable tested, was the packing material in the humidifier. Packing material should generally be of such size and shape as to provide a high contact surface and low pressure drop. The choice of packing material tends to have an effect on the thermal efficiency and productivity of the unit. Examples include Raschig rings, Berl saddles, Pall rings, Lessing rings, Prym rings, meshed curtains, wooden slats, wooden shavings and fleeces made of Polypropylene or honey-comb paper as used by some researchers.

1.7. Efficiency and Production Improvement Studies in Solar Desalination

Progressing studies and methods to improve the fresh water production rate and system efficiency are required to make the solar MEH class of desalination units commercially viable. A brief review of improvisation studies with reference to solar desalination in general is presented in the following sections.
### Table 1.1. Studies on Closed-Air/Open-Water Cycle System

<table>
<thead>
<tr>
<th>Ref.</th>
<th>Unit Features</th>
<th>Parameters Varied</th>
</tr>
</thead>
<tbody>
<tr>
<td>Younis et al.</td>
<td>Solar Pond HDD Forced Air Circulation Packing Material-Meshed Curtains</td>
<td>Air flow rate has a significant effect on a collected amount of fresh water. Study of different packing material required for improving performance.</td>
</tr>
<tr>
<td>Abdel Salam et al.</td>
<td>Two closed loops: air and water loops Forced Air circulation</td>
<td>Inlet water and air dry-bulb temperature to the cooling tower varied. Water flow rate through the cooling tower and air cooler varied. Increase of inlet water and air dry-bulb temp: to the humidifier and increasing humidifier size will increase system productivity, and decrease energy required in pumping water and air. Air recirculation is essential for raising system performance.</td>
</tr>
<tr>
<td>Farid et al.</td>
<td>Closed-Air Cycle Natural and Forced Circulation Humidifier packing-wooden slats</td>
<td>Effect of water flow rate is significant on the heat and mass transfer coefficients in the condenser and humidifier. Effect of air flow rate is small. Natural circulation preferred. Simulation program required to conduct detailed study</td>
</tr>
<tr>
<td>Farid and Al-Hajaj</td>
<td>Closed-Air Cycle Forced Circulation Humidifier packing-wooden shavings</td>
<td>Forced circulation not suitable due to high consumption of electrical power. Forced convection is practical with wind energy. Decreasing water flow rate causes more efficient evaporation and condensation. Decreasing water flow rate below 70 kg.h(^{-1}) reduces the performance factor due to an expected decrease in efficiency of collector at elevated temperature. Production of 12 L.m(^{-2}).d achieved.</td>
</tr>
<tr>
<td>Nawayseh et al.</td>
<td>Unit in Jordan tested Closed-Air Cycle Forced and Natural Circulation Humidifier packing-wooden packing</td>
<td>Simulation Program developed to describe performance of units Significant effect of water flow rate on unit production with two opposite effects: lower evaporation and condensation efficiency and higher collector efficiency Effect of air flow rate negligible. Natural convection preferred. Surface area of condenser varied-significant improvement in production if condenser area is doubled. Humidifier surface area &gt;5.6 m(^2) can increase production insignificantly Increase in evaporator and condenser surface area increases productivity significantly</td>
</tr>
</tbody>
</table>
continuation of Table 1.1.

<table>
<thead>
<tr>
<th>Authors</th>
<th>Study Description</th>
<th>Results/Findings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Al-Hallaj et al.</td>
<td>Two Units (bench scale and pilot) built and tested in Jordan. Closed-Air Cycle</td>
<td>Condenser Area varied from 0.6 m² in bench unit with single condenser and 8.0 m² in pilot unit with double condenser. Humidifier inlet temp. 60°C - 63°C. No significant improvement with forced circulation. Significant effect only at temp: of approximately 50°C. Production of pilot MEH Unit increases with night operation using rejected water from humidifier. Productivity estimated 8 L.m⁻².d⁻¹. Large mass of outdoor unit (approx: 300 kg) is negative factor. Use of lighter material for construction proposed. Night time operation recommended.</td>
</tr>
<tr>
<td></td>
<td>For Natural Circulation</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Humidifier packing-wooden slats</td>
<td></td>
</tr>
<tr>
<td>Nawayseh et al.</td>
<td>Two Units in Jordan and One in Malaysia studied. Closed-Air Cycle Natural Circulation</td>
<td>Due to larger contact area of humidifier and condenser, productivity improved significantly in the unit in Malaysia compared with that in Jordan. Increase in humidifier area was 136%, single condenser area 122.5% and double condenser area was 11.25%. Humidifier cross-section of unit in Malaysia reduced by 39% to that of unit in Jordan. But complete wetting not achieved. Computer simulation for design required to achieve optimisation of unit. Effect of water is significant on heat and mass transfer coefficients in humidifier and condenser. Effect of air flow rate small.</td>
</tr>
<tr>
<td>Nawayseh et al.</td>
<td>Units in Jordan and Malaysia studied</td>
<td>Simulation program developed to correlate with experimental results. Effect of air flow rate found insignificant. Increasing water flow rate was found to decrease production due to lower evaporation. However, reducing water flow rate to extreme values was found to lower production due to drop in collector efficiency. Simulation shows fast convergence of temperatures and production rate to final values.</td>
</tr>
<tr>
<td>Muller-Holst et al</td>
<td>Pilot Solar MEH unit at Canary Islands and desalination unit at Sfax, Tunisia studied. Closed-Air Cycle Natural Circulation Evaporator and condenser located in same unit Evaporator Packing Material-fleeces made of Polypropylene</td>
<td>Simulation tool for optimising collector field and storage developed at ZAE Bayern. Cost of water determined using laboratory based simulation criteria. Laboratory unit yields reduced specific energy demand due to better evaporation surfaces and thinner flat plate heat exchangers on the condensation side of the unit. Simulation results for optimisation of 24-hour operation to be verified in field tests.</td>
</tr>
</tbody>
</table>
continuation of Table 1.1.

| Muller-Holst et al $^{54}$ | Simulation of solar thermal seawater desalination system using TRNSYS. | Simulation results compared with data from pilot plant in Fuerteventura, which were in good agreement. Increasing evaporator inlet temp: causes rising distillate volume flow. Higher distillate volume flow achieved naturally with increasing brine volume flow (load flow > 600 L.h$^{-1}$). Best condenser inlet temperature found to be 40°C. Simulation tool for the desalination unit to be used in unit configuration with storage tank. |

Table 1.2. Studies on Open-Air/Closed-Water Cycle System

<table>
<thead>
<tr>
<th>Ref.</th>
<th>Unit Features</th>
<th>Parametric Analysis</th>
</tr>
</thead>
<tbody>
<tr>
<td>Khedr $^{30}$</td>
<td>Open-Air System</td>
<td>Studies cover: Water to air mass ratios range from 60 to 80 and the concentration ratio between 2 and 5 at temp: difference between 10°C -16°C, along the liquid for dehumidification 76% energy consumed in humidifier recovered by condensation Increase of concentration ratio to 5 can reduce make-up water and rejected brine by approx: 58% and 24% respectively HD Process has significant potential for small capacity desalination plants as small as 10 m$^3$.d$^{-1}$</td>
</tr>
<tr>
<td>Dai and Zhang $^{44}$</td>
<td>Closed-Water System Forced Circulation Packing material-honey comb paper</td>
<td>Thermal efficiency and water production increases with increase of mass flow rate of water to humidifier Effect of rotation speed of fan on thermal efficiency studied. The lower the temp: of inlet water to humidifier, the smaller the optimum rotation speed. Optimum mass flow rate of air exists. Higher or lower mass flow rate of air is not recommended for increasing both water production and thermal efficiency Thermal efficiency and water production increase significantly with temp: of inlet water into humidifier. Productivity achieved approximately 6.2 kg.m$^{-2}$.d$^{-1}$</td>
</tr>
</tbody>
</table>

1.7.1. Improvements in Evaporator Performance

Takada and Drake $^{55}$, carried out research on the applications of an improved high-performance evaporator. Development of an evaporator that can achieve a high performance even when operating at low temperature is one of the keys to efficient
utilisation of low-grade waste heat and solar energy. An improved horizontal-tube thin-film evaporator with a high heat transfer coefficient working at low temperatures is described in their study. The conventional single-pass, horizontal-tube, thin-film evaporator has brine evaporating from one side of the heat exchanger surface and vapour condensing on the other, although having a high heat transfer coefficient, it still shows a rapid decline in performance as the working temperature is reduced. The proposed system has a tube bundle with a multi-pass arrangement for the vapour, which condenses inside the tubes. The first (bottom) pass has the greatest number of tubes and the number decreases by about half in each successive pass to the top of the bundle. Salt water is sprayed evenly onto the top of the bundle and descends as a thin film over the outside of the tubes. The purpose of the gradually diminishing number of tubes in each successive pass is to maintain a high vapour velocity in the tubes, which would otherwise decrease as the mass vapour flow decreased due to partial condensation. In the top pass, non-condensable gases are finally sub-cooled in the exit region of the tubes where they accumulate and are ready for venting. The three special features of this evaporator are self-desuperheating, blowing out the condensate due to a high vapour exit velocity and good separation of condensate. Among the applications mentioned for the use of this highly-improved evaporator, is its use in a solar heat multi-effect distiller located at Takami Island, Japan, and operated for 2 years, as reported in 1983. The pilot plant has 16 effects and a maximum summer capacity of 16 tons.day\(^{-1}\). The heat collecting system operates independently of the distiller and automatically starts collecting heat whenever sunshine is available. The distiller begins operating automatically when the water temperature in the heat accumulator reaches 75\(^{\circ}\)C in summer (62\(^{\circ}\)C in winter), and the accumulator stores enough heat to enable operation to continue overnight and during cloudy weather for several days in succession. The temperature range for seawater is from approximately 29\(^{\circ}\)C in summer to 11\(^{\circ}\)C in winter. The plant is designed to perform efficiently with a temperature difference of only 39\(^{\circ}\)C between heat source and cooling water. The typical operating data for an average winter day of partial sunshine is reported as follows:

| Heating Water Inlet Temperature | 53.0\(^{\circ}\)C |
| Heating Water Outlet Temperature | 45.8\(^{\circ}\)C |
| Evaporating Temperature in 1\(^{\text{st}}\) Effect | 37.5\(^{\circ}\)C |
| Evaporating Temperature in 16\(^{\text{th}}\) Effect | 19.5\(^{\circ}\)C |
| Seawater Inlet Temperature | 12.7\(^{\circ}\)C |
| Seawater Outlet Temperature | 17.4\(^{\circ}\)C |
| Heating Water Flow Rate | 48.5 Tons.day\(^{-1}\) |
| Product Water Flow Rate | 6.82 Tons.day\(^{-1}\) |

From the above data, it is shown that each evaporator (effect) works with a mean temperature difference as low as 1.2\(^{\circ}\)C.

Recently, Kalogirou\(^{56}\), conducted a study for the design of a low-cost evaporator. The new type of evaporator proposed, was the spray-type, which causes spraying of the seawater into fine droplets to evaporate the water. The cost was mentioned to be relatively low due to the absence of heat exchangers. With proper design, the spray-
mode of evaporation was superior to both the pool-boiling and thin-film evaporation. It was observed that the rate of evaporation was mainly influenced from the droplet size and temperature. Typical expected results are presented in the study, which prove the viability of the system. The complete system consists of the evaporator, a solar collector 1 m$^2$ in area and pumps modelled with TRNSYS, which gives an output of 11.2 m$^3$y$^{-1}$ of fresh water. This figure is comparatively better than the best enhanced solar still. The maximum collector outlet temperature is approximately 70°C and the maximum production of the unit at this condition is about 15.8 L.h$^{-1}$.

A new technique has been developed at the University of Bahrain to improve the efficiency of horizontal solar desalination stills by improving the efficiency of both evaporating and condensing zones and a new cheap method of storing excess solar energy during the day and for the continuation of the process at night, has been suggested by Rahim$^{57}$. Exhaustive data collected over a few years has been analysed and presented to show the effectiveness of the new techniques in improving the efficiency of small solar desalination units. One of the features mentioned in this new technique for solar stills, is the extraction of water vapour from the evaporating zone before it is allowed to reach the glass cover where it is then forced to condense inside copper tubes, which are held at relatively lower temperatures than that of the water vapour produced. In an experimental rig, a fan-condenser set-up was introduced to suck out the water vapour and force it through L-shaped copper tubes. These tubes, which act as a condenser unit, were kept at lower temperatures at the bottom of the tank. Advantages of using a forced condensing system over a conventional free condensing system are presented in the study. For the non-insulated experimental unit, the average efficiency for the free condensing without using the fan was 19.41%. With the introduction of the forced condensation facility, the average efficiency increased to 31.1%. Taking into account the electrical energy to the small fan, the efficiency of the unit still averaged 29.55%.

To improve the performance of the evaporating zone, an aluminium sheet, painted black at the top surface and thermally insulated at the bottom, was placed about 20 mm below the surface of brackish water, which when introduced allowed the heat storing capability at sunshine hours to be utilised for the continuation of the process during the night. Utilisation of the heat absorbed by the back wall of the evaporator improved the efficiency of the system by about 29%. This modification has effectively increased the surface area of the evaporator zone without changing its physical dimension.

1.7.2. Improvements in Condenser Performance

Efficiency-improving techniques, introduced for the condensing zone are the separation of the evaporating and condensing zones in order to keep the evaporating condensing zone as hot and as cold as possible, respectively. This also allows insulation of the evaporating zone so as to reduce the heat losses to the surroundings and to shade the condensing unit to keep it as cool as possible. Investigation into the design of the copper tube of a condensing unit for higher efficiency was carried out. For high air-velocity (approx: greater than 5 m.s$^{-1}$), it was found that the straight pipe without bending had better performance. At low flow rates of up to 4.5 m.s$^{-1}$, and a bath temperature of 27°C, pipes of 2 m length gave better condensation than pipes of 4 m length.
1.7.3. Studies on Convection Systems

An experimental study has been carried out by Khalifa\textsuperscript{58}, to compare the performance of natural and forced natural domestic solar water heaters. Several measurements have been made for both cases, which includes: the collector water inlet and outlet temperatures, mass flow rate, tank temperature, ambient temperature and the solar insulation. The main parameters for the solar collector are calculated for the natural and forced convection systems.

Tests were carried out on a south-oriented solar water heater that incorporated two flat plate solar collectors tilted at an angle of 45° to the horizontal and a 170 litre thermal storage tank. Each solar collector had a 1.42 m\textsuperscript{2} absorber plate contained in an insulated case and covered with a 4 mm single glass cover. The solar absorber is made from ten extruded aluminium finned tubes. A dye-injection method was used to measure the mass flow rate of the water in the system in the case of the natural circulation system.

It is observed that the hourly efficiency of the forced system is much greater than that of the natural system. The hourly efficiency for both cases tended to decrease with time due to an increase in the overall loss coefficient of the collector as the mean absorber temperature was increased. Despite the fact that the solar power incident on the forced and natural circulation systems throughout the test hours is very close (8565 W and 8759 W, respectively), the useful energy collected by the forced circulation system was found to be greater by about 68% (5266 W compared to 3122 W). Therefore, it has been experimentally determined from tests, that the efficiency through the working hours of the forced circulation system was 35% to 80% higher than that of natural circulation system. The use of the forced circulation system may be recommended when the improvement in the system performance offers the extra complexity and cost of the forced circulation system.

Earlier Abdel-Salam et al\textsuperscript{52}, conducted a study on a HD desalination system. The effect of air recirculation was observed in their study. It was determined that air recycling causes about 20% reduction in the required size of the cooling tower at the inlet water temperature of 60°C to the cooling tower. It was also observed that air recycling would cause about 33% reduction in the air-cooler size. An increase of 16% and 37% in the productivity factors of humidifier and air cooler, respectively, resulted due to the recirculation of air. Overall, air recycling causes approximately an 18% increase in system performance.

Other methods to improve the production rate have been studied, with the most important method of a 24-hour operation with storage capability being mentioned in this study. Many investigators proposing different systems of heat storage and heat recovery have carried out studies with encouraging results being displayed. Some of the results are briefly described below.

1.7.4. Heat Recovery Systems

Recently, Schwarzer\textsuperscript{59}, proposed a new solar thermal desalination system with heat recovery. Because of the heat recovery process, a higher thermal performance than the
conventional still-type solar distiller is achievable. Additionally, a continuous water-flow through the stages of the unit avoids salt accumulation. The disadvantage mentioned, was that the cost of installation was higher when compared to the still-type unit. The basic system components are a flat plate collector and a desalination tower. The working fluid in this system was oil. The system was powered by solar energy and installed on the northeast coast of Brazil. Results indicate that the unit can reach a water production rate of 25 L.m⁻².d for a value of 4.8 kWh.m⁻².d of solar radiation. This represents a rate of 5.2 L.kWh⁻¹.m⁻² and a factor of 5-6 times greater than the tank-type distiller.

A practical scale solar desalination system harnessing only solar energy as the heat source from solar collectors and the power source from solar cells is in operation at the Al Azhar University in Gaza (Abu-Jabal et al.⁶⁰) The basic principle of the unit is that when the air in the system is evacuated, the water boils at low temperature. When the conditions are of higher temperatures of approximately 40°C and lower temperatures of approximately 5°C, solar radiation can be used many times for the distillation by re-use of the latent heat of condensation of steam for the next evaporation of steam at a few Kelvin degrees lower. The smaller temperature difference between an evaporator and a condenser makes it possible to get more fresh water. It was determined that distilled water productivity for this system is higher than other solar desalination processes.

1.7.5. Heat Storage Systems

It has been mentioned earlier in this study that certain researchers investigated the use of hot water storage tanks and running the desalination units for a continuous period of 24-hours to enhance the productivity of fresh water. In other studies carried out by Voropoulos et al.⁶¹, it was found that coupling a solar still with a hot water tank, generally doubles the distilled water output in the 24-hour period, as a result of continuous heating of basin water from tank water. Increases are higher at night than in the day, since at night, differences in water and cover temperatures are generally higher, resulting in higher production rates. It was observed from studies that there is a significant increase in water productivity from the system when the solar still is coupled with a storage tank, with almost double the productivity being noticed. The increase in night productivity is almost 60% - 180% greater than during the day when it is 30% - 100%. Also, in the case of coupling with a storage tank, 40% of the 24 h production was during the day period while the remainder (60%), was at night.

The effect of thermal storage has been discussed in this study. Earlier, studies conducted on test units and pilot plants by Muller-Holst et al.⁵⁸, ⁵⁴, ⁴⁴ have been mentioned. Geroifi et al.⁶², in their study on solar desalination units in Australia mentioned the effect of thermal storage. Solar assisted systems without thermal storage and operating 24-hours per day will typically have a solar contribution of only 20% - 30%. In their study, a 100 m³.d⁻¹ solar-assisted plant powered by a flat plate collector with 1600 m² area without thermal storage, gave a 22% solar contribution. With the aid of TRNSYS, a “utility factor” versus storage volume chart was plotted which showed that 150 m³ of storage raised the solar contribution to 37%.
1.7.6. Recent Research on Solar Collectors

The solar collector is the main component of the solar desalination unit and any improvement in its efficiency will have a direct bearing on the water production rate and the product cost. In a relatively new study by Abu-Qudais et al.\textsuperscript{63}, a convex type solar heater was designed and its performance studied. The main purpose of a convex solar collector was to accomplish static tracking for the sun’s radiation without additional cost. Two adjacent types of collector: the flat plate and convex type with the same projected area were constructed and tested under the same conditions. The efficiency and useful energy for the convex type collector was found to be considerably greater than the flat plate collector. An additional 45% energy gain was achieved from the convex type compared to the flat plate collector.

1.8. Conclusion

A comprehensive review of the solar humidification-dehumidification technique, and the various desalination units based on this principle have been presented in this study. Solar desalination plants based on this technique have yet to be commercially implemented and a detailed understanding and compilation of all relevant information of the process is another step in that direction. A detailed review of the other desalination processes based on the humidification-dehumidification principle would also help in improving the design of current solar-based HD units. System improvement studies along with state-of-the-art advances in evaporator, condenser and collector design have been reported in this study, which would help in the implementation of productivity enhancement techniques for current and future installations. Prior to implementing techniques in improving unit performance, it is necessary to optimise the MEH unit by optimising component size and to understand the effect of feed water flow rate. Along with the thrust towards unit optimisation, the economic viability of these units is also important and to understand the status of solar-based HD units in the current desalination market scenario. Part 2.0 and Part 3.0 of this report focuses on these issues and presents a simulation of system performance and a comprehensive cost review and comparison respectively.

1.9. Outlook

Although solar desalination units based on the humidification-dehumidification principle have been in existence for a while, more work needs to be performed to increase the productivity and decrease the cost of product water of these units. Work has been initiated to add thermal storage modules to the MEH units. Alternative sources of energy could be utilised to heat water stored for nocturnal use in a 24-hour operation. Waste heat from industry and waste heat from fuel cells are some of the sources of “free energy”, which could help increasing productivity of the MEH units at no additional expense except for the thermal storage units. Further studies and simulation and process improvement design of MEH units with a 24-hour operation and thermal storage modules, should be considered in the future as a critical step for the commercialisation of these units.
2. MATHEMATICAL MODELLING OF SOLAR DESALINATION UNIT WITH HD CYCLE

2.1. Background

Extensive work has been done to simulate the single-basin still at the Indian Institute of Technology, Delhi, and other places. Malik et al.\textsuperscript{1}, and recent publications, describe the basic models used for single-basin stills. However, only Farid and co-workers reported a rigorous detailed model that can be used to describe the full performance of the desalination units working on the humidification-dehumidification principle. Most of the models available in the literature constitute simple material and energy balances with heat and mass transfer calculations to predict the evaporation rate. Beckman\textsuperscript{45}, has developed a simplified theoretical model for an open-air/closed-water system. The assumption of the liquid film temperature being equal to the average air temperature, made in the model limited its application.

Muller-Holst et al.\textsuperscript{54}, have used a simulation program (TRNSY) to study the effect of varying the operating conditions of the unit constructed in the Canary Islands. The simulation was based on an open-water/closed-air system operating in 24-hour mode using storage tanks.

The only two parameters, which have been investigated, are the evaporator inlet temperature and load. By changing these parameters, it was possible to influence and optimise the thermal efficiency of the system for a determined solar power input. The production was improved by operating the unit at higher evaporator temperatures due to the more efficient evaporation and condensation at such conditions. However, the drop in the efficiency of the collector at higher temperatures would limit such improvement. The collectors were replaced by a waste heat source at night. The operating temperature in the unit was not allowed to increase beyond 85°C because the plate–type condenser was made of Polypropylene. The simulation model for the distillation unit with a collector field and without a thermal storage tank was validated by comparing the measured data of the pilot plant at Fuerteventura with simulation results. The parameters of the collector field were fitted by the best agreement of the solar yield method. Results of the simulation and the measured data for the daily distillate production were found to be in good agreement, however, the simulation was limited to the existing design of the unit with the objective to maximise its production or minimise production cost. There was no attempt to study the effect of varying the size of the components of the unit i.e. the surface area of the collector, condenser and humidifier.

2.2. Mathematical Modelling

As mentioned earlier, the only rigorous model available in the literature was that developed by Farid and co-workers. They developed a computer simulation program, which can be used to predict the performance of the humidification-dehumidification units operating on natural or forced draft air circulation.
Figure 2.1, is a simplified sketch of a typical multi effect humidification (MEH) unit.

Figure 2.1. Sketch of a natural draft air circulation MEH desalination unit (Nawyseh et al.\textsuperscript{37})

2.3. Energy and Mass Balances

Sea water at a temperature $T_1$ is first passed through the condenser to condense the water vapour and become preheated to temperature $T_2$ by the latent heat of condensation. It is further heated in a flat plate solar collector to the humidifier inlet temperature $T_3$. The saline water leaves the bottom of the humidifier at a temperature of $T_4$. The saturated air at the bottom of the unit is heated and further humidified as it passes through the humidifier, showing an increase in temperature from $T_5$ to $T_6$. The performances of the humidifier, condenser and solar collector were described by the following five equations (Nawyseh\textsuperscript{36}):

\begin{align*}
LCp_w(T_2 - T_1) + 0.5U_{\text{loss, unit}}\frac{T_5 + T_6}{2} - T_{\text{amb}} &= G(H_6 - H_5) \\
LCp_w(T_2 - T_1) &= U_{\text{cond, A}} \frac{(T_6 - T_5) - (T_5 - T_1)}{\ln \frac{T_6 - T_2}{T_5 - T_1}} \\
LCp_w(T_3 - T_4) - 0.5U_{\text{loss, unit}}\frac{T_5 + T_6}{2} - T_{\text{amb}} &= G(H_6 - H_5) \\
G(H_6 - H_5) &= KaV \left[ \frac{(H_3 - H_6) - (H_4 - H_5)}{\ln \frac{H_3 - H_6}{H_4 - H_5}} \right]
\end{align*}
\[ Q_{\text{heater}} = LCP_w(T_3 - T_2) \]  \hfill (2.5)

In the unsteady state simulation, the heat balance on the heater, defined by equation (2.5), is replaced by the heat balance on the solar collector.

\[ I_T A_{\text{col}} \eta_{\text{col}} = \frac{M_{\text{in}}(C_{\text{unit}})}{\Delta T} \left[ \frac{T_i + T_f}{2} \right]_{\Delta t} - \left[ \frac{T_i + T_f}{2} \right]_{\Delta t-1} + U_{\text{loss}} A_{\text{unit}} \left[ \frac{T_i + T_f}{2} - T_{\text{amb}} \right] + LCP_w (T_i - T_f) \]  \hfill (2.6)

The first term in equation (2.6), represents the useful energy obtained from the collector. The second term represents the energy stored (or lost) in the wall of the unit during a specified time interval; this includes the heat stored in the condenser plate and the tubes filled with the water. The third term, describes the total heat losses from the unit wall to the ambient, which is twice the heat loss term shown in Equations (2.1) or (2.3). In writing the above equations the following assumptions were made:

- The air enters the humidifier saturated and leaves saturated. This has been verified from the experimental measurements.
- The log-mean temperature difference of the temperature and enthalpy may be used to simplify the computation. This will avoid the requirement of point-to-point calculations in both the condenser and the humidifier, for each time increment, during the whole simulation time.
- In the calculations of the heat loss from the units, the inside wall was assumed to be at the same temperature of the humid air flowing along it, which is a reasonable assumption due to the heavy insulation used. For this purpose it was also assumed that the air temperature varies linearly from the bottom to the top of the unit. The same assumption was used in the calculations of the heat stored in the wall.
- In the natural air circulation mode, the air velocity was too small to be measured by any available method. In the simulation, an average value for the air velocity during the whole day was used. Further work would be required to develop an independent equation for predicting the air velocity in term of the buoyancy and the frictional losses through the humidifier and the condenser. However, a small error in the measured air velocity would have only a limited effect on the results of the simulation.

The solution of the above equations requires a prior knowledge of the heat and mass transfer coefficients. The empirical correlation used to predict the heat and mass transfer coefficients are discussed in the next paragraph. The heat loss coefficient \( U_{\text{loss}} \) was measured and calculated theoretically for the different units as described by Al-Hallaj\(^3\) and Nawayseh\(^6\). The measured value (1.0 W.m\(^{-2}\).K\(^{-1}\)), which corresponds to an insulation of 50 mm thick glass wool, was used in all the simulations. Table 2.1, shows the values of all the fixed parameters used in the original simulation program. These values will be changed in the simulation presented in this report and their effects will be analysed. The Enthalpy and the humidity of the saturated air were calculated using the following empirical correlation obtained from data reported by Stoecker and Jones\(^6\).

\[ H = 0.00585T^3 - 0.497T^2 + 19.87T - 207.61 \]  \hfill (2.7)
Equations (2.1) to (2.6) were solved using Newton’s method of solution for non-linear equations. The collector efficiency in equation (2.9) was introduced in the simulation program as a function of the ambient temperature, water inlet temperature and solar radiation, as reported by Al-Hallaj\textsuperscript{33} and Nawayseh\textsuperscript{36}:

$$\eta_{\text{col}} = 0.60 - 0.0078 \left( \frac{T_2 - T_{\text{amb}}}{I_T} \right)$$

(2.9)

The above efficiency includes the heat losses through the pipes connecting the desalination unit with the solar collector. The flow mal-distribution of water in the collector, due to the low water flow rate used, might have been another reason for the low efficiency. The flat plate solar collector efficiencies, as reported by Hsieh\textsuperscript{66}, are higher than those given by the above equation. However, the reported efficiencies were for the collector without its associated piping. The above equation is used in the simulation conducted in this report in order to provide conservative values of production.

2.4. Desalination Units Constructed in Iraq, Jordan and Malaysia

In this section, the desalination units constructed in Iraq, Jordan and Malaysia, are described based on the published work of Farid and Al-Hajaj\textsuperscript{32}, Farid et al.\textsuperscript{53}, Nawayseh et al.\textsuperscript{35}, Al-Hallaj et al.\textsuperscript{33}, Nawayseh et al.\textsuperscript{36}, Nawayseh et al.\textsuperscript{37} and Farid\textsuperscript{27}.

Special emphasis will be given to the simulation of the units and the heat and mass transfer correlation used to describe water evaporation and condensation in these units. The objective is to use this information to conduct new simulations that cover a wide range of operating condition, and will be illustrated in this report.

The first unit constructed by Farid and co-workers for water desalination, based on the humidification-dehumidification process was in 1990 at the city of Basrah, south of Iraq (Farid and Al-Hajaj\textsuperscript{32}). The unit was built, based on a closed-air, open-water cycle, with a forced air circulation using a blower. The humidifier and the condenser were made from 100 mm OD hard PVC pipes, which were connected to form a closed loop with the blower fixed at the bottom. The condenser, which was fixed in the PVC pipe, was made of a 27 m long copper pipe, mechanically bent to form a 4 m long helical coil. The preheated water in the condenser was further heated in 1.9 m x 1.0 m flat plate solar collector. The hot water leaving the collector was sprayed over the wooden shaving packing in the 2 m long humidifier. Figure 2.1, shows a schematic diagram of the desalination unit. During the first two hours of the day, water was circulated between the collector and the preheating tank to provide a small quantity of hot water for starting the operation.

The unit produced 12 L.m\textsuperscript{-2}.day\textsuperscript{-1}, which was about three times the production of the single-basin still under similar solar radiation conditions (Farid and Hamad\textsuperscript{5}). In order to define the desalination efficiency, a performance factor was defined as the ratio of heat utilised for evaporating the water to the incident solar energy. This performance
factor includes the efficiency of the collector, as well as that of the condenser and humidifier. Results showed that the values for the performance factor greater than one, at all operating condition, compared to less than 0.5, usually found for the basin still. A maximum daily performance factor of 1.6 was found to occur at a certain water flow rate, which was attributed to the strong effect of water flow rate on the efficiency of the collector, humidifier, and condenser. However, the pressure drop in the condenser and the humidifier was too high, increasing the electrical power consumption by the blower to a level that makes such a process, uneconomical.

Two units of different sizes were constructed and operated in Jordan, described by Al-Hallaj et al. and Farid et al. The pilot unit was 2 m high and was constructed from galvanised steel plates, while the bench scale unit was only 1 m high, and constructed from Plexiglas. Both of the units consisted of two vertical rectangular ducts connected at their upper and lower ends to form a closed loop for air circulation. In one of the ducts, a large surface condenser was fixed, while the other duct contained a large number of wooden slats fixed in a wooden frame to form a typical humidifying tower. A third unit was built in Malaysia, based on the same principle but with a different construction. The unit was 3 m high and was constructed from 316 mm outside diameter PVC pipes. Details of the unit are given by Nawayseh et al. and Termizi.

As illustrated in Figure 2.1, the hot water leaving the flat plate solar collector was sprayed over the packing using a simple distributor. The concentrated brine was rejected from the bottom of the humidifier section, while the desalinated water was withdrawn from the bottom of the condenser. The air was circulated in the unit either by natural draft or forced draft, using an electrical fan fixed at the upper section of the units. The unit constructed in Malaysia was operated with natural draft only. The three units were operated in a steady state mode using an electrical heater, and in an unsteady state mode, used heat obtained from a flat plate solar collector.

### 2.4.1. Humidifier Design

The humidifier was a typical cooling tower with wooden slat packing, as shown in Figure 2.2. In the two units constructed in Jordan, the wooden slats were fixed in a wooden rectangular frame with an inclination of 45° in multiple rows along the whole height of the frame. The dimensions of the frame in the pilot and bench units were 1.0 m x 0.18 m x 2.0 m and 0.3 m x 0.05 m x 1.0 m, respectively. These dimensions provided packing surface area of 14 m².m⁻³, for the pilot unit, and 87 m².m⁻³, for the bench unit, with minimum air pressure drop. The true surface areas of the packing were 5.6 m², and 0.96 m², respectively. In the 3 m high unit, constructed in Malaysia, the packing was made of thin wooden sheets combined in a cross arrangement to form six modules of 0.5 m height, shown in Figure 2.2. The modules were dropped from the top of the unit to form a 3 m high humidifier. This arrangement gave a packing surface area of 11.9 m² (58 m².m⁻³).

### 2.4.2. Condenser Design

In the desalination units constructed in Jordan, the condensers were constructed from 1.0 m x 0.3 m and 2.0 m x 1.0 m galvanised steel plates for the bench and pilot units, respectively. In the pilot unit, a copper tube of 10 mm OD and 18 m long was welded
to the galvanised plate in a helical shape. The tube’s outside diameter and length in the bench unit were 8 mm and 3 m, respectively. Either one or two condensers, connected in series, were fixed vertically in one of the ducts in both units. Figure 2.3, shows sketches of the condensers used.

In the pilot unit constructed in Malaysia, the condenser was simply a 3 m long cylinder having a diameter of 170 mm, made of galvanised steel plates. Ten longitudinal fins, of 50 mm height, were soldered to the outer surface of the cylinder and nine similar fins were soldered to the inner surface. Aluminium had not been used in the construction of the condenser due to some practical difficulties. The thickness of the plate used to make the cylinder and the fins was 1.0 mm. A 19.2 m long copper tube having a 10 mm outside diameter was soldered to the surface of the cylinder. The condenser was fixed vertically through the 316 mm diameter PVC pipe, which was connected to the humidifier section from the top and bottom by two short horizontal PVC pipes. The specifications of the components of the desalination units are given in Table 2.1.

2.5. Mass Transfer Coefficient in the humidifier

In this section, the method used in the determination of the mass transfer coefficient in the humidifier will be described as reported by Nawayseh et al. 36. The same correlation will be used for the new simulations presented in this report. It is common practice, in the design of cooling towers, to use the performance characteristic, $K a V / L$, defined as follows:

$$K a V / L = \int_{t_4}^{t_3} \frac{C p_w \ dT}{H_f - H_a}$$

(2.10)

where $K$ is a mass transfer coefficient (kg water evaporated.m$^{-2}$) and $a$ is the surface area of packing per unit volume. The integral may be evaluated from the measured temperature and the enthalpy of the water and air entering and leaving the humidifier. If the operating range of the temperature is small enough to allow the assumption of linear enthalpy-temperature saturation line, then the log mean driving force $(H_f - H_a)_{lm}$ may be used for the evaluation of the integral, otherwise numerical integration will be necessary.

The desalination units were run under steady state, by replacing the solar collector with an electrical heater of known power. In the forced air circulation mode, different air velocities were obtained by applying a variable AC power supply to the fan inside the unit, then, the desalination unit was run for few hours to reach steady state. Humidity of the air at the top and bottom of the unit were measured and found to be saturated. Water and air temperatures were measured at different locations using thermocouples and a multi-channel data acquisition unit. Water inlet flow rate was measured using a rotameter; while brine and desalinated water flow rates were measured using a graduated cylinder.
(a) The construction of the packing used in Malaysia

(b) The construction of the packing used in Jordan

Figure 2.2. Details of the humidifier’s packing used in the desalination units (Nawayseh)
Figure 2.3. Details of the condenser used in the desalination units
Table 2.1. Specifications of the condenser, humidifier, and solar collector used in the three desalination units

<table>
<thead>
<tr>
<th>Reference</th>
<th>Condenser surface area m²</th>
<th>Humidifier surface area m²</th>
<th>Humidifier surface area per unit volume m².m⁻³</th>
<th>Height of the unit m</th>
<th>Solar collector area m²</th>
<th>Uloss W.m⁻² K⁻¹</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nawayseh et al., ³⁶ (Pilot unit)</td>
<td>8.9</td>
<td>11.9</td>
<td>58</td>
<td>3</td>
<td>2.77</td>
<td>1.0</td>
</tr>
<tr>
<td>Al-Hallaj et al., ³⁴</td>
<td>4 (single) 8 (double)</td>
<td>5.6</td>
<td>14</td>
<td>2</td>
<td>2.0</td>
<td>1.0</td>
</tr>
<tr>
<td>Al-Hallaj et al., ³⁴</td>
<td>0.6 (single) 1.2 (double)</td>
<td>1.3</td>
<td>87</td>
<td>1</td>
<td>_</td>
<td>1.2</td>
</tr>
</tbody>
</table>

It was not possible to measure the velocity of the air using a hot wire anemometer because of the strong effect of the humidity on the measurements. In the natural draft air circulation, air velocity was too small to be measured accurately by any available method. It was decided to calculate the air velocity from the energy balance over the condenser or the humidifier, as given by equation (2.1) and (2.3).

Based on the steady state analysis, Nawayseh et al. ³⁶, had developed the following correlation for the mass transfer coefficient in the humidifier. These correlations were found to describe the mass transfer favourably in all the humidifiers constructed in Jordan and Malaysia:

\[ KaV/L = 1.19 \left( \frac{L}{G} \right)^{-0.66} \quad \text{for natural draft} \quad 1 < \frac{L}{G} < 8 \]  \hspace{1cm} (2.11)

\[ KaV/L = 0.52 \left( \frac{L}{G} \right)^{-0.16} \quad \text{for forced draft} \quad 0.1 < \frac{L}{G} < 2 \]  \hspace{1cm} (2.12)

### 2.6. Heat Transfer in the condenser

In order to utilise the latent heat of condensation of the water efficiently, the condenser area must be made large. The overall heat transfer coefficient of the condenser is expected to be small due to:

- The low velocity of the air circulated in the unit, even in the forced circulation, which was necessary to avoid an excessive pressure drop
- The large reduction in the condensation heat transfer coefficient due to the mass transfer resistance occurring in the process of condensation of water vapour with non-condensable air
- The low waterside heat transfer coefficient, due to the low water flow-rate per unit condenser area. However, a small diameter tube can be used to overcome this problem

The finned-type condenser, designed for the unit constructed in Malaysia has a large surface area, but is difficult to construct. In the simulation done in this report, the
design of the condenser used with the unit constructed in Jordan is adopted due to its simple design. It is assumed that a stack of such flat condensers may provide a very large surface area. The following expression was used to define the overall heat transfer coefficient. The copper tube resistance was assumed negligible.

\[
\frac{1}{UA_c} = \frac{1}{h_w A_r} + \frac{1}{\eta_f A_r h_c} \tag{2.13}
\]

The fin efficiency, \(\eta_f\), is the usual fin efficiency defined in heat transfer textbooks

\[
\eta_f = \frac{\tanh \sqrt{\frac{2h_c}{h_t k L}}} \sqrt{\frac{2h_c}{h_t k L}} \tag{2.14}
\]

The characteristic fin dimension, \(L\), was taken equal to half the tubes spacing. The air-side heat transfer coefficient, \(h_c\) is a condensation coefficient that includes a significant mass transfer resistance due to the presence of the non-condensable air. The values of \(h_c\) vary along the condenser height and hence a point-to-point calculation is required as discussed by Tanner et al.\textsuperscript{68}. However, in order to obtain an average value to be used in the simulation, the approximate analysis of Bell and Ghaly\textsuperscript{69}, was followed. In this method, the coefficient \(h_c\) is defined as follows:

\[
h_c = \frac{h_a}{Z} \tag{2.15}
\]

where

\(h_a\) is the convection heat transfer coefficient of the humid air flowing along the condenser surface by forced or free convection.

The condensation factor \(Z\), represents the ratio of the sensible to the total heat load in the condenser:

\[
Z = \frac{C_p a dT}{dH} \tag{2.16}
\]

Values of \(Z\) were obtained from the derivative of the enthalpy expression defined by equation (2.6), calculated at the average operating temperature. The above equations show that the condensation coefficient may vary from two to eight times that of the convection coefficient, depending on the operating temperature.

The waterside heat transfer coefficient was calculated from the known empirical correlation of flow in the pipe presented by Holman\textsuperscript{70};

\[
\text{Nu}_w = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} \tag{2.17}
\]

The water desalination rate was calculated from the difference between the estimated humidity of the air at the top and bottom of the unit.
2.7. Simulation Results for the Variations in Collector, Condenser, and Humidifier Areas on the Productivity of an MEH Unit

Simulation studies are carried out for the unit constructed in Malaysia with the same overall dimensions as described, but with varying sizes of collector, condenser and humidifier. The structured packing used in the humidifier is similar to that shown in Figure 2.2, while the condenser is of the plate-type used in the unit built in Jordan, as shown in Figure 2.3. It is assumed that a stack of these plate condensers are placed in the unit to provide the required large surface area without causing more restriction to the air flowing by natural draft circulation.

The solar radiation intensity used in the simulation (see Figure 2.8), based on that measured on a typical summer day in the Gulf (Farid and Hamad⁵). The feed water temperature is fixed at 25°C, while the ambient temperature is assumed to vary during the day reaching a maximum at noon. In all the simulations, the feed flow rate is changed whenever the collector area is increased or decreased so that the evaporator operates below 90°C as recommended.

Figure 2.4, and Table 2.2, shows the effect of increasing the collector and condenser areas on the desalinated water production rate for a humidifier area of 11.9 m². For a small collector area, the production increases slightly with the increase of the condenser area, while for a large collector area, the effect of condenser area is very significant. The simulation results show the importance of choosing the right sizes of condenser corresponding to a selected collector area.

Figure 2.4a. Effects of the condenser and collector surface areas on the daily production of the desalination unit, humidifier area = 11.9 m²
Figure 2.4b. Effects of the condenser and collector surface areas on the daily production of the desalination unit, humidifier area = 11.9 m\(^2\)

Table 2.2. The daily production of the desalination unit with various condenser and collector surface areas, humidifier area = 11.9 m\(^2\)

<table>
<thead>
<tr>
<th>Condenser Area (m(^2))</th>
<th>Collector Area (m(^2))</th>
<th>Production (kg.day(^{-1}))</th>
<th>Production (kg.day(^{-1}).m(^{-2}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>2</td>
<td>8.78</td>
<td>4.39</td>
</tr>
<tr>
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<td>4.64</td>
</tr>
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<td>6</td>
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<td>9.63</td>
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</tr>
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<td>10.37</td>
<td>5.18</td>
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<td>10.70</td>
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</tr>
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<td>27.75</td>
<td>4.63</td>
</tr>
</tbody>
</table>
continuation of Table 2.2.

<table>
<thead>
<tr>
<th>Collector Area</th>
<th>Humidifier Area</th>
<th>Production</th>
<th>Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>6</td>
<td>28.82</td>
<td>4.80</td>
</tr>
<tr>
<td>25</td>
<td>6</td>
<td>29.44</td>
<td>4.91</td>
</tr>
<tr>
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<td>6</td>
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<tr>
<td>4</td>
<td>12</td>
<td>30.88</td>
<td>2.57</td>
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<td>12</td>
<td>34.30</td>
<td>2.86</td>
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<td>36.71</td>
<td>3.06</td>
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<td>43.68</td>
<td>3.64</td>
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<td>4.04</td>
</tr>
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<td>12</td>
<td>51.36</td>
<td>4.28</td>
</tr>
<tr>
<td>25</td>
<td>12</td>
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</tr>
<tr>
<td>40</td>
<td>12</td>
<td>58.03</td>
<td>4.84</td>
</tr>
<tr>
<td>4</td>
<td>20</td>
<td>38.00</td>
<td>1.90</td>
</tr>
<tr>
<td>5</td>
<td>20</td>
<td>43.85</td>
<td>2.19</td>
</tr>
<tr>
<td>6</td>
<td>20</td>
<td>47.95</td>
<td>2.40</td>
</tr>
<tr>
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<td>20</td>
<td>60.40</td>
<td>3.02</td>
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<td>20</td>
<td>70.11</td>
<td>3.51</td>
</tr>
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<td>20</td>
<td>76.56</td>
<td>3.83</td>
</tr>
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<td>20</td>
<td>81.13</td>
<td>4.06</td>
</tr>
<tr>
<td>40</td>
<td>20</td>
<td>89.61</td>
<td>4.48</td>
</tr>
<tr>
<td>4</td>
<td>40</td>
<td>57.50</td>
<td>1.44</td>
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<td>40</td>
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<td>1.64</td>
</tr>
<tr>
<td>6</td>
<td>40</td>
<td>66.05</td>
<td>1.65</td>
</tr>
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<td>40</td>
<td>88.30</td>
<td>2.21</td>
</tr>
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<td>40</td>
<td>107.63</td>
<td>2.69</td>
</tr>
<tr>
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<td>40</td>
<td>121.60</td>
<td>3.04</td>
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<td>25</td>
<td>40</td>
<td>132.72</td>
<td>3.32</td>
</tr>
<tr>
<td>40</td>
<td>40</td>
<td>153.60</td>
<td>3.84</td>
</tr>
</tbody>
</table>

Figure 2.5 and Table 2.3, shows the combined effect of the condenser and humidifier areas on the production of the MEH unit employing 20 m$^2$ collector area.

Figure 2.5a: Effects of the condenser and humidifier surface areas on the daily production of the desalination unit, collector area = 20 m$^2$
Table 2.3. The daily production of the desalination unit with various condenser and humidifier surface areas, collector area = 20 m²

<table>
<thead>
<tr>
<th>Humidifier Area (m²)</th>
<th>Condenser Area (m²)</th>
<th>Production (kg.day⁻¹)</th>
<th>Production (kg.day⁻¹.m⁻²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.19</td>
<td>4</td>
<td>21.46</td>
<td>1.07</td>
</tr>
<tr>
<td>5.95</td>
<td>4</td>
<td>37.70</td>
<td>1.89</td>
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<tr>
<td>11.90</td>
<td>4</td>
<td>41.92</td>
<td>2.10</td>
</tr>
<tr>
<td>17.85</td>
<td>4</td>
<td>42.30</td>
<td>2.12</td>
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<tr>
<td>23.80</td>
<td>4</td>
<td>43.50</td>
<td>2.18</td>
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<tr>
<td>1.19</td>
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<td>25.35</td>
<td>1.27</td>
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<td>5.95</td>
<td>10</td>
<td>54.12</td>
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<tr>
<td>11.90</td>
<td>10</td>
<td>57.47</td>
<td>2.87</td>
</tr>
<tr>
<td>17.85</td>
<td>10</td>
<td>58.70</td>
<td>2.94</td>
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<tr>
<td>23.80</td>
<td>10</td>
<td>58.95</td>
<td>2.95</td>
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<td>20</td>
<td>27.22</td>
<td>1.36</td>
</tr>
<tr>
<td>5.95</td>
<td>20</td>
<td>63.67</td>
<td>3.18</td>
</tr>
<tr>
<td>11.90</td>
<td>20</td>
<td>73.87</td>
<td>3.69</td>
</tr>
<tr>
<td>17.85</td>
<td>20</td>
<td>76.30</td>
<td>3.82</td>
</tr>
<tr>
<td>23.80</td>
<td>20</td>
<td>76.90</td>
<td>3.85</td>
</tr>
<tr>
<td>1.19</td>
<td>40</td>
<td>28.10</td>
<td>1.41</td>
</tr>
</tbody>
</table>

Figure 2.5b: Effects of the condenser and humidifier surface areas on the daily production of the desalination unit, collector area = 20 m²

40
Both have a significant effect and it diminishes when the areas become large. The results show that the productivity of the unit will not improve significantly if the humidifier or the condenser is kept small. With a small condenser or humidifier surface area, the effect of increase in any of them on the productivity is very significant.

The combined effect of collector and humidifier surface areas is shown in Figure 2.6 and Table 2.4. The effect of collector is evident at any humidifier area, while the effect of the humidifier is significant only with a large collector area. It is interesting to note, that at a large collector area of 40 m$^2$, there is no need to increase the humidifier area beyond that of 12 m$^2$ if the condenser area is kept at 9 m$^2$. In order to increase productivity, the areas of both condenser and humidifier must be increased.

![Figure 2.6a: Effects of the collector and humidifier surface areas on the daily production of the desalination unit, condenser area = 9 m$^2$](image)

<table>
<thead>
<tr>
<th>Collector Area (m$^2$)</th>
<th>Humidifier Area (m$^2$)</th>
<th>Production (kg/day)</th>
<th>Eff. (m$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.95</td>
<td>12</td>
<td>77.36</td>
<td>3.87</td>
</tr>
<tr>
<td>11.90</td>
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<td>23.80</td>
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<td>60</td>
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<td>17.85</td>
<td>60</td>
<td>99.10</td>
<td>4.96</td>
</tr>
</tbody>
</table>

continuation of Table 2.3.
Table 2.4. The daily production of the desalination unit with various collector and humidifier surface areas, condenser area = 9 m$^2$

<table>
<thead>
<tr>
<th>Humidifier Area (m$^2$)</th>
<th>Collector Area (m$^2$)</th>
<th>Production (kg.day$^{-1}$)</th>
<th>Production (kg.day$^{-1}$.m$^{-2}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.19</td>
<td>2</td>
<td>2.72</td>
<td>1.36</td>
</tr>
<tr>
<td>2.38</td>
<td>2</td>
<td>5.07</td>
<td>2.53</td>
</tr>
<tr>
<td>4.76</td>
<td>2</td>
<td>8.04</td>
<td>4.02</td>
</tr>
<tr>
<td>7.14</td>
<td>2</td>
<td>9.33</td>
<td>4.67</td>
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<td>9.52</td>
<td>2</td>
<td>9.92</td>
<td>4.96</td>
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<tr>
<td>11.90</td>
<td>2</td>
<td>10.23</td>
<td>5.12</td>
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<tr>
<td>17.85</td>
<td>2</td>
<td>10.54</td>
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<td>20.23</td>
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<td>10.60</td>
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<td>1.19</td>
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<td>4.83</td>
<td>1.21</td>
</tr>
<tr>
<td>2.38</td>
<td>4</td>
<td>8.92</td>
<td>2.23</td>
</tr>
<tr>
<td>4.76</td>
<td>4</td>
<td>14.15</td>
<td>3.54</td>
</tr>
<tr>
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<tr>
<td>11.90</td>
<td>6</td>
<td>24.78</td>
<td>4.13</td>
</tr>
</tbody>
</table>
The effect of feed water flow rate on the production of the MEH unit is complex due to its combined effect on the operating temperature and efficiency of the collector, humidifier and condenser. Increasing the feed flow rate reduces the operating temperature, which will reduce the production of the unit. The heat and mass transfer coefficients in the humidifier and condenser decrease at low temperatures. However, the efficiency of the collector, humidifier and condenser increases at higher feed flow rates. In the humidifier, higher water flow rate will improve both mass transfer rate and the degree of wetting of packing (Nawayseh et al.'). The combined effect of the feed flow rate is probably the reason for the maximum production of 0.013 kg.s$^{-1}$ water flow rate, as shown in Figure 2.7. Figure 2.8, shows the accumulative production of the unit during the day at a specified operating condition, together with the solar radiation used in the simulation.
Figure 2.7. Effect of the feed water flow-rate on the daily production of the desalination unit

Figure 2.8. Solar intensity measurement at Arab Gulf countries, and the daily production of the desalination unit

Figure 2.9, shows the predicted temperature at the different locations in the unit as defined in Figure 2.1. The rise in water temperature through the condenser is higher than the corresponding increase in the solar collector, which is evident of the good utilisation of the latent heat of water condensation in a unit with only one effect. Figure 2.9, shows that the concentrated saline water rejected from the humidifier, leaves at a moderate temperature. The unit productivity may be increased by the re-use of this source of energy by operating the unit at night, as suggested by Farid and
Al-Hajaj\textsuperscript{32}, and Al-Hallaj et al.\textsuperscript{34}, and which has been proved in recent studies by Muller-Holst et al\textsuperscript{38,54}.

![Temperature distribution of the desalination unit](image)

**Figure 2.9. Temperature distribution of the desalination unit**

**Table 2.5. The daily production of the desalination unit with water flow-rate**

<table>
<thead>
<tr>
<th>Water flow-rate (kg.s(^{-1}))</th>
<th>Production (kg.day(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.009</td>
<td>13.71</td>
</tr>
<tr>
<td>0.009</td>
<td>14.05</td>
</tr>
<tr>
<td>0.010</td>
<td>14.3</td>
</tr>
<tr>
<td>0.011</td>
<td>14.47</td>
</tr>
<tr>
<td>0.012</td>
<td>14.56</td>
</tr>
<tr>
<td>0.012</td>
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<td>14.55</td>
</tr>
<tr>
<td>0.014</td>
<td>14.44</td>
</tr>
<tr>
<td>0.015</td>
<td>14.27</td>
</tr>
<tr>
<td>0.016</td>
<td>13.71</td>
</tr>
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<td>0.018</td>
<td>12.94</td>
</tr>
<tr>
<td>0.020</td>
<td>12.07</td>
</tr>
<tr>
<td>0.022</td>
<td>11.23</td>
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<tr>
<td>0.029</td>
<td>8.42</td>
</tr>
<tr>
<td>0.037</td>
<td>6.65</td>
</tr>
<tr>
<td>0.044</td>
<td>5.47</td>
</tr>
<tr>
<td>0.051</td>
<td>4.6</td>
</tr>
</tbody>
</table>
2.8. Conclusion

Three main parameters (collector area, humidifier surface area and condenser surface area) of the MEH Unit were varied and the effects studied. This is perhaps the first simulation program that shows the variation effects of the three parameters. Optimum collector, humidifier and condenser areas can be determined from this simulation, which could lead to a better design of MEH Units. The effect of the feed water flow rate on the productivity of the MEH Unit was also shown and which shows a maximum production capacity at the optimum water flow rate. The combined simulation results presented in this study would further assist in designing state-of-the-art MEH Units and thus provide an impetus to the commercialisation of such units.
3. ECONOMIC CASE FOR SOLAR-POWERED DESALINATION SYSTEMS

3.1. Introduction

Although desalted water may be produced using free energy such as solar or wind, the technology could still be expensive, depending on fixed charges and operating costs. The estimation or stipulation of an economic price is essential for poor small communities lacking fresh water, Belessiotis and Delyannis\(^{71}\). Although market prices for renewable energies are high, it could gradually become lower in the future and would be competitive with conventional energy sources. Costs for wind energy averages \(~5.0\ \text{¢.kWh}^{-1}\) for grid connected electricity, \(~7.0\ \text{¢.kWh}^{-1}\) for geothermal and \(~20-40\ \text{¢.kWh}^{-1}\) for solar power, Belessiotis and Delyannis\(^{71}\).

The advantage of using “free” energy is partly offset by increased amortisation costs; however, distillation with solar energy remains one of the most favourable processes for small capacity water desalting for remote regions where there is substantial solar radiation, lack of skilled personnel, erection and maintenance facilities. A comparison of the use of renewable energy for desalination compared with conventional systems (Reverse Osmosis and Multi Stage Flash), is presented (as a percentage of the total costs) by Heschl\(^{25}\), and is highlighted in Table 3.1.

<table>
<thead>
<tr>
<th>Type of Process</th>
<th>Investment Costs (%)</th>
<th>Operational Costs (%)</th>
<th>Energy Costs (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conventional (RO)</td>
<td>22-27</td>
<td>14-15</td>
<td>59-63</td>
</tr>
<tr>
<td>Conventional (MSF)</td>
<td>25-30</td>
<td>38-40</td>
<td>33-35</td>
</tr>
<tr>
<td>Renewable</td>
<td>30-90</td>
<td>10-30</td>
<td>0-10</td>
</tr>
</tbody>
</table>

The comparison shows that investment costs are the highest, but energy costs are the lowest in the case of renewable energy powered desalination systems.

A major hurdle in determining the cost of desalinated water from solar-powered systems is the limited data available for economic evaluation and cost estimation of these processes. The task of economic analysis of solar MEH units is not easy to accomplish, because cost information for these processes is even more limited. Solar energy based plants are capital-intensive and as prices for the construction materials are location specific, there is difference in cost of fresh water produced between regions. Some investigators have conducted cost estimation studies on solar powered desalination processes in general, which is presented in the following sections. A few researchers have also studied a combination of solar energy with conventional desalination techniques and the costs of these processes have also been presented in this study.
3.2. Solar Distillation

3.2.1. Solar Stills

In a study by Delyannis\textsuperscript{72}, it is reported that for smaller capacities of approximately less than 70 m\textsuperscript{3}.d\textsuperscript{-1}, the cost of water produced by conventional solar distillation is less than that of Vapour Compression (VC) and MSF plants. Because of the large area required for solar plants, the final water cost is sensitive to land values. When land cost is excluded, the cost of water for a simple still, with a production rate of 4 L.m\textsuperscript{2}.d\textsuperscript{-1} and a 20-year life period with an 8% interest rate, totals $2.88.m\textsuperscript{-3} (Madani\textsuperscript{73}).

In studies on greenhouse type solar distillation, Delyannis and Delyannis\textsuperscript{74} (52, 1985) have presented the price of water produced by this process. It is reported that the cost of water is affected negligibly by the size of the plant because the energy input system has to be increased proportionally to the desired output. It is observed that the cost of water increases linearly with increase in cost of the solar still. Economies of scale are not observed in this case, with cost remaining almost the same even for increasing plant capacity. A product water cost comparison (for conventional solar stills) for different collecting surface areas and varying plant capacity is presented in Table 3.2. The life of the stills was estimated to be 15 years, specific productivity 4 L.m\textsuperscript{-2}.d\textsuperscript{-1} and amortisation rate 10%.

<table>
<thead>
<tr>
<th>Collecting Surface Area (m\textsuperscript{2})</th>
<th>Plant Capacity (m\textsuperscript{3}.d\textsuperscript{-1})</th>
<th>Cost of Still ($m\textsuperscript{-2}$)</th>
<th>Investment ($)</th>
<th>Cost of Water ($m\textsuperscript{-3}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>250</td>
<td>1</td>
<td>28</td>
<td>6,838</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>56</td>
<td>13,676</td>
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</tr>
<tr>
<td></td>
<td></td>
<td>112</td>
<td>27,352</td>
<td>12</td>
</tr>
<tr>
<td>2500</td>
<td>10</td>
<td>28</td>
<td>68,380</td>
<td>3</td>
</tr>
<tr>
<td></td>
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<td>273,520</td>
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</tr>
<tr>
<td></td>
<td></td>
<td>56</td>
<td>683,800</td>
<td>6</td>
</tr>
<tr>
<td></td>
<td></td>
<td>112</td>
<td>1,367,600</td>
<td>12</td>
</tr>
</tbody>
</table>

Semiat\textsuperscript{75}, observed that one of the serious disadvantages of the solar still was that it requires approximately 250 m\textsuperscript{2} collecting surface area to produce 1 m\textsuperscript{3} of freshwater per day. This makes it inefficient even for the deserted arid zones, as this form of energy is available only one-third or one-quarter of the time. The alternate solar pond technique studied also suffers from technical drawbacks as well as low heat efficiency.

It is observed that solar stills do not exhibit economies of scale and that solar collector assisted solar desalination units present better economic viability compared to solar water heaters.
3.2.2. Cost Analysis Method for Solar Distillation Units

Delyannis and Delyannis\textsuperscript{74}, gives a cost analysis method for solar distillation units. Considering a mean lifetime of 20 years for the plant, the main components of the annual average cost of distilled water \(C (\text{\$m}^{-3})\), is given by the following equation:

\[
C = \frac{10I(IA + MR + TI) + 1000(Oc + Cs)}{A(Y_d + Y_c)} \quad (3.1)
\]

The average annual interest and amortisation rate, \(IA\), is a function of the annual interest rate (\%), \(r\), and is defined as:

\[
IA = r\left(1 + \frac{1}{1 + \frac{r}{100}}\right)^n - 1 \quad (3.2)
\]

Among the various types of solar stills in review, the multi-effect still presents the best productivity options. In a comparison study of all the solar still designs available Goosen et al.\textsuperscript{76}, concludes that the multi-wick double-effect solar still is the most economical design, irrespective of weather conditions, due to a high output for the least heat capacity of water in the wicks.

A collector assisted solar desalination system is presented as an investment alternative to solar water heater systems. A techno-economic analysis reported in the study by Fath\textsuperscript{9}, for both systems in the same economic environment and considering the same capacity, showed that the cost of energy from the distillation system is much less than the cost of energy obtained from the water heater, and that the annual operation cost of the solar water heater is much higher than that of the solar still because of the higher investment in the former. Costs of solar collector desalination units are presented in the following sections.

3.2.3. Solar-thermal Processes

Costs for solar-thermal desalination processes are presented by Teplitz-Semбитzky\textsuperscript{77}, for gate Information Services in their Technical Information W10e. Simple solar stills produce 2.5 L.m\(^{-2}\) - 4.5 L.m\(^{-2}\) of flat plate collector depending on the solar radiation and plant efficiency. The stills are available in modules for 5 L.d\(^{-1}\) - 8 L.d\(^{-1}\) and cost in the range of $490 - $690. Based on a collector cost of $315.m\(^{-2}\) and a capacity of 4 L.m\(^{-2}\).d, assuming a lifetime of 20 years for the plant with 5% interest rate, the specific distillation costs works out to $23.80.m\(^{-3}\).

In other studies on multi-effect stills, Rheinlander and Grater\textsuperscript{12}, present cost-data for hybrid operation with the aid of a ZSW compiled Excel workbook. The compilation is for typically 10 m\(^3\).d\(^{-1}\) output for 24-hour operation. The levelised water cost is calculated to be $28.80.m\(^{-3}\).
Multi-effect stills, consisting of collectors, storage module, and a desalination component have a higher GOR than simple stills, as mentioned previously. A unit tested by the Bavarian ZAE on the Canary Islands, on average, purified 12 L.m\(^{-2}\).d. It is estimated that mass production would reduce the costs of the desalination module. This would bring down the total distillation costs to $9.95.m\(^{-3}\) (over a 20 year lifetime with a 5% interest).

The German Fraunhofer Institute, expects the output of the still to increase under the conditions of using non-corroding polymer absorbers and the inclusion of thermal storage facilities for a 24-hour operation. The target is a daily production rate of 20 L.m\(^{-2}\) of collector area. This would bring down the total distillation costs below $9.m\(^{-3}\).

### Table 3.3. Summary of Water Cost for Simple and Multi-Effect Solar Stills

<table>
<thead>
<tr>
<th>Type</th>
<th>Capacity/Productivity</th>
<th>Cost of Water ($/m(^3))</th>
<th>Description</th>
<th>Ref:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar Stills</td>
<td>4 L.m(^{-2}).d</td>
<td>23.80</td>
<td>20 year lifetime, 5% interest rate</td>
<td>77.</td>
</tr>
<tr>
<td>Multi-Effect Still</td>
<td>10 m(^{3}).d(^{-1})</td>
<td>28.80</td>
<td>24-hour operation; hybrid</td>
<td>12.</td>
</tr>
<tr>
<td>Multi-Effect Stills</td>
<td>12 L.m(^{2}).d</td>
<td>9.95</td>
<td>storage module, 20 year lifetime, 5% interest rate</td>
<td>77.</td>
</tr>
<tr>
<td>Multi-Effect Stills *</td>
<td>20 L.m(^{2}).d *</td>
<td>&lt;9.0 *</td>
<td>non-corroding polymer absorbers, storage, 24-hour operation</td>
<td>77.</td>
</tr>
</tbody>
</table>

* predicted

It was observed that the use of thermal storage modules with a 24-hour operation reduced the cost of produced water substantially from between $20.m\(^{-3}\) - $30.m\(^{-3}\) to below $9.m\(^{-3}\). A summary of the cost of water for Solar-thermal processes mentioned above are tabulated in Table 3.3.

### 3.3. Combination of Different Solar Powered Systems

In an attempt to reduce water costs for desalination systems, various options were studied to combine solar energy collection systems with conventional desalination plants. Among several options to connect a seawater desalination system with a solar power plant, the combination of a thermal desalination system such as Multi Effect Distillation (MED) and a solar trough field as the heat source, was one of the most promising.

#### 3.3.1. Solar-MED Units

Glueckstern\(^{78}\), presented a detailed parametric study to compare the costs of partial or fully solar-powered desalting systems for large capacities of 20,000 m\(^{3}\).d\(^{-1}\) - 200,000 m\(^{3}\).d\(^{-1}\). He considered two solar technologies i.e. solar gradient pond and dual-purpose
solar electric power stations supplying low-pressure steam to thermal desalting systems. Two desalting techniques were considered: MED and hybrid MED/SWRO systems. The first used thermal process heat from the solar field and a relatively small supply of commercial electricity for pumping purposes. The second used solar energy only for the MED portion of the hybrid system while the SWRO portion used commercial electricity. The hybrid concept therefore represents a partial rather than a full solar system. On comparison, solar pond powered desalting systems were found to have a significant potential to be cost-effective if favourable site conditions exist. Only for low specific solar field cost and/or high commercial electricity prices would the full solar options be more cost competitive than the partial solar options. A comparison of the final product water costs from the different techniques is presented in Table 3.4.

Hoffman\textsuperscript{79}, has presented a comprehensive cost analysis based on operational data for solar-driven plants. Four potential combinations of seawater desalination and solar energy collection systems within 100,000 m\textsuperscript{3}.d\textsuperscript{-1} plant capacities were studied and costs compared. Costs presented in the study range from $0.67.m\textsuperscript{-3} to $1.44.m\textsuperscript{-3}. The cheapest option determined from his study was the solar pond Low Temperature MED (LT-MED), at a cost of $0.67.m\textsuperscript{-3}.

### Table 3.4. Cost comparison of solar pond powered desalination with conventional SWRO (Glueckstern\textsuperscript{78})

<table>
<thead>
<tr>
<th>System Type</th>
<th>SWRO</th>
<th>SP-MED</th>
<th>SP-HYB</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capacity, m\textsuperscript{3}.d\textsuperscript{-1}</td>
<td>20,000</td>
<td>200,000</td>
<td>20,000</td>
</tr>
<tr>
<td>Investment (mil. $)</td>
<td>20.0</td>
<td>160.0</td>
<td>48.0</td>
</tr>
<tr>
<td>Specific Investment ($\cdot.m\textsuperscript{3}.d$)</td>
<td>1000</td>
<td>800</td>
<td>2400</td>
</tr>
<tr>
<td>Unit Water Cost ($\cdot.m\textsuperscript{-3}$)</td>
<td>0.77</td>
<td>0.66</td>
<td>0.89</td>
</tr>
</tbody>
</table>

In a relatively early laboratory study of solar-powered multi-effect distillation systems, Tleimat\textsuperscript{80}, determined that a solar boiler producing saturated steam in combination with an evaporator and condenser resulted in minimum water costs ranging from $2.15.m\textsuperscript{-3} to $4.70.m\textsuperscript{-3}, for seawater feed. With an increase in steam temperature, the minimum water cost decreased from $3.60.m\textsuperscript{-3} to $2.05.m\textsuperscript{-3}, with a productivity of 151 L.m\textsuperscript{-2}.

A solar desalination system composed of a 14-effect MED plant was hooked up to a field of solar parabolic trough collectors during Phase I of the operation at the Plataforma Solar de Almeria from 1988 to 1994 (Milow and Zarza\textsuperscript{81}). The unit is still in operation and has a production capacity of 72 m\textsuperscript{3}.d\textsuperscript{-1}. Cost analysis, shows that implementation of an absorption heat pump lowers the cost considerably and a cost of $2.m\textsuperscript{-3} could be obtained for large plants.
In a more recent study of solar desalination, Sagie et al\textsuperscript{82}, presented cost analysis for plant size of three different capacities. They utilised a new collector technology developed by Solel Solar Systems that is coupled to an MED process, modified by IDE Technologies to match the solar steam characteristics. For the specific case analysed in detail, replacing the LT-MED with the new combination increases the Economic Ratio (ER) from 7 to 16, while the installation costs increase by only 60\%. The three plant sizes studied are 1000 m\textsuperscript{3}.d\textsuperscript{-1}, 10,000 m\textsuperscript{3}.d\textsuperscript{-1} and 100,000 m\textsuperscript{3}.d\textsuperscript{-1}. Applying the proposed model, distillate costs for a solar powered plant along the Red Sea coast for the small size plant is $1.2.m^{-3}$ for solar-only plant with a large capacity steam storage, and $1.1.m^{-3}$ for hybrid plant using $0.18.kg^{-1}$ diesel oil when solar-steam is not available. For the medium size plant, the Solar-MED plant would produce distillate at $0.92.m^{-3}$ and in the last case the distillate cost would be $0.69.m^{-3}$ (including land cost) for an available site close to the Israel’s Mediterranean shore. These costs are competitive with grid-powered RO, when electricity cost is about ₤6.5.kWh\textsuperscript{-1}.

For a solar-powered MED plant operated by solar collectors or a solar pond, and for feed water of salinity in the range of 5000 ppm - 35,000 ppm, Goosen et al\textsuperscript{76}, suggested an equation to estimate the cost of distilled water $C$ (in $\.m^{-3}$):

$$
C = \frac{N(IA + 1)}{868(1 + N)} \left[ \frac{1.246(2 + N)C_e}{h(\Delta T - \alpha N)} + 1 \right] + \frac{0.199(IA + 1)C_e}{I_s(1 + N)} + \frac{0.266(IA + 1)}{1 + N} + \frac{7.54}{1 + N}
$$

(3.3)

In other configurations, solar multiple-effect evaporator systems were studied, details of some of which are mentioned below.

In 1987, Gunzbourg and Fremont\textsuperscript{82}, presented the case of a solar seawater desalination plant with an average output of 40 m\textsuperscript{3}.d\textsuperscript{-1} on the island of La Désirade in the French Caribbean. The plant consisted of a 670 m\textsuperscript{2} tubular collector area, with 100 m\textsuperscript{3} hot water storage tank, and a 14 cell effect multi-effect evaporator unit. However, no economic analysis has been presented in this study. But Madani\textsuperscript{72}, states in his study, that for similar plants operating under similar conditions, the cost of product water was between $2.15.m^{-3}$ - $4.70.m^{-3}$. These results were obtained for temperatures of approximately 60°C - 65°C and lower costs could be obtained at a higher evaporator operating temperature of 88°C, but at the expense of risking scale formation requiring feed treatment.

Geroﬁ et al\textsuperscript{62}, presented a case of distillation systems in Australia, where the plant’s entire thermal energy requirements are provided by solar energy. A dynamic computer simulation model was developed for two solar-driven systems-one using flat-plate collectors (FPC) and a 76°C Multiple Effect Evaporator (MEE) plant, and the other using evacuated tubular collector (ETC) and a 95°C MEE plant. Various combinations of collector area and thermal storage volume were considered, and an optimal value determined in each case. For the particular case chosen (100 m\textsuperscript{3}.d\textsuperscript{-1}), the FPC system was found to be superior, giving an overall water cost of approximately $4.00.m^{-3}$, compared to the ETC system which gave an overall water cost of approximately $5.10.m^{-3}$. The costs were compared with those from a basin type solar still, where the cost of product water from an Australian-manufactured solar still was
determined to be approximately $8.75.m^{-3}$ for a 20 m$^3$.d$^{-1}$ plant. This cost, however, did not include operating costs. As mentioned earlier in this study, solar stills have few economies of scale. Hence the cost for the 20 m$^3$.d$^{-1}$ plant would also apply to a 100 m$^3$.d$^{-1}$ plant. Collector/plant systems, however, do exhibit economies of scale, and hence for a smaller capacity (example below 20 m$^3$.d$^{-1}$), solar stills could still be more cost-effective than solar-driven systems described in the Australian study.

Recently, a study of the economic feasibility of small solar MED seawater desalination plants for remote arid regions was conducted by El-Nashar$^{84}$. Three system configurations were investigated, which included a conventional system using a steam generator to provide steam for the Multi Effect Stack (MES) evaporator and a diesel generator to provide pumping power, a solar-assisted system using solar thermal collectors to provide hot water for the evaporator and a diesel generator for pumping power, and a solar stand-alone system using solar thermal collectors for the evaporator heat requirement with a solar PV array to provide electrical energy for pumping.

Conclusions drawn from the study by El-Nashar$^{84}$, state that for the second configuration of solar thermal-diesel-MES system, water cost depended mainly on the collector cost and the cost of diesel oil. At a low collector cost of $200.m$^{-2}$, and for diesel fuel costs of $10/GJ and $20/GJ, the cost of product water varied between $5.m^{-3} - $5.5.m^{-3}$ and at the highest collector cost of $400.m^{-2}$ the cost varies between $6.2.m^{-3} - $6.7.m^{-3}$. Evaporator capacity is stated to be 500 m$^3$.d$^{-1}$. Economies of scale are again presented in this case, with the cost of water reducing for increasing plant size. For a plant size of 100 m$^3$.d$^{-1}$, and for fuel cost of $10/GJ, water costs vary between $8.3.m^{-3} - $9.3.m^{-3}$, compared to $3.4.m^{-3} - $4.4.m^{-3}$ for a 1000 m$^3$.d$^{-1}$ plant size. At prevailing fuel costs, the cost of water from the stand-alone solar system (configuration 3), varies between $8.6.m^{-3} - $9.9.m^{-3}$ for a 100 m$^3$.d$^{-1}$ plant. The cost of water from the conventional system varies between $7.3.m^{-3} - $8.3.m^{-3}$, and approaches the costs from a solar system at high fuel costs, when the price of the PV panels is set at $6.Wp^{-1}$.

### 3.3.2. Solar MSF and Solar RO Systems

In further studies aimed at utilising solar energy for powering desalination systems, Suri et al.$^{85}$, analysed the application of photovoltaic (PV) and low-grade thermal energy using the solar pond system. In the first comparison using an MSF plant, three systems have been considered in their study, a conventional system (cogeneration electricity and water production plant), a partial solar system powering the MSF plant (thermal energy from solar pond and electricity from commercial source), and a full solar system powered MSF plant (thermal energy from solar pond and electricity from an organic Rankine power cycle (ORPC) or PV). A desalination plant capacity of 1 m$^3$.d$^{-1}$ of water production and annual plant utilisation factors of 75% and 90% have been considered for solar-based and the conventional systems, respectively. A comparison is presented in Table 3.5.
Similar studies of a Reverse Osmosis (RO) unit were conducted. Three systems were studied in this case, a conventional system, an ORPC system coupled to a solar pond powering the RO plant and a PV system coupled to the RO unit. The plant capacity considered was 1 m$^3$.d$^{-1}$. A comparison is presented in Table 3.6.

Water production using conventional RO system is about 26% less than from conventional MSF system. Solar-based technologies cannot compete with conventional systems as inferred from this comparison. Partially solar-powered MSF systems produce water at the lowest cost for all solar-energy-based technologies. Further reduction in the construction cost of the solar pond or increase in its energy extraction efficiency would make water production cheaper. PV cost was assumed to be $14.Wp^{-1}$ in the study, and a reduction in this cost would result in a reduction in water cost.

The effect of different parameters on the cost of distilled water from a solar still coupled to a PV-powered RO desalination plant has been presented by Goosen et al.\textsuperscript{76} The results are shown in Table 3.7. Considering different parameters of lifetime and interest rate, one would obtain varying costs. The cheapest option presented is for a lifetime of 20 years for the still, with a 5% interest rate, which yields a product water cost of $2.99.m$^{-3}$. 

### Table 3.5. Cost of water production using conventional and solar-powered MSF systems, Plant Capacity: 1m$^3$.d$^{-1}$ (Suri et al.\textsuperscript{85})

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Type</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Commercial System</td>
</tr>
<tr>
<td>Annual Water Production (m$^3$)</td>
<td>328</td>
</tr>
<tr>
<td>Cost of Water Production ($m^{-3}$)</td>
<td>1.75</td>
</tr>
</tbody>
</table>

### Table 3.6. Cost of water production using conventional and solar-powered RO systems, Plant Capacity: 1m$^3$.d$^{-1}$ (Suri et al.\textsuperscript{84})

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Type</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Commercial System</td>
</tr>
<tr>
<td>Annual Water Production (m$^3$)</td>
<td>328</td>
</tr>
<tr>
<td>Cost of Water Production ($m^{-3}$)</td>
<td>1.30</td>
</tr>
</tbody>
</table>
Table 3.7. Effect of different parameters on the cost of distilled water obtained from a proposed solar still plant coupled to a PV-powered RO desalination plant (Goosen et al.76)

<table>
<thead>
<tr>
<th>No.</th>
<th>Year</th>
<th>Rate of Interest</th>
<th>Annual Maintenance Cost ($)</th>
<th>Capital Recovery Factor</th>
<th>Annual Cost ($)</th>
<th>Yield (L.$(L.\text{-}1))</th>
<th>Annual Cost ($.(\text{m}^3))</th>
<th>Water Cost ($.(\text{m}^3))</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>10</td>
<td>0.05</td>
<td>7601</td>
<td>0.128</td>
<td>9336</td>
<td>174</td>
<td>6.44</td>
<td>5.60</td>
</tr>
<tr>
<td>2</td>
<td>10</td>
<td>0.08</td>
<td>8849</td>
<td>0.149</td>
<td>8882</td>
<td>183</td>
<td>6.13</td>
<td>5.33</td>
</tr>
<tr>
<td>3</td>
<td>15</td>
<td>0.05</td>
<td>5701</td>
<td>0.090</td>
<td>5703</td>
<td>285</td>
<td>3.94</td>
<td>3.47</td>
</tr>
<tr>
<td>4</td>
<td>15</td>
<td>0.08</td>
<td>6948</td>
<td>0.117</td>
<td>7186</td>
<td>226</td>
<td>4.96</td>
<td>4.27</td>
</tr>
<tr>
<td>5</td>
<td>20</td>
<td>0.05</td>
<td>4751</td>
<td>0.08</td>
<td>4856</td>
<td>334</td>
<td>3.35</td>
<td>2.99</td>
</tr>
<tr>
<td>6</td>
<td>20</td>
<td>0.08</td>
<td>6057</td>
<td>0.120</td>
<td>6391</td>
<td>254</td>
<td>4.41</td>
<td>4.00</td>
</tr>
</tbody>
</table>

3.3.3. RO Units Powered by Renewable Energy

In a study by gate Information Services (German Appropriate Technology Exchange), Germany, Teplitz-Sembitzky77, compares the various options available for powering RO plants with renewable energy. Table 3.8, gives the details of these options:

- 855 $.kW^{-1}, 20 years, 5% interest, annual O&M equivalent to 4% of investment costs. The range of costs for a given average wind speed reflects different wind speed distributions
- 5400-6525 $.kW^{-1}, 20 years, 5% interest, annual O&M equivalent to 3% of investment costs
- Parabolic troughs; 2925-4050 $.kW^{-1}, 20 years, 5% interest, annual O&M equivalent to 3% of investment costs
- Excluding storage or back-up costs; the RO-plant consumes 6 kWh.m^{-3}
- 1575-2070 $.kW^{-1}, 120 years, 5% interest, 25% plant factor, 5% interest, annual O&M equivalent to 6% of investment costs
- 100-1,000 kW; 200-1,000 meter reservoir depth; 100°C-140°C
- Energy- and fixed (capital) costs of RO. Fixed costs of large-scale RO with conventional power: 0.59 $.m^{-3}. Fixed costs of RO powered with renewable energy (< 2,500 m^3.day^{-1}): 0.70 $.m^{-3} (1350 $.m^3.day^{-1}, 20 years, 5% interest, annual O&M equivalent to 8% of investment costs, 85% plant factor)
Table 3.8. Cost of RO-plants powered on the basis of renewable energy
Teplitz-Sembitsky

<table>
<thead>
<tr>
<th>Energy Source</th>
<th>Power Generation Costs ($\text{MWh}^{-1}$)</th>
<th>Energy Costs of Desalination ($\text{m}^{-3}$)</th>
<th>Total Costs of Desalination ($\text{m}^{-3}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wind$^4$</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4 m.s$^{-1}$</td>
<td>108-239</td>
<td>0.65-1.40</td>
<td>1.35-2.10</td>
</tr>
<tr>
<td>6 m.s$^{-1}$</td>
<td>45-59</td>
<td>0.27-0.35</td>
<td>0.97-1.05</td>
</tr>
<tr>
<td>8 m.s$^{-1}$</td>
<td>27-32</td>
<td>0.16-0.19</td>
<td>0.86-0.89</td>
</tr>
<tr>
<td>Solar</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Photovoltaics$^3$</td>
<td>342-410</td>
<td>2.05-2.45</td>
<td>2.75-3.15</td>
</tr>
<tr>
<td>Solarthermal$^3$</td>
<td>185-257</td>
<td>1.10-1.54</td>
<td>1.80-2.23</td>
</tr>
<tr>
<td>Tidal$^5$</td>
<td>81-104</td>
<td>0.50-0.63</td>
<td>1.19-1.32</td>
</tr>
<tr>
<td>Geothermal$^6$</td>
<td>54-315</td>
<td>0.32-1.90</td>
<td>1.02-2.58</td>
</tr>
<tr>
<td>Conventional Power Plant</td>
<td>27-45</td>
<td>0.16-0.27</td>
<td>0.75-0.85</td>
</tr>
</tbody>
</table>

The cost of product water from all the solar-powered desalination units studied thus far, are tabulated and summarised in Table 3.9.

A review of the economics of solar-stills and solar-powered desalination units has been conducted in this study. However, the main objective was to obtain definite cost figures related to the Humidification-Dehumidification and Multi-Effect Humidification processes. As mentioned earlier, limited work has been carried out covering the economics of solar desalination and particularly the HD and MEH process economics. A comprehensive review of the cost of product water obtained by these processes has been covered in the following sections.

3.4. Economics of Solar HD and MEH Process

As previously mentioned in this study, the solar multiple-effect humidity process is the most efficient among all the solar desalination processes due to the recovery of heat of condensation for return to the system. In 1968, Garg et al.$^{23}$, estimated costs for desalinated water by the HD technique. They reported, that the solar still is suitable up to 90,000 L capacity, above which the HD technique would be more suitable. Based on very large plant capacities of one million gallons, they predicted the cost of water by the HD technique to be $0.14\text{m}^{-3}$ - $0.17\text{m}^{-3}$.

According to a study of MEH process by Grune.$^{26}$, in 1970, the best estimates for unit cost of product water based on the available information, it indicated that a 0.1 mgd plant could convert saline water at approximately $0.26\text{m}^{-3}$, and plants (10 mgd and larger) may convert seawater below $0.26\text{m}^{-3}$.

These estimates of plant size and product water cost, however, do not appear to be feasible given the realisation of the actual desalination costs over the years, and the application to small plant sizes of 10 m$^3$.d$^{-1}$ - 100 m$^3$.d$^{-1}$ for MEH based units.
### Table 3.9. Summary of Water Cost from Different Solar-Powered Desalination Units

<table>
<thead>
<tr>
<th>Plant Type</th>
<th>Capacity/Productivity</th>
<th>Cost of Product Water (S.m⁻³)</th>
<th>Description/Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar-MED</td>
<td>20,000 m³.d⁻¹</td>
<td>0.89</td>
<td></td>
</tr>
<tr>
<td></td>
<td>200,000 m³.d⁻¹</td>
<td>0.71</td>
<td></td>
</tr>
<tr>
<td></td>
<td>100,000 m³.d⁻¹</td>
<td>0.67-1.44</td>
<td>Solar-Pond-MED²⁸</td>
</tr>
<tr>
<td></td>
<td>151 L.m².d⁻¹</td>
<td>2.05</td>
<td>Solar Boiler-MED²⁰</td>
</tr>
<tr>
<td></td>
<td>40 m³.d⁻¹</td>
<td>2.15-4.70</td>
<td>Solar hot water storage-MED²¹</td>
</tr>
<tr>
<td></td>
<td>1000 m³.d⁻¹</td>
<td>1.20</td>
<td>Solar Only with storage²²</td>
</tr>
<tr>
<td></td>
<td>1000 m³.d⁻¹</td>
<td>1.10</td>
<td></td>
</tr>
<tr>
<td></td>
<td>10,000 m³.d⁻¹</td>
<td>0.92</td>
<td>Solar-Fuel-Hybrid²³</td>
</tr>
<tr>
<td></td>
<td>100,000 m³.d⁻¹</td>
<td>0.69</td>
<td></td>
</tr>
<tr>
<td>Solar-MED</td>
<td>100 m³.d⁻¹</td>
<td>4.00</td>
<td>Thermal Storage²⁶</td>
</tr>
<tr>
<td>Solar-MED</td>
<td>100 m³.d⁻¹</td>
<td>5.10</td>
<td></td>
</tr>
<tr>
<td>Solar-MED</td>
<td>100 m³.d⁻¹</td>
<td>8.3-9.3</td>
<td>Solar Thermal-Diesel Systems²⁴</td>
</tr>
<tr>
<td></td>
<td>500 m³.d⁻¹</td>
<td>5-6.70</td>
<td></td>
</tr>
<tr>
<td></td>
<td>1000 m³.d⁻¹</td>
<td>3.40-4.40</td>
<td>Complete Solar Based²⁵</td>
</tr>
<tr>
<td>Solar-MSF</td>
<td>1 m³.d⁻¹</td>
<td>2.84</td>
<td></td>
</tr>
<tr>
<td>Solar-RO</td>
<td>1 m³.d⁻¹</td>
<td>1.79</td>
<td>Partial Solar²⁶</td>
</tr>
<tr>
<td></td>
<td>-</td>
<td>12.05</td>
<td>Solar-PV System²⁶</td>
</tr>
<tr>
<td></td>
<td>-</td>
<td>5.70</td>
<td>Solar Pond-RO²⁶</td>
</tr>
<tr>
<td></td>
<td>-</td>
<td>2.99</td>
<td>Solar-PV²⁶</td>
</tr>
<tr>
<td></td>
<td>-</td>
<td>1.80-2.23</td>
<td>Solarthermal²⁶</td>
</tr>
<tr>
<td></td>
<td>-</td>
<td>2.75-3.15</td>
<td>Solar-PV²⁷</td>
</tr>
</tbody>
</table>

In later studies, Sousa and Janisch²⁷, estimated the cost of the Multi-Cycle solar driven desalination module (MCSSDDM), in their work on solar desalination using the multi-cycle concept. The cost of a MCSSDDM system was divided into three subsystems, collector field, solar tank and a rain tower. The cost of a conventional solar still was fixed at 100 units.m⁻³, and the cost of the MCSSDDM was given as:

Collector field: 100  
Solar Tank : 20  
Rain Tower : 100  

Total : 220 units.m⁻³

The lifetime for a solar still was assumed to be 20 years and that of the MCSSDDM, 15 years. The cost for the conventional solar still was then determined as 6.85 units.m⁻³ and that for the MCSSDDM, as 6.19 unit.m⁻³. The latter cost does not include electrical energy costs, to be calculated at 10 kWh.m⁻³. It was determined that the multi-cycle
concept permits approximately 30% reduction in the investment costs of solar desalination but the cost of the water remains unchanged.

In other work on the HD process, Madani\textsuperscript{73}, studied the dehumidification process in a laboratory-scale experimental study. Water cost was calculated by considering a backward curved blade fan with air flow rate of 5.27 m\textsuperscript{3}.s\textsuperscript{-1}. Calculating on the basis of experimentally derived enthalpy based heat transfer coefficient, 5h.d\textsuperscript{-1} operation, $0.1$.kWh\textsuperscript{-1}, a 20 year lifetime and with a 8% interest rate, the water cost was $1.9$.m\textsuperscript{3} in the month of August and $6.97$.m\textsuperscript{3} in February.

Further studies on desalination by humidification-dehumidification, were carried out by Khedr\textsuperscript{30} in 1993. He estimated the economics of the process considering a unit life of 10 years, and 85% load factor. From the chart depicted in his study, it can be determined that the process is economical for a unit size of 10m\textsuperscript{3}.d\textsuperscript{-1} and higher. An approximate cost of product water $3.3$.m\textsuperscript{3} can be inferred for a 10 m\textsuperscript{3}.d\textsuperscript{-1} plant and approximately $2.4$.m\textsuperscript{3} for a 30 m\textsuperscript{3}.d\textsuperscript{-1} unit.

The present value method was employed by Chaibi\textsuperscript{29}, to determine the cost effectiveness of the desalination plant studied by him. The present value cost (PVK) can be determined by the following equations:

\begin{equation}
PVK = I_o + \frac{(P + OM)}{PVF}
\tag{3.4}
\end{equation}

\begin{equation}
PVF = \text{Present Value Factor} = \frac{(1 + i)^n - 1}{i(i+1)^n}
\tag{3.5}
\end{equation}

The product cost of the unit production is given by:

\begin{equation}
P = \frac{PVK}{\sum_{i=1}^{n} E_i} \tag{3.6}
\end{equation}

where

- $E_i$ : production of the fresh water (m\textsuperscript{3}.y\textsuperscript{-1})
- $I_o$ : investment cost of the plant
- $OM$ : maintenance cost of the system
- $P$ : personnel cost
- $i$ : rate of interest (%)
- $n$ : life of the system (years)

From his study, Chaibi\textsuperscript{29}, determined a cost of $21.5$.m\textsuperscript{3} for a product of 2000 m\textsuperscript{3} of fresh water.
3.4.1. Recent Cost Studies for MEH Units

In more recent studies, Muller-Holst et al.\textsuperscript{38, 54}, simulated plant conditions in the laboratory to predict water costs for the desalination unit. Details of the unit are already described earlier in this study. The simulated conditions (with thermal storage) were created for comparison with the plant installed at Fuerteventura, which was without thermal storage. Storage size was 4.1 m\textsuperscript{3}, with collector area of 32.8 m\textsuperscript{2}. Under given conditions of flow rates and inlet temperatures, the predicted water price for the simulated system was as high as $45.m^{-3}$.

In July 1997, a pilot plant with thermal storage was investigated and measured in detail by the ZAE Bayern. The unit had a 2 m\textsuperscript{3} storage facility, with a collector area of 38 m\textsuperscript{2}. Daily distillate production of one module was 505 litres, and operated for 20 hours a day. The resulting water price from the test results at the site gives an astounding cost of $80.m^{-3}$. However, it is mentioned that significant cost reduction is possible in serial production.

In a later study, the cost of water determined from simulations, shows that the distillate costs can be forced down below $15.m^{-3}$ in solar operation using an optimised plant design (Muller-Holst\textsuperscript{54}). In the scope of the “SODESA” Project mentioned in their study, ZAE Bayern were in the process of developing a simulation tool for the TRNSYS simulation program to calculate and design the components of the desalination unit mentioned earlier. Under simulated and calculated boundary conditions for a thermal storage coupling facility with a 24-hour operation, the estimated distillate costs are $11.m^{-3}$.

3.4.2. Investment and Operational Costs

Comparing cost of investment for the HD process with other conventional techniques, Garg et al.\textsuperscript{23}, presented a cost of $353.m^{-3}$ for the HD process compared with $325.m^{-3}$ for the MEE plant at Freeport, USA and $353.m^{-3}$ for the flash evaporation plant at San Diego.

Another comparison was made by El-Dessouky\textsuperscript{86}, in a study utilising waste heat from a gas turbine powered HD desalination unit. The comparison of this process is shown in Table 3.10. For the comparison shown in the table, the author mentioned that the cost includes only the equipment capital investment. Performance ratio of the MSF plant was assumed to be 7, feed water for the RO plant was 43,000 ppm, and all the dual-purpose plants were based on gas turbines.

As seen in the table, the HD process powered by waste heat has the lowest specific cost compared with other conventional processes based on waste heat.

As can be seen from the cost comparisons mentioned above, the investment costs of the HD process are comparable with other conventional processes.
Table 3.10. Comparison of HD Process (with waste heat) with other processes (El-Dessouky [86])

<table>
<thead>
<tr>
<th>Parameter/Process</th>
<th>Specific Cost ($/m^3.d)</th>
<th>Specific Power (kJ/kg)</th>
<th>Water to Power Ratio (kg/kWh)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Humidification/Dehumidification</td>
<td>287</td>
<td>61.26</td>
<td>3.9772</td>
</tr>
<tr>
<td>Single MSF</td>
<td>1451.18</td>
<td>294.21</td>
<td>-</td>
</tr>
<tr>
<td>Dual MSF</td>
<td>3647.4</td>
<td>96.519</td>
<td>23.919</td>
</tr>
<tr>
<td>Single RO without energy recovery</td>
<td>1022.8</td>
<td>50.420</td>
<td>-</td>
</tr>
<tr>
<td>Single RO with energy recovery</td>
<td>1127.4</td>
<td>33.619</td>
<td>-</td>
</tr>
<tr>
<td>Dual RO without energy recovery</td>
<td>1906.3</td>
<td>36.20</td>
<td>36.819</td>
</tr>
<tr>
<td>Dual RO with energy recovery</td>
<td>2225.8</td>
<td>27.20</td>
<td>49.19</td>
</tr>
</tbody>
</table>

3.5. Cost of Other HD Processes

For comparison, the cost of other desalination processes based on similar principles as the MEH process can be briefly reviewed. Such similar processes include the Dewvaporation technique utilising air as a carrier gas and the aero-evapocoondensation process.

3.5.1. Dewvaporation

As mentioned, an innovative and relatively new non-traditional heat efficient tower process, referred to as “Dewvaporation” has been studied by Beckman [45]. This process used air as the carrier-gas to evaporate water from saline feeds and dew form as pure condensate at constant atmospheric pressure.

Summarising the Dewvaporation technique studied by Beckman [45], the demonstration bench-scale tower used for desalination of brackish water had an average GOR of 7.8, while seawater desalination was demonstrated with a GOR of 7.5. Accounting for certain losses, it is mentioned that the GOR could be increased to 16.7 and 20. The bench-size database allows design for and demonstration of the Dewvaporation technology at 1000 gallon per day capacity. The projected capital cost as mentioned for a 3.79 m^3.d^1 (1000 gallon per day) unit based on laboratory data has been estimated at $369, which includes water heater, two pumps, one air fan (pumps and fan require 0.46 kWh per day of electricity) and a manufacturers gross margin of 30%. The capital cost includes a 594.6 m^2 of heat transfer wall. Therefore, it has been calculated that the total inclusive operating cost would be $0.88.m^-3 of condensate. Using solar energy or atmospheric steam waste heat would lower the operating cost to approximately $0.39.m^-3 of condensate. Unit dimensions mentioned are 1.22 m x 1.22 m x 2.44 m height. Enhanced heat transfer coefficients were found only with entrance region effects and not used in the final cost studies by Beckman [45].

The operating costs mentioned, are dominated by the natural gas fuel cost ($0.10.m^-3.d), which represents 74% of the total cost of $369. As mentioned earlier, fuel costs can be reduced by less expensive fuels, higher GOR values and by the use of solar energy or waste heat. Waste heat sources and solar, could be essentially free.
However, solar will have an additional capital charge in the operating cost category. A solar collector operating at 85°C could be used to evaporate (not boil) water into the air stream at the top of the desalination tower. Assuming that, the collector could absorb 315.5 W.m\(^{-2}\), and the collector cost was $53.8.m\(^{-2}\), then during daylight hours, the heat from the collector could be used as an additional capital cost of $343. The capital cost would increase from $0.12 earlier to $0.24 as the fuel cost is reduced from $0.65 to zero. The operating cost would then reduce from $0.88.m\(^{-3}\) to $0.34.m\(^{-3}\) for the solar application. Table 3.11, shows the cost summary of desalination plant capacities.

### 3.5.2. Carrier Gas Process

A capital cost comparison is conducted for the carrier gas process (CGP), which is described earlier in this study. A comparison for large systems is shown in Table 3.12.

**Table 3.11. Cost summary of Desalination Plant with Different Capacities (Beckman\(^{45}\))**

<table>
<thead>
<tr>
<th>Production m(^3).day(^{-1})</th>
<th>GOR</th>
<th>Capital Cost*</th>
<th>Operating Cost*</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Natural Gas</td>
<td>Waste Heat</td>
</tr>
<tr>
<td>1.90</td>
<td>24</td>
<td>544</td>
<td>0.62</td>
</tr>
<tr>
<td>3.79</td>
<td>12</td>
<td>369</td>
<td>0.88</td>
</tr>
<tr>
<td>7.57</td>
<td>6</td>
<td>289</td>
<td>1.40</td>
</tr>
</tbody>
</table>

* $.m\(^{-3}\).d (Operating Cost Includes Capital Charge)

**Table 3.12. Capital Cost Comparison for Large Systems ($m\(^{-3}\).d) (Larson et al.\(^{49}\))**

<table>
<thead>
<tr>
<th>Process</th>
<th>Brackish water</th>
<th>Seawater</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carrier Gas</td>
<td>355</td>
<td>384</td>
</tr>
<tr>
<td>RO</td>
<td>265</td>
<td>1182</td>
</tr>
<tr>
<td>MSF</td>
<td>N/A</td>
<td>1111</td>
</tr>
</tbody>
</table>

Table 3.12, indicates that the CGP would have greatest cost advantage in seawater applications. Operating cost comparison is shown in Table 3.13.
Table 3.13. Operating Cost Comparison for Large Systems (S.m⁻³)
(Larson et al.⁴⁹)

<table>
<thead>
<tr>
<th>Carrier Gas Process (Brackish or Seawater)</th>
<th>0.63</th>
</tr>
</thead>
<tbody>
<tr>
<td>High Temperature Heat</td>
<td>0.67</td>
</tr>
<tr>
<td>Solar Energy</td>
<td>0.51</td>
</tr>
<tr>
<td>Ambient Air Heat</td>
<td>0.19</td>
</tr>
<tr>
<td>Discharge Heat</td>
<td>0.19</td>
</tr>
<tr>
<td>Reverse Osmosis</td>
<td>0.40-0.53</td>
</tr>
<tr>
<td>Brackish Water</td>
<td>1.05-1.59</td>
</tr>
<tr>
<td>Seawater</td>
<td>1.05-1.59</td>
</tr>
<tr>
<td>MSF (Seawater Only)</td>
<td>1.05-1.59</td>
</tr>
</tbody>
</table>

Although the economic comparison of the CGP with other processes poses an encouraging picture for this process, additional study is required to make this process commercially viable and to be able to replace conventional techniques of desalination.

3.5.3. Aero-Evapo-Condensation Process

In the aero-evapo-condensation, described earlier in this study, Bourouni⁴⁸, estimates the cost of water for the case of using geothermal energy. The process is energy intensive, and in the case of the energy being supplied by fuel, the cost of energy accounts for 84% of overall production costs. In this specific case, the cost of evaporated water is evaluated at $4.8.m⁻³. But as the unit operates on geothermal energy, the cost drops to $1.2.m⁻³.

3.6. Case for Cost Reduction

As can be observed from the cost figures mentioned, there is a remarkable difference between the cost of product water calculated and reviewed by different investigators. One major common hurdle in reducing costs and the main reason for high water costs; was the cost of the solar collector in all the units investigated. The solar collector unit constitutes almost 28% of the total costs (Garg et al.²²). Another prohibitive cost mentioned, is that large amounts of air needed to be recirculated and which leads to high energy costs due to pumping. Methods to reduce the cost of produced water have been analysed, which includes development of more efficient collectors and implementation of thermal storage facilities with the solar desalination units. Additional information of the costs of solar collectors and implementation of storage facilities are given in proceeding sections.

3.6.1. Solar Collectors and Thermal Storage

All studies of solar desalination reviewed thus far, state, that the single largest prohibitive cost in the process is the cost of the solar collector unit. This has already been mentioned earlier in this study. The impact of the collector cost can be seen from the study by Samad⁴⁸, in which he describes the operational problems in solar
desalination plants and suggests remedies to overcome such difficulties. In his study, it is mentioned that the minimum water cost is very sensitive to the collector cost, and that a drop in collector cost to 50% of its basic value could result in a drop in water cost from $4.77\,m^{-3}$ to $3.03\,m^{-3}$.

### 3.6.1.1. Types and Cost of Collectors

The most commonly used collectors are the Flat Plate Collectors (FPC), Evacuated Thermal Collectors (ETC), paraboloidal trough concentrating collectors and solar ponds.

Paraboloidal trough collectors, collect only direct solar radiation and require auxiliary energy for tracking which could be expensive in remote regions. The other collectors can utilise direct and diffuse radiation.

In regions where solar ponds are applicable, and where fairly large collector areas are required, they would appear to be the most cost effective of all collector options because of their relatively low capital cost per unit area and because of the substantial thermal storage provided. However, certain site prerequisites would need to be met to make solar ponds economically viable.

FPC is commercially produced in large quantities and in a wide variety of designs mainly for hot water production. Initial capital cost is not prohibitively high and ranges from $80\,m^{-2}$ - $250\,m^{-2}$ (Belessiotis and Delyannis\textsuperscript{70}), depending on the material used and the country of origin. FPC have low exit temperatures and are combined efficiently with solar stills, or may be combined with small capacity MED, but efficiencies are low.

ETC are more suitable for conventional distillation plants, as MED or Thermal Vapour Compression (TVC). They are more expensive than FPC and range from $300\,m^{-2}$ - $550\,m^{-2}$. Initial capital costs are high but operational costs are relatively low, these are more suitable for large capacities compared with low temperature FPC.

The choice of the collector would have a great impact on the final cost of the product water. Currently, the prices of the collectors are relatively high and only a reduction in these prices would enable the cost of desalinated water to drop to economical levels, which would allow solar-thermal units to compete with conventional desalination techniques.

An alternative to the FPC, could be a new convex type solar collector that has been mentioned earlier in this study and which could be more favourable to the flat plate collector. The use of the new tubeless collectors leads to a 45% greater energy gain than the flat plate collectors.

### 3.6.1.2. Thermal Storage

It has been reported earlier in this study, that coupling solar desalination units with a thermal storage facility improves productivity and reduces the cost of product water. Muller-Holst et al\textsuperscript{38, 54} have studied the effects of implementing a 24-hour operation with thermal storage in their investigation. The self-sufficient pilot plant at
Fuerteventura, showed a maintenance free working for 5 years. Investigation of the actual improved laboratory distillation unit yields a reduced specific energy demand of 90 kWh.m$^{-3}$ - 100 kWh.m$^{-3}$. This is achievable by means of better evaporation surfaces and thinner flat plate heat exchangers on the condensation side of the distillation unit. This step reduces the water cost by 20%. Further cost reduction by at least half the original cost could be achieved by implementing storage capabilities. The test plant, with storage implementation in Tunisia, gains 500 L.d$^{-1}$ distillate output with one distillation module and storage implementation, which ultimately leads to a reduction in water cost.

Therefore, it is important to find ways to reduce the collector cost and implement storage facility and find other methods to improve system efficiency in order to achieve cost reductions so that solar desalination could become competitive with conventional desalination techniques in the future and not merely for small capacity decentralised use in arid regions of the world.

3.7. Conclusions

A summary of the studied solar HD and solar MEH desalination costs are tabulated in Table 3.14, Table 3.15 and Table 3.16.

### Table 3.14. Summary of Estimated Capital Cost for Different Types of HD Systems

<table>
<thead>
<tr>
<th>Process</th>
<th>Capital Cost ($ .m$^{-3}$.d)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar HD System (Estimated)</td>
<td>353</td>
</tr>
<tr>
<td>CGP Process</td>
<td>384</td>
</tr>
<tr>
<td>Dewvaparation Process</td>
<td>369</td>
</tr>
</tbody>
</table>

### Table 3.15. Summary of Cost of Water from HD and MEH Processes

<table>
<thead>
<tr>
<th>Capacity/Productivity</th>
<th>Cost of Water ($ .m$^{-3})</th>
<th>Ref:</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 mgd</td>
<td>0.14-0.17*</td>
<td>23</td>
</tr>
<tr>
<td>0.1 mgd</td>
<td>0.26*</td>
<td>26</td>
</tr>
<tr>
<td>10 mgd</td>
<td>&lt; 0.26*</td>
<td>26</td>
</tr>
<tr>
<td>30 m$^3$.d$^{-1}$</td>
<td>3.30*</td>
<td>30</td>
</tr>
<tr>
<td>2000 m$^3$</td>
<td>21.50</td>
<td>29</td>
</tr>
<tr>
<td>437.2 L.d$^{-1}$</td>
<td>45</td>
<td>38</td>
</tr>
<tr>
<td>505 L.d$^{-1}$</td>
<td>80</td>
<td>38</td>
</tr>
<tr>
<td>15</td>
<td>15</td>
<td>54</td>
</tr>
<tr>
<td>11</td>
<td>11</td>
<td>54</td>
</tr>
</tbody>
</table>

* predicted
Table 3.16. Operational Cost for Other HD Processes with Solar Energy Usage

<table>
<thead>
<tr>
<th>Process</th>
<th>Operational Cost (S.m$^3$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>CGP Process</td>
<td>0.67</td>
</tr>
<tr>
<td>Dewvaporation Process</td>
<td>0.34*</td>
</tr>
</tbody>
</table>

* Cost includes Capital Charges

From the cost figures studied and summarised in Table 3.15, it can be stated, that there is no strong basis for judgment of a single cost or even a narrow range of water cost for the Solar Humidification-Dehumidification process. Further work needs to be carried out, especially in the cost estimation area, so that an estimation of the actual cost can be obtained. From the capital and operational costs mentioned in Table 3.14 and Table 3.16, respectively, the HD process does appear economically attractive. From the last tier (energy costs) of the cost breakdown shown in Table 3.1, accounts for only 0%-10% of total costs, the Solar HD process may be a suitable replacement to all other forms of solar desalination techniques in the smaller capacity range.
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69

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71