TRANSIENT MODELING OF A NEW SOLAR DISTILLATION UNIT FOR REMOTE AREAS

by

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A Thesis Presented to the Faculty of the American University of Sharjah College of Engineering in Partial Fulfillment of the Requirements for the Degree of

Master of Science in Mechanical Engineering

Sharjah, United Arab Emirates

December 2016
Approval Signatures

We, the undersigned, approve the Master’s Thesis of Muhammad Mustafa Muhammad Iqbal.

Thesis Title: Transient Modeling of a New Solar Distillation Unit for Remote Areas

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Acknowledgements

I would like to take this chance to thank Dr. Mehmet Fatih Orhan and Dr. Hassan Fath, my advisors, for their guidance and valuable recommendations throughout my thesis completion. I would also like to express my gratitude towards their understanding, persistence, help and motivation they provided me with throughout the course of my thesis. Additionally, I would like to thank the MCE department for giving me the chance to get a graduate teaching assistantship at the American University of Sharjah. The culmination of this work would not have been possible without the support of my family and friends.
Dedication To

My parents, who gave me the freedom to choose the path of my own as a source of inspiration and motivation

My brothers, sisters, friends and wife, for their unconditional support and encouragement to achieve this goal
Abstract

The concept of lack of fresh water in our daily life is hard to imagine in recent times with its increasing demand. Lack of fresh water supplies has become one of the major current societal concerns, especially for people living in remote areas, where there is limited or no resources of fresh water. Large amount of fresh water is produced using the conventional desalination technologies like thermal and membrane process. The conventional desalination technologies for large water production are Multi Stage Flash (MSF), Multi Effect Desalination (MED), Vapor Compression (VC) and Reverse Osmosis (RO). Big shares of Thermo and/or electrical energies are utilized using these processes. These technologies require high quality operation and maintenance. They require the usage of fossil fuels, which causes negative environmental effects. In order to minimize these effects and especially for remote areas, where the demand of fresh water is less, renewable energy integrated with desalination is the optimum solution. This thesis presents the transient modeling of a novel solar desalination system for remote areas. The new system consists of Solar Still (SS) integrated with multi effect Humidification-Dehumidification (HDH) with built in solar absorber. As the conventional solar still distillation has drawbacks of low specific productivity due to the loss of condensation energy as well as the adverse effect in environment due to brine discharge, the aim is to model and design a solar driven integrated SS-HDH-Absorber desalination unit that can produce the fresh water capacity of 50 Liter with zero brine discharge (ZBD) for a single family use. The main objective is to enhance the unit’s water production by utilizing the waste latent heat of condensation of the still and solar absorber thermal energy for additional water production in addition to recovering brine salts as by product. This will lead to produced water cost reduction while maintaining the simplicity of the system operation and maintenance. A numerical code is developed in MATLAB to simulate the integrated system and study the effect of various environmental, design and operational parameters on the unit’s productivity.

Search Terms: Solar Desalination, Solar Still, Humidification-dehumidification, Solar Absorber, Transient Modeling
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Nomenclature

A  Area (m²)
C_p  Specific Heat (J/kg K)
D_h  Hydraulic diameter
Dt  Time step (sec)
d  Ropes diameter (m)
HDH  Humidification-Dehumidification
h  Specific Enthalpy (J/kg)
h_f_g  Latent heat (J/kg)
I  Solar intensity (W/m²)
L  Liter (m³)
M  Combined mass (kg)
N  Number of Ropes
N.F.  Number of fins
P  Vapor pressure (Pa)
Pr  Prandtl’s number
Q  Heat flux (W/m²)
Re  Reynolds number
RH  Relative humidity
SS  Solar Still
T  Temperature (°C)
t  time (sec)
v  Wind velocity (m/s)

Greek Letters

α  Absorptivity
θ  Angle
σ  Boltzmann constant = 5.67×-8 (W/m² K⁴)
ρ  Density (kg/m³)
Δ  Difference
μ  Dynamic viscosity (kg/m s)
ε  Emissivity
ν  Kinematic viscosity (m²/s)
τ  Transmissivity
## Subscripts

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Chapter 1: Introduction

1.1. Water Shortage in Rural Areas

Fresh water is the essence of life and is the most important constituent of the environment. Along with energy, water is important for sustaining life in the modern world and is a basic human requirement for domestic, industrial and agriculture purposes. It is also the basis for social well-being of people [1-2]. Water is, however, the most abundant resource available on earth, covering more than two thirds of the planet’s surface. Although the availability of water is abundant in the world, yet 97% of it is salty, whereas the rest is fresh water (3%). Less than 1% of fresh water is within human reach [3-4]. Even considering the small ratio is indeed adequate to support life and vegetation on earth. The crisis related to fresh water is a global problem after climate change, which is mainly due to industrial and other sources of pollution. Having access to clean and drinkable water is a major problem faced by many countries internationally. Hence, tremendous efforts are now required to address the aforementioned water problems.

In most of the remote and desert regions, fresh water is very limited i.e. scarce. This is true for the Mediterranean basin, Middle East and North Africa (MENA) region, where many big cities and small villages suffer from lack of fresh water enormously. However, at the same time, they are blessed with salinized water and solar energy. As water scarcity has become a major problem, desalination of saline water has become one of the most sought alternative water treatment methods to support fresh water shortage around the world, especially in rural areas of inadequate fresh water. Low density remote areas lack not only fresh water, but also electrical energy sources, while renewable energy sources are enormous and are mostly referring to solar radiation. For these regions, solar desalination can be a moderate solution for their needs [5-8].

1.1.1. Desalination. Desalination is the separation of salt from saline (sea or brackish) water to produce fresh water, which consists of low salt concentration, and brine concentration [9]. Desalination of brackish and sea water, which is used to provide the needed drinking water, not only fulfills basic social needs, but also prevents any serious impact on the environment. Figure 1 show the global installed desalination based on different sources of feed water.
It is observed that 58% of the desalination plants are fed by sea water, followed by brackish water (23%). Desalination processes provide fresh water to many communities, for domestic and industrial purposes, which helps enhancing the socio economic development in several developing countries, namely Middle East and the like, where fresh water is always in the form of scarcity. Hence, as a result there has been a dramatic worldwide increase in the number and capacity of desalination processes and plants and extensive Research and Development activities, especially currently with renewable energy [11].

There are several types of desalination processes and they can be subcategorized into thermal and membrane processes. Among them are Multi Stage Flash, known as MSF and Multi Effect Desalination (MED), Vapor Compression (VC) and Reverse Osmosis (RO). Figure 2 shows the classification of water desalination technology in general. These technologies are relatively expensive, especially in the case of small amount of fresh water production. They require high quality operation and maintenance. On the other hand, these technologies require conventional energy sources such as hydrocarbon fuels, which have an adverse impact on the environment. In addition, fuel being a finite resource also attributes to limiting its long term usage. In order to secure the future, renewable energy (RE) based desalination is the ideal solution. Thus, if RE is integrated with desalination processes, it could counteract problems of water case, pollution and depletion of resource, especially in the remote areas.
1.1.2. Renewable energy and desalination. There are several reasons for the suitability of RE (solar, wind, geothermal) for saline water desalination in remote areas. These include:

- Remote or arid regions are blessed by RE sources as compared to conventional energy supply.
- The operations and maintenance of RE systems in remote areas are easier than conventional energy ones.
- It is convenient in terms of economic aspect to implement RE driven desalination system as the RE resources is available at the plant site.

RE and desalination are two different technologies which can effectively be combined in various forms as shown in Table 1 and Figure 3.

Table 1: Recommended renewable energy–desalination combinations [12]
RE driven desalination system is a stand-alone system, which means there is no electricity connection grid. At present, through research and development stage, RE-desalinated systems are implemented as pilot size applications, where the capacities range from few m$^3$/day up to 100m$^3$/day. Selecting the most suitable RE-driven desalination technology is a tedious task as it depends on several factors that include plant size, cost, feed water salinity and the like. For larger capacity, wind or geothermal is the best choice and it can be used to drive conventional desalination plant such as MED or MSF, but these plants face a severe problem with regards to operation and maintenance (O & M) requirements and scaling problems, which can be a tedious task to setup in remote areas. Not all the combinations shown in the Figure 3 are suitable for remote regions as some of the combinations are more suited for large-scale applications.

Based on the environmental condition in GCC area, solar driven desalination is the best choice to study as the amount of solar energy available is enormous in this region. Figure 4 shows the current possible configuration of solar energy with different types of plants in percentages. It can be seen that PV is considered proper solution for small applications in sunny regions. PV is mainly used with reverse osmosis (RO) desalination plant. In terms of economical aspect, RE desalination is relatively more expensive than the conventional technology in terms of capital cost;
however, as the cost of materials is decreasing, installing cost is decreasing for such types of PV-RO technology.

Figure 4: Solar energy configurations [12]

A direct solar desalination system is the best option for remote area and very small plant size (up to 100 L/day) because of its design and O & M simplicity. Small PV panel could be used to generate electricity for pumps and other needed components. PV cooling (known as PV/T) is needed to improve PV electrical efficiency and the recovery of the cooling thermal energy. PV covering is also needed to be free from dust and maintenance cost, which is generally the drawback of PV. More detailed discussion of direct solar desalination system is given in the next chapter.

1.2. Objective

The objectives of this study are based on the published work involving solar still configuration. Solar still is known for producing low cost fresh water productivity. Recently several solar stills integration methods are being proposed to enhance the productivity of the system as compared to the conventional solar still. Solar still integrated with HDH is a relatively new concept. In this thesis, SS is integrated with i- Two stage HDH system and ii- Built in finned solar absorber to reach our objectives of designing small desalination unit of higher efficiency and lower specific water cost (L/$). The main outcomes of the thesis are summarized below:

- To design a novel small size Solar Desalination unit for remote areas of; i- simple design; ii- operational and maintenance (O&M) and iii- of low cost.
- To develop a distillation system that can enhance the production of water in solar stills through the recovery of latent heat of condensation and thermal
energy. This is carried out by integrating solar still with Multi Effect HDH process and built-in finned solar absorber systems.

- To do an analysis based on transient one dimensional system, which will be carried out by solving the governing equation for the different components of the systems:
  - Still system
  - Two effects HDH system
  - Built-in finned solar absorber system
- To study the effect of different design, operation and environmental parameters on the distillation system performance and productivity.
- To design a 50 L/day distillation unit for a single family use as a module to larger plant.

### 1.3. Thesis Organization

Chapter 1 introduces the various desalination technologies and summarizes the processes that are suitable in the remote areas. In Chapter 2, a literature review is carried out on active and passive forms of solar still configuration, humidification-dehumidification process, as well as on solar still integration methods with different configuration. Chapter 3 discusses the proposed system, model equations and assumptions that were carried out in the design process. The equations in each component of the model are analyzed using MATLAB and validation with published work is also presented. In Chapter 4, results of the designed model are presented for the summer and winter days. A parametric study is also carried out on different designs, environmental and operational parameters to optimize the design system. Finally, Chapter 5 presents the conclusion and recommendations for future work.
Chapter 2: Literature Review

2.1. Introduction

Fresh water is obtained by a process called natural distillation (hydrological cycle). In this process, water evaporates from the surface of the oceans and sea water due to lower density and the wind speed. The vapor rises and condenses as rain to form rivers and other surface water bodies. Most places with water problems are located in dry or remote areas. In these regions, solar energy is readily available for most of the year. Solar energy is a pollution free resource which is abundant in nature and is everlasting. In this thesis, two processes that utilize solar energy, Solar Stills (SS) and Humidification-Dehumidification (HDH), will be addressed.

2.2. Thermal Solar Desalination

Thermal solar energy is one of the most promising applications of renewable energy to seawater desalination, as it is suitable for arid and sunny regions. Solar distillation is a relatively simple treatment for saline water. The process is similar to the hydrological cycle. As a technology, it is economical, effective and has minimum environmental impact. It is attractive to use such solar distillation system in remote location as it requires minimal maintenance in terms of cleaning and salt removal.

A solar thermal desalination system can be classified into two methods; direct and indirect. In direct method, the distiller and the solar collector are coupled as one and known as Solar Still (SS). Solar stills as a direct process is of interest to remote area deployment as the requirements in these areas are simple. In these areas, sea or brackish water is available, but pure water is limited. On the other hand, conventional energy sources are limited in these areas, and hence direct solar distillation is the only process that can be used to convert available water into drinkable one.

In indirect method, thermal energy is collected by solar collectors, or solar based ponds. They drive conventional desalination methods that include MSF, MED and MVC. These technologies could be driven by a flat plate, parabolic trough or solar collectors [13], however, the use of conventional technologies is not feasible in the operation and maintenance of small scale fresh water production in remote area. Direct method using SS is, therefore, considered to be the best option for remote areas.
2.3. Solar Still

2.3.1. Introduction. Solar stills are not commercialized; however, for a few individual units and is suited for small production systems where the freshwater demand is less than 200 m³/day [14]. It is a simple device which can convert saline water into drinking one via evaporation and condensation. The basic principle of solar still is similar to the greenhouse effect as the solar energy enters the SS through the transparent glass or plastic which is slope in shape and heats a basin of sea water. The basin is well insulated to maintain the maximum energy and reduce heat loss. The heated water evaporates and is condensed on the cooler glass cover and droplets are collected. The distilled water obtained is of high quality because all the salts, components such as inorganic and organic are left behind.

Besides having such simple and attractive features, one of the main drawbacks for this desalination system is low thermal efficiency and low productivity. The reasons are: firstly, the condensation heat loss on the glass surface, and secondly its daily variation due to sinusoidal trend, which changes environmental conditions of solar intensity and ambient temperature, thus affecting production efficiency. A general rule of thumb is that SS produces only 4-6 L/m²/day of freshwater and has an efficiency of about 30-40% [19-22]. Hence, it should be a design with inexpensive materials to avoid initial capital cost. Conventional stills have a high specific capital cost ($/m³) but it is balanced off by the operational cost. There are other disadvantages to conventional stills such that it is highly vulnerable to extreme conditions, where there is a risk of formation of algae on the surface and accumulation of dust on glass covers which can also reduce the penetration of solar irradiation.

As a result of the aforementioned problems, several types of solar still have been evolved. Among them are single or multiple stills as will be explained in Section 2.3.2. Only the basin type stills using single effect distillation have been used for the supply of large quantities of water for communities and other purposes. A number of designs and modification of the solar still have been tested for improving productivity by many researchers on passive and active configuration. They studied some factors influencing the SS performance which include: glass cover slope, base insulation, basin water depth, and use of dye. In addition, climatic conditions such as wind speed, ambient temperatures and solar radiation, as shown in Figure 5, were also studied. A
number of researchers have carried out experimental methods to assess the productivity and efficiency, but they are quite expensive to set up and time consuming. On the other hand, many have developed and proposed mathematical models to illustrate theoretical analysis on solar still.

2.3.2. Classification of solar stills. Solar stills can be classified as either single effect or multi-effect stills. Further, based on the source of heating, they are
classified as passive or active as shown in Figure 6. In passive solar still, water is heated up in the basin by the direct solar irradiation. In the active solar still, thermal energy is externally collected and is then fed into the basin of a passive solar still for faster evaporation. The external source could be solar collector or waste thermal energy. Be it passive or active still, basin construction materials play an important role for the productivity of the water obtained. The construction of the basin could be made of either plastic or metal sheets. Common metals used are copper, aluminium and steel. As thermal conductivity is the important property related to the evaporation of the water, copper and aluminium are considered to be the most efficient in terms of conductivity, but are comparatively expensive as compared to galvanized steel or any plastic material [19-20].

2.3.3. Passive solar stills. It is the simplest and the most practical design to produce distilled water. Experimental based and mathematical modeling are carried out to study the performance of various passive designs; single and double sloped conventional still, with and without supplementary reflectors, and of single and double glass cover were studied by many researchers. A schematic diagram of simple basin type solar still (single effect) is shown in Figure 7. This is the original solar still construction, which consists of typical single glazing cover over the water surface, where heat energy exchange takes place. According to [21], these stills have an efficiency of 30 to 40%, with typically producing distillate water of 4-6 L/m²/day.

![Figure 6: Classification of solar stills (SS) [17]](image-url)
2.3.3.1. **Cover material of thickness.** Material selection plays an important role in deciding the overall output of fresh water and the efficiency of the still. Among them, still cover is of the most component aspects. There are two types of materials, glass and plastic. Glass is preferred, but is expensive. Phadatre and Verma [22] evaluated the effect of using plastic glass in a single basin slope solar still as shown in Figure 8. The output obtained was 2.1 L/m$^2$/day for the water depth of 2cm, which is comparatively less as compared to the glass still.

In addition to the glass materials, SS glass shape plays a pivotal role in water productivity. Tayeb [23] did the experimental study for four different types of glass cover of the same effective absorption area shown in Figure 9. They are a- incline flat glass cover, b- a semi-sphere cover, c- a bi-layer semi sphere cover and d- an arch cover respectively. It was found that the conventional incline surface (a) gives the best result among others.
2.3.3.2. Single and multi-effect SS. A number of researches have evaluated the effect of single effect versus the multi effect solar still with numerous configurations for the passive design. Tiwari et al. [24] made a comparison for the single slope basin and double slope basin. In their experiments, they found that the single slope gives a better output result as compared to double slope if the experiment is carried out in cold climate. However, the opposite was reported in warm conditions. Karaghouli and Alnaser [25] experimentally studied two single and double basin stills as shown in Figure 10 and 11. Both stills have the same basin area of 0.45 m² and the basin sides were also insulated with Styr bore material of thickness 2.5 cm. The experiments were conducted for a period of 5 months and it was recorded that maximum production is during the month of June as the solar radiation was higher. The maximum production was 3.91 L/m²/day for the month of June. In the case of not using insulation, the daily production amount in the month of June was 3.13 L/m²/day. Overall, it was observed that the still production increased by 40% in case of double basin still as compared to single basin still. Similarly, Rajaseenivasan et al. [26] did a comparative study on single and double slope basin depth of water, storage material and porous absorbers. Window glass of 4 mm thickness was used and was coated in black to ensure maximum absorption. Both stills were fabricated using mild steel, having thickness of 1.4 mm and 50 mm insulated thickness. The covers were inclined to 30 degrees. From the experiments, it was observed that although the capital cost gets higher, the production of fresh water increases by 85 % in the case of
double basin slope for the same condition. Also, using porous materials or energy storing materials further enhances the distillate output. In addition to this, in comparison with different material, mild steel basin materials give better production as compared to jute cloth and clay pot as shown in the Table 2.

Table 2: Comparison of productivity of single and double basin still [26]

<table>
<thead>
<tr>
<th>S. no</th>
<th>Basin condition</th>
<th>Total production (ml/day)</th>
<th>% increase</th>
<th>% increase (with single basin still)</th>
<th>% increase (with single basin still at 8 cm depth)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Single basin</td>
<td></td>
<td>Double basin</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>8 cm</td>
<td>1330</td>
<td>ref</td>
<td>2685 ref</td>
<td>95.86</td>
</tr>
<tr>
<td>2</td>
<td>6 cm</td>
<td>1395</td>
<td>4.66</td>
<td>2700 3.98</td>
<td>93.54</td>
</tr>
<tr>
<td>3</td>
<td>4 cm</td>
<td>1490</td>
<td>9.52</td>
<td>3205 7.80</td>
<td>92.17</td>
</tr>
<tr>
<td>4</td>
<td>2 cm</td>
<td>1610</td>
<td>17.40</td>
<td>2990 12.80</td>
<td>85.71</td>
</tr>
<tr>
<td>5</td>
<td>Clay pot – facing down</td>
<td>1905</td>
<td>21.50</td>
<td>3650 14.60</td>
<td>79.94</td>
</tr>
<tr>
<td>6</td>
<td>Clay pot – facing up</td>
<td>1755</td>
<td>24.20</td>
<td>3700 15.10</td>
<td>74.53</td>
</tr>
<tr>
<td>7</td>
<td>Waste cotton piece</td>
<td>1920</td>
<td>19.40</td>
<td>3805 15.40</td>
<td>85.75</td>
</tr>
<tr>
<td>8</td>
<td>Jute cloth</td>
<td>1775</td>
<td>25.10</td>
<td>3300 21.50</td>
<td>87.04</td>
</tr>
<tr>
<td>9</td>
<td>Black cotton cloth</td>
<td>1850</td>
<td>26.10</td>
<td>3510 25.80</td>
<td>89.73</td>
</tr>
<tr>
<td>10</td>
<td>Mild steel pieces</td>
<td>1940</td>
<td>31.40</td>
<td>3380 27.21</td>
<td>84.53</td>
</tr>
</tbody>
</table>

Figure 10: Single basin water solar-still [25]

Figure 11: Schematic of double-solar basin [25]
2.3.3.3. Water depth. Various researches investigated the effect of water depth on productivity. Ahsan et al. [27] and Zurigat et al. [28] evaluated water productivity with various depths of 1.5, 2.5 and 5 cm and concluded that water production in inversely proportional the depth of water. Similarly, Suneja et al. [29] investigated the effect of water depth on double basin solar still and it was also found that productivity increases with decreasing water depth. Tiwari et al. [30] concluded that at a minimum water depth, higher productivity is obtained due to an increase in water temperature and evaporation rate.

2.3.3.4. Supplementary reflectors. Another way to improve productivity is by using internal and external reflectors. Tanaka [31] modified the conventional still by using the internal and external reflectors as shown in Figures 12. The basin was overall made of plywood material. Both the side and backs walls were covered with the reflector, which was 1.8 mm thick and the mirror was made of stainless steel. It was found that using reflectors increased the daily productivity of the basin still about 70 to 100%. Khalifa and Ibrahim [32] investigated effects of incline angles of internal and external reflectors at angles 0° (vertical), 10°, 20° and 30° on the productivity. It was further observed that daily water production increased by the use of the reflectors at an optimum angle.

Figure 12: Solar still with internal and external reflector [31]

2.3.3.5. Basin dyes. Several researchers have attempted improving productivity by increasing water’s absorption rate in the basin. As still basin partially reflects solar radiation, the loss can be reduced by increasing the coefficient of absorption of water presents in the basin, by adding dye. Rajnavshi [33] evaluated the effect of adding dyes: green, red and black napthlamine. It was found that black
napthlamine was the most suitable because it increased evaporation dramatically due to increase in water temperature on the upper layer, which in turn increases the evaporation rate. It was concluded that approximately 29% of output distillate is increased with black dye. Similarly, Akash et al. [34] experimentally studied the effect of absorbing materials (black dye, ink, matt, none) with double slopes basin. From their experiments, using black dye gave the maximum water productivity as it absorbs maximum solar radiation.

Similarly, Nijmeh et al. [35] studied the effect of adding a- salts (potassium permanganate (KMnO₄) and potassium dichromate (K₂Cr₂O₇)), b- violet dye and c- charcoal on still efficiency. It was found that violet dye and potassium permanganate (KMnO₄) enhanced the still efficiency to maximum amount of 19.2% and 18.7%, respectively, as shown in Figure 13.

![Figure 13: Different absorption materials [35]](image)

2.3.3.6. Energy storage. Energy storage materials can be placed on the still to enhance productivity at night time when the solar energy is not available. Some of these storing materials include rubber, gravels, aluminum sheet, and glass. Sakthivel and Shanmugasundaram [36] studied single SS with black granite gravel as the storage medium. The gravel absorbs the excess energy during noon time and increases the temperature difference between the water and glass. It was found that the basin still produces an increased amount of water of 17 to 20% as compare to the conventional still without energy storage material. On the other hand, gravel is very cheap; hence the modification cost is almost the same.

The use of Phase Change Materials (PCM) in SS is used by El-sebaii et al. [37] who did an experimental and mathematical model for a single slope basin, (Figure 14). It was found that the productivity nearly increases the output by 85% on a given day of summer compared to the conventional solar still. Another important
material usage is of basin liner material. The basin liner materials absorb better solar radiation. Badran [38] found that the still productivity increased by 51% when asphalt basin liner and sprinkler were applied to the still.

Nafey et al. [39] investigated different energy storing materials such as black rubber sheet of 10 mm and black gravel of 20-30 mm in size, as shown in Figure 15. The black rubber sheet improved the productivity by 20% and black gravel did by 19%. Farshad and Askan [40] integrated the SS with built in sandy heat reservoir. This led to an increase in water glass temperature difference and an increase in the productivity by 10%.

2.3.4. Active solar stills. Several methods were incorporated for the system to act as active SS. Many researchers have tried different approaches to enhance productivity by integrating extreme solar heater with SS.

2.3.4.1. Solar collector. Raj and Tiwari [41] conducted an experiment on a solar still fabricated with flat plate under forced air condition, as shown in Figure 16. It was found that the average daily production was 24% higher than that of a simple basin solar still. Further, Raj et al. [42] added dye and black jute cloth to increase the absorptivity to the similar experiment. It was found that the output distillate increases further. Badran et al. [43] fabricated a single stage double sloped basin still with a flat plate absorber and found that an additional output distillate of 2.3 L/m²/day with an additional efficiency of 22.26% as compared to SS without flat plate.
2.3.4.2. Parabolic trough. Zenab and Ashraf [44] studied experimentally and theoretically single solar still integrated with parabolic trough and a heat exchanger. In their experiments, fresh water productivity increased by around 18% as compare to passive solar still.

In another study by Ahmed et al. [45], solar still was integrated with water sprinkles and cooling for the climate condition of Kuwait city. Three different experiments configuration were used: The first one was the passive conventional system, the second was the active system still integrated with water sprinkles and the third one was with cooling fan. Their results showed that increasing the fan air speed from 1.2 m/s to 3 m/s, increases the productivity form 8% to 15.5%. When using the sprinkers at intervals of specific time of 10 min and 20 min, the productivity was raised from 16% to 32%.

![Figure 16: Solar still fabricated with flat plate [41]](image)

2.3.4.3. PVT. Shiv Kumar and Arvind Tiwari [46] integrated single SS with hybrid Photo Voltaic Thermal energy (PV/T) system for the remote areas, as shown in Figure 17. Photovoltaic (PV) integrated with flat plate is used to generate electricity so that the water can be pumped from the storage system to the solar still. It was found that the active hybrid SS system gives better productivity than the passive during winter season. The maximum daily yield of water in passive was 2.26 L/day and when using active system, it was 7.22 L/day. On the other hand, Gajendra Singh et al. [47] evaluated the similar pattern experiment but with using double slope SS under natural and forced convection. It was found that the production of water increased 1.4 times than the single slope hybrid PVT active solar still.
2.4. Humidification-Dehumidification Process

Humidification-Dehumidification (HDH) process has been developed while trying to solve the major problem of solar still low productivity and to minimize heat of condensation energy loss. HDH process basically results in an increase in the overall efficiency of the distillation process with solar energy, as shown in Figure 18.

In HDH process, warm unsaturated air is in contact with water under the conditions provided, so that a desired level of humidity is reached, which is followed by removal of vapor from the humidified air that passes the cooling unit. The process is similar to the solar still, however, in the solar still the process of heat absorption, evaporation and condensation takes place in the same unit, whereas HDH operation is being engineered in a way that different components are needed as solar collector, humidifier tower and condenser. In the three parts, solar collector heats the air; in humidifier the intake air humidity is increased to near saturation, and when the air passes through the condensing unit, the vapor is stripped out to form product water.
Although the efficiency is generally increased, the capital cost of such process is also increased because of extra individual components. HDH techniques have the following advantage. The efficiency of energy utilization is increased due to the flexible control over the evaporation and condensation operation. On the other hand, both running and capital cost are increased due to using more components and the possibility of increased energy losses. Similar to SS, HDH has the following advantages:

- Very efficient method to produce water for families in remote areas.
- It is operated under low temperature and pressure, and therefore requires cheaper materials for fabrication.
- As air-water is indirect contact, heating surface is protected from corrosion or scale deposits.
- The process is also cheaper compared to other commercial desalination processes like MSF, RO or MED.

HDH systems are broadly categorized under three parts. The three broadly categorized based on the type of energy used, cycle configuration and type of heating systems [48].

1) Forms of energy such as solar, thermal, geothermal or hybrid systems.
2) Cycle configuration- There are two types of cyclic configurations; closer-air-open water (CAOW) and closed-water-open-air (CWOA) cycle. In CAOW cycle air is circulated in the closed loop for the humidification and dehumidification process by either natural or forced convection, whereas in CWOA air is circulated in the open loop cycle as shown in Figures 19 and 20 respectively.
3) Heated water or air is used as a type of heating systems

CAOW (Figure 19) is widely researched where the saline water is fed with through pre-heater, then to external solar heater while the air is heated by the latent heat of evaporation from the humidifier. Air flows at a rate proportional to the amount of evaporated water. The moist air streams flow to the dehumidifier where the vapor gathers on the cooler surface of the condenser. The energy is recovered by feed water in the dehumidifier. The air circulates in a closed loop where the pressure is maintained at atmospheric level.
Figure 20 shows CWOA where the feed water is preheated in the dehumidifier before it enters the solar collector to form the close loop. The air, when released from the dehumidifier, is not reverted back; hence this form is open air loop process. One of the HDH drawbacks is that the efficiency depends upon humidification process i.e. if the humidification process is not efficient enough then the coolant water temperature that is entered to the dehumidifier increases. Hence, it affects the dehumidification process of the moist air, affecting fresh water production.

Since 1990, a lot of work has been done in regards to the improvement of the HDH system, from conventional single effect HDH system to multi-effect, as well as the integration of HDH techniques with solar still.

Müller-Holstet [49] patented a small scale seawater desalination system (0.5–2 m³/day). The system claimed to operate as standalone and maintenance free and can be performed without electricity. The system developed, shown in Figure 18, can be run by either solar energy or waste energy from diesel. Hallaj et al. [50] studied experimentally solar desalination HDH configuration where air circulated, heated and humidified by hot water from flat plate solar collector. Two different units were construed of different sizes but of same materials. It was concluded that the
productivity of these units was found to be much higher than those of the single SS. Zhani and Ben Bacha [51] carried out a similar investigation using better design solar HDH system, which consists of flat plate solar air collector, flat plate solar water collector and humidification/dehumidification setup. They found that fresh water production increases as the solar intensity increases and it gives greater production of fresh water than the conventional SS.

Fath and Ghazy [52] investigated performance of simple solar desalination using HDH process that consists of solar air heater, humidifier and dehumidifier, as shown in Figure 21. They concluded that the productivity increases when the solar intensity increases and wind velocity decreases. The feed flow rate has no effect of the productivity; and moreover, dehumidifier effectiveness has no contribution to the productivity, which can be very vital in terms of economical design of the system.

Fath et al. [53] carried out a numerical study to investigate the transient thermal performance of a HDH solar still as shown in Figure 22. The still consists of several basins carrying the saline water. The still was divided into two chambers—evaporation chambers, where the air circulated is heated and the evaporated water humidified and condensation chambers, where the air carrying vapor is cooled and dehumidified for fresh water formation. It was concluded that water production increased with increasing solar intensity, ambient temperature, and initial saline water temperature and basin absorptivity. The still productivity was 5.2 L/m²/day, which is quite higher as compared to the conventional solar still basin. Furthermore, it was noticed that the water production has no significant effect with the change in evaporation and condensation area, increasing wind velocity, condenser emissivity and changes in saline water mass. In addition to this, Fath et al. [54] studied natural air circulation for the same HDH distiller. The output productivity was similar to the one found in the previous study [60]. It was further found that using lower air flow
rate has no effect on productivity and efficiency of the system. Natural circulation is more desirable as it minimizes the operational and capital cost.

Figure 22: Schematic of HDH [54]

Efat Chafik [55] patented a complete research and development (R&D) project that can aid as a new process for seawater desalination. The main feature developed in his research was that heating and humidification process can be performed in several stages or multi effect. This will allow air to load with high amount of humidity and reduce the free circulation of air around the system. Hence, the effect of multi-effect came into consideration as a source of better water production.

Zamen et al. [56] evaluated experimentally for two stage multi-effect HDH to improve the fresh water productivity as shown in Figure 23. The collector area was 80 m² for the two-stage process. For the experimental two stage HDH effect, it was found that the fresh water production can reach 7.25 (L/m²/day) which is about 40% more than the single stage effect. The cumulative productivity can be at least increased by 20% as compared to single stage. As per the effect of stages on the daily productivity, there is hardly any increment in productivity of stages 3 and 4 (1% and 4%) respectively, as shown in Figure 24.

Figure 23: Three stage multi-effect humidification [56]
Adewale Giwa et al. [57] studied solar HDH desalination technology with the use of photovoltaic (PV) for the production of fresh water and improvement of PVT electric power generation efficiency. They found that small scale PVT-HDH desalination process resulted in water production in the range of 0-0.528 L/m² and there was 83.6% decrease in environmental impacts when compared with photovoltaic-reverse osmosis (PV-RO) system as shown in Table 3.

Yuan et al. [58] experimentally investigated solar HDH system to produce a capacity of 1000 L/day. The system consists of solar air heater of 100 m², solar water collector of 12 m² and humidifier and dehumidifier unit along with pre and post treatment system. They managed to successfully obtain a production of 1200 L/day, having the average intensity of solar radiation of 550 W/m².

Cemil Yamal and Ismail Solmus [59] investigated a double-pass flat plate along with two glass covers. The results of the experimental study showed that under certain operating conditions, the system productivity decreases about 15% if double-pass solar air heater is not used. In addition, productivity of the system increases with increasing the feed water mass flow rate and quantity of water inside the storage tank. However, productivity of the system remains approximately the same when the air
mass flow rate is increased. Moreover, increasing the cooling water mass flow rate results in the improvement of the productivity of the system investigated.

Similarly, Hamed et al. [60] also studied mathematically and experimentally a solar HDH desalination unit that consists of solar water heater tube collector with a humidifier and dehumidifier. It was found that when using HDH system in a typical desalination system of indirect method (solar water heater tube collector), the average productivity increases to 11 (L/m²/day). Also, the unit productivity increases as water temperature entering the humidifier increases.

2.4.1. Integrated SS-HDH. Koning et al. [61] developed a solar still with HDH integration. The evaporation and condensation chambers were separated and the collector surface was expanded to form a cube instead of a flat surface. Additional mirrors were added to increase the solar energy input and for wind protection. This model, later popularized as Aqua Solaris, on the Island of Bonaire is guaranteed to right now create 40 liters of freshwater for each square meter surface zone at a surrounding temperature of 30 °C for an expected existence of 20 years. Every unit can supply the freshwater needs of an extensive family at low expenses, and the water source can be anything from ocean water to water dirtied with substantial metals and mineral toxic substance.

Fath et al. [62] investigated numerically a novel standalone direct solar distillation system that integrates SS with HDH. In their study, both forced and natural air circulation was considered under the typical weather of Abu Dhabi, UAE. The system product was around 10 L/m²/day which is almost double the amount from the conventional still. The authors concluded that natural air circulation can replace forced circulation which can reduce the capital as well as the operating cost. An extension of Fath et al. [62] is, therefore, needed to enhance the HDH air circulation by adding thermal chimney with built-in finned solar absorber for thermal energy generation.
Chapter 3: System Description

3.1. Introduction

The proposed SS-HDH-FINNED SOLAR ABSORBER system will be described in this chapter. The modeling will be a follow up of the work carried out in [62-63]. A numerical code will be developed based on the analysis in MATLAB in order to simulate the integrated system and study the effect of various environmental, design and operational parameters on the unit’s productivity and efficiency. The integrated distillation system will be studied under force convection during the typical climate of UAE.

3.2. Integrated System Description

Figure 25 shows the real dimensions model of SS-HDH-Solar absorber system.

![Figure 25: Proposed system (units in mm)](image)
Figure 25 illustrates the basic configuration of the proposed solar desalination system that consists of single slopped solar still with a built in two effects HDH system which acts as both dehumidifier and energy storage system. Solar finned absorber is added into the unit to further enhance temperature and better heat transfer as convection. Both, thermal energy generated from absorber and solar stills heat of condensation are recovered to drive the HDH circulating air and as a result, enhance the unit performance and productivity by evaporating more water in both the stills and the HDH system.

Solar energy enters the system through the glass cover (2), (3) absorbed by the saline water in the basin (4). Heated saline water partially evaporates, and vapor condenses on the glass cover (3) and the condensate (D2) represents the SS production (PW 1). Condensing energy is also recovered by air in the channel (1) - (2) and circulated upward to the first humidifier (4) where it is partially cooled and humidified. Air continues to flow upward along the thermal chimney, cool the solar finned absorber. The air (5) is heated and flows over the main humidifier 2 (6) to further carry more humidity. A secondary dehumidifier (7) is added to ensure humidity condensation and recovery.

- The humidifiers are a set of flexible cotton rope of large surface area to ensure humidification and rate air circulate.
- The single slope SS vertical wall is a wick absorber to further preheat the feed flow into the SS.

3.3. System Modeling Assumptions

- Each sub-component is taken as lumped.
- The HDH system operation is initially studied under forced air circulation.
- All vapor losses through the joints are neglected.
- Solar, wind and ambient conditions of Abu Dhabi city are used.

3.4. Governing Equation

Transient mass and energy balance for each sub components will be considered as follows.

3.4.1. Winter condition. Solar intensity, ambient temperature and wind velocity of winter condition were obtained using Eq. 1, 2 and 3 [63].

\[
I_{sun} = 1.228 \times 10^{-23} t^6 - 1.483 \times 10^{-18} t^5 + 6.463 \times 10^{-14} t^4 - 1.206 \times 10^{-9} t^3 + 6.692 \times 10^{-6} t^2 + 0.06574 \times t - 38.29
\] (1)
\[ T_{amb} = 2.90072 \times 10^{-27} t^6 - 8.044511 \times 10^{-22} t^5 + 8.335309 \times 10^{-17} t^4 - \\
3.782771 \times 10^{-12} t^3 + 5.694766 \times 10^{-8} t^2 + 4.347677 \times 10^{-4} t + \\
11.54281 \]  
(2)

\[ V_{wind} = 2.108465 \times 10^{-28} t^6 - 1.191010 \times 10^{-22} t^5 + 1.966906 \times \\
10^{-17} t^4 - 1.323923 \times 10^{-12} t^3 + 3.477846 \times 10^{-8} t^2 - \\
1.945428 \times 10^{-4} t + 1.307941 \]  
(3)

3.4.2. Summer condition. Solar intensity, ambient temperature and wind velocity of winter condition were obtained using Eq. 4, 5 and 6 [63]. The relative humidity Eq. 7 obtained from the data in Microsoft Excel’s polynomial curve fit.

\[ I_{sun} = 2.04407 \times 10^{-15} t^4 - 2.115872 \times 10^{-10} t^3 + \\
5.503613 \times 10^{-6} t^2 - 2.001969 \times 10^{-3} t - \\
3.519359 \]  
(4)

\[ T_{amb} = 2.384193 \times 10^{-27} t^6 - 7.363871 \times 10^{-22} t^5 + 8.597613 \times \\
10^{-17} t^4 - 4.556763 \times 10^{-12} t^3 + 9.619542 \times 10^{-8} t^2 - \\
1.930423 \times 10^{-4} t + 30.65434 \]  
(5)

\[ V_{wind} = -2.546024 \times 10^{-27} t^6 + 6.52121 \times 10^{-22} t^5 - 6.210357 \times \\
10^{-17} t^4 + 2.724245 \times 10^{-12} t^3 - 5.79122 \times 10^{-8} t^2 + \\
6.441281 \times 10^{-4} t + 0.5685121 \]  
(6)

\[ RH = -1.82046 \times 10^{-22} t^5 + 2.62677 \times 10^{-17} t^4 - 5.57872 \times 10^{-13} t^3 + \\
3.64342 \times 10^{-8} t^2 + 5.43926 \times 10^{-4} t + 0.7537008529 \]  
(7)

3.4.3. Still basin. The basin transient temperature and energy balance for the still basin is given below, Figure 26. The transient temperature of water basin is given by Eq. 8.

\[ \frac{dT_{water \ basin}}{dt} = \sum Q_{basin} / (M_{basin} C_{p_{basin}}) \]  
(8)

where,

\[ \sum Q_{basin} = Q_{abs \ basin} + Q_{ref \ basin} + Q_{fw} - Q_{rad \ water\_g3} - Q_{conv \ water\_g3} - Q_{eva \ water\_g3}, \]  
(9)

\[ Q_{abs \ basin} = I A_{basin} \alpha_{basin} \tau_g^2, \]  
(10)

\[ Q_{ref \ basin} = I \cos(90 - \theta) \ r A_{ref} \tau_g^2 \alpha_{basin}, \]  
(11)

\[ Q_{fw} = M_{evap \ water} * C_{p_w} * T_{hum1}, \]  
(12)

\[ Q_{rad \ water\_g3} = \sigma \left( \frac{1}{E_{R1}} \right) \left[ (T_{water \ basin} + 273)^4 - (T_{g3} + 273)^4 \right], \]  
(13)

where,

\[ E_{R1} = (1 - \varepsilon_{basin})/(\varepsilon_{basin} A_{water \ basin}) + (1/F_{12}) + (1 - \varepsilon_g)/(\varepsilon_g A_{g3}), \]  
41
\( F_{12} = \text{shape factor} \)

\[ Q_{\text{conv\,water\,.g3}} = H_{\text{conv\,water\,.g3}}A_{\text{water\,basin}}(T_{\text{water\,basin}} - T_{g3}) , \quad (14) \]

where,

\[ H_{\text{conv\,water\,.g3}} = 0.884 \times \left( \frac{(T_{\text{water\,basin}} - T_{g3}) + (P_{\text{water}} - P_{g3}) \times \left( \frac{T_{\text{water\,basin}} + 273}{268900 - P_{\text{water}}} \right)^{0.33}}{0.33} \right) \]

\[ Q_{\text{evap\,water\,.g3}} = H_{\text{evap}}A_{\text{water}}(P_{\text{water}} - P_{g3}) \quad (15) \]

where,

\[ H_{\text{evap}} = 0.016 \times H_{\text{conv\,water\,.g3}} \]

\[ P_{\text{water}} = 1000 \times \left( 0.14862 \times T_{\text{water\,basin}} - 0.0036526 \times T_{\text{water\,basin}}^2 \right. \]
\[ + \left. 0.0001124 \times T_{\text{water\,basin}}^3 \right) \]

\[ P_{g3} = 1000 \times \left( 0.14862 \times T_{g3} - 0.0036526 \times T_{g3}^2 + 0.0001124 \times T_{g3}^3 \right) \]

and,

\[ M_{\text{evap\,water}} = \frac{Q_{\text{evap\,water}}}{h_f g} \quad (16) \]

---

**3.4.4. System glass covers 1.** The temperature and energy balance on the still glass cover \( g1 \) is calculated using the energy balance equations stated below, Figure 27. The transient temperature of glass cover 1 is given by Eq. 17.

\[ \frac{dT_{g1}}{dt} = \frac{\sum Q_{g1}}{M_{g1}C_{pg}} \quad (17) \]

where,

\[ \sum Q_{g1} = Q_{abs\,g1} + Q_{rad\,g3\,g1} + Q_{conv\,air\,g1} - Q_{conv\,g1\,amb} - Q_{rad\,g1\,sky} \]

\[ Q_{abs\,g1} = I A_{g1} \alpha_g \]

\[ Q_{conv\,g1\,amb} = H_{amb}A_{g1}(T_{g1} - T_{amb}) \]

---
\[ H_{amb} = 0.664 \times k \times v_{wind} \times Pr\ amb^{1/3} / \sqrt{\mu / \rho \times \sqrt{L_{g1}}} \],

\[ Q_{rad\ g1,\ sky} = \sigma \varepsilon g A_{g1} \left( (T_{g1} + 273)^4 - (T_{sky} + 273)^4 \right), \]

\[ Q_{rad\ g3,\ g1} = \sigma \left( \frac{1}{ER2} \right) \left( (T_{g3} + 273)^4 - (T_{g1} + 273)^4 \right), \]

and,

\[ Q_{conv\ air,\ g1} = H_{conv\ air,\ g1} A_{g1} (T_{air} - T_{g1}), \]

where,

\[ T_{air} = (T_{a1} + T_{a2}) / 2, \]

\[ T_{a2} \text{ is defined in the Air 1-2 section} \]

\[ H_{conv\ air,\ g1} = Nu \frac{k}{D_h}, \]

\[ Re = \frac{(m_{air} \times D_h)}{(b_g \times W \times u)}, \]

Where,

\[ D_h = \text{hydraulic diameter between glass 1 and 3} \]

\[ b_g = \text{space between glass 1 and 3} \]

\[ W = \text{Width of the system} \]

\[ \mu = \text{dynamic viscosity} \]

For Laminar Flow Nusselt Number:

If Reynold number less than 2800

\[ Nu = 7.54 + (0.03 \times D_h Re Pr / L_{g3}) / \left( 1 + 0.016 \left( D_h Re Pr / L_{g3} \right)^{2/3} \right) \]

else

For Turbulent Flow Nusselt Number:

friction factor \( f = (0.79 \times ln Re - 1.64)^{-2} \)

\[ Nu = (f / 8 \times (Re - 1000) \times Pr) / (1 + 12.7 \times (Pr^{2/3} - 1) \times \sqrt{f / 8}) \]

Figure 27: Energy balance between glass 1 and 3
3.4.5. Still glass covers 3. The temperature and energy balance on the still glass cover g3 is calculated using the energy balance equations stated below, Figure 28. The transient temperature of glass cover 3 is given by Eq. 24.

\[
dT_{g3} / dt = \sum Q_{g3} / (M_{g3} C_{pg})
\]  

(24)

where,

\[
\sum Q_{g3} = Q_{abs g3} + Q_{rad water g3} + Q_{conv water g3} + Q_{evap water g3} - Q_{conv g3 air - Q_{rad g3 g1} - Q_{dist}},
\]  

(25)

\[
Q_{g3} = I A_{g3} \alpha_{g} \tau_{g3},
\]  

(26)

\[
Q_{conv g3 air} = H_{conv air} A_{g3} (T_{g3} - T_{air}),
\]  

(27)

\[
Q_{rad g3 g1} = \sigma \left( \frac{1}{ER^2} \right) \left( (T_{g3} + 273)^4 - (T_{g1} + 273)^4 \right),
\]  

(28)

where,

\[
ER^2 = \left( \frac{2}{\varepsilon_g} - 1 \right);
\]

and,

\[
Q_{dist} = M_{evap water} C_{pw} T_{g3}
\]  

(29)

\(Q_{rad water g3}, Q_{conv water g3}\) and \(Q_{evap water g3}\) are given in the energy balance of transient temperature in Eq. 15.

**Figure 28: Energy balance of glass 3**

3.4.6. Air 1-2. The humid air flow energy between glass 1 and 3 and the temperature balance is given by Eq. 30 and 31.

\[
h_{a2} = h_{a1} + \left( Q_{conv g3 air - Q_{conv air g1}} \right) / m_{air},
\]  

(30)

\[
w_1 = w_2,
\]
and
\[ T_{a2} = T_{a1} + \left( Q_{\text{conv}g3,\text{air}} - Q_{\text{convair},g1} \right) / \left( m_{\text{air}} C_{p\text{air}1} \right) \] (31)
where,
\[ C_{p\text{air}1} = C_{p\text{dryair}} + C_{p\text{water vapor}w1} \]

3.4.7. Chimney humidifier-absorber glass covers 2. The transient temperature and energy balance of g2 can be calculated using the energy balance equations stated below, Figure 29. The transient temperature of glass cover 1 is given by Eq. 32.
\[ \frac{dT_{g2}}{dt} = \sum Q_{g2} / (M_{g2} C_{pg}), \] (32)
where,
\[ \sum Q_{g2} = Q_{\text{abs} g2} + Q_{\text{rad} \text{hum} g2} + Q_{\text{convair} g2} + Q_{\text{rad} \text{pvt} g2} + Q_{\text{convpvtair} g2} - Q_{\text{conv} g2,\text{amb}} - Q_{\text{rad} g2,\text{sky}}, \] (33)
\[ Q_{\text{abs} g2} = I \cos(90 - \theta) A_{g2} \alpha_g, \] (34)
\[ Q_{\text{conv} g2,\text{amb}} = H_{\text{amb}}A_{g2}(T_{g2} - T_{\text{amb}}), \] (35)
\[ Q_{\text{rad} g2,\text{sky}} = \sigma \varepsilon_g A_{g2} \left( (T_{g2} + 273)^4 - (T_{\text{sky}} + 273)^4 \right), \] (36)
\[ Q_{\text{rad} \text{hum} g2} = \sigma \left( \frac{1}{ER3} \right) \left( (T_{\text{hum}} + 273)^4 - (T_{g2} + 273)^4 \right), \]
where,
\[ ER3 = (1 - \varepsilon_{\text{hum}})(\varepsilon_{\text{hum}}A_{\text{hum}}) + (1/F_{21}) + (1 - \varepsilon_g)(\varepsilon_gA_{\text{hum}g2}), \]
\[ F_{21} = \text{shape factor} \]
\[ A_{\text{hum}} = \text{length of rope} \ast \text{width} = 0.5 \ast A_{g2}, \]
\[ A_{\text{hum}g2} = A_{\text{hum}} \]
\[ Q_{\text{rad} \text{pvt} g2} = \sigma \left( \frac{1}{ER4} \right) \left( (T_{\text{pvt}} + 273)^4 - (T_{g2} + 273)^4 \right), \] (37)
where,
\[ ER4 = (1 - \varepsilon_{\text{abs}})(\varepsilon_{\text{abs}}A_{\text{absorber}}) + (1/F_{22}) + (1 - \varepsilon_g)(\varepsilon_gA_{\text{abs}g2}) \]
\[ F_{22} = \text{shape factor} \]
\[ A_{\text{absorber}} = \text{length of absorber} \ast \text{width of absorber} \]
\[ A_{\text{abs}g2} = A_{\text{absorber}} \]
\[ T_{ah} = (T_{a2} + T_{a4})/2 \]
\[ T_{ah} \text{ is the average air between point 2 and 4 of humidifier 1} \]
\[ T_{a4} \text{ is given in the Air 2-4 section} \]
\[ T_{ap} = (T_{a4} + T_{a5})/2 \]
$T_{ap}$ is the average air between point 4 and 5 of absorber plate $T_{a5}$ is given in the Air 5-6 section

$$Q_{\text{conv air}g2} = H_{\text{conv air}g2}(A_{\text{hum}})(T_{ah} - T_{g2}), \quad (38)$$

and,

$$Q_{\text{conv pva}irg2} = H_{\text{conv air}g2}(A_{\text{absorber}})(T_{ap} - T_{g2}), \quad (39)$$

where,

$$H_{\text{conv air}g2} = Nu \frac{k}{D_h}$$

$$Re = \frac{(m_{air} * D_{h2})}{(Area * u)}$$

$D_{h2} = \text{hydraulic diameter between glass 2 and humidifier 1}$

$Area = (b_h * L) - A_{rope}$

$L = \text{length of the system}$

$\mu = \text{dynamic viscosity}$

For Laminar Flow Nusselt Number:

If Reynold number less than 2800

$$Nu = 7.54 + (0.03 D_h Re Pr / L_{g3}) \left(1 + 0.016 (D_h Re Pr / L_{g3})^{2/3}\right)$$

else

For Turbulent Flow Nusselt Number:

friction factor $f = \left((0.79 * ln Re) - 1.64\right)^{-2}$

$$Nu = \left(f / 8 * (Re - 1000) * Pr\right) / \left(1 + 12.7 * (Pr^{2/3} - 1) * \sqrt{f / 8}\right)$$
3.4.8. Humidifier 1. The humidifier (1) accounts for part of the distillate formed outside the basins are shown in Figure 30. The transient temperature of humidifier 1 is given by Eq. 40.

$$\frac{dT_{\text{hum}}}{dt} = \sum Q_{\text{hum}}/(M_{\text{hum}} C_{p\text{hum}}) \quad (40)$$

where,

$$\sum Q_{\text{hum}} = Q_{\text{abs hum}} + Q_{\text{feed}} - Q_{\text{rad hum.g2}} - Q_{\text{conv hum.air}} - Q_{\text{evap hum.air}}, \quad (41)$$

$$Q_{\text{abs hum}} = I \cos(\pi(90 - \theta)/180)(A_{\text{hum}} \sigma_{\text{hum}} \tau_{g}), \quad (42)$$

$$Q_{\text{conv hum.air}} = H_{\text{conv air.g2}} A_{\text{hum}} (T_{\text{hum}} - T_{\text{air}}), \quad (43)$$

$$T_{\text{air}} = (T_{a4} + T_{a2})/2$$

$$Q_{\text{rad hum.g2}} = \sigma \left( \frac{1}{E_R^3} \right) \left( (T_{\text{hum}} + 273)^4 - (T_{g2} + 273)^4 \right), \quad (44)$$

$$Q_{\text{evap hum.air}} = M_{\text{evap}} h_{fg,hum}$$

$$M_{\text{evap}} = H_{\text{mass}} * A_{\text{hum}} * (\rho_{\text{saturated steam}} - \rho_{\text{humidity}})$$

$$RH = \text{Relative humidity of air}$$

$$H_{\text{mass}} = Sh * D/L_{\text{hum}}$$

Diffusivity, $D = 1.87e - 10 \times (T_{avg} + 273.15)^{2.072} \times F$

Schmidt number $Sc = v / D$

Average temperature in the humidifier $T_{avg} = (T_{\text{hum}} + T_{\text{air}})/2$

Laminar Flow:

If Reynold number is less than 2800

$$Nu = 7.54 + (0.03 * D_{\text{hum}} * Re \times Pr/L_{g2})/(1 + 0.016 \times ((D_{\text{hum}} * Re * Pr/L_{g2})^{2/3}))$$

Sherwood number, $Sh$

$$Sh = 7.54 + (0.03 * D_{\text{hum}} * Re \times Sc/W_{g2})/(1 + 0.016 \times (D_{\text{hum}} * Re * Sc/L_{g2})^{2/3}))$$

Turbulent Flow:

$$f = (0.79 \times \log(Re) - 1.64)^2$$

$$Nu = (f/8 \times (Re - 1000) \times Pr)/(1 + 12.7 \times (Pr^{2/3} - 1) \times \sqrt{f/8})$$

$$Sh = (f/8 \times (Re - 1000) \times Sc)/(1 + 12.7 \times (Sc^{2/3} - 1) \times \sqrt{f/8})$$

and,

$$Q_{\text{feed}} = M_{\text{evap}} * T_{\text{amb}} * C_{p\text{w}} \quad (46)$$
3.4.9. Air 2-4. The humid air flow energy, temperature balance and the specific humidity at the exit of humidifier 1 are given by Eq. 47, 48 and 49.

\[ h_{a4} = h_{a2} + \left( Q_{\text{conv hum air}} + Q_{\text{evap hum air}} - Q_{\text{conv air g2}} \right) / m_{\text{air}}, \]  
\[ T_{a4} = \left( h_{a4} - 250 \cdot 1000 \cdot w_4 \right) / C_{p_{\text{air4}}}, \]  
or  
\[ T_{a4} = \text{from saturated moist air table} \]
\[ C_{p_{\text{air4}}} = C_{p_{\text{dry air}}} + C_{p_{\text{water vapor}}} w_4 \]
and,  
\[ w_4 = w_1 + M_{\text{evap}} / m_{\text{air}} \]  

3.4.10. Solar absorber. The energy balance equation in solar absorber is given below in Figure 31. The transient temperature of solar absorber is given by Eq. 50.

\[ \frac{dT_{\text{abs}}}{dt} = \sum Q_{\text{abs}} / (M_{\text{abs}} C_{p_{\text{abs}}}) \]  

where,  
\[ \sum Q_{\text{solarabs}} = Q_{\text{abs solar}} - Q_{\text{radabs g2}} - Q_{\text{convabs air}}, \]  
\[ Q_{\text{abs solar}} = I \cos(\pi (90 - \theta) / 180) A_{\text{abs}} \alpha_{\text{abs}} \tau_g, \]  
\[ Q_{\text{convabs air}} = H_{\text{conv air g2}} (A_{\text{abs}}) \ast (T_{\text{abs}} - T_{\text{air}}), \]  
\[ T_{\text{air}} = (T_{a4} + T_{a5}) / 2 \]  
and,
\[ Q_{\text{rad} \, \text{abs} \, g2} = \sigma \varepsilon_{\text{abs}} A_{\text{abs}} \left( (T_{\text{abs}} + 273)^4 - (T_{g2} + 273)^4 \right), \]  

(53)

### 3.4.11 Air 4-5

The humid air flow energy, temperature balance and the specific humidity at the exit of absorber are given by Eq. 54 and 55.

\[ h_{a5} = h_{a4} + \left( Q_{\text{conv} \, \text{abs} \, \text{air}} - Q_{\text{conv} \, \text{abs} \, \text{air} \, g2} \right)/m_{\text{air}}, \]  

(54)

\[ w_5 = w_4, \]  

and,

\[ T_{a5} = T_{a4} + \left( Q_{\text{conv} \, \text{abs} \, \text{air}} - Q_{\text{conv} \, \text{abs} \, \text{air} \, g2} \right)/C_{p\text{air}5} \times m_{\text{air}} \]  

(55)

where,

\[ C_{p\text{air}5} = C_{p\text{dry} \, \text{air}} + C_{p\text{water} \, \text{vapor}} \times w_5 \]

### 3.4.12 Humidifier 2

The energy balance equation in humidifier 2, which corresponds to additional water production is given below, Figure 32. The transient temperature of humidifier 2 is given by Eq. 56.

\[ \frac{dT_{\text{hum}1}}{dt} = \Sigma Q_{\text{hum}1} / (M_{\text{hum}1} C_{p\text{hum}1}) \]  

(56)

where,

\[ \Sigma Q_{\text{hum}1} = Q_{\text{feed}1} + Q_{\text{conv} \, \text{abs} \, \text{air}} - Q_{\text{conv} \, \text{hum}1 \, \text{air}} - Q_{\text{evap} \, \text{hum}1 \, \text{air}} \]  

(57)

\[ Q_{\text{abs} \, \text{hum}} = I \cos(\pi(90 - \theta)/180) A_{\text{hum}} \alpha_{\text{hum}} \tau_g, \]  

(58)

\[ Q_{\text{conv} \, \text{hum}1 \, \text{air}} = H_{\text{conv} \, \text{air} \, g2} A_{\text{hum}1} (T_{\text{hum}1} - T_{\text{air}}), \]  

(59)

\[ T_{\text{air}} = (T_{a5} + T_{a6})/2, \]  

and,

\[ Q_{\text{conv} \, \text{abs} \, \text{air}} = H_{\text{conv} \, \text{air} \, g2} A_{\text{abs}} (T_{\text{abs}} - T_{\text{air}}), \]  

(60)
### 3.4.13 Air 5-6.
The humid air flow energy, temperature balance and the specific humidity at the exit of humidifier 2 are given by Eq. 61, 62 and 63.

\[
h_{a6} = h_{a5} + \left( Q_{\text{conv hum1_air}} + Q_{\text{evap hum1_air}} - Q_{\text{conv abs_air}} \right) / m_{\text{air}}, \quad (61)\]

\[
T_{a6} = \left( h_{a6} - 2501000 \ w_4 \right) / C_{p_{\text{air6}}}, \quad (62)
\]

where,

\[
C_{p_{\text{air6}}} = C_{\text{dry air}} + C_{\text{water vapor}} \ w_6
\]

and,

\[
w_6 = w_5 + M_{\text{evap}} / m_{\text{air}} \quad (63)
\]

### 3.4.14 Dehumidifier.
The energy balance equation in dehumidifier 2 is given below, Figure 33. The transient temperature of dehumidifier is given by Eq. 64.

\[
T_{\text{dehum}} = (T_{\text{in}} + T_{\text{out}}) / 2 \quad (64)
\]

Where,

\[
T_{\text{in}} = T_{\text{amb}},
\]

\[
T_{\text{out}} = T_{\text{in}} + \Delta T_{\text{dif}}
\]

\[
T_{a7} = T_{\text{in}} + 0.5,
\]

\[
\Delta T_{\text{dif}} = 1^\circ
\]

\[
Q_{\text{cond air dehum}} = (h_{a4} - h_{a5}) / m_{\text{air}} \quad (66)
\]

\[
M_{\text{cond dehum}} = Q_{\text{cond dehum}} / h_{fg}
\]

And,

\[
m_{\text{water}} = Q_{\text{cond dehum}} / ( (T_{\text{out}} - T_{\text{in}}) \ * C_{p_{\text{w}}}) \quad (67)
\]
3.4.15 Air 6-7. The humid air flow energy and the specific humidity at the exit of dehumidifier are given by Eq. 68 and 69.

\[
\begin{align*}
    h_{a7} &= h_a7 - Q_{\text{cond air,dehum}} / m_{\text{air}} \\
    w_5 &= w_6 - M_{\text{cond air,dehum}} / m_{\text{air}}
\end{align*}
\]
Chapter 4: Results and Discussion

4.1. Introduction

This chapter presents the results of the SS-HDH solar distillation system performance and productivity, in addition to the temperature distribution and energy balances for various components. Furthermore, the validation of SS-HDH performance results is done with a published work to ensure that the results obtained are in precise manner. The main source of energy in our system is the incident solar energy. The sun intensity (or irradiation) is absorbed first by the main systems based on their absorptivity and then is transferred and lost through various heat transfer processes such as evaporation, convection, condensation and radiation. In the studied system, there are four main absorbers that influenced the unit performance; still basins, solar fins absorbers and humidifier 1 and 2.

Firstly, the still basin gets heated up as the transmitted solar energy is absorbed due to its high absorptivity which eventually heats up the basin saline water. In this, the heated water evaporates due to the temperature and pressure gradient between the air and water, and then partly condenses as distillate on the cooler side of the glass cover glass 3 (g3) and loses the latent heat of condensation. This increases the temperature of glass 3 and convects part of its energy to the air flowing in duct (g3-g1). The moving air transfer the part of its energy to the glass 1, which eventually loses energy to the ambient and by convection and radiation.

Similarly, solar intensity heats up the humidifier compartment, which is sprayed with saline water droplets. The humidifier consist of a set of ropes gets heated up and evaporate humidifier water to the flowing air across it. Humidifier also radiates to the glass 2 and convects part of the energy to the hot flowing air. The circulate air gains humidity and heats up the glass (g2) by convection. Similarly, like g1, g2 loses energy to the ambient.

Similarly, absorption of solar irradiation heats up the finned solar absorber as well. The humidified air passes the finned solar absorber. The absorber gets heated up and increases the air flowing temperature as it convects part of the energy to the flowing air and to the glass 2. The glass 2 loses this energy to the ambient as with case of humidifier 1. The finned surface increases the overall surface area for the heat transfer to take place by convection. Eventually, this reduces the temperature of the plate as well. The absorber also loses some of the energy by radiation to the glass 2.
The moving air transfers some of the energy to the humidifier 2, which is situated in the SS-HDH back side. Hence, the form of input energy is from the convection from moving air and input feed from the spraying water. The humidifier 2 loses part of the energy as convection and part by evaporation.

4.2. Assumptions

Figures 34 to 39 show typical environmental data obtained from Masdar, Abu Dhabi [62]. The reference equations for the figures were obtained using polynomial curve fitting in Microsoft excels and plotted in MATLAB as input parameters to study the model under UAE climatic conditions.

4.2.1. Winter condition. Solar intensity, ambient temperatures and wind velocity is obtained as the environmental data and is plotted on y-axis respectively. The x-axis of the figures represents the time and for winter the starting time is at zero which represents the sunrise at 7am. The mid-day for winter is at 12pm, which is at 5th of the hour and the sunset at 6pm, which is at 11 hours. The time axis is further extended beyond sunset to observe the effect of stored energy of the system per day i.e. till 24 hours.

Figure 34: Solar intensity during winter in UAE
Figure 35: Ambient temperature during winter in UAE

Figure 36: Wind velocity during winter in UAE
4.2.2. **Summer condition.** For summer the starting time at zero represents the sunrise at 5:30 am. The mid-day for summer is at 12:30 pm, which is at 7\textsuperscript{th} of the hour and the sunset at 7:30 pm, which is at around 13.6 hours. The time axis is also further extended beyond sunset to observe the effect of stored energy of the system per day i.e. till 24 hours.

![Figure 37: Solar intensity during summer in UAE](image1)

![Figure 38: Ambient temperature during summer in UAE](image2)
Table 4 shows the reference parameters for the present analysis. Rest of the detailed parameters is given in the appendix.

**Table 4: Modeling assumptions**

<table>
<thead>
<tr>
<th>REFERENCE CASE</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Difference between condenser Inlet (T_in) and outlet temperature (T_out)</td>
<td>1 °C</td>
</tr>
<tr>
<td>2. Mass flow rate of circulating air</td>
<td>0.5 kg/s</td>
</tr>
<tr>
<td>3. Temperature of air at dehumidifier exit T_a7</td>
<td>T_amb + 0.5 °C</td>
</tr>
<tr>
<td>4. Temperature of dehumidifier (tubes) T_dehum</td>
<td>(T_in + T_out)/2</td>
</tr>
<tr>
<td>5. Basin absorptivity</td>
<td>0.9</td>
</tr>
<tr>
<td>6. Basin area</td>
<td>6.0 m²</td>
</tr>
<tr>
<td>7. Glass transmissivity</td>
<td>0.95</td>
</tr>
<tr>
<td>8. Water height in basin</td>
<td>3 cm</td>
</tr>
</tbody>
</table>
4.3. Validation of the Results

The following results are the comparison with the published results as a validation of the modeling results of SS-HDH based. Design and operations conditions are calculated, and run by the analysis to get the results to ensure that the present methods of program are in the right direction.

Figures (40-46) show (a) temperatures, energy and total production graph over various components in SS-HDH published work [66] and (b) are based on the analysis performed for validation purpose. The validation was performed for the summer conditions. The temperatures obtained were almost identical based on the data gathered. However, a slight deviation will always be there as knowing detailed design and operational condition of published results were not fully possible. Furthermore, the energy balance for various components such as glass cover 2, 3 and basin were identical. In addition to this, specific humidity and the overall productivity were in close range as well. The result helps us to carry the design and analysis of the proposed two stage SS-HDH-absorber system.
Figure 41: Air Temperature distribution. (a) Published result, (b) Validated result

Figure 42: Basin - water energy balance. (a) Published result, (b) Validated result
Figure 43: Energy balance on glass g3. (a) Published result, (b) Validated result

Figure 44: Energy balance on glass g2. (a) Published result, (b) Validated result
Figure 45: Specific humidity of air. (a) Published result, (b) Validated result

Figure 46: Total production in kg. (a) Published result, (b) Validated result
4.4. Winter Condition Results of Discussion

For the winter conditions, various results were obtained for i- temperature distribution of the different components, ii- air temperatures, iii- energy balance for different components, iv- specific humidity and v- the overall productivity. More detailed results calculation for various components and design are given in the appendix.

4.4.1 Temperature distribution. Temperature difference is considered to be the driving force for any two constituents. Change in temperature is directly related to the change or measure in energy flows in a system discussed in this section. Figure 47 shows the temperature distribution of the different stationary components in the system during the winter conditions. Initially, all components are taken to be at ambient temperatures as the reference. Total sun hour for winter is approximately 11 hours during winter. From the figure it is observed that maximum temperature is of the basin because of the absorptivity, followed by absorber and glass g3. This shows the amount of energy flows from basin to the ambient condition. The maximum value of temperature difference between the basin and the glass g3 can be obtained at midday and is 13.6°C and varies at an average value of around 5-7°C. The difference between basin and glass g3 acts as the source of driving energy for the evaporation, and hence this causes evaporation and condensation at the surface of glass g3. The least amount of temperature is absorbed in humidifier 1 as it consists of humid ropes, which absorb most of the energy before it gets heated up. This temperature difference results in an energy extraction as heat of evaporation/condensation, while the rest is lost. Temperature of finned solar absorber shows dramatic increase because of its nature of high absorptivity until midday. After midday as sun sets, the source of energy diminishes.

Figure 48 shows temperature distribution of humidifier 1. Humidifier 1 temperature is greater than glass g2 temperature till 4th hour. And during this time, energy flow direction is from humidifier 1 to the glass. However, after this the air temperature increases more than the humidifier temperature. Hence, after 4th hour, humidifier 1 temperature falls well below the glass and air as this is again due to the substantially high temperature difference between the air temperature at the entrance and exit to the humidifier 1, which is due to increased convective heat transfer from glass g3 to the air temperature, T_a2. This further increases the average air
temperature. This is further resulted from higher temperature difference between water basin and glass 3. After the sunset at 11th hour the dip in temperature is because the sun sets as it’s the major source of energy. Hence, humidifier 1 now loses energy at a rate higher than it gains. The air in the humidifier is at the highest temperature and convects heat to the glass and the humidifier 1 as can be observed from the figure. In case of humidifier 2 as shown in Figure 49, it shows the earliest rise and maximum rise because the solar absorber plate enhances the temperature of the incoming air towards humidifier 2. The trend is similar to humidifier 1, but it rises further above the ambient point as well because absorber plate gets heated as well, which increases the inlet air temperature for absorber. However, then again, it loses its energy after the 7th hour due to substantial high air temperature difference of air $T_{a5}$ and $T_{a6}$ and sun set, as these can be observed in Figure 49.

![Figure 47: Temperature distribution of various components](image1)

![Figure 48: Temperature distribution of the humidifier 1](image2)
Figure 50 explains distribution of air temperature in the system. Initially, air is in the ambient condition in the system (point 1). This air gains sensible as it flows between the two glasses (g1) and (g3) and follows a temperature profile as shown. T_a2. T_a2 shows the maximum temperature as it flows around the duct. T_a4 is the exit of the humidifier1 temperature and it is lower because the energy is gained by humidifier as well, which substantially decreases the temperature at the exit. T_a5 shows the second highest temperature because this is due to the usage of absorber plate, which enhances convection and subsequently rises air temperature. The humid air then flows to the dehumidifier losing a part of its energy as condensation energy which is gained by the cooling water flowing in the dehumidifier. Hence, T_a6 and T_a7 form a lower profile as shown in Figure 52.
4.4.2. Energy distribution. Figure 51 represents the energy flows of glass g1. The x-axis of the figure shows the time in hours and the y-axis represents the energy (heat fluxes) measured in watt. The solar intensity is measured in watt (W)/m², based on the absorption component respective area. Qabs represents the amount of energy absorbed by the glass 1, which is 5% based on absorptivity, from the total incident solar energy. The maximum solar intensity at mid-day in winter is around 700 W/m² between 7th and 8th hour. Qconv amb and Qrad sky represent the amount of energy lost to the ambient by convection and radiation from the glass surface respectively. Among the total energy received, small amount of energy (4 to 4.5%) is lost as convective heat transfer to the ambient and a very small amount (1%) is as lost as radiation to the sky. Qrad g3 and Qconv air is the amount of energy gained by the glass 1 from glass g3. Qrad g3 is the radiation from the glass g3 and is 1% and Qconv air is the air flowing between glass 1 and 3 and is 2 to 3% approximately. Since the glass transfers heat to the air flowing in the duct, there is an inverted nature of the curve for Qconv_airg1 as it goes negative and positive based on either it loses energy or gains from the glass.

![Figure 51: Energy flow on glass 1](image)

Figure 52 illustrates the energy balance on the basin and the water. It can be observed that the solar radiation absorbed by the basin accounts for most of the energy gained (70%) from the total incident solar energy while the evaporative heat transfer lost (73%) to the glass g3 from basins accounts for most of the total solar energy as well. The evaporative energy from the basin is transferred to the glass 3 as the energy of condensation. Rest of the energy is transferred via convection and
radiation to glass 3 from basin and is minimum amount between 1 to 3% and small 1% accounts to the conduction losses through the insulation in basin. Energy from reflection accounts around 20% of the total energy incident to the basin; the energy input from the reflection increases the overall productivity and is explained under total productivity.

Figure 52: Energy balance on basin and water

Figure 53 shows energy balance on glass 3. Most of the energy gained by glass 3 is by condensation as it is regarded as the condenser for the solar stills. The energy absorbed by glass 3 is around 9%. It loses maximum energy as convective heat transfer to the flowing air. More than 50% is lost as convection (Qconvection glass 3 to air) as shown in the figure. A little less than 50% is gained as evaporation (Qevaporation), which is basically condensed and is collected. Besides, rest of the energy flow is between 5 to 10% of the total indents solar energy.

Figure 53: Energy flow on glass 3
Figure 54 illustrates the energy balance of the humidifier 1. Absorption of the solar irradiation accounts for around 20% of total incident flow to the humidifier while most of the energy transfer is due to evaporation of water from ropes to air i.e. 47% around, which is based on maximum number of humid ropes used. Hence, this is the energy gain for the humid air. The energy by convection heat transfer is positive at first as the energy transfer takes place from humidifier ropes to the air till the 4th hour as explained earlier in Figure 51. After that, the energy direction gets reversed i.e. convective heat transfer takes place from air to the humidifier till the sun sets, where after that it changes direction until the 24th hour. This was earlier explained that the humidifier temperature increases till 4th hour and then decreases till the sunset. Radiation loses to the glass is almost negligible compared to the other energy flows as radiation takes place in the front part of the ropes.

![Figure 54: Energy flow on humidifier 1](image)

Figure 55 shows the energy balance of the glass g2. Most of the energy is lost to the ambient through convection loss, and the energy gained is by absorption of solar irradiation.

Figure 56 shows the energy content of the finned solar absorber plate. The energy input is the absorption due to the sun intensity and it accounts for around 65% of the total incident due to its high absorptivity. Q radiation from absorber to glass is small. Apart from this, Q convection from absorber to air is high due to the high convective area (two sides) of the finned absorber.

Figure 57 show the energy content of the humidifier 2. Here, the input is fed and the convective energy transferred from the solar absorber to the air as there is no
passage for solar intensity due to the insulated wall. Hence again, the maximum energy transferred in terms of loss is due to the evaporation of water to the air. The convection heat transfer is different as to the case of humidifier 1. Here it is negative initially and gets positive because energy is transferred from the humidifier 1 to the air. Also due to the usage of the solar absorption plate, it helped in receiving high amount of energy that made the humidifier 2 temperature high as compared to air for most of the time until solar absorber plate stops receiving solar energy.

This humid air when it enters dehumidifier is taken to saturated condition. Hence the air is at saturation curve. It then loses its energy by condensing on the dehumidifier along the saturated curve. The overall energy gained in dehumidifier is always less than still basin and that can be explained by the productivity of water obtained from dehumidifier, shown in Figure 59.

Figure 55: Energy flow on glass 2

Figure 56: Energy flow on solar absorber
Figure 58 illustrates the specific humidity of air at different positions in the still; w1 is the humidifier entrance humidity which is assumed to be at ambient conditions same as discussed before. Here, w4 and w6 are the specific humidity of air at the humidifier 1 and 2 exit (saturated condition), whereas w7 is the dehumidifier exit air state.

4.4.3. Water production daily. Figure 59 illustrates solar still productivity for solar still basin and condenser during winter case. 29 L/6m²/day productivity was obtained without reflection and 38 L when using reflector in the basins and 13 L in
the dehumidifier. This accounts for a total of 51 L, which also helps us to design for 50 L per day as our objectives.

4.5. Summer Condition Results of Discussion

The summer results were obtained as similar to winter conditions and they are temperature distribution of the different components, air temperatures, energy balance for different components, specific humidity and the overall productivity. The detailed summer results calculation for various components and design has been attached in the appendix.

4.5.1. Temperature distribution. Figure 60 shows the temperature distribution of the different stationary components in the system during the summer conditions. Total sun hour for summer is approximately 14 hours. The result is similar to the case of winter results. From the figure, it is observed that maximum temperature is of the basin because of the absorptivity, followed by absorber and glass 3. The least amount of temperature is absorbed in humidifier 1 as it consists of humid ropes, which absorb most of the energy before it gets heated up. The temperature difference between the basin and the glass g3 is at the maximum value of 10°C at midday and varies at an average value of around 5°C. This temperature difference results in an energy extraction as heat of evaporation/condensation, while the rest is lost.
Figure 61 shows distribution of humidifier 1 temperature. The figure is not exactly similar to winter case as per the humidifier 1 graph. Here, initially humidifier 1 loses its temperature as the energy is gained by the humidifier 1 from the air. Then with time, its temperature increases and the energy flow direction becomes from humidifier 1 through the air to the glass, but only for the first three hours. Then, humidifier 1 temperature falls well below the glass and air temperature as this is again due to the substantially high temperature difference between the air temperature at the entrance and exit to the humidifier 1, thus leading to high average air temperature. This trend is again due to the increased convective heat transfer from the glass g3 to the air temperature, T_a2; this is further resulted from higher temperature difference between water basin and glass 3. After the sunset at 14th hour the dip in temperature is because the sun sets as it is the major source of energy. Hence, humidifier now loses energy at a rate higher than it gains. In case of humidifier 2 as shown in Figure 62, it shows the earliest rise and maximum rise because the solar absorber plate enhances the temperature of the incoming air towards humidifier 2. The trend is similar to humidifier 1, but it rises further above the ambient point as well because absorber plate gets heated as well, which increases the inlet air temperature for absorber. However, then again, it loses its energy after the 7th hour due to substantial high air temperature difference of air T_a5 and T_a6 and also due to sunset.
Figure 63 explains distribution of air temperature in the system. Initially, air is the ambient condition in the system (point 1). This air gains sensible as it flows between the two glasses (g1) and (g3) and follows a temperature profile as shown,
T_a2. T_a2 shows the maximum temperature as it flows around the duct. T_a4 is the exit of the humidifier temperature and it is lower because the energy is gained by humidifier as well, which substantially decreases the temperature at the exit. T_a5 shows the second highest temperature because this is due to the usage of absorber plate, which enhances convection and subsequently rises air temperature. The humid air then flows to the dehumidifier losing a part of its energy as condensation energy which is gained by the cooling water flowing in the dehumidifier. Hence, T_a6 and T_a7 form a lower profile as shown in the figure.

4.5.2. Energy distribution. Figure 64 represents the energy flows of glass g1. The maximum solar intensity at mid-day in summer is around 880 W/m² at 2.40pm (between 7th and 8th hour). Qabs as similar to winter is the amount of energy absorbed by the glass 1 from the total incident solar energy which is 5% of the total incident energy. Similarly, the same amount of 5% is lost from glass to the ambient as Q_conv_g1amb. The convection to air is around 2 to 3% of the total energy. Since, the glass transfers heat to the air flowing in the duct and hence there is an inverted nature of the curve for Q_conv_airg1 as it goes negative and positive based on either losing energy or gaining from the glass. Radiation from glass 3 to glass 1 is almost minimal. The radiation from glass 1 to sky is around 2% of the total energy.
Figure 65 illustrates the energy balance on the basin and the water. It can be observed that 78% of total incident is absorbed by basin while the evaporative heat transfers from basin to glass 3 accounts for 90%. The evaporative energy from the basin is transferred to the glass 3 as the energy of condensation. Rest of the energy (connective and radiation), which is transferred via convection and radiation to glass 3 from basin, is in between 3 to 4%.
Figure 66 shows energy balance on glass 3. Like winter results, most of the energy, which is gained by glass 3, is due to condensation. It accounts for 60% of the total energy flowing to glass 3. Convective transfer to the flowing air is also 58% of the total energy flowing out of the glass 3. Beside these, rest of the energy accounts around 10% and below of the total indents solar energy.

Figure 67 illustrates the energy balance of the humidifier 1. Similar to the winter results, absorption of the solar irradiation accounts only small 21% of total incident flow to the humidifier while most of the energy transfer is due to evaporation of water which is around 80%. The energy by convection heat transfer is mainly negative and large as humidifier gains the energy from the flowing air, except for the first 3rd hours, where the energy is transferred from the humidifier to the air as similarly shown in the temperature distribution of humidifier 1 in Figure 63. Radiation losses to the glass are almost negligible.

Figure 68 shows the energy balance of the glass g2. The major energy flows are due to the absorption from solar radiation as abs g3 and convection losses to the ambient.

Figure 69 shows the energy content of finned solar absorber plate for the summer case. The result is very similar to the winter case, where the maximum is absorbed by the plate due to its high absorptivity. Q convection is also high due to
high surface area of the finned (two sides). And radiation effect from the absorber plate is conservably small.

Figure 70 shows the energy content of the humidifier 2. Here again like winter case, input is fed and the Q convective energy transferred from the finned solar absorber to the air. The maximum energy transferred is due to the evaporation and convection because of high surface area due to the total number of ropes used. The convection heat transfer energy is negative. It is negative because of the solar absorber plate, which increases the average air temperature and transfers the energy from the air to the humidifier 2.
Figure 69: Energy flow on solar absorber

Figure 70: Energy flow on humidifier 2
Figure 71 illustrates the specific humidity of air at different positions in the system; $w_1$ represents specific humidity at inlet, which is based on the ambient based on the ambient conditions. Specific humidity, $w_4$ and $w_6$ of air is at the humidifier 1 and 2 exit (saturated condition), whereas $w_7$ is the specific humidity of the dehumidifier exit air.

![Figure 71: Specific humidity of air at different still position](image)

4.5.3. Water production daily. Figure 72 illustrates SS-HDH productivity for the case of summer. Water production in the still alone accounts for most of the productivity as compared to humidifier. 40.2 L/$m^2$/day productivity was obtained without reflection and 51 L when using reflector in the basin and 16.6 L in the dehumidifier. This gives a total productivity of around 67.6 L, which also helps the targeted unit of 50 L per day for family use as our objectives. As we have seen the case of single humidifier productivity previously in validation section, thus having two humidifier and solar absorber plate increases productivity significantly over 90%. The results also agree with the literature study that using two humidifier increases the productivity to maximum.

Water production in summer is 33% greater than in winter, overall. However, even though both cases fulfill our objectives, parametric study will be carried out for winter and summer case for different basin absorber materials and for different
operational condition to optimize the design and ensure better productivity with lighter SS-HDH unit.

4.6. Parametric Study-Winter

This section will present a parametric study of the SS-HDH unit performance under varying operational and design parameters for winter case.

4.6.1. Operational parameters. Operational parameters can be changed during the units daily operational so as to alter the productivity. In this thesis, two studied parameters are the initial mass (height) of water in the still basin and the mass flow rate of circulating air.

4.6.1.1 Initial water height in solar still basin. Changing initial water level shows a tremendous change in the overall productivity. Higher the water, higher will be weight of the basin, and the productivity reduces on. Figure 73 shows a large increase in total productivity by 15% at a water height of 0.5 cm and 11% for 1 cm. However, with 5 cm water height, the productivity decreases by approximately 15% as compared to 3 cm.
4.6.1.2 Mass flow rate of circulating air. Mass flow rate of the circulating air impacts the still performance and its economy. From Figure 74, increasing flow rate decreases the productivity, as can be seen by 10% when the mass flow rate increased from 0.5 kg/sec to 0.8 kg/sec. Similarly, when the flow rate was decreased from 0.5 to 0.1 kg/sec and 0.05 kg/sec, the productivity was increased by 11% approximately. Increasing flow rate increases the moisture content capacity of air instantly. Thus, reducing the rate of evaporation in the still due to high heat transfer rate from glass cover g3, reduces the delta temperature between glass 3 and the basin. Hence, radiation and convective losses increase as a result. This is an important factor to be considered when designing the forced air convection. The results help to reach an optimum values which introduces the possibility of natural circulation of air, giving the edge over a capital cost as no fans would be needed for the air circulation.
4.6.2. **Effect of design parameters.** Design parameters help in better still configuration in terms of materials usage, which helps to optimize the design weight and cost. Several parameters were studied such as basin materials, glass thickness, glass type, solar absorber materials and the temperature difference between the inlet and outlet water across the dehumidifier. The impact of alternative parameters like the absorptivity of the basin and basin insulation thickness has already been reported within the literature to be insignificant.

4.6.2.1. **Temperature difference in the dehumidifier.** Temperature difference between the inlet and outlet of cooling water determines the amount distillate produced in the condenser. This factor has hardly had significant effect on the productivity as shown in Figure 75.

![Figure 75: Effect of temperature difference in condenser](image)

4.6.2.2. **Basin still material.** Different material affects the weight and productivity of the system. This is because the existing design still was found to be heavy to maneuver from place to place. As an alternative to conventional stainless steel, basin still was made with black coated cotton to see the change in weight and productivity. From Figure 76, it was found that the productivity slightly increases from 50.8 kg to 51.78 kg, and the weight of SS-HDH unit significantly reduces to 181.55 kg from 328.28 kg. This is a tremendous decrease in weight by 81% with having almost insignificant change in productivity.
4.6.2.3. **Stainless steel basin thickness as parameter.** Different basin thickness resulted in insignificant change in the total productivity as shown in Figure 77. However, it resulted in weight reduction of the SS-HDH unit. For 1 mm basin thickness, mass of the SS-HDH unit was 229.43 kg, and for 2 mm and 3 mm, solar still mass was 278.86 kg and 328.2 kg respectively.

4.6.2.4. **Glass thickness as parameter.** Different glass thickness resulted in an insignificant change in the total productivity as shown in Figure 78. However, it resulted in weight reduction of the overall system. Total weight of the glass reduced to 68.07 kg for 1 mm thickness from 272.23 kg of 3 mm glass thickness. This ensures weight of the overall system can be reduced.
4.6.2.5. Absorber material. Solar finned absorber was studied with different materials on the system productivity and weight. Absorber was made with cotton rope materials to study against conventional aluminum absorber. From Figure 79, it was observed that the productivity decreases by 4% when using cotton rope instead of finned aluminum plate. However, the weight reduces too to 2.56 kg using cotton material from 16.2 kg of aluminum sheet.

4.6.2.6. Glass covers material. Parametric study was studied on the glass cover type. Plastic cover Plexiglas [65] was compared against the conventional glass as shown in Figure 80. It was found that the productivity reduced to 43.5 kg from 48.2
kg for the case of plastic cover. However, the mass glass covers were reduced to 115 kg from 271 kg.

![Glass cover materials](image1)

**Figure 80: Glass cover materials**

### 4.7. Parametric Study-Summer

This section will present a parametric study of the SS-HDH unit performance under varying operational and design parameters for summer case.

#### 4.7.1. Operational parameters

Like winter case, two studied parameters are the initial mass (height) of water in the still basin and the mass flow rate of circulating air.

**4.7.1.1. Initial water height in solar still basin.** Changing initial water height shows an overall change in the productivity. Figure 81 shows an increase in total productivity by 7.5% at a water height of 0.5 cm and nearly 6% for 1 cm. The increment is not the same for winter case, but still significant enough to increase the productivity and reduce the overall weight of SS-HDH unit.

![Effect of water height on still productivity](image2)

**Figure 81: Effect of water height on still productivity**
4.7.1.2 Mass flow rate of circulating air. Mass flow rate of the circulating air also shows the similar results as of winter case on the still performance and its economy. From Figure 82, it can be seen that total productivity increases by 12% when the mass flow rate was decreased from 0.5 kg/sec to 0.1 kg/sec. The change was almost insignificant when further reduced from 0.1 kg/sec to 0.05 kg/sec; hence 0.1 kg/sec of mass flow rate of circulating air can be optimum. Also, reduction of air flow rate supports the target goal for natural air circulation and for eliminating fan capital of running costs.

Figure 82: Effect of mass flow rate

4.7.2. Effect of design parameters. Similar to winter, several parameters were studied such as basin materials, glass thickness, glass type, solar absorber materials and the temperature difference between the inlet and outlet water across the dehumidifier.

4.7.2.1. Temperature difference in the dehumidifier. Temperature difference between the inlet and outlet of cooling water had insignificant effect on the productivity as shown in Figure 83.

Figure 83: Effect of temperature difference in condenser
4.7.2.2. Basin still material. For summer also, the change in the productivity was insignificant using cotton material as compared to stainless steel as shown in Figure 84; but the overall mass of solar still reduces tremendously.

![Figure 84: Effect of different materials in solar still](image)

4.7.2.3. Stainless steel basin thickness as parameter. Different basin thickness resulted in insignificant change as the case of winter in the total productivity as shown in Figure 85. However, it resulted in weight reduction of the solar still.

![Figure 85: Effect of basin thickness](image)
4.7.2.4. Glass thickness as parameter. Different glass thickness resulted in insignificant change in the total productivity as shown in Figure 86. However, it resulted in weight reduction of the overall system as stated in the winter case.

![Figure 86: Effect of glass thickness](image)

4.7.2.5. Absorber material. Solar finned absorber was studied with different materials on productivity and weight as in the case of winter. From Figure 87, it was found that the productivity decreases by 15% when using cotton rope instead of finned aluminum plate. Although the weight reduces significantly, in summer it affects productivity dramatically.

![Figure 87: Effect of different materials on absorber plate](image)

4.7.2.6. Glass covers material. Plastic cover Plexiglas was compared against the conventional glass as shown in Figure 88. It was found that the productivity reduced by 11%.

![Time (hr) Liter (m$^3$)](image)
4.8 Combine System- Optimum case study

In this section, based on parameters study results, optimum factors were taken to maximize the productivity, as well as, to ensure cost effective design by ensuring light weight design. Table 5 shows the combine parameters that were taken for the combine system of winter and summer.

Table 5: Combine parameters

<table>
<thead>
<tr>
<th>Operational Condition</th>
<th>Design Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flow rate</td>
<td>Basin still material</td>
</tr>
<tr>
<td>0.1 kg/sec</td>
<td>Basin thickness</td>
</tr>
<tr>
<td>Water height</td>
<td>Glass thickness</td>
</tr>
<tr>
<td>0.005 m</td>
<td>Finned absorber material</td>
</tr>
<tr>
<td></td>
<td>Cover</td>
</tr>
</tbody>
</table>

Aforementioned are the factors that were chosen for the combine system. Dehumidifier inlet and outlet temperature has no effect on productivity. Optimum flow rate and water height of 0.005 m and 0.1 kg/sec chosen had significant increase.
in productivity. For the design factors, basin still material of cotton shows almost insignificant change in total productivity while reducing the mass of the system greatly. Hence, cotton was chosen. Basin thickness and glass thickness have no effect in the total productivity. Hence 1mm is the ideal for reducing the weight of the unit for the same productivity. As for the solar absorber material, the conventional aluminium was favorable as compared to cotton, because using cotton materials, the mass of SS-HDH unit was reduced, but productivity reduced dramatically. Also, the change in overall mass using cotton as absorber was also less. Cover should be made of glass as plastic decreases penetration of solar radiation.

4.8.1. Combine winter and summer system. Choosing the optimum parameters resulted in significant increase in total productivity and decrease in weight of the total SS-HDH unit. Figures 92 and 93 show the total productivity of winter and summer combine parameter results respectively. For winter, it can be observed that the total productivity increased case from 51 L to 67.73 L i.e. increased by 31%. And for the summer case, the productivity increased from 67.6 L to 85.1 L i.e. increased by 26%. Table 6 shows the change in total mass for the SS-HDH unit. From the table we can see that the total mass of SS-HDH unit reduced to 131.797 L from 486.597 L for without water, showing a dramatic decrease of 73%. The usage of cotton material and smaller thickness also ensures reduced capital cost of the system. Therefore, based on the parametric study, we were able to figure out best possible combination for enhancing productivity and low weight.

Table 6: Comparison of mass

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Conventional mass (kg)</th>
<th>Parameter mass (kg)</th>
</tr>
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<tbody>
<tr>
<td>Mass of glass 1</td>
<td>109.2</td>
<td>26.5</td>
</tr>
<tr>
<td>Mass of glass 2</td>
<td>56</td>
<td>14</td>
</tr>
<tr>
<td>Mass of glass 3</td>
<td>110</td>
<td>27.6</td>
</tr>
<tr>
<td>Mass of basin (No water content)</td>
<td>148.28</td>
<td>0.48</td>
</tr>
<tr>
<td>Mass of absorber</td>
<td>18.35</td>
<td>18.35</td>
</tr>
<tr>
<td>Mass of humidifier ropes (No water content)</td>
<td>44.767</td>
<td>44.767</td>
</tr>
<tr>
<td>Total Mass</td>
<td>486.597</td>
<td>131.7</td>
</tr>
</tbody>
</table>
Figure 89: Winter combined parameters result

Figure 90: Summer combined parameters result
Chapter 5: Conclusion and Future Work

This thesis attempted to provide a novel small size solar desalination unit for remote areas which is simple in design, operation and maintenance, and of low cost. The most important objective was to improve the fresh water production by recovering the solar still waste heat of condensation. In order to achieve this, Solar Still (SS) was integrated with a two-stage Humidification-Dehumidification (HDH) system and a built in finned solar absorber to reach our objectives of designing small desalination unit of 50 Liter (L)/day and lower specific water cost (L/$). The system was designed and modeled using AutoCAD and simulated in MATLAB. The system was studied under forced convection with data obtained from Abu Dhabi. From the analysis, it was found that the system productivity is 38 L/6m²/day in basin for winter, while the dehumidifier produces 13 L per day. Similarly, for summer, the productivity in basin was 51 L/6m²/day and 16.6 L in the dehumidifier. In both the cases, the objective of 50 L was obtained. This gives a total productivity of 51 L in winter and 67.6 L in summer. However, the weight of the system was the concern, and hence parametric study was run to illustrate the possible materials as design parameters and possible operational parameters were run to simply the design and operation in real conditions. For summer and winter conditions, the still productivity was affected by height of water in the basin and the mass flow rate of air through the duct as operational parameters. Furthermore, as design parameters, it was found that if the still was made from cotton, it can enhance the productivity and reduce the weight of the system. Similarly, basin thickness and glass thickness have no effects on the productivity, hence should be minimum at all cases. Furthermore, changing absorber materials from aluminium to cotton and glass cover materials to plastic, not only reduces the mass of the overall system, but also significantly reduces productivity. Hence, convention materials are good in their aspects. Moreover, the condenser water inlet temperature has no effect on the productivity. Finally, the combined parameters resulted in 31% and 26% increase in winter and summer productivity respectively and a dramatic decrease of 73% in total weight of the system.

Some of the future recommendations are:

- Simulating the whole system in a closed air natural convection. As per the literature, natural convection can result in similar productivity and can minimize the capital cost of fan.
• Manufacturing the system and validating the model results for better accuracy.
• The system can be studied with the addition of energy storage system like PCM in the dehumidifier to further ensure that the system operates at night time after the sunset.
References


[57] A. Giwa, H. E. S. Fath, and S. W. Hasan, “Humidification-dehumidification desalination process driven by photovoltaic thermal energy recovery (PV-HDH) for


Appendix A

System properties, dimensions and input data

### Basin

<table>
<thead>
<tr>
<th>PARAMETERS</th>
<th>CALCULATION</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Basin properties (Stainless Steel 316L)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Length of basin</td>
<td></td>
<td>1.2 m</td>
</tr>
<tr>
<td>Width of basin</td>
<td></td>
<td>5 m</td>
</tr>
<tr>
<td>Basin Reflector Height</td>
<td></td>
<td>1 m</td>
</tr>
<tr>
<td>Thickness of SS basin</td>
<td></td>
<td>0.001 m</td>
</tr>
<tr>
<td>Area of basin</td>
<td>Length × width</td>
<td>6 m²</td>
</tr>
<tr>
<td>Basin Density (SS)</td>
<td>[68]</td>
<td>8238 kg/m³</td>
</tr>
</tbody>
</table>

### Water properties

<table>
<thead>
<tr>
<th>PARAMETERS</th>
<th>CALCULATION</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density (water)</td>
<td></td>
<td>1000 kg/m³</td>
</tr>
<tr>
<td>Specific heat (SS)</td>
<td>[68]</td>
<td>468 J/kg K</td>
</tr>
<tr>
<td>Specific heat (water)</td>
<td>[68]</td>
<td>4187 J/kg K</td>
</tr>
<tr>
<td>Basin-Water Absorptivity,</td>
<td>[68]</td>
<td>0.9</td>
</tr>
<tr>
<td>Emissivity</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Height of water in basin</td>
<td></td>
<td>0.01 m</td>
</tr>
</tbody>
</table>

Mass- SS Basin\( (M_b) \)
\[
M_b = \text{Area of basin} \times \text{Thickness of SS basin} \times \text{density(steel)}
\]
\[49.428 \text{ Kg}\]

Mass-Water\( (M_w) \)
\[
M_w = \text{Area of basin} \times \text{Height of water} \times \text{density(water)}
\]
\[60 \text{ Kg}\]

Total Mass\( (M = M_w + M_b) \)
\[
M = \text{Mass-Basin} + \text{Mass-Water}
\]
\[109.28 \text{ Kg}\]

Specific heat of combined
\[
\left( M_w \times C_{pw} + M_b \times C_{pb} \right) / M
\]
\[2507.1 \text{ J/kgK}\]

### Reflector

<table>
<thead>
<tr>
<th>PARAMETERS</th>
<th>CALCULATION</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reflectance ( (\rho) )</td>
<td></td>
<td>0.9</td>
</tr>
<tr>
<td>Width of reflector</td>
<td></td>
<td>4.5 m</td>
</tr>
<tr>
<td>Area of Reflector</td>
<td>Length × width</td>
<td>4.5 m²</td>
</tr>
</tbody>
</table>
## Glass Properties

<table>
<thead>
<tr>
<th>PARAMETERS</th>
<th>CALCULATION</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Glass</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Density (glass)</td>
<td>[68]</td>
<td>2800 kg/m³</td>
</tr>
<tr>
<td>Specific heat</td>
<td>[68]</td>
<td>750 J/kg K</td>
</tr>
<tr>
<td>Emissivity</td>
<td>[68]</td>
<td>0.05</td>
</tr>
<tr>
<td>Transmissivity</td>
<td>[68]</td>
<td>0.95</td>
</tr>
<tr>
<td>Absorptivity</td>
<td>[68]</td>
<td>0.05</td>
</tr>
<tr>
<td>Air</td>
<td>Mass flow rate of air</td>
<td>0.5 Kg/s</td>
</tr>
<tr>
<td>Density of air</td>
<td></td>
<td>1.16 Kg/m³</td>
</tr>
<tr>
<td>Air gap length-(bₐ)</td>
<td></td>
<td>0.03 m</td>
</tr>
<tr>
<td>hydraulic diameter of glass duct -Dₕ</td>
<td>$\frac{4 \times (b_g \times \text{width})}{2 \times (b_g + \text{width})}$</td>
<td>0.0596 m</td>
</tr>
<tr>
<td>Glass gap cross section area</td>
<td>$\text{Len of g1} \times b_g$</td>
<td>0.15 m²</td>
</tr>
<tr>
<td>Velocity in glass gap</td>
<td>mass of air Density(air) × (gap cross sectional)</td>
<td>2.8688 m/s</td>
</tr>
<tr>
<td>Reynold number in glass gap</td>
<td>$\frac{\text{density air} \times D_h \times \text{Velocity}}{\text{dynamic viscosity}}$</td>
<td>10,386 (turbulent)</td>
</tr>
<tr>
<td>Width of glass 2, Humidification and finned absorber-(bₕ)</td>
<td></td>
<td>0.1 m</td>
</tr>
<tr>
<td>$d_{in}$</td>
<td></td>
<td>0.005 m</td>
</tr>
<tr>
<td>hydraulic diameter of humidifier (Dₕ2)</td>
<td>$\frac{4 \times ((b_h \times \text{Len of g2}) - (\text{No. of ropes})}{(2 \times (b_h + \text{Len of g2})) + (\text{No. of ropes})}$</td>
<td>0.0226 m</td>
</tr>
<tr>
<td>Humid cross section area</td>
<td>$(\text{Len of g2} \times b_h) - (\text{No. of ropes}) \times \left(\frac{\pi}{4} \times d_{in}^2\right)$</td>
<td>0.4198 m²</td>
</tr>
<tr>
<td>Air velocity in humidifier</td>
<td>mass of air Density(air) × (Humid cross section)</td>
<td>1.0247 m/s</td>
</tr>
<tr>
<td>Reynold number in humidifier</td>
<td>$\frac{\text{mass of air}}{\text{density air} \times D_{h2} \times \text{Velocity}}$</td>
<td>1,405.2 (Laminar)</td>
</tr>
</tbody>
</table>
**Glass 1**

<table>
<thead>
<tr>
<th>PARAMETERS</th>
<th>CALCULATION</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length of the cover ((g_1))</td>
<td></td>
<td>1.651 m</td>
</tr>
<tr>
<td>Thickness</td>
<td></td>
<td>0.004 m</td>
</tr>
<tr>
<td>Area of the cover ((g_1))</td>
<td>(\text{Len of cover} \times \text{width})</td>
<td>8.2550 m²</td>
</tr>
<tr>
<td>Area of the sides (two sides)</td>
<td>(2 \times (0.5 \times \text{Length of basin} \times \text{Height}))</td>
<td>1.2 m²</td>
</tr>
<tr>
<td>Total Area</td>
<td>(\text{Area of the cover} + \text{Area of the sides})</td>
<td>9.4550 m²</td>
</tr>
<tr>
<td>Mass</td>
<td>(\text{Density(glass)} \times \text{Total Area} \times \text{thickeness})</td>
<td>105.8960 Kg</td>
</tr>
</tbody>
</table>

**Glass 3**

<table>
<thead>
<tr>
<th>LENGTH OF THE COVER ((g_3))</th>
<th>1.732 M</th>
</tr>
</thead>
<tbody>
<tr>
<td>Width (wid)</td>
<td>5 m</td>
</tr>
<tr>
<td>Thickness</td>
<td>0.004 m</td>
</tr>
<tr>
<td>Area of the cover ((g_3))</td>
<td>8.66 m²</td>
</tr>
<tr>
<td>Area of the sides (two sides)</td>
<td>(2 \times (0.5 \times \text{Length of basin} \times \text{Height}))</td>
</tr>
<tr>
<td>Total Area</td>
<td>9.86 m²</td>
</tr>
<tr>
<td>Mass</td>
<td>110.432 Kg</td>
</tr>
</tbody>
</table>

**Glass 2**

<table>
<thead>
<tr>
<th>LENGTH OF THE COVER ((g_2))</th>
<th>1 M</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thickness</td>
<td>0.004 m</td>
</tr>
<tr>
<td>Total Area</td>
<td>5 m²</td>
</tr>
<tr>
<td>Mass</td>
<td>56 Kg</td>
</tr>
</tbody>
</table>

**Finned Solar Absorber**

<table>
<thead>
<tr>
<th>PARAMETERS</th>
<th>CALCULATION</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material properties</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Density (Aluminium sheet)</td>
<td>[68]</td>
<td>2702 kg/m³</td>
</tr>
<tr>
<td>Specific heat</td>
<td>[68]</td>
<td>903 J/kg K</td>
</tr>
<tr>
<td>Absorptivity</td>
<td>[68]</td>
<td>0.90</td>
</tr>
<tr>
<td>Emissivity</td>
<td>[68]</td>
<td>0.90</td>
</tr>
<tr>
<td>Length absorber</td>
<td></td>
<td>0.30 m</td>
</tr>
<tr>
<td>Width absorber</td>
<td></td>
<td>5 m</td>
</tr>
<tr>
<td>Thickness abs. (Thk.)</td>
<td></td>
<td>0.003 m</td>
</tr>
<tr>
<td>Area of absorber</td>
<td>(\text{Length abs.} \times \text{width abs.})</td>
<td>1.50 m²</td>
</tr>
</tbody>
</table>
Volume of absorber \( \text{Area of absorber} \times \text{Thickness abs.} \quad 0.0045 \text{ m}^3 \)

Mass of absorber \( \text{Density} \times \text{Volume of absorber} \quad 12.159 \text{ Kg} \)

Fins length (F.T) \( 0.3 \text{ m} \)

Fins width (F.W) \( 0.02 \text{ m} \)

Fins thickness \( 0.001 \text{ m} \)

Spacing \( 0.05 \)

Number of fins \((N.F)\) \( \left( \frac{\text{width}}{\text{spacing}} \right) - 1 \quad 99 \)

Fins cross sect area back sides \( N.F \times \text{Fins width} \times \text{Fins thickness} \quad 0.0020 \text{ m}^2 \)

Volume of Fins \( \text{Area Fins (cross sect)} \times \text{Fins length} \quad 0.000594 \)

Mass of fins \( \text{Density} \times \text{Volume of Fins} \quad 1.605 \text{ Kg} \)

Total Mass \( \text{Mass of fins} + \text{Mass of absorber} \quad 13.764 \text{ Kg} \)

Total Area (H.T convection) \( ((2 \times \text{Fins width} \times \text{Fins length}) \times N.F)) + (2 \times \text{Area of absorber}) \quad 4.1880 \text{ m}^2 \)

hydraulic diameter of finned absorber \( \left( \frac{4 \times \left( \left( b_h \times \text{width} \right) - \left( \text{Thk.} \times \text{wid} \right) + \left( N.F \times F.W \times \text{Fins Thk.} \right) \right)}{2 \times \left( b_h + \text{width} \right) + 2 \times \left( \text{Thk.} + \text{wid} \right) + \left( N.F \times \left( 2 \times F.W \right) + (F.T) \right) } \right) \quad 0.0796 \text{ m} \)

Finned-absorber air flow cross section area \( (b_h \times \text{width}) - (\text{Thk.} \times \text{width} + (\text{Fins cross sect area})) \quad 0.4830 \text{ m}^2 \)

Air velocity in absorber gap \( \frac{\text{mass flow rate of air}}{\text{Density(air)} \times (\text{F.C.S})} \quad 0.998 \text{ m/s} \)

Reynold number in finned solar gap \( \frac{\text{density air} \times D_{h3} \times \text{Velocity}}{\text{dynamic viscosity}} \quad 3,655.3 \) (Laminar)

**Humidifier**

<table>
<thead>
<tr>
<th>MATERIAL PROPERTIES (POLYESTER ROPEs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length of the rope ( 0.5 \text{ m} )</td>
</tr>
<tr>
<td>Ropes Diameter ( d_{in} ) ( 0.005 \text{ m} )</td>
</tr>
<tr>
<td>Rows spacing (Centre to center) ( 0.010 )</td>
</tr>
<tr>
<td>Column spacing (Centre to center) ( 0.011 )</td>
</tr>
<tr>
<td>No. of rows of the ropes ( \left( \frac{b_h}{\text{spacing}} \right) - 1 \quad 9 )</td>
</tr>
<tr>
<td>No. of column of the ropes ( \left( \frac{\text{width}}{\text{spacing}} \right) - 1 \quad 454 )</td>
</tr>
<tr>
<td>Number of ropes (N) ( \text{Rows of the ropes} \times \text{Column of the ropes} \quad 4086 )</td>
</tr>
</tbody>
</table>
### Humidifier Area (for H.T. by convection + evaporation)

\[ \pi \times d_{in} \times \text{Len. of the rope} \times N \]

| Cross section area of rope | \( \text{No. of ropes} \times \left( \frac{\pi}{4} \times d_{in}^2 \right) \) | 0.0802 m² |
| Total Volume of ropes | \( \text{Area of ropes} \times \text{Length} \) | 0.0401 m³ |

| Absorptivity, Emissivity | [68] | 0.9 |

| Density (Polyester) | [68] | 150 kg/m³ |

| Polyester Specific heat \( (C_p_h) \) | [68] | 1021 J/kg K |

| Mass of ropes (Without Water) \( M_h \) | \( \text{Vol. of ropes} \times \text{Density} \) | 55.9593 Kg |

| Density (water) | | 1000 kg/m³ |

| Specific heat water \( (C_p_w) \) | | 4187 J/kg K |

| Water layer Outer Diameter \( (d_o) \) | | 0.0055 m |

| Cross-section Area of water layer | \( \text{No. of ropes} \times \frac{\pi}{4} \times (d_o^2 - d_{in}^2) \) | 0.0168 m² |

| Volume of water layer | \( \text{Area of water} \times \text{Len. of Ropes} \) | 0.0084 m³ |

| Mass of Water layer \( (M_{w}) \) | \( \text{Vol. of water} \times \text{Density(water)} \) | 8.4240 Kg |

| Total mass \( (T_m = M_w + M_h) \) | \( \text{Mass of Water layer} + \text{Mass of ropes} \) | 64.3833 Kg |

| Specific heat of Humidifier | \( (M_w \times C_p_w + M_h \times C_p_h) / T_m \) | 1564.7 J/kg K |

### Overall system weight

<table>
<thead>
<tr>
<th>COMPONENTS</th>
<th>MASS (KG)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass of glass 1</td>
<td>105.896 kg</td>
</tr>
<tr>
<td>Mass of glass 2</td>
<td>56 kg</td>
</tr>
<tr>
<td>Mass of glass 3</td>
<td>110.432 kg</td>
</tr>
<tr>
<td>Mass of basin (dry)</td>
<td>49.428 kg</td>
</tr>
<tr>
<td>Mass of basin (wet)</td>
<td>109.28 kg</td>
</tr>
<tr>
<td>Mass of humidifier 1+2 (dry)</td>
<td>111.92 kg</td>
</tr>
<tr>
<td>Mass of humidifier 1+2 (wet)</td>
<td>128.76 kg</td>
</tr>
<tr>
<td>Mass of absorber</td>
<td>13.760 kg</td>
</tr>
<tr>
<td>Overall system mass (dry)</td>
<td>447.436</td>
</tr>
<tr>
<td>Overall system mass (wet)</td>
<td>523.744</td>
</tr>
</tbody>
</table>
Vita

Muhammad Mustafa was born and brought up in the United Arab Emirates. He completed his O-Level from Our Own English high School, Fujairah, UAE in 2005 and A-level from Karachi, Pakistan. He then moved to Malaysia in 2008 to complete his undergraduate studies in Bachelor of Mechanical Engineering in 2012. He was awarded first class Honor degree for his outstanding and consistency academic achievements. He worked in Malaysia as a Mechanical CAD Engineer for almost a year before moving back to UAE, where he worked in MUC Engineering consultancy as an engineer and draftsman for a year. He then joined Master in Mechanical engineering program at the American University in 2014. While pursuing master he worked as a teaching assistant as part of the graduate Assistantship program until May 2016.