Investigations on Air-cooled Air Gap Membrane Distillation and Radial Waveguides for Desalination

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ABSTRACT

This thesis presents investigations on air-cooled air gap membrane distillation for desalination and the application of radial waveguides based on total internal reflection for solar thermal desalination.

Using an air-cooled design for an air gap membrane distillation (AGMD) process may result in significantly lower energy requirements for desalination. Experiments were conducted on AGMD module to study the effect of air gap, support mesh conductivity and hydrophobicity, condensing surface hydrophobicity. A novel modular design was used in which modules could be used in a series configuration to increase the flux value for the distillate. The output from the series configuration was found to be about three times the production from a single pass water-cooled system with the same temperature difference between the saline and clear water streams. The results also indicated that the mesh conductivity had a favorable effect on the flux value whereas the hydrophobicity of the mesh had no significant effect. The hydrophobicity of the condensing surface was favorable on two accounts: first, it led to an increase in the flux of the distillate at temperatures below 60 °C and second, the temperature difference of the saline feed when it enters and leaves the module is lower which can lead to energy savings and higher yields when used in a series configuration.

The second part of the thesis considers use of low-cost radial waveguides to collect and concentrate solar energy for use in thermal desalination processes. The optical-waveguide-based solar energy concentrators are based on total internal reflection and minimize/eliminate moving parts, tracking structures and cost. The use of optical waveguides for thermal desalination is explored using an analytical closed-form solution for the coupled optical and thermal transport of solar irradiation through a radial planar waveguide concentrator integrated with a central receiver. The analytical model is verified against and supported by computational optical ray tracing simulations. The effects of various design and operating parameters are systematically
investigated on the system performance, which is quantified in terms of net thermal power delivered, aperture area required and collection efficiency. Design constraints like thermal stress, maximum continuous operation temperature and structural constraints have been considered to identify realistic waveguide configurations which are suitable for real world applications. The study provides realistic estimates for the performance achievable with radial planar waveguide concentrator-receiver configuration. In addition to this, a cost analysis has been conducted to determine the preferred design configurations that minimize the cost per unit area of the planar waveguide concentrator coupled to the receiver. Considering applications to thermal desalination which is a low temperature application, optimal design configuration of waveguide concentrator-receiver system is identified that result in the minimum levelized cost of power (LCOP).
Depleting reserves of fresh water and deteriorating quality of naturally occurring water reserves has led to growing scarcity of potable water. The severity of this water crisis has made it necessary to explore other sources of potable water. The abundance of seawater makes it rewarding to explore desalination of seawater as a source of potable water. This thesis presents investigations on the use of air-cooled air gap membrane distillation (AGMD), which is a filtration technique which can be used to remove salt and other impurities from seawater, for desalination. Radial waveguide can be used for concentrating solar energy on a smaller surface, which in turn can be used to raise the temperature of a fluid passing through that surface. These waveguides can be used to heat up the seawater for the solar thermal desalination process.

Using an air-cooled design for an air gap membrane distillation process may result in significantly lower energy requirements for desalination. A novel modular design was used in which modules could be used in a series configuration to increase the output of the potable water. The output from the series configuration was found to be about three times the output from a single pass water-cooled system with the same temperature difference between the saline and clear water streams.

The second part of the thesis considers use of low-cost radial waveguides to collect and concentrate solar energy for use in thermal desalination processes. The optical-waveguide-based solar energy concentrators minimize/eliminate moving parts, tracking structures and cost. The use of optical waveguides for thermal desalination is explored using an analytical closed-form solution for the coupled optical and thermal transport of solar irradiation through a radial planar waveguide concentrator integrated with a central receiver. The analytical model is verified against and supported by computational optical ray tracing simulations. The effects of various design and operating parameters are systematically investigated on the system performance. Design constraints like thermal stress, maximum continuous operation temperature and structural
constraints have been considered to identify realistic waveguide configurations which are suitable for real world applications. In addition to this, a cost analysis has been conducted to determine the preferred design configurations that minimize the cost per unit area of the planar waveguide concentrator coupled to the receiver. Considering applications to thermal desalination which is a low temperature application, optimal design configuration of waveguide concentrator-receiver system is identified that result in the minimum levelized cost of power (LCOP).
# Table of Contents

Chapter 1. Introduction  

Chapter 2. Air-cooled Air Gap Membrane Distillation Module  
2.1. Introduction  
2.2 Experimental setup and the AGMD module  
2.3 Superhydrophobic surface preparation and contact angle measurement  
2.4. Results and Discussion  
2.4.1 Validation of results  
2.4.2 Effect of air gap  
2.4.3 Effect of support mesh conductivity  
2.4.4 Effect of hydrophobicity of support mesh and condensing surface with small air gap  
2.4.5 Effect of hydrophobicity of support mesh and condensing surface with larger air gap  
2.4.6 Effect of air gap and hydrophobicity of the mesh on the saline feed temperature and the cumulative yield in series configuration  
2.5. Conclusions  

Chapter 3. Radial Waveguides for Solar Thermal Desalination  
3.1. Introduction  
3.2 Analytical model for the radial waveguide  
3.3 Simulation and parametric study for the radial  
3.4 Loss mechanisms associated with waveguides  
3.5 Cost model for the radial waveguide  
3.6 Analysis Methodology  
3.7. Results and Discussion  
3.7.1 Parametric Analysis based on Simulations  
3.7.1.1 Effect of Grating size
3.7.1.2 Effect of Waveguide Radius 30
3.7.1.3 Effect of Receiver Radius 31
3.7.1.4 Effect of Waveguide Thickness 32

3.7.2. Parametric Analysis based on Analytical Model 32
3.7.3 Economic Analysis 38

3.8. Conclusions 43

Chapter 4. Conclusions and Future Work 44

Bibliography 46
List of Figures

Figure 1. (a) Schematic illustration of the AGMD apparatus diagram for the experiments and the parametric study; and (b) photograph of the setup. 07

Figure 2. (a) Exploded CAD drawing, (b) photograph of the AGMD air-cooled module used in the experiments. 08

Figure 3. The prepared superhydrophobic surface along with the equivalent drop shape and water contact angle on the prepared superhydrophobic coatings surface. 10

Figure 4. Comparison of the results for the air-cooled AGMD module with the conventional AGMD module results in Warsinger et al [25]. 11

Figure 5. Flux versus Saline Feed Temperature at different air gaps, d. 12

Figure 6. The effect of support mesh with different thermal conductivities on the flux. 13

Figure 7. Effect of hydrophobicity of the support mesh and the condensing surface on the flux when air gap, d = 0.7 mm. 14

Figure 8. (a) Effect of hydrophobicity of the support mesh and the condensing surface on the flux when air gap, d = 5.0 mm. (b) Cumulative effect of the hydrophobicity of the condensing surface and air gap, d on the flux. 15

Figure 9. Saline Feed Temperature versus the respective stages during the series configuration trials and the effect of hydrophobicity of the condensing plate and the air gap, d on the same. 16

Figure 10. Cumulative Yield for the respective stages during the series configuration trials and the effect of hydrophobicity of the condensing plate and the air gap, d on the same. 17

Figure 11. The overall flux obtained in series configuration with the control surface and the hydrophobic surface at different air gaps and comparison to a single pass water-cooled AGMD module with the same ΔT [25]. 18

Figure 12. Schematic illustration of the radial waveguide concentrator integrated to a central receiver, and control volume showing the influx and out fluxes of heat. 20
Figure 13. (a) Simulation setup for the radial waveguide (b) rays undergoing total internal reflection in the waveguide (c) cross section of the scattering surface.

Figure 14. Waterfall chart showing the various loss mechanisms for solar energy concentration through a waveguide.

Figure 15. Effects of grating dimension on the (a) collection efficiency for the range 60 to 150 µm (b) percentage of rays collected (c) collection efficiency for the range 78 to 150 µm (d) absorption and decoupling losses.

Figure 16. Effects of waveguide radius on the collection efficiency, energy collected and absorption and decoupling losses.

Figure 17. Effects of receiver radius on the collection efficiency, energy collected and absorption and decoupling losses.

Figure 18. Effects of waveguide thickness on the collection efficiency, energy collected and absorption and decoupling losses.

Figure 19. Validation of the analytical model by plotting the variation of collection efficiency, energy collected and energy absorbed with waveguide radius from the analytical model and the TracePro simulations.

Figure 20. Influence of incident irradiance, waveguide thickness, convective heat transfer coefficient and waveguide radius on the spatial distribution of temperature rise within the radial waveguide.

Figure 21. Effects of incident solar irradiance ($I_0$) and waveguide radius ($R_{wg}$) on (a) net thermal power delivered, (b) aperture area, and (c) collection efficiency for HTF temperature of $T_F = 100$ °C. The values for waveguide thickness ($t$) and receiver diameter ($D_G$) are kept constant at 10 mm and 40 mm, respectively.

Figure 22. (a) Effects of incident solar irradiance ($I_0$) and waveguide radius ($R_{wg}$) on maximum temperature difference between the waveguide and ambient air obtained for HTF temperature of $T_F = 100$ °C. The values for waveguide thickness ($t$) and receiver diameter ($D_G$) are kept constant at 10 mm and 40 mm, respectively. (b) Effect of waveguide radius ($R_{wg}$) and waveguide...
thickness \( (t) \) on the stress induced in the waveguide for a fixed incident solar irradiance \( (I_0) \) of 1000 W/m\(^2\). The red line shows the maximum allowable stress which is 8 MPa.

**Figure 23.** (a) Effects of incident solar irradiance \( (I_0) \) and waveguide thickness \( (t) \) on the maximum allowable waveguide radius \( (R_{wg-max}) \) for (a) when convective heat transfer coefficient \( (h) \) is 2.5 W/m\(^2\)K (b) when convective heat transfer coefficient \( (h) \) is 5 W/m\(^2\)K.

**Figure 24.** Effects of waveguide thickness and radius on (a) cost per unit aperture area of the radial waveguide concentrator-receiver system. Influence of waveguide thickness and incident irradiation intensity on (b) minimum waveguide-receiver cost per unit aperture area and the corresponding (c) optimal waveguide radius obtained for \( \Delta T_{max}^* = 80 \) °C.

**Figure 25.** (a) Effect of waveguide thickness and radius on levelized cost of power. (b) Effect of waveguide radius and incident irradiance values \( (I_0) \) on levelized cost of power.

**Figure 26.** Effect of waveguide thickness and incident irradiance values \( (I_0) \) on (a) levelized cost of power and (b) optimal waveguide radius. (c) Effect of waveguide thickness and convective heat transfer coefficient \( (h) \) on levelized cost of power.
List of Tables

Table 1. Thermophysical properties of optical glass (ZK7) used in this study 27

Table 2. Preferred planar waveguide design based on minimum LCOP for $\Delta T_{max} = 80$ °C. The minimum values of the objective function are italicized and listed in bold face 41
Chapter 1: Introduction

There is a growing scarcity of potable water due to the depleting reserves of fresh water and deterioration in quality of the naturally occurring water reserves [1-3]. This is leading to a looming water crisis which necessitates exploring alternative sources of potable water for the future. About 96.5% of all the water on earth is seawater. Of the remaining 3.5% only about 1% is in the form of freshwater lakes, rivers and ground water with the rest being frozen water in the form of polar ice caps, glaciers, ice and snow. The shear abundance of seawater when compared to freshwater makes it rewarding to explore the possibilities of using it as a source of potable water. Seawater desalination is especially useful in coastal areas as there is no additional cost of transportation involved. Reverse osmosis has been extensively used for desalination in many countries across the world [4-7]. Other methods such as membrane distillation, electro-dialysis, and solar stills are also being explored for desalination including some methods in which renewable energy like solar, wind, wave and geothermal energy are being employed [8-9]. Solar thermal energy can be used as the source for energy for many of the desalination methods and can result in lower costs [9]. The cost for desalination depends on the type of feed water, the desalination method used and the type of energy used [10]. Developing a desalination method that meets the need of potable water at the lowest cost, has low energy requirements and is preferably powered by a renewable source of energy is critical.

Membrane distillation is a promising process which could be extensively used for desalination in the future [11-16]. Membrane distillation can be used with brackish water and water containing other impurities. The main advantage of membrane distillation over reverse osmosis is that modules based on membrane distillation are resistant to fouling [17-18]. Direct Contact Membrane Distillation (DCMD), Air Gap Membrane Distillation (AGMD), Sweeping Gas Membrane Distillation (SGMD) and Vacuum Membrane Distillation (VMD) are the configurations which are used in membrane distillation. In DCMD, the hot feed solution is in direct contact with a membrane which is in direct contact with the cold solution. A sweeping gas which is usually inert is used to sweep the vapor which is on the permeate side of the membrane in SGMD. In VMD, a vacuum is created on the permeate side of the membrane to direct the vapor out of the module for condensation. AGMD is a membrane distillation configuration in which the feed solution is in direct contact with the membrane and there is an air gap separating the membrane and the condensing surface. The air gap between the membrane and the
condensing surface leads to a decrease in heat loss due to conduction [19-20]. However, there is an additional resistance to mass transfer due to the air gap which is a disadvantage associated with this process. It has been shown in the past that AGMD is the most energy efficient of all membrane distillation configurations [20-23].

The energy associated with pumping the hot saline feed water and the cooling fluid forms a substantial part of the energy requirements for AGMD process. Hence, using an air-cooled system instead of using the conventional water-cooled configuration may lead to a system with lower energy requirements which will in turn lead to lower running costs and has been studied as a part of present work. The condensing plate can be placed in such a way that it faces the windward side which will lead to cooling through buoyancy-induced convection. There have been studies in the past in which the internal temperatures of an AGMD module has been studied [24]. These studies have been used in this study when comparing an air-cooled system to the more conventional water-cooled one.

Employment of a superhydrophobic condensing surface has been shown to result in a substantial increase of permeate flux in past for the conventional AGMD configuration with a cooling channel [25-26]. This can be attributed to the fact that hydrophobic jumping droplet condensation has better heat transfer coefficients which results in faster condensation and an increased permeate flux [26-29]. Warsinger et al. [25] has conducted parametric study for a two channel AGMD system in which he studied the effect conductivity and hydrophobicity of support mesh, hydrophobicity of the condensing surface and air gap. Using the modules in series configuration can lead to greater yields as there is recovery of latent heat of permeate and the energy associated with the saline flowing out of one module is utilized in the next module.

With the objective of reducing the costs associated with desalination, air-cooled AGMD modules which have been assembled in series configuration are being studied in this paper. A parametric study in which the effect of air gap and hydrophobicity and conductivity of the support mesh for an air-cooled single channel system is being conducted as part of this study. The effect of having a hydrophobic condensing surface on the permeate flux is also being studied for the air-cooled module and in the case when the modules have been placed in series configuration.
The second part of the thesis is motivated by the objective of making the desalination process fully sustainable by using solar energy concentration systems to meet the thermal needs of a desalination system.

Solar energy harvesting can be done by either optically concentrating the energy or by using photovoltaic cells to convert the solar energy into electrical energy. In concentrated solar thermal (CST) systems, light collecting elements which are basically large mirror surfaces to focus sunlight on a smaller area is used. It can be used for applications such as power generation, thermal desalination, etc. Currently, in utility scale CST power plants parabolic troughs or flat mirrors (heliostats) are used which track the sun diurnally to concentrate sunlight onto the receiver [30,31]. Concentrated Solar thermal systems which use solar tracking are effective, but they suffer from major drawbacks like tracking error caused by wind loading, tracking cost, and the high capital cost of the mirrors and support structure [30]. To achieve the tracking motion and to support the heavy mirrors or the troughs, support structure and pylon, drive systems, and wiring are needed which account for almost 30–40% of the total cost [31]. Active tracking requires the heliostats or troughs to move. This movement needs them to be spaced apart to limit shading losses, thus leading to poor land-use efficiency [30-33]. The drawback with implementing wide spread use to solar energy is the high costs associated with it. To achieve competitive costs, solutions which eliminate the need for active tracking of the sun are required [30,34].

Optical waveguide based approach which utilizes total internal reflection (TIR) to concentrate sunlight on a smaller area is a promising approach. Planar waveguide solar concentrators are being explored in the area of photovoltaic based solar energy harvesting systems to focus solar irradiation on to the small area of solar cells [35-45]. Concentrated solar thermal systems based on total internal reflection can be based on either of the following two approaches: (a) luminescent solar concentrators (LSC) and (b) micro-optics solar concentrators with coupling features, which is the focus of this study. LSC consists of a waveguide dispersed with luminescent molecules (such as organic dyes, quantum dots, etc.) that absorb the incident light and re-emit at longer wavelengths. Some of the re-emitted radiation undergoes total internal reflection and is guided towards the receiver [41-45]. In a TIR based micro-optics solar concentrator, the incoming sunlight is focused by a lens array onto localized scatterers in the waveguide, which guides the light rays to the receivers by total internal reflection. Non-
sequential ray tracing studies and laboratory proof-of-concept demonstrations of micro-optic concentrators fabricated for concentrating photovoltaics are being conducted [35-47]; this concept has not yet been applied to CST applications. Due to its lightweight nature the mounting and positioning of the waveguides does not require a lot of structural support, which in turn simplifies structural design and reduces cost. Since, there is no active tracking involved in this concept the overall cost of the system is low and the land-use efficiency is also better as there are no shading concerns [35]. It has been shown that these systems have the ability to focus the sunlight throughout the day with only a few centimeters of lateral movement [46,47]. This leads to considerable reduction in the tracking cost. Fixed axis concentrators which have no tracking mechanism can be used as seasonal concentrators [48], especially for lower temperature applications. Self-tracking planar concentrators based on thermal phase change actuator [49] and light induced refractive index change [50], that can achieve high levels of concentration over a wide acceptance angle (> 40 degrees) are also being explored.

TIR based optical waveguides have been explored for concentrated photovoltaics applications [35-50] but there have been very few investigations of the concept for concentrated solar thermal applications. The numerical efforts of in this field have been mostly limited to optical analysis based on ray trace modeling [35-37,39,48,49], with a few exceptions of recent experimental effort in analyzing the temperature distribution within a luminescent waveguide for photovoltaic applications [51]. However, to implement the TIR concept in CST applications an integrated optical and thermal analysis of the system is required. In this thesis, an analytical closed form solution for the coupled optical and thermal transport in a TIR based ideal planar waveguide concentrator integrated with central receiver is presented. In this study the possibility of using a radial waveguide which concentrates the light towards the centrally placed receiver instead of the conventional planar waveguide design which has the receiver at the ends of the waveguide has been explored. The radial design could lead to better geometric concentration ratio as the light rays are focused on a smaller surface and better collection efficiency as the rays travel for a shorter distance before they are collected at the receiver. The spatial distribution of temperature and the irradiation reaching the receiver has been obtained based on various design and operating parameters. The results guide us towards effective design of the system based on the considerations of thermal stress, wind loading, net thermal power delivered to the receiver, collection efficiency and aperture area requirement. A simple cost analysis is presented, and a
new performance metric namely, levelized cost of power (LCOP) is introduced to provide useful insights into optimal waveguide concentrator-receiver configurations that yield the minimum levelized cost of power (LCOP) based on the tradeoff between cost and thermal power delivered to the receiver. This study also provides a feasible path forward towards meeting the cost and performance objectives for applications such as, thermal based desalination [52,53], and CST power generation.
Chapter 2. Air-cooled Air Gap Membrane Distillation Module

2.1. Introduction

Experimental setups, the AGMD module and superhydrophobic surface preparation are described in this chapter. Further in this chapter, the results from the different configurations have been discussed followed by the conclusions that can be inferred from these results. The validation of results is followed by the parametric study in which the effect of the various parameters on the performance of the process is studied.

2.2 Experimental setup and the AGMD module

The schematic of the experimental setup to perform various optimization and parametric studies on considered AGMD module is shown in Figure 1. The feed water is stored in 5 gallon tank and an immersion water heater with a thermostat is used to keep the feed water at a temperature which is within ± 0.2 °C of the desired temperature. The feed water is pumped to the AGMD module by a revolution water pump (12 VDC, 7.5 Amps). It had a maximum flow rate of 3 GPM and the shutoff pressure of 3.8 bars. A flowmeter was used to measure the flow rate of the feed water. Thermocouples were used to monitor the water temperature at the inlet and outlet and the condensing plate temperature. Instead of using a channel behind the condensing plate, a fan (5 VDC, 0.5 Amps) was placed on the other side of the condensing plate and forced convection was used for cooling purposes.

The effect of membrane material, pore size, thickness, porosity and membrane support on permeate flux for an AGMD process has been studied [54,55]. Bigger pore size, lesser thickness and absence of support structure results in increase in permeate flux. But lack of support structure can result in a fragile membrane which will need to be replaced often. Also, the pore size cannot be very large as it will let impurities through the pores. These considerations led us to choose a polytetrafluoroethylene (PTFE) membrane with propylene support structure, a pore size of 0.45 μm and thickness, t of 160 μm to 230 μm for our experimental module. The membranes for the experiments were supplied by Membrane Solutions, Shanghai, China. The membrane in the module was held in place using an acrylic piece and a mesh. Copper, Aluminum, Steel and Plastic meshes were used for this purpose to study the effect of conductivity oh the support mesh on permeate flux. The AGMD module which was used in the experiments was made from
acrylic. The module contains a saline channel, an air gap and an air-cooled condensing surface. The feed channel is 65 mm X 63 mm X 12.7 mm. The channel is serpentine and has six walls with width 5 mm each to guide the saline feed through the channel. The serpentine channel helps to maximize the time for which the feed comes in contact with the membrane and the pressure on the saline side of the membrane. Alsaadi et al. [54] has shown that the saline residence time inside the AGMD module has a positive effect on the flux.

Figure 1. (a) Schematic illustration of the AGMD apparatus diagram for the experiments and the parametric study; and (b) photograph of the setup.
Copper plates were used as condensing surfaces. The vapor which passed through the membrane upon condensing on the back plate was allowed to drip down under the effect of gravity and collected in a flask. The air gap was created by inserting combination of acrylic sheets and gaskets of different thicknesses between the membrane and the condensing plate. When the saline feed water exits the module, its temperature is recorded and it is released back into the feed water storage tank. The ppm level of impurities in the feed water was kept in a range of 150 – 170 ppm by adding water.

![Figure 2. (a) Exploded CAD drawing, (b) photograph of the AGMD air-cooled module used in the experiments.](image)

The effects of various parameters and conditions on the performance of an air-cooled air gap membrane distillation module were studied in this experimental study. The Feed water temperature was varied from 40 to 70 °C and its effect on the permeate flux was studied. Flow rate of about 2 L/min was maintained by controlling the current input to the circulation pump. The volume of the distillate collected in the flask (V) was measured for a given time (t) to calculate the permeate flux. The effective area of the membrane (A) was also used in this calculation.

\[
Flux = \frac{V}{At} \tag{1}
\]

The system was made in form of modules which could be placed in series configuration. This resulted in a scalable design in which the output could be increased or decreased by adding or
removing modules. Effect of hydrophobicity of the copper condensing surface and the hydrophobicity and conductivity of the support mesh were also studied during the trials.

2.3 Superhydrophobic surface preparation and contact angle measurement

In recent times, there have been attempts to develop Graphene based hydrophobic coatings [56-59]. In this study, we have focused on using copper based coatings on the condensing plate. Copper based superhydrophobic surfaces as considered in this study were prepared through an electrodeposition based process to generate inherent and durable superhydrophobic coatings as described by Jain and Pitchumani [60]. Electrodes, copper sheet/mesh and platinum mesh were cleaned with acetone and deionized water to remove any dirt and grease from the surface and employed as working electrode and anode, respectively. Two step electrodeposition process as described by Haghdooost and Pitchumani [61], where application of high potential is followed by a small potential to obtain a multiscale and stable deposit, was employed. The electrolyte was an aqueous solution containing CuSO\(_4\) (1 M) and H\(_2\)SO\(_4\) (0.5 M). Potentistatic electrodeposition at 1.1 V, followed by electrodeposition at 0.15 V for 10 secs, was performed and inherent copper based deposit was obtained on considered copper back plate and copper mesh. Furthermore, deposition time was optimized to obtain the deposit thickness of about 30µm on the considered sample. After electrodeposition, surfaces were rinsed with acetone and deionized water and dried with nitrogen gas.

As a final step, the as-prepared surfaces were modified through immersion in 0.02 M methanol solution of stearic acid at room temperature for 24 hours. The modified samples were then washed with methanol to remove any residual organic acid on the surface, followed by washing with deionized water and then dried for further characterization and study.

Multiscale cauliflower-shaped morphology was obtained on the prepared superhydrophobic surface, as observed in Fig. 3. Fig. 3 also shows the equivalent drop shape and water contact angle on the prepared superhydrophobic coatings surface. An aggressive fractal texturing with globular asperities at multiple scales leads to the observed superhydrophobicity with contact angles of 159° for the prepared superhydrophobic copper back condensing plate. Presence of multiscale asperities and superhydrophobic nature of the coating ensured the high nucleation density and dropwise and jumping droplet condensation on the superhydrophobic condensing back plate.
Figure 3. The prepared superhydrophobic surface along with the equivalent drop shape and water contact angle on the prepared superhydrophobic coatings surface

2.4. Results and Discussion

2.4.1 Validation of results

The design and performance of the basic air-cooled AGMD module was validated against previously studied conventional AGMD system design by Warsinger et al [25]. Flux output was compared for a basic AGMD configuration with a plastic mesh as a membrane support and pure copper condensing surface as a back plate. The inconsistencies between presented and previously studied designs such as use of air cooling instead of conventional water cooling channel and different air gap thicknesses, did not allow a straightforward comparison of flux output between two designs. Furthermore, with an increase in saline feed temperature, condensing plate temperature increases, which leads to an improved convective heat transfer from condensing plate to ambient air. Hence, it was not feasible to get a consistent temperature difference between hot saline feed liquid and condensing back plate for the studied configuration at various saline feed temperatures, unlike a conventional water-cooled systems where a constant temperature difference can be maintained for various inlet saline feed temperatures.
To normalize the effect of inconsistencies and allow the validation of designed AGMD configuration, a corrected flux value is specified and considered. Corrected flux approximately modifies the actual flux values to obtain the flux for any different air gap thickness and temperature difference, assuming a linear effect of these parameters and can be given as:

\[
\text{Corrected Flux} = \text{Real Flux} \times \frac{d}{d_{[25]}} \times \frac{\Delta T_{[25]}}{\Delta T}
\]  

(2)

where, \( d = \) air gap [mm]

\( d_{[25]} = \) air gap [mm] used by Warsinger et al [25]

\( \Delta T = \) Temperature difference between the condensing surface and the saline channel

\( \Delta T_{[25]} = \) Temperature difference between the condensing surface and the saline channel as reported by Warsinger et al [25]

Real flux and corrected flux values as obtained from the present study for various feed temperatures are compared with the flux output for the conventional AGMD setup described by Warsinger et al [25] and are shown in Figure 4. A similar trend of real flux values between both the studies can be observed from the same. In addition, a small variation of about 5-10% between corrected flux for present study and flux output as reported by Warsinger et al, validates the designed setup and various design and performance parameters. Based on this validation, the air-cooled AGMD system has been optimized through various experimental parametric studies as described in following subsections.

**Figure 4.** Comparison of the results for the air-cooled AGMD module with the conventional AGMD module results in Warsinger et al [25].
2.4.2 Effect of air gap

A parametric study to study the effect of various system and process variables on the performance of the air-cooled module was conducted. Figure 5 shows the effect of air gap dimensions on the flux for the process. Flux is the parameter which has been used to gauge the performance of the membrane distillation process. The unit for flux is L/m²hr, where the amount of condensate formed is measured in liters and the area of the membrane exposed for the droplets to cross over is measured in m². As can be seen from Figure 5, flux increases somewhat linearly with increase in saline feed temperature. The flux values are highest for the case when the air gap is smallest and it becomes noticeably lower when the air gap increases. It has been shown that an increase in air gap results in decrease in mass flux due to an additional resistance to mass transfer [62].

Figure 5. Flux versus Saline Feed Temperature at different air gap, d.

2.4.3 Effect of support mesh conductivity

The effect of mesh conductivity on the flux was studied using Plastic, Steel, Aluminum and Copper meshes. The meshes are essential for holding the membrane in place and reduce bulging of the membrane due to the pressure difference on the two sides. The meshes had an open area in the range of 65% to 66%, so that they do not interfere with the vapor transfer across the
membrane. As can be seen from Figure 6, the module with the Copper mesh performs the best and the plastic mesh results in the lowest yields. The steel and aluminum mesh have flux values between copper and plastic.

It is seen that the modules with meshes that have higher conductivity have higher yields. This is due to better heat conduction across the air gap [25-28]. The highly conductive mesh surfaces reduce resistance to heat transfer in the air gap. When the saline feed temperature is above 60 °C, the air gap in the module gets flooded and the effect of the mesh conductivity is more pronounced at these temperatures.

![Graph](image.png)

**Figure 6.** The effect of support mesh with different thermal conductivities on the flux

### 2.4.4 Effect of hydrophobicity of support mesh and condensing surface with small air gap

Hydrophobic condensing plate and hydrophobic mesh surface also affect the yields for the process. Figure 7 shows the flux values for the different condensing plate and mesh combinations for a small air gap (0.7 mm) configuration. When a control mesh is used which is a normal copper mesh, the modules with hydrophobic back plate have better flux values at lower temperature. This has been attributed to hydrophobic jumping droplet condensation [25-28].

At temperatures above 60 °C, the module with the control back plate surface performs better. This is due to the fact that at these temperatures, the air gap is flooded and both the control surface and the hydrophobic surface behave similarly. The hydrophobic surface has lower yields
at high temperatures because of the fact that thermal conductivity of the condensing plate reduces due to the hydrophobic coating, which results in higher temperature for the back plate and this leads to a lower ΔT which in turn leads to lower yields.

The hydrophobicity of the mesh has a negative effect on the yield which can be seen in Figure 7, this is due to the fact that the mesh is not a condensing surface. The hydrophobic mesh has lower conductivity when compared to control copper mesh and this leads to the lesser yield.

![Graph showing flux vs saline feed temperature](image)

**Figure 7.** Effect of hydrophobicity of the support mesh and the condensing surface on the flux when air gap, d = 0.7 mm.

### 2.4.5 Effect of hydrophobicity of support mesh and condensing surface with larger air gap

When the air gap is increased from 0.7 mm to 5.0 mm, the hydrophobic condensing surface performs better than the control surface, as can be seen in Figure 8a. The yield from the module with the control surface improves considerably when the temperature goes beyond 60 °C. Flooding at high temperatures negates the positive effects of hydrophobicity like jumping droplet condensation while the lower conductivity of the hydrophobic surface affects the yield negatively. The hydrophobic mesh surfaces have the same effect as they had in the case of lesser air gap.

The results here show that the conductivity of the mesh, the air gap and the hydrophobicity of the condensing plate have a significant effect on the flux values for the system. The hydrophobicity of the mesh has no significant positive effect on the yield. Figure 8b shows the
effects of both air gap and hydrophobicity of the condensing surface on the yield. From the figure, it can be seen that even though the hydrophobic surface performs better than the control surface at large air gap, the yield it provides is lower than the yield of the control surface module at lesser air gap.

Figure 8. (a) Effect of hydrophobicity of the support mesh and the condensing surface on the flux when air gap, \(d = 5.0\) mm (b) Cumulative effect of the hydrophobicity of the condensing surface and air gap, \(d\) on the flux
2.4.6 Effect of air gap and hydrophobicity of the mesh on the saline feed temperature and the cumulative yield in series configuration

Although, the yield from the control surface and small air gap configuration is highest at high temperature ($70 \, ^\circ C$), using a hydrophobic surface has its own advantages even at such high temperatures, even though the flux is slightly lower. Because of the lower conductivity of the hydrophobic back plate, the temperature drop for the saline water after it passes through the module is less when compared to the control back plate case. This can be useful as lower drop in temperature will lead to lesser heating requirement when it goes to the hot reservoir, which can lead to lesser energy requirements.

The lower drop in saline temperatures can also be useful when the modules are placed in series and the saline coming out of one module is the saline feed for the next module. It can be seen that in such a series configuration, the temperature drop is more in the case of control surface and small air gaps. Large air gap cases have a smaller drop in the saline feed temperature for the consecutive modules.

![Figure 9](image.png)

**Figure 9.** Saline Feed Temperature versus the respective stages during the series configuration trials and the effect of hydrophobicity of the condensing plate and the air gap, d on the same
To study how the different configurations work in a series configuration, the exit saline temperature of the first module was used as the feed saline temperature for the second module and so on. The trials were conducted till the saline feed temperature dropped to below 50 °C, as the flux reduces considerably after that. Figure 9 shows the saline feed temperature for the respective stages. As can be seen from the figure, the feed temperature decreases faster for the control back plate and small air gaps. The corresponding cumulative yields can be seen in Figure 10. The cumulative yield increases sharply for the two configurations with small air gap but as the number of stages increase, the flux readings for hydrophobic surface back plate configuration surges ahead, as it has higher yield for later stages because the temperature drop is less for consecutive stages in this case and lower temperature drop means higher saline feed temperature for a given stage. The saline feed temperature directly affects the flux and thus, it results in higher cumulative flux for hydrophobic surfaces. The highest cumulative flux is generated for the case with hydrophobic condensing plate and small air gap of 0.7 mm. Even the hydrophobic condensing surface with large air gap of 5.0 mm case has better cumulative flux values than the control condensing surface cases.

![Figure 10](image.png)

**Figure 10.** Cumulative Yield for the respective stages during the series configuration trials and the effect of hydrophobicity of the condensing plate and the air gap, d on the same
When this series configuration of this air-cooled system, in which the temperature of saline feed water is allowed to drop by 20 °C, is compared to single pass systems which employ a cooling channel with ΔT of 20 °C, the cumulative flux for the series configuration is about 3 times that of the single pass system in spite of the limitations of the air-cooled system. This can be seen in Figure 11.

**Figure 11.** The overall flux obtained in series configuration with the control surface and the hydrophobic surface at different air gaps and comparison to a single pass water-cooled AGMD module with the same ΔT [25].

### 2.5. Conclusions

The air-cooled air gap membrane distillation system works well even though it has lower energy requirements when compared to the water-cooled system. The modular design helps us in improving the flux by adding modules in series to utilize the lower difference in temperatures of saline feed and the saline when it exits the module. The hydrophobic surface helps in improving the flux as well as reduces the temperature drop for the saline water as it passes through the module.

The conductivity of the support mesh had a significant effect on the yield from the setup. The copper mesh with its conductivity of about 401 W/m-K resulted in the highest flux values.
followed by the Aluminum mesh and steel mesh. The plastic mesh resulted in about 30% lesser flux values. Hydrophobicity of the support mesh did not have as great an impact on the flux values as the conductivity values. In fact in certain cases the hydrophobic mesh resulted in lower yields when compared to the normal copper mesh.

Increasing the air gap resulted in lower flux values. The saline temperature reduced less in the case of increased air gap but the flux but it was not enough to result in an increased overall flux even in the series configuration. The hydrophobic surface works well even for the larger air gaps but the flux is lower when compared to small air gap setup.

In series configuration, the hydrophobic surface with the small air gap works best and results in about three times more yield when compared to the single pass water-cooled system.
Chapter 3. Radial Waveguide for Solar Thermal Desalination

3.1. Introduction

The analytical closed form solution, the simulation method for the ray trace analysis, description of the loss mechanisms associated with the waveguides and a cost model for the radial waveguide are described in this chapter. Later in the chapter, a parametric study based on the ray trace simulation and a parametric study based on the analytical model has been presented. This is followed by a cost analysis and the optimal radial waveguide-receiver configuration that minimizes LCOP. The key findings and conclusions have been stated towards the end.

3.2 Analytical model for the radial waveguide

The design of a waveguide collector and its integration to the receiver is critical for CST applications. Figure 12 shows the schematic of a radial waveguide integrated to a linear receiver, also called a heat collection element, such as that commonly employed in parabolic trough CST power plants. The heat collection element (HCE) consists of an absorber pipe carrying heat transfer fluid (HTF) surrounded by an evacuated glass tube envelope. The waveguide radius and thickness are denoted by $R_{wg}$, $t$ respectively; while the radius of the heat collection element is denoted by $R_{sec}$.

Figure 12. Schematic illustration of the radial waveguide concentrator integrated to a central receiver, and control volume showing the influx and out fluxes of heat
The waveguide is assumed to be made of the same material as that of the glass envelope of the receiver so that they have similar thermal coefficient of expansion for a perfect hermetic sealing. In an actual waveguide based solar concentration system, an array of cylindrical lenses forms the top layer that focuses the incident solar rays on to a coupling structure etched on the bottom layer of the waveguide. The rays reflected by the coupler at angles exceeding the critical angle would be totally internally reflected and transmitted to the two edges of the waveguide into the HCEs.

An analytical model is developed for the theoretically ideal waveguide concentrator depicted in Fig. 12. The goal is to develop an analysis that would present the theoretical maximum realizable performance of a waveguide based concentrator, which would form a target to achieve in practical desalination applications. The analysis assumes the following:

(a) Perfect focusing of the solar irradiation by the lenses on to the coupling structure, i.e. no focal dispersion of the rays at the focal point resulting in zero losses due to rays missing the coupler or coupled at an undesirable angle.
(b) Perfect total internal reflection with no escape cone losses.
(c) No waveguide decoupling loss due to the likelihood of the propagating rays striking a subsequent coupling feature during TIR.
(d) Stable and constant optical properties of the waveguide material over operating temperature range.
(e) Monochromatic incident light (optical properties invariant of wavelength).

For a perfectly transparent material exhibiting lossless TIR, all the incident irradiation should reach the two receivers coupled to the edges of the waveguide. Nevertheless, even transparent materials absorb some of the irradiation due to the extinction coefficient of the material that will attenuate the irradiation flux propagated to the ends. Considering that ray propagation is along $r$-direction only, with the incident light assumed to scatter uniformly over $0 \leq \phi \leq \frac{\pi}{2}$ (Fig. 12), the irradiation through an angle $d\phi$ is $I_0 (2\pi r \, dr) \frac{d\phi}{\pi/2}$. The attenuated intensity of solar irradiation reaching the receiver due to absorption coefficient ($\alpha$) of the waveguide material is dependent on the path length of the solar rays traversing through the waveguide. The path length ($\lambda$) of light
entering at \( r \) through \( dr \) is \( \lambda = \frac{r-R_{rec}}{\sin \phi} \). The total irradiation, \( I_t \) reaching a receiver through an edge can then be expressed as:

\[
I_t(2\pi r_t t) = \frac{I_0}{2\pi R_{rec} t} \int_{\phi=0}^{\phi=\pi} \int_{r=R_{rec}}^{r=R_{wg}} e^{-\alpha(r-R_{rec})/\sin \phi} \cdot dr \cdot d\phi
\]  

which, when integrated over the outer receiver radius \( R_{rec} \) and outer waveguide radius \( R_{wg} \) results in the following expression:

\[
I_t = \frac{2I_0}{\pi R_{rec} t} \int_{0}^{\frac{\pi}{2}} \left( \frac{\sin \phi}{\alpha} \right)^2 \left( 1 - e^{-R^*} \right) - \left( \frac{\sin \phi}{\alpha} \right) \left( R_{wg} e^{-R^*} - R_{rec} \right) \cdot d\phi
\]

where \( R^* = \frac{\alpha(R_{wg}-R_{rec})}{\sin \phi} \). Assuming all the incident irradiation energy are transferred as heat to the heat transfer fluid (HTF) in the HCE at a thermal efficiency of \( \eta_R \), the thermal power \( P_t \) delivered to the linear receiver per unit waveguide based on a simple energy balance follows:

\[
P_t = \dot{m}_F C_F (T_{F,\text{out}} - T_{F,\text{in}}) = N_{wg} \times I_t (2\pi R_{rec} t) \cdot \eta_R
\]

where \( N_{wg} \) is the number of waveguides, \( \dot{m}_F \) is the mass flow rate of the HTF, \( C_F \) is the specific heat of the HTF, \( T_{F,\text{in}} \) is the HTF inlet temperature and \( T_{F,\text{out}} \) is the HTF outlet temperature. The thermal efficiency of the receiver \( \eta_R \) is given by:

\[
\eta_R = \beta \tau - \frac{h(2R_{rec} + 2t) \times (T_{Rec} - T_{amb})}{I_t \times t}
\]

where \( \beta \) and \( \tau \) are the absorptivity and transmissivity of the receiver glass envelope, respectively; \( h \) is the convective heat transfer coefficient, \( T_{Rec} \) is the receiver glass envelope temperature, and \( T_{amb} \) is the ambient temperature.

The governing equation for steady state temperature distribution along the radius of the waveguide concentrator can be obtained from the conservation of energy principle applied to the differential control volume, \( dr \), in Fig. 12. Referring to Fig. 12, \(-k \frac{dT}{dr}\) is the heat transfer rate due to thermal conduction, \( k \) is the thermal conductivity of the material, and \( h \) is the convective heat transfer coefficient due to thermal interaction with ambient air. The appropriate form of the energy conservation equation is as follows:
\[ \frac{d^2T}{dr^2} + \frac{1}{r} \frac{dT}{dr} - \frac{2h}{kt} (T - T_{amb}) + \frac{\bar{g}'''}{k} = 0 \]  

(6)

where \( \bar{g}''' \) is the volumetric generation due to the irradiation absorption within the waveguide, which is assumed to be uniformly distributed within the volume of the waveguide:

\[ \bar{g}''' = \frac{l_0 \pi (R_{wg}^2 - R_{rec}^2) - l_t (2\pi R_{rec} t)}{\pi (R_{wg}^2 - R_{rec}^2) t} \]  

(7)

Solution to Eq. (6) can be obtained as a combination of general solution and particular solution leading to the following expression:

\[ \theta(r) = C_1 I_0(r^*) + C_2 K_0(r^*) + \frac{\bar{g}'''}{m^2 k} \]  

(8)

where \( r^* = m r \), \( m = \sqrt{\frac{2h}{kt}} \), \( \theta = T - T_{amb} \). \( I_0 \) and \( K_0 \) are zeroth order modified Bessel function of the first and second kind, respectively. \( C_1 \) and \( C_2 \) are the coefficients to be determined from appropriate boundary conditions. Since the waveguide concentrator is bonded to the glass envelope of the receiver, the temperature at the inner radius of the waveguide concentrator is the receiver glass envelope temperature as identified by Eq. (9a). The outer edge of the waveguide is an active convective tip. Nevertheless, since, \( R_{rec} \gg t \), an adiabatic boundary condition is assumed and a corrected outer radius of \( R_{rec} + t/2 \) is substituted in the resulting final expression to capture the effect of convective loss at the outer edge on the temperature distribution within the waveguide [63]. The boundary conditions can be mathematically expressed as:

\[ \theta(r = R_{rec}) = \theta_{rec} \]  

(9a)

\[ \frac{d\theta}{dr} (r = R_{wg}) = 0 \]  

(9b)

The evaluation of coefficients \( C_1 \) and \( C_2 \) require only two boundary conditions. Eqs. (9a) and (9b) are utilized to determine the coefficients. The acquired expression for the steady state temperature distribution along the radial of the planar waveguide concentrator in dimensionless form is:

\[ \frac{\theta(r)}{\theta_{rec}} = R_g + [1 - R_g] \left\{ \frac{K_1(r_0^*).I_0(r^*) + I_1(r_0^*).K_0(r^*)}{K_1(r_0^*).I_0(r^*) + I_1(r_0^*).K_0(r^*)} \right\} \]  

(10)
$I_1$ and $K_1$ are first order modified Bessel function of the first and second kind, respectively; and

$$R_g = \frac{\bar{g}'''}{m^2 k \theta_{Rec}}$$

where $\theta_{Rec} = T_{Rec} - T_{amb}$. A quick verification by substituting $\bar{g}''' = R_g = 0$ in the acquired solution, Eq. (10) verifies the validity of the solution as it reduces to the expression for temperature distribution in a radial fin as reported in Incropera et al. [63]

### 3.3 Simulation and parametric study for the radial waveguide

The thermal and optical analysis of the waveguide was conducted using TracePro. The lens array and waveguide were designed in CAD software with the required specification and then imported into TracePro for the analysis. The lens array and waveguide were assigned properties of Schott BK7 and Schott F2 glass respectively. The lens array and the waveguide were assembled as shown in Figure 13a. The rays would pass through the lens array and converge towards the scattering surface and then undergo total internal reflection as shown in figure 13b. The grating surface is made as shown in figure 13c and it consists of 6 reflecting surfaces. The angle included between the reflecting surface is kept constant at 120°. The source for the radiation is placed above the lens surface and the intensity and wavelength of the radiation is varied for the different studies. The rays have been traced for the various combinations of parameters and the collection efficiency at the center of the radial waveguide along with the total energy collected is recorded. The energy incident on the lens surface is kept constant at 1000 W/m². The decoupling loss which occurs when a ray undergoing total internal reflection is incident on the scattering surface and then escapes the waveguide is also recorded. The energy lost due to absorption while travelling through the waveguide material is also recorded.

### 3.4 Loss mechanisms associated with waveguides

The energy collected at the center of the radial waveguide is only a part of the energy incident on the lens surface. There are losses associated with every step from the light striking the lens surface to the point when the light strikes the energy absorbing surface. The Figure 14 shows the various loss mechanisms involved in energy collection through waveguides.
Figure 13. (a) Simulation setup for the radial waveguide (b) rays undergoing total internal reflection in the waveguide (c) cross section of the scattering surface

Figure 14. Waterfall chart showing the various loss mechanisms for solar energy concentration through a waveguide.

The first stage of energy loss is when the rays strike the lens surface. At this point, some of the energy is reflected back into the atmosphere. The extent of this loss can be limited by using antireflective coatings like MgF₂. A single layer of antireflective MgF₂ coating is used in this
case and it has limited the loss at this stage to 0.6% for each of the lens surfaces and since there are two lens surfaces, top and bottom, the total loss due to reflection at the lens surface is about 1.2%. There is a small amount of energy lost due to absorption in the lens material. This loss is usually insignificant as the rays travel for a very small distance through the lens which is roughly equal to the thickness of the lens array. The air gap between the lens surface and the waveguide, which is essential to make sure that the scattering surface coincides with the lens focus, leads to some loss due to absorption of the radiation in air. A considerable amount of energy losses occur due to absorption through the waveguide material. This loss will depend on the size of the waveguide which directly affects the distance the rays travel before they are absorbed by the energy absorbing surface at the center of the waveguide. The loss due to reflection at the top surface of the waveguide is limited by using the antireflective MgF$_2$ coating. There are two more loss mechanisms in which the rays escape the waveguide. First is the loss which occurs when the rays passing through the lens array are not focused on the scattering surface properly and directly pass through the waveguide. This loss is prominent when the either the waveguide is not aligned properly with the lens array or the scattering surface is small and all rays passing through the lenses cannot be focused on the scattering surface. The second prominent loss in which the rays escape the waveguide is the decoupling loss. It occurs when a ray undergoing total internal reflection and travelling through the waveguide strikes the scattering surface and then decouples as the angle of incidence becomes less than the critical angle and it passes through the waveguide surface and escapes. This loss increases as the size of the waveguide increases as the rays travel longer to reach the absorbing surface and because of this the probability of it undergoing decoupling increases. The size of the scattering surface has an effect on this loss. The probability of a ray undergoing decoupling increases as the size of the scattering surface increases and this leads to greater losses. Therefore, the size of the scattering surface has to be optimized as it has a great effect on the overall collection efficiency of the waveguide.

### 3.5 Cost model for the radial waveguide

The present work also incorporates a simple cost model to evaluate the optimal design configurations based on minimizing cost. The total cost of the system ($C_t$) comprises of both receiver and waveguide subsystems, which can be expressed in the form of cost per unit aperture area ($C''$) as:
\[ C'' = \frac{C_t}{N_{wg} \cdot \pi (R_o^2 - R_i^2)} = \frac{C_R' \cdot (2t + 2R_o)}{\pi (R_o^2 - R_i^2)} + \frac{\bar{C}_G \cdot \rho_G \cdot t}{c_{W_G}} \]  

where \( C_R' \) is the cost per unit width of the receiver ($/m), \( \bar{C}_G \) is the cost per unit mass of the waveguide material ($/kg), \( C_R'' \) is the receiver cost per unit aperture area ($/m^2), \( C_{W_G}'' \) is the waveguide cost per unit aperture area ($/m^2) and \( \rho_G \) is the density of the waveguide material. In addition to the cost per unit area, a levelized cost of power (LCOP) is also evaluated to identify configurations that yield cost reductions, while simultaneously maximizing the net thermal power delivered to the receiver. The levelized cost of power (LCOP) in $/W is expressed as:

\[
\text{LCOP} = \frac{C''}{\eta_c \cdot I_o} = \frac{C_R' \cdot (2t + 2R_o)}{P_t} + \frac{\bar{C}_G \cdot (\rho_G \cdot t) \cdot \pi (R_o^2 - R_i^2)}{P_t} \]

where \( \eta_c \) is the collection efficiency defined as \( \eta_c = \frac{P_t}{I_o \cdot \pi (R_o^2 - R_i^2)} \). I_o is the incident solar irradiance and \( P_t \) is the net power delivered to the receiver. The cost per unit length of the receiver (\( C_R \)) is taken as 150 $/m based on state-of-art Ultimate Trough evacuated tube receiver concept [64], the density of the waveguide material (ZK7) is tabulated in Table 1 and the cost of glass is taken to be 2.25 $/kg [65]. It is to be noted that the cost of support structures and pylons, drive systems, electronics and control, foundations, etc., which are a major part of parabolic trough collector cost [66] are negligible for waveguide-receiver system due to its compact, lightweight nature and possible non-tracking implementation. Hence, although the cost model introduced here is intended to be simple, it is a fairly accurate representation of large-scale planar waveguide concentrator-receiver system.

**Table 2.** Thermophysical properties of optical glass (ZK7) used in this study

<table>
<thead>
<tr>
<th>Properties</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density, ( \rho ) [kg/m^3]</td>
<td>2490.0</td>
</tr>
<tr>
<td>Thermal conductivity, ( k ) [W/m-K]</td>
<td>1.1</td>
</tr>
<tr>
<td>Thermal stress factor, ( \varphi ) [MPa/K]</td>
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</tr>
<tr>
<td>Absorption coefficient, ( \alpha ) [m^-1]</td>
<td>1.4</td>
</tr>
<tr>
<td>Transformation temperature, ( T_g ) [°C]</td>
<td>539.0</td>
</tr>
</tbody>
</table>
3.6 Analysis Methodology

The present study evaluates the influence of the two design or material parameters—waveguide thickness \((t)\) and radius \((R_{wg})\)—and the operating parameter namely, the incident irradiation \((I_0)\) on the steady-state dimensionless temperature distribution \((\theta)\) and net thermal power delivered \((P_t)\). The radius \((R_{wg})\) and thickness \((t)\) of optical glass waveguide, such as BK7 or ZK7, is varied between 0.1 m to 2 m, 3 mm to 50 mm, respectively, while the solar spectral averaged absorption coefficient is \(\sim 1.4 \text{ m}^{-1}\) [67,68]. Table 1 provides the thermophysical properties of ZK7 glass obtained from literature [67-69]. The incident solar irradiation \((I_0)\) is varied between 500 W/m\(^2\) to 1000 W/m\(^2\) to investigate radial waveguide designs for various geographical locations.

It is reasoned that the waveguide design will be limited by the maximum temperature within the waveguide material that occurs when the convective heat transfer between the waveguide and ambient is at its lowest. Accordingly, the convective heat transfer coefficient, \(h\), is chosen to be that of buoyancy-induced convection from a heated, horizontal rectangular surface to a quiescent ambient [63], which evaluates to \(\sim 2.5\) to \(5.0 \text{ W/m}^2\cdot\text{K}\). The receiver glass radius \((R_{wg})\) is assumed to be 40 mm, similar to that of commercially available Schott’s parabolic trough receivers [71,72]. As a result, the parametric range of receiver glass temperature is determined from the literature [71,72] that characterizes the trend and magnitude of receiver glass temperature as a function of heat transfer fluid temperature \((T_F)\) within the absorber tubes. Since the receiver glass envelope is essentially opaque to infra-red energy, all the thermal losses from the receiver must pass, via conduction, through the glass envelope. As a result, the radiant and convective thermal losses from the receiver are directly related to the glass envelope temperature \((T_{Rec})\), regardless of what happens inside the absorber tube.

Based on the data obtained from Price et al. [71] for air speed of 0 m/s and ambient temperature of 35 °C the correlation for the receiver glass envelope temperature \((T_{Rec} \text{ in °C})\) as a function of HTF temperature \((T_F \text{ in °C})\) inside the absorber tubes can be expressed as:

\[
T_{Rec}[^\circ \text{C}] = 51.96 \times 10^{-5} \cdot T_F^2 - 89.63 \times 10^{-3} \cdot T_F + 53.36
\]

(13)

For desalination applications, the required HTF temperature is \(\sim 100 \text{ °C}\) [53], corresponding to which the receiver temperature based on the correlation is 50 °C. With increase in wind speed,
the glass envelope temperature will decrease due to forced convection cooling and Ref. [71] also provides the characteristic curves for various wind speeds. However, based on the same rationale discussed above for the choice of minimum convective heat transfer coefficient value, the design of the waveguide concentrator will be limited by the maximum material temperature corresponding to quiescent (zero wind speed) conditions, which is of interest in this study.

3.7. Results and Discussion

3.7.1 Parametric Analysis based on Simulations

3.7.1.1 Effect of Grating size

The grating size affects the overall collection efficiency and hence the energy collected at the collection surface. Decoupling losses and the losses due to the rays passing through the waveguide are greatly affected by the grating size. The grating is square in shape and its cross section is as shown in Figure 13c. It consists of six reflecting surfaces and all of them have an included angle of 120°. For this study, the side of the square scattering surface is varied from 60 µm to 150 µm. The other parameters for the waveguide and lens array were kept constant. The waveguide radius was 60 mm and the receiver radius was 30 mm. The thickness of the waveguide was 5 mm, the focal length of the lens array was 19.09 mm and the pitch of the lens in the hexagonal lens array was kept constant at 4.5 mm. The graph in Figure 15a shows the variation of collection efficiency with grating dimension. It can be seen that the collection efficiency and the energy collected increases sharply from 60 µm to 78 µm. This is because of the fact that a huge percentage of rays pass through the waveguide without getting totally internally reflected at 60 µm. It can be seen in Figure 15b that there is a sharp decrease in the number of rays that pass through the waveguide when the grating size is increased from 60 µm to 78 µm. From 78 to 150 µm, the percentage of rays passing through the waveguide decreases at a lesser rate. It can also be seen from the same graph that the decoupling losses steadily increase as the grating size is increased from 60 to 150 µm. In the Figure 15c which shows the variation of the collection efficiency as the scattering dimension varies from 78 to 150 µm shows that the highest collection efficiency is achieved when the scattering dimension is 120 µm. For values higher than 120 µm, the value of the collection efficiency starts decreasing due to higher decoupling losses. The variation of absorption and decoupling loss is shown in Figure 15d. The
absorption loss is somewhat constant which is expected in this case since there is no variation in the dimensions of the waveguide which is the major parameter on which the absorption loss depends. The decoupling loss on the other hand increases steadily as the scattering dimension increases.

**Figure 15.** Effects of grating dimension on the (a) collection efficiency for the range 60 to 150 µm (b) percentage of rays collected (c) collection efficiency for the range 78 to 150 µm (d) absorption and decoupling losses

### 3.7.1.2 Effect of Waveguide Radius

For this study, the waveguide radius was varied from 100 mm to 500 mm. The other parameters for the waveguide and lens array were kept constant. The scattering surface dimension was 120 µm. The thickness of the waveguide was 5mm, the focal length of the lens array was 19.09 mm and the pitch of the lens in the hexagonal lens array was kept constant at 4.5 mm. The waveguide radius affects the collection efficiency and the energy collected in different
ways. The energy collected as can be seen from the graph in Figure 16 increases as the waveguide radius increases. This is due to the fact that the total energy incident on the waveguide surface increases as the area of the waveguide exposed to the sun increases. The collection efficiency decreases as the size of the waveguide increases because the rays have to travel for a longer distance to reach the collection surface and this leads to greater absorption and decoupling losses which can be seen from the graph in Figure 16.

Figure 16. Effects of waveguide radius on the collection efficiency, energy collected and absorption and decoupling losses

3.7.1.3 Effect of Receiver Radius

The receiver radius has a substantial effect on the efficiency and the energy collected by the waveguide. All parameter other than the receiver radius were kept constant. The pitch of the hexagonal lens array was kept constant at 2.38 mm, the waveguide thickness at 5 mm, waveguide radius at 570 mm, the scattering surface dimension at 120 µm and the focal length at 16.83 mm. The receiver radius was varied from 10 mm to 50 mm. The effect on the collection efficiency and the collected energy can be seen in the graph in Figures 17. Both plots follow a similar trend. They increase, first reach a maximum value and then decrease. This is due to competing forces. The absorption and decoupling losses decrease when the receiver radius is increased as the distance which the rays have to travel decreases. At the same time an increase in the receiver radius leads to a part of radiation being lost as they are incident on the part of the waveguide which is hollow.
Figure 17. Effects of receiver radius on the collection efficiency, energy collected and absorption and decoupling losses

3.7.1.4 Effect of Waveguide Thickness

The absorption loss leads to generation of heat in the waveguide and that in turn leads to temperature rise in the waveguide. The temperature of the waveguide cannot be allowed to rise unchecked as the material which is being used as the waveguide, BK7 in this case has a specified temperature range in which it can be used. Increasing the thickness of the waveguide distributes the heat generated in the waveguide to a greater volume and so, the rise in temperature is less. The effect of varying the waveguide thickness in the range on 1 to 5 mm is not very significant on the collection efficiency or the losses as can be seen from the plots in Figure 18. Thus the only reason for increasing the waveguide thickness is to keep the temperature in check and within the working limits of the material.

Figure 18. Effects of waveguide thickness on the collection efficiency, energy collected and absorption and decoupling losses

3.7.2 Parametric Analysis based on Analytical Model

The analytical model is first verified against the results from the ray trace simulations and the results have been plotted in Figure 19. The values of collection efficiency, energy collected and
the energy absorbed by the waveguide material were plotted for a case in which the waveguide radius was varied from 100 to 500 mm and all other parameters were kept constant. The collection efficiencies from the two were within 1.5% of each other and the energy collected values were almost same. The energy absorbed values predicted by the analytical model is higher than that being predicted in the simulations and that is because a few loss mechanisms like the energy lost due to reflection at the surfaces and absorption in the lens array is ignored to simplify the model. The over estimation of the energy absorbed by the waveguide material leads to safer design in terms of thermal stress and temperature limitations. This analytical model has been used for the parametric study and the design studies which will be discussed in this section.

![Graphs showing the variation of collection efficiency, energy collected and energy absorbed with waveguide radius from the analytical model and the TracePro simulations.]

**Figure 19.** Validation of the analytical model by plotting the variation of collection efficiency, energy collected and energy absorbed with waveguide radius from the analytical model and the TracePro simulations.
Figure 20 shows the spatial variation of the Temperature rise, ΔT for the different convective heat transfer coefficients, radius of waveguide, and thickness of waveguide and incident irradiation.

Figure 20. Influence of incident irradiance, waveguide thickness, convective heat transfer coefficient and waveguide radius on the spatial distribution of temperature rise within the radial waveguide

The temperature increases from the center of the waveguide to the ends and reaches a maximum value for a given set of parameters asymptotically. This is true in the case of low temperature applications in which the receiver temperature is kept at 50 °C. The cases in which the intensity of the incident irradiation is low, the temperature is highest at the central receiver and decreases towards the ends and stabilizes to a minimum temperature. This can be seen in the case where the irradiation is 500 W/m² and 600 W/m². In these cases, the effect of convective cooling over power the effect of energy absorption by the waveguide. The temperature rise is
less in case of thicker waveguides as the heat capacity of the waveguide increases because of the higher volume. The temperature rise is also dependent on the convective heat transfer coefficient and as the convective heat transfer coefficient is increased, the temperature rise decreases. In larger waveguides, even though the total incident irradiation is higher, its effect on the temperature rise is counteracted by the increase in heat capacity of the waveguide. The amount of energy absorbed the waveguide increases in larger waveguides due to the fact that the rays have to travel longer through the waveguide material to reach the absorbing surface. This leads to higher temperature rise in larger waveguides when compared to smaller waveguides.

Figure 21a shows the net thermal power delivered ($P_t$) to the receiver as a function of radius for various incident solar irradiation for low temperature applications ($T_F = 100 \, ^\circ\text{C}$). The thickness of waveguide is kept constant at 10 mm. As expected, $P_t$ increases with increase in incident solar irradiation. The trade-off between the exponential decay in totally internally reflected irradiation due to absorption and the increase in total incident irradiation with increase in radius results in the net delivered thermal power to increase asymptotically in Figure 21a.

Figure 21b shows the required aperture area per unit thermal power delivered ($A$), which is the ratio of waveguide area and net thermal power delivered ($P_t$). The required aperture increases with increase in radius due to increased absorption loss of the solar irradiation within the waveguide, despite the increase in collection area, which is also inferred from the asymptotic increase in $P_t$ in Fig. 21a. Therefore, there exists a maximum desired waveguide radius based on the asymptotic increase in $P_t$ and increase in aperture area ($A$). For constant radius, aperture requirement reduces with increase in incident irradiance. As observed in Fig. 21a, the net thermal power delivered approaches zero for $R_{wg} < 0.2 \, \text{m}$, due to the greater thermal losses from the receiver in comparison to the irradiation reaching the receiver, which ultimately leads to very high aperture requirements (Fig. 21b). This has a direct consequence on the collection efficiency also, which is analyzed in Fig. 21c. The collection efficiency drastically decreases for smaller radii ($R_{wg} < \sim 0.3 \, \text{m}$). This is attributed to the decrease in net thermal power delivered to the receiver (Fig. 21a) due to increased thermal losses from the receiver. Hence, there is an additional requirement of a minimum waveguide radius to realize high collection efficiencies (Fig. 21c) and low aperture area requirements (Fig. 21b).
Figure 21. Effects of incident solar irradiance \( (I_0) \) and waveguide radius \( (R_{wg}) \) on (a) net thermal power delivered, (b) aperture area, and (c) collection efficiency for HTF temperature of \( T_F = 100 \) °C. The values for waveguide thickness \( (t) \) and receiver diameter \( (D_G) \) are kept constant at 10 mm and 40 mm, respectively.

Figure 22a illustrates the maximum temperature difference between the waveguide made of optical glass ZK7 and ambient \( (\Delta T_{\text{max}}) \) for HTF temperature of 100 °C and an ambient temperature of \( T_{\text{amb}} = 30 \) °C. From known material properties, namely the absorption coefficient and thermal conductivity, the maximum temperature difference depicted in Fig. 22a is obtained as a function of radius and solar irradiation intensity, for a fixed thickness of 10 mm. Figure 22a shows that the maximum temperature difference \( (\Delta T_{\text{max}}) \) increases with increase in incident irradiation intensity. For a given solar irradiance, \( \Delta T_{\text{max}} \) increases and asymptotes to a constant
value with increase in radius due to the exponential attenuation of solar irradiation with increase in path length.

The maximum radius of the waveguide is limited by the critical stress limit which is 8 MPa. Figure 22b shows the variation of stress generated in the waveguide with the waveguide radius and thickness. The stress values increase with an increase in the waveguide radius because of higher temperature rise due to increased absorption in the waveguide. The increase in stress is at a lower rate in waveguides which have greater thickness. This is due to the slower temperature rise in thicker waveguides. Thus, for a given thickness, there is a maximum value up to which the waveguide radius can be increased.

Figure 22. (a) Effects of incident solar irradiance ($I_0$) and waveguide radius ($R_{wg}$) on maximum temperature difference between the waveguide and ambient air obtained for HTF temperature of $T_F = 100 ^\circ$C. The values for waveguide thickness ($t$) and receiver diameter ($D_G$) are kept constant at 10 mm and 40 mm, respectively. (b) Effect of waveguide radius ($R_{wg}$) and waveguide thickness ($t$) on the stress induced in the waveguide for a fixed incident solar irradiance ($I_0$) of 1000 W/m$^2$. The red line shows the maximum allowable stress which is 8 MPa.

Figure 23 shows how the maximum waveguide radius varies with variation in thickness and the incident solar irradiation. It is limited by the structural and thermal stresses. The thermally induced stress limits the radius value after a particular thickness and the increases in maximum radius with thickness becomes very slow after that. The maximum radius value is lower in case of higher irradiation value which is due to greater temperature rise which is associated with higher irradiation. The convective heat transfer coefficient also plays an important role in
deciding the maximum radius. Higher convective heat transfer coefficient allows for a larger waveguide as the temperature rise is less in this case.

Figure 23. (a) Effects of incident solar irradiance \( I_0 \) and waveguide thickness \( t \) on the maximum allowable waveguide radius \( R_{wg-max} \) for (a) when convective heat transfer coefficient \( h \) is 2.5 W/m\(^2\)K (b) when convective heat transfer coefficient \( h \) is 5 W/m\(^2\)K.

3.7.3 Economic Analysis

Figure 24a shows the cost per unit aperture area \( C' \) of the radial waveguide concentrator-receiver configuration as a function of radius for various thicknesses. It is observed that the cost per unit aperture area of the radial waveguide concentrator-receiver configuration decreases with increase in radius due to the inverse relation of \( C' \) with \( R_{wg} \) [Eq. (11)], while increasing with thickness due to increase in waveguide material. For a given incident irradiance, the minimum waveguide cost that satisfies both the thermal stress and structural constraints is observed for the maximum permissible waveguide radius, which varies with incident irradiance \( I_0 \), \( \Delta T_{max} \) and waveguide thickness \( t \).

Figure 24b depicts the variation of minimum waveguide cost per unit aperture area with waveguide thickness and incident irradiation for \( \Delta T_{max} = 80 \) °C. The corresponding optimal waveguide radius that yields the minimum waveguide cost is shown in Fig. 24c. The optimal waveguide radius that yields the minimum waveguide cost coincides with the maximum permissible waveguide span based on both thermal stress and structural considerations. For a given incident irradiation, the minimum cost initially decreases with increase in thickness, reaches a minimum before increasing with further increase in thickness (Fig. 24b). For smaller
thickness values, the receiver cost component—the first term in Eq. (11)—dominates, and $C''$ decreases with increase in thickness due to the subsequent increase in optimal radius (Fig. 24c).

**Figure 24.** Effects of waveguide thickness and radius on (a) cost per unit aperture area of the radial waveguide concentrator-receiver system. Influence of waveguide thickness and incident irradiation intensity on (b) minimum waveguide-receiver cost per unit aperture area and the corresponding (c) optimal waveguide radius obtained for $\Delta T_{\text{max}} = 80 \, ^\circ\text{C}$.

However, after a certain thickness value, the waveguide cost component—the second term in Eq. (11)—dominates, leading to an increase in the cost per unit aperture area of the receiver-concentrator configuration with increase in thickness. The optimal waveguide thickness that yields the lowest cost of the planar waveguide-receiver configuration is identified by the circular markers for $\Delta T_{\text{max}} = 80 \, ^\circ\text{C}$ in Fig. 24b, which is observed to decrease with increase in incident irradiation intensity. As the irradiance increases, the least cost per unit area is obtained for
thinner waveguides (Fig. 24b) to compensate for the restriction of maximum permissible waveguide radius to lower values (Fig. 24c) based on thermal stress and structural requirements. Therefore, as irradiance increases, the minimum C’’ steadily increases due to smaller values of optimal radius [Eq. (11)].

Figure 25a shows the levelized cost of power [Eq. (12)] as functions of thickness and radius for application temperature requirements of 100 °C, and incident irradiance of 1000 W/m². For smaller radius values, the LCOP is high due to low thermal power delivered (Fig. 21a). As the waveguide radius increases, the LCOP decreases due to the increase in thermal power delivered (Fig. 21a). With further increase in waveguide radius, the LCOP reaches a minimum (Fig. 25a) and then increases again due to the increase in cost [Eq. (12)], while the net thermal power delivered levels off (Fig. 21a). In Fig. 14a, it is also observed that the LCOP increases with increase in thickness due to the corresponding increase in cost with increase in material weight content of waveguide [Eq. (12)]. Figure 25b shows the LCOP as a function of radius and incident irradiance for a fixed thickness of 5 mm. It is seen that the LCOP increases significantly with a decrease in incident irradiance due to the decrease in thermal power delivered (Fig. 21a).

**Figure 25.** (a) Effect of waveguide thickness and radius on levelized cost of power. (b) Effect of waveguide radius and incident irradiance values (I₀) on levelized cost of power.

The values of radius and thickness that minimize the LCOP, while also satisfying the design constraints based on thermal and structural stress considerations, have been presented in Table 2. The optimal waveguide configuration that yields the least LCOP for different incident irradiance (pertaining to different geographical conditions) and convective heat transfer coefficients is
presented in the table. The application temperature requirements of $T_F = 100 \, ^\circ C$ corresponds to low temperature desalination [52]. The corresponding waveguide-receiver cost in $\$/m$^2$, aperture area required per unit thermal power delivered in m$^2$/W and collection efficiency are also shown in Table 2. The optimal waveguide concentrator configurations identified also satisfy the design requirements based on thermal and structural stress constraints.

**Table 2.** Preferred planar waveguide design based on minimum LCOP for $\Delta T_{max} = 80 \, ^\circ C$. The minimum values of the objective function are italicized and listed in bold face.

<table>
<thead>
<tr>
<th>Incident Irradiance, $I_0$ [W/m$^2$]</th>
<th>Radius, $R_{wg}$ [m]</th>
<th>Thickness, $t$ [mm]</th>
<th>Cost, $C''$ [$/m$^2$]</th>
<th>Levelized Cost of Power LCOP, $\frac{C''}{A}$ [$/W]</th>
<th>Aperture Area, $A$ [m$^2$/W]</th>
<th>Collection Efficiency, $\eta_c$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>A. Low temperature thermal desalination ($T_F = 100 , ^\circ C$)</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>600</td>
<td>0.5</td>
<td>10</td>
<td>252.09</td>
<td>1.077</td>
<td>0.0043</td>
<td>39.01</td>
</tr>
<tr>
<td>800</td>
<td>0.5</td>
<td>10</td>
<td>252.09</td>
<td>0.781</td>
<td>0.0031</td>
<td>40.35</td>
</tr>
<tr>
<td>1000</td>
<td>0.5</td>
<td>10</td>
<td>252.09</td>
<td>0.612</td>
<td>0.0024</td>
<td>41.16</td>
</tr>
<tr>
<td><strong>B. High temperature CST power generation ($T_F = 400 , ^\circ C$)</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>600</td>
<td>0.6</td>
<td>12.5</td>
<td>233.23</td>
<td>1.481</td>
<td>0.0063</td>
<td>26.25</td>
</tr>
<tr>
<td>800</td>
<td>0.6</td>
<td>12.5</td>
<td>233.23</td>
<td>0.988</td>
<td>0.0042</td>
<td>29.51</td>
</tr>
<tr>
<td>1000</td>
<td>0.6</td>
<td>12.5</td>
<td>233.23</td>
<td>0.741</td>
<td>0.0032</td>
<td>31.46</td>
</tr>
</tbody>
</table>

It can be seen from Fig. 26a that the minimum LCOP decreases with increase in irradiance due to the increase in net thermal power delivered, and correspondingly, the aperture area requirement also decreases. The optimal waveguide radius increases with increase in thickness up to a point and then starts decreasing. This is due to the stress constraints. The optimal radius is lower for higher incident irradiance. These trends can be seen in Fig. 26b. Fig. 26c shows the effect of convective heat transfer coefficient on the minimum LCOP. The minimum LCOP value reaches its lowest value at a smaller thickness for $h = 2.5 \, W/m^2K$ when compared to $h = 5 \, W/m^2K$. 
In general, for \( \Delta T_{\text{max}}^* = 80 ^\circ \text{C} \), the optimal design configuration with least LCOP is obtained for \( R_{\text{wg}} = 0.5 \) m and \( t = 10 \) mm for different irradiance values for low temperature applications and \( R_{\text{wg}} = 0.6 \) m and \( t = 12.5 \) mm for high temperature applications (Table 2) and the corresponding cost per unit area is 252 $/m^2$ for low temperature applications and 233 $/m^2$ for high temperature applications. The least LCOP decreases with increase in irradiance due to the increase in net thermal power delivered, and correspondingly, the aperture area requirement also decreases. Comparing Tables 2A and B, the least LCOP increases with increase in application temperature requirements due to increased thermal losses at the receiver that reduce the net thermal power delivered. For instance, the least LCOP obtained for \( I_0 = 1000 \) W/m\(^2\) increases from 0.612 $/W$ for \( T_F = 100 ^\circ \text{C} \) (Table 2A) to 0.741 $/W$ for \( T_F = 400 ^\circ \text{C} \) (Table 2B).
Concomitantly, the aperture area requirement also increases and the collection efficiency is observed to decrease. Overall, the planar waveguide design configuration with the least LCOP, smallest aperture area requirement and highest collection efficiency is obtained for applications that require low temperature and regions with high incident irradiance (Table 2).

### 3.8. Conclusions

An analytical closed-form solution was developed for the coupled optical and thermal transport of incident solar irradiation within a radial planar waveguide concentrator integrated to a central receiver. A systematic parametric study was conducted to establish the relationship between the various design and operating parameters on the system performance which is quantified in terms of net thermal power delivered to the receiver from the waveguide, collection efficiency and aperture area requirement.

The utility of results in guiding practical designs was demonstrated for a planar waveguide concentrator made of optical ZK7 glass. Feasible design and operational envelopes of the radial planar waveguide concentrator-receiver system are reported based on the structural (wind loading) and thermal (thermal stress, and maximum continuous operation temperature limit) constraints.

The results from the analytical model are backed up by the TracePro simulations. Parametric study is conducted using the ray trace software which gives clear trends about how the energy collected, collection efficiency, absorption and decoupling loss depend on the parameters like waveguide thickness, waveguide radius, receiver radius and the grating size.

A simple cost analysis was also developed and the preferred design configurations within the feasible regime that minimize the LCOP ($/W) of the disruptive, planar waveguide solar thermal concentrator for two example applications of low temperature desalination and concentrated solar thermal power generation are reported in Table 2.
Chapter 4. Conclusions and Future Work

The performance of the air-cooled air gap membrane distillation system is comparable to the water-cooled systems even though it has lower energy requirements. The modular design which has been proposed here helps in increasing the capacity of the system by adding more modules, when required. It also helps in better utilization of the thermal energy of the heated saline water by passing the water through multiple passes till the temperature of the water decreases to a point where it is not suitable for the process. The superhydrophobic surface helps in improving the flux. It also reduces the temperature drop for the saline water as it passes through the channel.

Of the parameters which were studied, the conductivity of the support mesh had a significant effect on the yield from the setup. The copper mesh which has a conductivity of about 401 W/mK resulted in the highest flux values followed by the Aluminum mesh and steel mesh. The plastic mesh resulted in about 30% lesser flux values. Increasing the air gap resulted in lower flux values. In series configuration, the hydrophobic surface with the small air gap works best and results in about three times more yield when compared to the single pass water-cooled system.

Further studies should be conducted to model and improve the yields for the air-cooled AGMD module. Long term performance of the module in natural environment where it is cooled by the ambient air should be studied. Effect of the structure of the saline channel on the yield can also be studied. Dimensionless form of the results should also be looked at in future. Using regression analysis on the collected data, equations can be developed which predict the flux values at different values of air gap, conductivity of mesh and saline feed temperature.

To make the desalination process fully sustainable, it is proposed that a solar energy concentration system be used to meet the thermal needs of the desalination system. Thus, an analytical model for a radial waveguide based solar concentration system was developed. An analytical closed-form solution for the waveguide was developed for the coupled optical and thermal transport of incident solar irradiation within a radial planar waveguide concentrator integrated to a central receiver. A systematic parametric study was conducted to establish the relationship between the various design and operating parameters on the system performance which is quantified in terms of net thermal power delivered to the receiver from the waveguide, collection efficiency and aperture area requirement.
Feasible design and operational limits for the radial planar waveguide concentrator-receiver system are reported based on the structural and thermal (thermal stress, and maximum continuous operation temperature limit) constraints.

A separate parametric study is conducted using the ray trace software which gives clear trends about how the energy collected, collection efficiency, absorption and decoupling loss depend on the parameters like waveguide thickness, waveguide radius, receiver radius and the grating size.

A cost model was also developed and the preferred design configurations that minimize the levelized cost of power ($/W) of the radial planar waveguide solar thermal concentrator for both low and high temperature applications are reported.

In future, endeavors should be made to manufacture the proposed radial waveguide and pair it initially with the AGMD module and later use it in other low and high temperature applications. A multi-variable optimization analysis using an optimization algorithm can be conducted for the different dimensions of the radial waveguide and the lens array. Studies should also be conducted to develop passive tracking techniques and apply it to this system. Other materials which have lower absorption coefficients and higher working temperature can also be explored to improve the collection efficiency of the system and decreases the LCOP.
Bibliography


