Design of a Fixed Boiler for Enhanced Solar Steam Generation

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ABSTRACT

While large-scale solar photovoltaic and thermal plants have been constructed, and small-scale photovoltaic installations are widespread, small-scale thermal plants are largely confined to water heating.

This thesis is concerned with a household-scale solar thermal plant producing mechanical power from a steam engine. This requires matching solar radiation capture onto the surface of a boiler in which water is heated to produce steam at a pressure (and hence temperature) which is useable in a practical steam engine.

The geometric problems of solar energy capture have been analysed to define the required multiple mirror system. Current analytical methods are inadequate to predict accurately the heat transfer within a simple vertical tube boiler, so a scaling law has been developed which would allow experimental sizing of a small model boiler and its scaling up to a production unit.

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1 LCDR, School of Aerospace, Civil & Mechanical Engineering. ZACM4450 Mechanical Engineering Project and Thesis.
DISCLAIMER

This thesis has been written in partial fulfillment for the requirements for the degree of Bachelor of Mechanical Engineering. It is the result of a period of research and analysis by the author while a student of the University of New South Wales. Views expressed do not represent the views of the University College, the University or the Australian Defence Force Academy.
NOMENCLATURE

\( a_s \)  
Solar azimuth angle (°)

\( h_s \)  
Solar hour angle (°)

\( h \)  
Height of mirror (m)

\( w \)  
Width of mirror (m)

\( q \)  
Local heat flux (W/m\(^2\))

\( p_{sat} \)  
Local saturation pressure (N/m\(^2\))

\( x \)  
Local vapour quality (Fraction vapour/ total by mass; 0 ≤ x ≤ 1)

\( m \)  
Mass flow rate (kg/s)

\( d_i \)  
Tube internal diameter (m)

\( c_{pL} \)  
Specific heat liquid (kJ/kg.K)

\( k_L \)  
Thermal conductivity liquid (W/m.K)

\( ACME \)  
School of Aerospace, Civil, and Mechanical Engineering

\( ADFA \)  
Australian Defence Force Academy

\( L \)  
Latitude (°)

\( R \)  
Radial spacing for mirror placement

\( A \)  
Azimuth spacing for mirror placement

\( I \)  
Extra-terrestrial solar radiation (W/m\(^2\))

\( K \)  
Local extinction coefficient

\( I_{b,N} \)  
Instantaneous beam solar radiation per unit length normal to the suns ray

\( C_a \)  
Clearness Number

\( C_s \)  
Cost of a solar energy system ($/kW.hr)

\( C_O \)  
Initial cost of the solar energy system equipment ($/m\(^2\))

\( Q \)  
Solar energy received per annum

\( T \)  
Expected life of the solar energy system (Years)

\( T_{wall} \)  
Local tube wall temp (K)

\( S \)  
Suppression boiling factor

\( F \)  
Two phase multiplier

\( R e_{tp} \)  
Reynolds number two phase

\( R e_L \)  
Reynolds number liquid

\( X_a \)  
Martinelli parameter

\( P r_L \)  
Prantl number liquid

\( \zeta \)  
Solar Declination (°)

\( \alpha \)  
Solar altitude angle (°)

\( \alpha_t \)  
Boiler height measure from the centre of the front mirror (m)

\( \eta_C \)  
Efficiency of the solar energy system

\( \alpha_{tp} \)  
Two phase flow boiling heat transfer coefficient

\( \alpha_{nb} \)  
Nucleate boiling heat transfer coefficient

\( \alpha_{cb} \)  
Two phase forced convection coefficient

\( \alpha_{FZ} \)  
Nucleate boiling heat transfer coefficient

\( \alpha_L \)  
Liquid phase convective heat transfer coefficient

\( \rho_G \)  
Density of saturated steam (kg/m\(^3\))

\( \rho_L \)  
Density of water (kg/m\(^3\))

\( \mu_G \)  
Viscosity of saturated steam (N.s/m\(^2\))

\( \mu_L \)  
Viscosity of water (N.s/m\(^2\))

\( \sigma \)  
Surface tension (N/m)
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I. Introduction

A. Background

The inevitable exhaustion of fossil fuels coupled with the adverse greenhouse effect has forced governments throughout the world to seek alternative energy sources, of which solar energy would seem an obvious choice. However, the requirement to efficiently capture the solar radiation has rendered the use of the available technologies out of reach of the rural community.

The National Science Foundation in testimony before the US Senate interior committee stated that “Solar energy is an essential inexhaustible source potentially capable of meeting a significant portion of the nation’s future energy needs with a minimum of adverse environmental consequences. . . . The indications are that solar energy is the most promising of the unconventional energy sources. . . .” The encouraging assessment of the potential of solar energy relies on overcoming considerable technical and economical boundaries.

Another challenge continually confronted by designers is that the distribution of solar radiation globally is geographically discriminatory. Australia rates as one of the prime locations for solar power generation (see Figure 1). Considering that the sunlight that falls on Australia each year is approximately 15,000 times that of the nation’s annual energy needs it should be inevitable that Australia will adopt this emergent technology.

Solar thermal energy technology is the capturing of solar flux for conversion to thermal energy in the form of heat, whereas photovoltaics convert solar flux directly into electricity. Solar thermal collectors are divided into three categories:

- Low temperature collectors- flat plate or black tube arrangements used primarily for swimming pool heating.
- Medium temperature collectors- flat plate type collectors used for the residential and commercial hot water production.
- High temperature collectors- utilise concentrated solar flux via a mirror or lens arrangement for the production of steam for power production.

It is the intention of this thesis to use the latter of the aforementioned collectors for the consideration of a household-scale solar thermal plant producing mechanical power from a steam engine.

![World Solar Energy Map](http://www.inforse.dk/europe/dieret/Solar/asolarirrad.gif)

Figure 1. World Solar Energy Map

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The integration of a steam engine into the system will convert thermal energy in the form of steam to the mechanical power required to generate electricity. A steam engine of acceptable thermal efficiency requires a relatively high pressure and therefore a relatively high temperature (when compared to that attainable from 1 square metre solar flux), and hence requires magnification/concentration of the solar flux.

The generation of electricity via the use of a steam engine offers good durability and maintenance with the absence of the exotic materials required for photovoltaic cells.

B. Aim

The aim of this thesis was to design a scaled fixed boiler model whose configuration was chosen as a theoretical optimum for enhanced steam generation utilising multiple steered plain mirrors, which simultaneously irradiate the boiler. Such a fixed boiler can be arranged in any orientation, and the size of plane mirrors can be chosen to give the most efficient output for a given boiler power.

C. Scope

The ability to calculate the quantity and quality of solar energy available at specific geographic location is fundamental when considering the design of any solar energy system. It is intended that the information collected will benefit the rural sector however due to the availability of data, all calculations conducted are based on data for the Canberra, Australia region. The exchange of energy from steam to physical power was considered using commercially available conventional data with some recommendation made within the text.

Figure 2. System Fundamentals
Figure 2 identifies the requirements to ensure the system operates as designed. The multiple solar tracking plane mirrors concentrate sunlight onto the surface of a boiler in which water is converted to steam that drives the power conversion subsystem, producing electricity or optionally shaft power.

However, the success or failure of the design is measured against the customer requirements, as per Fig. 3. Correctly identifying the power requirement, time of year required, number of hours required and the system location dictates the arrangement and orientation of the overall plant. The power required is a specific consideration of the boiler only, where as the time of the year required and number of hours power required per day are principle considerations for the geometric analysis and have a marginal influence on the boiler considerations.

The geometric analysis involves the calculation of the mirror arrangement dependant on the aforementioned variables. It is to be noted that the solar capture considerations are independent of the final boiler design, relying on the number of mirrors designated to attain the associated flux. The arrangement of mirrors and placement of the boiler relative to the mirrors and the terrain is decided during the geometric analysis. The site qualification requires sufficient beam normal radiation, which is the beam radiation that comes from the sun and passes through the planet's atmosphere without deviation and refraction.

Consequently, appropriate site locations are normally situated in arid to semi-arid regions of which Australia is recognized as a prime location for future development. A value for the available captured solar flux is fed back to the boiler design process to decide the feasibility of the design.

The flux available at a particular time of day and at a particular time of the year will vary and therefore the customers requirement for power will dictate the amount of mirrors allocated for a specific location. Consequently the calculation of solar flux is independent of the system design, i.e. if the customer demands 10 kW of power; this specifies the required boiler size and the solar flux required for that particular power. However the duration and time of year for which that power is required determines the size and arrangement mirror system to provide the calculated solar flux. Only with the interaction of the boiler and the captured solar flux is the customer power requirement achievable, realising the calculations for the amount of solar flux required is independent of the initial customer requirements but a function of the boiler.

Depending on the suitability/quantity of the flux, the boiler design process for the boiler will continue with scale modeling based on the scaling law derived within this thesis. When the model has fulfilled the scaled requirements of the customer the boiler construction and eventual placement continues.
Customer Requirements

- Power required.
- Number of hours power required per day.
- Time of year when power is required.
- Location where power is required.

Boiler Considerations

- Design boiler to match required power output.
- Scale model to ascertain solar flux required.

Geometric Analysis - Mirror System

- Longitude and latitude of mirrors.
- Location terrain.

Boiler Construction and Placement

- Boiler fulfils the power output.
- Power available at required times.

Figure 3. System Analysis
D. Objectives

The objective was to initially analyse boiler performance as a function of orientation, season, time of day and weather conditions. From these calculations, the best orientation for both boiler and mirrors was selected for a specified output. The requirement to analyse the scale dependency of the boiler led to investigation into the heat transfer of a vertical tube heated uniformly along its length. From this a scaling law was derived for application with model testing and full-scale boilers.

E. Limitations

The energy density of solar radiation incident on a surface depends on the angle between the normal to the surface and the solar ray and the local energy density associated with the solar ray – which is in turn affected by absorption, dispersion and diffusion of the atmosphere. All of this is dependant on the actual size of the mirrors required to generate the solar flux for the boiler surface. The generated usable solar flux determines the maximum size, maximum mass flow rate and therefore output of the boiler.

F. Summary

The culmination of this study was to provide a scaling law to allow the successful testing and trialing of a model fixed boiler specifically for the design conditions stated in the client brief. The availability of literature on the development of solar technology is plentiful, however few consider its use for small, single user applications. The literature review provided the basis of the theory for the establishment of a design that was modified to suit the specified design criteria. Understanding the sun-earth geometric relationship, methods of solar collection, thermal conversion and multi-phase heat transfer theory allowed for the successful completion of this thesis.

II. Solar Geometry

The purpose of this thesis was to establish a framework and survey existing knowledge that formed the foundation used for the design of the fixed boiler. The review focused on the requirements of a fixed boiler employing multiple steered mirrors for steam generation to allow for a 5 kW output. The initial development of the boiler was dependant on an understanding of the fundamentals of solar radiation and the application of this to efficiently capture and utilize the solar radiation. A thorough understanding of heat transfer, in particular forced convection boiling was required for the development of an accurate mathematical model.

A. Solar Geometric Considerations

Before considering the design and construction of the boiler, it was necessary to make allowance for the various position of the sun at different times of the day and differing seasons as a function of the geographic location. The variation of solar intensity caused by the inclination of the earth’s axis relative to the ecliptic plane of the Earth’s orbit is of great importance when considering the design of a fixed boiler. The angle between the ecliptic plane and the earth’s equatorial plane is 23.45° (See Fig. 2).
The angle between the earth-sun line and the plane through the equator is the Solar Declination, $\delta_s$.

$$\delta_s = 23.45^\circ \sin \left[ \frac{360(284 + n)}{365^\circ} \right]$$

(1)

where $n$ is the day number for the calendar year with January 1 being $n = 1$.

Solar azimuth angle $a_s$ and the solar altitude angle $\alpha$ are the terms required to calculate the sun's position on the celestial sphere.

$$\sin \alpha = \cos L \cos \delta_s \cos h_s + \sin L \sin \delta_s$$

(2)

Where $L$ is latitude and $h_s$ is $15^\circ \times$ (hours from local solar noon)

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Figure 4. Location of Ecliptic Plane

Figure 5. The Solar Azimuth Angle and the Solar Altitude Angle

---

The solar azimuth angle (Eq. 3) is measured clockwise on the horizontal plane, from the north pointing coordinate axis to the projection of the sun’s central ray.

\[
\sin a_s = \frac{\cos \delta \sin h_s}{\cos \alpha}
\]  

(3)

Using these equations 1, 2 and 3, values for consideration in future development of boiler orientation with regard to solar position are identified in Table 1.

<table>
<thead>
<tr>
<th>Date</th>
<th>Day Number</th>
<th>Declination Angle (ζs)</th>
<th>Canberr Latitude 8:00am (hs=60°)</th>
<th>Solar Noon (hs=0)</th>
</tr>
</thead>
<tbody>
<tr>
<td>30-Jan</td>
<td>30</td>
<td>-18.0</td>
<td>34.5 88.5</td>
<td>72.9 0</td>
</tr>
<tr>
<td>1-Mar</td>
<td>60</td>
<td>-8.4</td>
<td>29.2 79.0</td>
<td>63.2 0</td>
</tr>
<tr>
<td>31-Mar</td>
<td>90</td>
<td>3.2</td>
<td>22.1 68.9</td>
<td>51.6 0</td>
</tr>
<tr>
<td>29-Apr</td>
<td>120</td>
<td>14.1</td>
<td>14.8 60.3</td>
<td>40.7 0</td>
</tr>
<tr>
<td>29-Jun</td>
<td>150</td>
<td>21.5</td>
<td>9.7 54.8</td>
<td>33.3 0</td>
</tr>
<tr>
<td>28-Jul</td>
<td>210</td>
<td>18.7</td>
<td>8.5 53.6</td>
<td>31.6 0</td>
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<tr>
<td>27-Aug</td>
<td>240</td>
<td>9.3</td>
<td>11.7 56.9</td>
<td>36.2 0</td>
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<td>26-Sep</td>
<td>270</td>
<td>-2.2</td>
<td>18.1 64.0</td>
<td>45.6 0</td>
</tr>
<tr>
<td>27-Oct</td>
<td>300</td>
<td>-13.3</td>
<td>25.5 73.5</td>
<td>57.1 0</td>
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<td>26-Nov</td>
<td>330</td>
<td>-21.1</td>
<td>32.0 83.7</td>
<td>68.1 0</td>
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<td>26-Dec</td>
<td>360</td>
<td>-23.4</td>
<td>36.1 88.2</td>
<td>76.0 0</td>
</tr>
</tbody>
</table>

**Figure 6. Declination Angle Defined by Date**
Figures 7, 8 and 9 illustrate that the time of year along with the time of day has a significant input into the maximum collectable solar flux and because of these variations, it also affects the overall amount of solar concentration required to adequately heat the boiler surface.

The area of mirror when compared to the area of boiler is affected by the declination angle, solar altitude and solar azimuth. As the power requirement increases so does the solar flux required and therefore the number of mirrors increases to match the solar concentration demand. It is possible to operate at lower power, i.e. reduced solar flux, however when considering the operational arrangement of a steam engine the only variable is the mass flow rate, coupled with the inability to alter the steam quality with seriously affecting the operation of the whole steam plant. The reduction of solar flux would render the system insufficient to meet the customers requirement.

B. Solar Collection

The determining factor for optimum fixed mirror layout is the cosine efficiency of each mirror. This efficiency is dependant on the sun’s position and the location of each individual mirror relative to the boiler surface.

The efficiency of each mirror determines the size required and therefore the number of mirrors necessary to obtain the flux at the boiler surface.
The use of trigonometry (as per Fig. 9) has assisted with populating Table 2. Based on the figures it can be seen that the greater the distance from the normal of the boiler surface, the larger the area of mirror required as a fraction of the solar flux received.

The trend from Table 2 continues with the fraction of solar flux required per 1 square metre of mirrors ranging from 7% to maximum of 71%. To attain 71% of flux per square metre at 14 m differential east-west, the mirror is required to be 100m from the plane of the boiler surface.
Table 2. Fraction of Usable Solar Flux as a Function of Boiler and Mirror Position

<table>
<thead>
<tr>
<th>Height Differential $h$</th>
<th>North to South Differential $d$</th>
<th>East to West Differential $e$</th>
<th>Ground Distance $n$</th>
<th>Straight Line Length of ray $k$</th>
<th>Angle between boiler and ground $\theta$</th>
<th>$\Psi = \tan^{-1}(n/d)$</th>
<th>Fraction of 1 solar flux per m$^2$</th>
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<tbody>
<tr>
<td>1</td>
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<td>0.71</td>
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<td>4</td>
<td>2</td>
<td>4.47</td>
<td>4.58</td>
<td>77.40</td>
<td>48.19</td>
<td>0.67</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>4</td>
<td>5.66</td>
<td>5.74</td>
<td>79.98</td>
<td>54.74</td>
<td>0.58</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>6</td>
<td>7.21</td>
<td>7.28</td>
<td>82.10</td>
<td>60.98</td>
<td>0.49</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>8</td>
<td>8.94</td>
<td>9.00</td>
<td>83.62</td>
<td>65.91</td>
<td>0.41</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>10</td>
<td>10.77</td>
<td>10.82</td>
<td>84.70</td>
<td>69.63</td>
<td>0.35</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>12</td>
<td>12.65</td>
<td>12.69</td>
<td>85.48</td>
<td>72.45</td>
<td>0.30</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>14</td>
<td>14.56</td>
<td>14.59</td>
<td>86.07</td>
<td>74.64</td>
<td>0.26</td>
<td></td>
</tr>
</tbody>
</table>

Figure 10. East to West Differential Vs Percentage Solar Flux
One possible solution to overcome this shortfall is to place larger mirrors in the solar collection field. However, this resolution brings with it considerations of shadowing and blocking. Shadowing occurs at low sun angles when a mirror casts its shadow on a mirror located behind it. Therefore, not all the solar flux is reaching the boiler.

The amount of shadowing and blocking is a dependant on the mirror spacing, boiler height from the mirror plane and the sun’s angle (See Fig 12).

To minimize the effect of blocking reflected light and shading of mirrors a layout of optimised field position was developed.

\[ R = h(1.44 \cot \alpha_i - 1.094 + 3.068\alpha_i - 1.1256\alpha_i^2) \]  \hspace{1cm} (4)

and

\[ A = w(1.749 + 0.6396\alpha_i + \frac{0.2873}{\alpha_i - 0.04902}) \]  \hspace{1cm} (5)

---

7 Stine W. B., Geyer M., *Power from the Sun*, Figure 10.9
8 Dellin, T.A., Fish M.J. and Young C.L. *A Users Manual for DELSOL2.*
where \( h \) and \( w \) are the height and width of the mirror and \( \alpha \) is the boiler height measure from the centre of the front mirror.

![Radial Stagger Mirror Layout Pattern](image)

**Figure 13. Radial Stagger Mirror Layout Pattern**

III. Production of Mechanical Power

The conversion of the captured radiant energy to heat is the primary task of this thesis. However, before considering potential designs for the boiler, the temperature level required and the amount of converted energy must be known to match a conversion scheme to a specified output.

![Quasiturbine Steam Model](image)

**Figure 14. Quasiturbine Steam Model**

A. Engine Selection

To allow further calculation with regard to the required output of the boiler, a decision based on the estimations provided in figure 15, was made to utilise the Quasiturbine (Fig. 14) as the engine of choice. The Quasiturbine is a pressure driven, continuous torque and symmetrically deformable spinning wheel.
The available documentation supports the selection of this engine due to its expected efficiency (Fig. 16), its lightweight, its compact design, its low maintenance and relatively simple construction when compared to other products on the market.

![Quasiturbine Efficiency Comparisons](image1)

**Figure 15. Quasiturbine Efficiency Comparisons**

To facilitate the aforementioned engine, the steam pressure required is approximately 200psi (1.4 MPa). Therefore the temperature within the waterside is to be in the vicinity of 194°C (Fig. 16).

![Temperature Vs's Pressure](image2)

**Figure 16. Temperature/Pressure curve for saturated steam**

IV. Collector Design Considerations

The design of the solar collector/boiler is defined by its application. It is envisaged that the completed design will benefit the rural sector and therefore the design is to reflect that purpose. Table 3 shows the working

---

9 Saint- Hilaire G., “Quasiturbine Low RPM High Torque Pressure Driven Turbine for Top Efficiency Power Modulation”, Figure 3

temperatures for typical system available at present. To reach the required temperatures utilising the fixed concentrator arrangement, it is intended to increase the concentration ratio accordingly.

Table 3. Typical Temperature and Concentration Range of the Various Solar Thermal Collector Technologies

<table>
<thead>
<tr>
<th>Type of Collector</th>
<th>Concentration Ratio</th>
<th>Typical Working Temperature Range (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flat Plate Collector</td>
<td>1</td>
<td>≤70</td>
</tr>
<tr>
<td>High Efficiency Flat Plate Collector</td>
<td>1</td>
<td>60 - 120</td>
</tr>
<tr>
<td>Fixed Concentrator</td>
<td>3 - 5</td>
<td>100 - 150</td>
</tr>
<tr>
<td>Parabolic Trough Collector</td>
<td>10 - 50</td>
<td>150 - 350</td>
</tr>
<tr>
<td>Parabolic Dish Collector</td>
<td>200 - 500</td>
<td>250 - 700</td>
</tr>
<tr>
<td>Central Receiver</td>
<td>500 -&gt; 3000</td>
<td>500 -&gt; 1000</td>
</tr>
</tbody>
</table>

Because of their simple construction and minimal maintenance issue, the boiler design will follow that of a flat plate collector. In general, a flat plate collector consists of an absorber surface (usually a dark, thermally conductive surface), a trap for re-radiation losses from the absorber surface (glazing), a heat transfer medium and a form of insulation behind the absorber surface (Fig.17).

Figure 17. Cut Away of a Flat Plate Solar Collector.\(^\text{12}\)

Utilisation of a solar tracking system and correctly calculating number of mirrors necessary to match the solar flux requirement at the specified time of day will ensure the boiler output is adequate for the efficient use of the Quasiturbine.

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\(^{11}\) Goswami D.Y., Kreith F., Kreider J.F., Op cit, p. 82.

\(^{12}\) Ibid, p. 89
Figure 18: Performance of Typical Flat Plate Solar Collectors

Figure 18 illustrates typical performance for a number of different types of flat-plate collectors. This diagram does not consider the use of concentrated radiation and therefore relies on direct and scatter radiation flux for its heat source.

A. Black Body Radiation

Radiation, convection and conduction must be considered when calculating the thermal efficiency and performance of a solar collector system. Radiation is the dominant loss and can be greatly reduced by application of a selective coating. Figure 19 shows that black chrome, when integrated into a flat plate solar collector can greatly improve the efficiency.

Figure 19: Radiation as a Function of Temperature

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13 Stine W. B., Geyer M., Op cit, Figure 6.4
V. Further Solar Considerations

The value of terrestrial radiation differs from that of extra-terrestrial radiation. To calculate the value of terrestrial radiation as a function of extra-terrestrial radiation, historical data is required for analysis. As radiation passes through a transparent medium, such as the atmosphere, it decreases in intensity. Since the atmosphere consists of multiple components whose concentrations vary as a function of time and location, determining the consistency of the atmosphere for the estimation of terrestrial solar radiation is unreliable.

A. Estimation of Terrestrial Radiation

As extra-terrestrial solar radiation, $I$, enters the atmosphere a large percentage is reflected back into space, a part is absorbed by air and water vapour and a quantity is scattered by molecules of water vapour, aerosols and dust particles (Fig. 20).

![Figure 20. Breakdown as Solar Radiation Enters the Atmosphere](image)

The solar constant or total solar irradiance is the constant expressing the amount of solar radiation reaching the Earth from the sun, approximately 1370 watts per square meter. It is not, in fact, truly constant and variations are detectable.

The intensity of the solar radiation ($I$) is depleted once it enters the atmosphere at a rate dependant on the weather conditions and atmospheric make up for that particular global location. If $K$ is the local extinction coefficient of the atmosphere then $I_{b,N}$ is the instantaneous beam solar radiation per unit length normal to the suns ray.

Where:

$$I_{b,N} = C_n I e^{-k/sin\alpha} \quad (6)$$

And $C_n$ (Clearness Number) is a parameter to account for variations in local conditions from average sea level conditions.

---


B. Costing

Solar energy may be free, however the equipment required to capture and convert the energy into a workable medium is not. For solar energy to be a viable option in the energy market, it must be first economically feasible.

To calculate the cost of a solar energy system \( (C_s) \), neglecting the interest charges on capital,

\[
C_s = \frac{C_O}{Q T \eta_C} = \frac{\text{$/kW \cdot hr}}{}
\]

Where:
- \( C_O \) is initial cost of the solar energy system equipment ($/m^2);
- \( Q \) is the solar energy received per annum (W/m²);
- \( T \) is the expected life of the solar energy system (years);
- \( \eta_C \) is the efficiency of the solar energy system.

![System Cost vs Solar Flux](image)

**Figure 21. System Cost Compared with Available Solar Flux**

Figure 21 illustrates that installing the exact same solar energy system in a different location with a half the mean irradiance doubles the effective cost of the system.

VI. Steam Generation

Due to the complexity of the geometry and variation of the mirror angles, the calculations associated with the actual irradiated surface of any one of the boiler tubes is very complicated.

The complexity increases with the addition of the conduction problem in the solid materials of the tubes and any fins placed between them. Finally, there is the complication within the tube due to the inclusion of both single and two phase heating region within.

Therefore, the construction of an ANSYS model or the like, which would attempt to analyses the situation accurately is very difficult to create. An alternative is to derive a model and then scale up from that model, therefore justifying the requirement for a scaling law.

The biggest complication to arise in terms of scaling is the internal flow, i.e. the way in which the water is converted by the hot walls into steam, therefore accurately depicting the two-phase area. This problem dominates the available literature. Therefore I have produced an analysis based on the best available modelling of the two-phase flow. This data is only available for uniform heating however it is expected that the scaling will remain the same.
The same scaling law would apply for a simple heating regime, with an equivalent mix of external heat source, conduction and then internal convection (both single phase and two phase) that has been calculated by using the Chen correlation when compared to the real life complicated scenario.

The intention of the analysis was to attain the scaling effects or the scaling law for a simple heating model. The same model applies for the much more complicated calculation therefore allowing for the application of the simple model with an assumed mass flow rate and particular radiation pattern for analysis of the full size version.

It is worth noting that the actual calculation will be much more complicated than the simple model due to particular assumptions made, including of a constant solar flux and the backside of the tubes will not be perfectly insulated.

However the derived scaling law does not differ dramatically from that of the real life scenario because providing the mix is similar the scaling law should be approximately the same, considering the scaling law is dominated by the internal convection and should remain similar for the real life situation.

**VII. Heat Transfer Within a Vertical Tube**

The derivation of the scaling law required the consideration of a vertical tube heated uniformly over its length with a known heat flux and with sub cooled liquid entering at its base. This prescribed heat flux boundary condition for a one-dimensional situation is the simplest case and was calculated in the initial stages of the MATLAB code (in Appendix 1). However a thorough understanding and consideration was required prior to considering the more complex situations. At some predetermined mass flow rate the liquid is to be totally evaporated over the length of the tube.

![Figure 22. Conceptual Representation of Various Flow Patterns](image)

---

Figure 22 is a conceptual representation of the various flow patterns encountered over the tube length, the qualitative temperature profile and the corresponding heat transfer regions.

The liquid (water) enters at the base of the tube and whilst being heated to its saturation temperature, the wall temperature remains below the condition necessary for nucleation. At this stage heat transfer is single-phase convective heat transfer to the liquid (Region A). At some point along the tube, the condition adjacent to the wall allows the stable formation of vapor from wall nucleation sites. Initially, vapor formation takes place in the presence of sub-cooled liquid (Region B) and this heat transfer region is known as sub-cooled nucleate boiling. In the sub-cooled nucleate boiling region, (B), the wall temperature remains essentially constant at a few degrees above the saturation temperature, while the mean bulk fluid temperature is increasing to the saturation temperature.

The transition between regions B and C, the sub-cooled nucleate boiling region and the saturated nucleate boiling region, is clearly defined from the thermodynamic viewpoint. It is the point at which the liquid reaches the saturation temperature. Vapor generated in the sub-cooled region is present at the transition between regions B and C (x = 0); thus, some of the liquid must be sub-cooled to ensure that the liquid bulk (mixing cup) enthalpy equals that of saturated liquid. This effect occurs as a result of the transverse temperature profile in the liquid and the sub-cooled liquid flowing in the center of the channel will only reach the saturation temperature at some distance downstream of the point, x = 0. In the regions C to G, a variable characterizing the heat transfer mechanism is the thermodynamic mass "quality" (x) of the fluid.

![Figure 23. Variation of Heat Flux](image)

Figure 23 shows the various regions of two-phase heat transfer in forced convective boiling in terms of heat flux, steam quality and temperature.

As the quality increases in the saturated nucleate boiling region a point may be reached where transition in the heat transfer mechanism occurs. The transition occurs between the process of "boiling" and the process of "evaporation". This transition is preceded by a change in the flow pattern from bubbly or slug flow to annular flow (regions E and F). In the latter regions the thickness of the thin liquid film on the heating surface is often such that the effective thermal conductivity is sufficient to prevent the liquid in contact with the wall being

---

18 Corradini, Prof M. L., *Fundamentals of Multiphase Flow*, Fig. 6.2b
superheated to a temperature, which would allow bubble nucleation. Heat is carried away from the wall by forced convection in the film to the liquid film-vapor core interface, where evaporation occurs. Since nucleation may be completely suppressed, the heat transfer process may no longer be called "boiling". The region beyond the transition has been referred to as the two-phase forced convective region of heat transfer (Regions E and F).

A. Chen Correlation

There are numerous methods for predicting two-phase flow boiling heat transfer coefficient ($\alpha_{tp}$) in vertical tubes. However by definition:

$$\alpha_{tp} = \frac{q}{(T_{wall} - T_{sat})}$$

(8)

where $q$ is the local heat flux from the tube wall into the fluid, $T_{sat}$ is the local saturation temperature at the local saturation pressure $p_{sat}$ and $T_{wall}$ is the local tube wall temperature.

Flow boiling models, including the Chen correlation, consider two heat transfer mechanisms to be the basis for any derivation: nucleate boiling heat transfer ($\alpha_{nb}$) and the convective boiling heat transfer coefficient ($\alpha_{cb}$).

Chen proposed the first flow boiling correlation that received widespread acceptance. The correlation covers both the saturated nucleate boiling region $\alpha_{nb}$ and the two-phase forced convection region $\alpha_{cb}$.

It is assumed that both nucleation and convective mechanisms occur to some degree over the entire range of the correlation and that the contributions made by the two mechanisms are:

$$\alpha_{tp} = \alpha_{nb} + \alpha_{cb}$$

(9)

References:
20. Corradini, Op Cit, Fig. 6.2a
He concluded that the steeper temperature gradient in the liquid near the tube wall partially suppressed the nucleation of boiling site therefore minimizing its effect on nucleate boiling. He also considered that the velocity of the liquid is increased, when compared to the single-phase flow due to the vapor formed in the evaporation process. To account for this he derived the following expression to compensate for these effects on the two-phase flow boiling heat transfer coefficient.

\[ \alpha_p = \alpha_{FZ} S + \alpha_L F \]  \hspace{1cm} (10)

The equation of Forster and Zuber (1955)\(^{21}\), was taken as the basis for the evaluation of the nucleate boiling component,

\[ \alpha_{FZ} = 0.00122 \left[ \frac{k_L^{0.79} c_{pl}^{0.45} \rho_L^{0.49}}{\sigma^{0.5} \mu_L^{0.29} \rho_L^{0.24} \rho_G^{0.24}} \right] \Delta T_{sat}^{0.24} \Delta \rho_{sat}^{0.75} \]  \hspace{1cm} (11)

and \( S \) is a suppression boiling factor defined as the ratio of the mean superheat to the wall superheat and is represented as a function of the local two-phase Reynolds number:

\[ S = \frac{1}{1 + 0.00000253 \text{Re}_p^{1.17}} \]  \hspace{1cm} (12)

where \( \text{Re}_p = \text{Re}_L F^{1.25} \)

\[ \text{Figure 25. Suppression Boiling Factor Vs Reynolds Number}^{22} \]

\(^{21}\) Forster, H.K. and Zuber, N., *Dynamics of Vapour Bubble Growth and Boiling Heat Transfer*, p 535

\(^{22}\) Corradini, Op Cit, Fig. 6.7
The parameter $F$ is a two-phase multiplier and is a representation of the increase in the liquid phase convection due to the two-phase flow.

$$F = 2.35 \left( \frac{1}{X_n} + 0.213 \right)^{0.736}$$  \hspace{1cm} (13)

The Martinelli parameter $X_n$ is used to define the two phase effect on convection:

$$X_n = \left( \frac{1-x}{x} \right)^{0.9} \left( \frac{\rho_G}{\rho_L} \right)^{0.5} \left( \frac{\mu_L}{\mu_G} \right)^{0.1}$$  \hspace{1cm} (14)

When $1/X_n \leq 0.1$, the value for $F$ is equal to 1.

![Figure 26. Increase in the Liquid Phase Convection Due to the Two-Phase Flow](image)

The liquid phase convective contribution coefficient $\alpha_L$ is given by a modified Dittus Boelter correlation:

$$\alpha_L = 0.023 \text{Re}_L^{0.8} \text{Pr}_L^{0.4} \left( \frac{k_L}{d_L} \right)$$  \hspace{1cm} (15)

The liquid Reynolds number $\text{Re}_L$ is:

---

23 Corradini, Op Cit, Fig. 6.6
Where the value for the local vapor quality, \( x \), and the mass flow rate, \( \dot{m} \), and internal diameter of the tube, \( d_i \), are given. The liquid Prandtl number is defined as:

\[
Pr_L = \frac{c_p \mu_L}{k_L}
\]  

(17)

B. Derivation of the Scaling Law

Based on the Chen correlation and basic single-phase theory a MATLAB code was developed to calculate various values from given inputs. The initial code seemed rudimentary from the onset, however with the introduction of more variables became quite complicated. The code has been written specifically for the aforementioned criteria of saturated steam at 194°C and 200psi, however it has the capability to be expanded to allow for further development at other temperatures and pressures.

To allow for the code to model the vertical tube arrangement under various operating conditions, numerous variables are required to be determined by the user. These inputs include:

- Water Inlet Temperature;
- Solar Flux delivered;
- Tube Diameter (set for commercially available sizes);
- Delta T (the difference in wall temperatures);
- Mass Flow Rate.

By varying the mass flow rate and solar flux it was possible to calculate, by iteration the delta T required for each commercially available sized copper tube. Substituting in the calculated delta T the required overall tube length could be determined for a required temperature accounting for the single and two-phase regions.

By allotting an original mass flow rate of 0.0025 kg/s and a variation of heat flux ranging from 2500 W/m\(^2\) to 10000 W/m\(^2\) the developed MATLAB code populated Table 4.

<table>
<thead>
<tr>
<th>Mass Flow (kg/s)</th>
<th>Inlet Temp (°C)</th>
<th>Heat Flux (W/m(^2))</th>
<th>Delta T (°C)</th>
<th>Tube Diameter ID -&gt; OD (mm)</th>
<th>Overall Tube Length (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0025</td>
<td>20</td>
<td>10000</td>
<td>1.863</td>
<td>10.66-&gt;12.70</td>
<td>14.66</td>
</tr>
<tr>
<td>0.0025</td>
<td>20</td>
<td>7500</td>
<td>1.612</td>
<td>10.66-&gt;12.70</td>
<td>19.54</td>
</tr>
<tr>
<td>0.0025</td>
<td>20</td>
<td>5000</td>
<td>1.314</td>
<td>10.66-&gt;12.70</td>
<td>29.32</td>
</tr>
<tr>
<td>0.0025</td>
<td>20</td>
<td>2500</td>
<td>0.928</td>
<td>10.66-&gt;12.70</td>
<td>58.64</td>
</tr>
</tbody>
</table>

Table 4. Mass flow rate of 0.0025 kg/s illustrating variation of overall tube length.

<table>
<thead>
<tr>
<th>Mass Flow (kg/s)</th>
<th>Inlet Temp (°C)</th>
<th>Heat Flux (W/m(^2))</th>
<th>Delta T (°C)</th>
<th>Tube Diameter ID -&gt; OD (mm)</th>
<th>Overall Tube Length (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00125</td>
<td>20</td>
<td>10000</td>
<td>1.863</td>
<td>10.66-&gt;12.70</td>
<td>7.32</td>
</tr>
<tr>
<td>0.00125</td>
<td>20</td>
<td>7500</td>
<td>1.612</td>
<td>10.66-&gt;12.70</td>
<td>9.77</td>
</tr>
<tr>
<td>0.00125</td>
<td>20</td>
<td>5000</td>
<td>1.314</td>
<td>10.66-&gt;12.70</td>
<td>14.66</td>
</tr>
<tr>
<td>0.00125</td>
<td>20</td>
<td>2500</td>
<td>0.928</td>
<td>10.66-&gt;12.70</td>
<td>29.32</td>
</tr>
</tbody>
</table>

Table 5. Mass flow rate of 0.00125 kg/s illustrating variation of overall tube length.
When halving the mass flow rate whilst maintaining the same heat flux, it can be seen that the overall tube length is approximately halved as shown in Table 5. A variation in the inlet temperature had little to no effect on the overall tube length, for example by doubling the inlet temperature to 40 °C, therefore reducing the liquid temperature range from 194°C - 20°C to 194°C - 40°C found a shortening of the overall tube length of only 7mm.

Regardless of the specific heat reducing by the extra 20°C, the latent heat remains far greater than the specific heat and therefore it is a very small fraction of the overall heat transfer.

![Figure 27. Heat Flux Vs Overall Tube Length](image)

Tables 6-8 show commercially available tube diameters under varying mass flow rates with constant heat flux and therefore constant delta T. Figure 28 illustrates the collected data plotted on common axis’s.

**Table 6. Commercially Available Tube Diameters – Mass Flow Rate 0.00125 kg/s**

<table>
<thead>
<tr>
<th>Mass Flow (kg/s)</th>
<th>Inlet Temp (°C)</th>
<th>Heat Flux (W/m²)</th>
<th>Delta T (°C)</th>
<th>Tube Diameter (ID -&gt; OD (mm))</th>
<th>Overall Tube Length (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00125</td>
<td>20</td>
<td>10000</td>
<td>1.863</td>
<td>4.53-&gt;6.35</td>
<td>17.25</td>
</tr>
<tr>
<td>0.00125</td>
<td>20</td>
<td>10000</td>
<td>1.863</td>
<td>6.12-&gt;7.94</td>
<td>12.77</td>
</tr>
<tr>
<td>0.00125</td>
<td>20</td>
<td>10000</td>
<td>1.863</td>
<td>7.79-&gt;9.53</td>
<td>10.03</td>
</tr>
<tr>
<td>0.00125</td>
<td>20</td>
<td>10000</td>
<td>1.863</td>
<td>10.66-&gt;12.7</td>
<td>7.33</td>
</tr>
<tr>
<td>0.00125</td>
<td>20</td>
<td>10000</td>
<td>1.863</td>
<td>13.44-&gt;15.88</td>
<td>5.81</td>
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<td>20</td>
<td>10000</td>
<td>1.863</td>
<td>16.21-&gt;19.05</td>
<td>4.82</td>
</tr>
<tr>
<td>0.00125</td>
<td>20</td>
<td>10000</td>
<td>1.863</td>
<td>22.14-&gt;25.4</td>
<td>3.53</td>
</tr>
<tr>
<td>0.00125</td>
<td>20</td>
<td>10000</td>
<td>1.863</td>
<td>28.49-&gt;31.75</td>
<td>2.74</td>
</tr>
<tr>
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<td>20</td>
<td>10000</td>
<td>1.863</td>
<td>34.84-&gt;38.1</td>
<td>2.24</td>
</tr>
<tr>
<td>0.00125</td>
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<td>10000</td>
<td>1.863</td>
<td>47.54-&gt;50.8</td>
<td>1.64</td>
</tr>
<tr>
<td>0.00125</td>
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<td>10000</td>
<td>1.863</td>
<td>60.24-&gt;63.5</td>
<td>1.3</td>
</tr>
</tbody>
</table>
Table 7. Commercially Available Tube Diameters – Mass Flow Rate 0.0025 kg/s

<table>
<thead>
<tr>
<th>Mass Flow (kg/s)</th>
<th>Inlet Temp (°C)</th>
<th>Heat Flux (W/m²)</th>
<th>Delta T (°C)</th>
<th>Tube Diameter (ID -&gt; OD (mm))</th>
<th>Overall Tube Length (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0025</td>
<td>20</td>
<td>10000</td>
<td>1.863</td>
<td>4.53-&gt;6.35</td>
<td>34.49</td>
</tr>
<tr>
<td>0.0025</td>
<td>20</td>
<td>10000</td>
<td>1.863</td>
<td>6.12-&gt;7.94</td>
<td>25.53</td>
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<tr>
<td>0.0025</td>
<td>20</td>
<td>10000</td>
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<td>7.79-&gt;9.53</td>
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<td>10000</td>
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<td>14.66</td>
</tr>
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<td>10000</td>
<td>1.863</td>
<td>13.44-&gt;15.88</td>
<td>11.63</td>
</tr>
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<td>20</td>
<td>10000</td>
<td>1.863</td>
<td>16.21-&gt;19.05</td>
<td>9.64</td>
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<td>7.06</td>
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<td>28.49-&gt;31.75</td>
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<td>60.24-&gt;63.5</td>
<td>2.59</td>
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</tbody>
</table>

Table 8. Commercially Available Tube Diameters – Mass Flow Rate 0.005 kg/s

<table>
<thead>
<tr>
<th>Mass Flow (kg/s)</th>
<th>Inlet Temp (°C)</th>
<th>Heat Flux (W/m²)</th>
<th>Delta T (°C)</th>
<th>Tube Diameter (ID -&gt; OD (mm))</th>
<th>Overall Tube Length (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.005</td>
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<td>1.863</td>
<td>4.53-&gt;6.35</td>
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<td>6.12-&gt;7.94</td>
<td>51.07</td>
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<td>23.25</td>
</tr>
<tr>
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<td>10000</td>
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<td>60.24-&gt;63.5</td>
<td>5.19</td>
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</tbody>
</table>

The graph below clearly shows the linear trend of the overall tube length against tube diameter. This is highlighted for tube arrangement 5 (Inside diameter of 13.44mm and outside diameter of 23.25mm), at a mass flow rate of 0.00125 kg/s the required overall tube length to reach 194°C is 5.81 m. When compared to the larger mass flow rates, the overall tube length remains proportional to the increase in mass flow rate.

![Overall Tube Length Vs's Dia. for a Fixed Mass Flow and Flux](image)

Figure 28. Overall Tube Length Vs Diameter

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C. Assumptions Used in the Derivation Process

To allow for the derivation of the model and simplify the analysis of the scaling law particular assumptions were made. These included:

1. Uniform heat flux; and
2. Perfect insulation on the backside of the boiler tube.

The derived theoretical solution is not realistic in its application to the more complicated real life scenario, however the inclusion of these assumptions should not have an effect that changes the overall relationship of the scaling law. In fact, it is expected that the effect on the scaling law to be proportional to the real life more complicated scenario.

VIII. Summary

By identifying the customer requirements at the onset of the design process, through the course of my research I have been able to calculate the required mirror surface area and mirror arrangement to provide the concentration of solar flux required for the specific steam output throughout differing seasonal conditions.

The common component between the geometric analysis and derived scaling law is the usable solar flux. As the final product of the mirror field, the usable solar flux also defines the boiler dimensions and water mass flow rate to ensure efficient production of steam and therefore power.

Based primarily on the solar flux, the scaling law will allow for further testing via modeling. Once the accuracy of the scaling law has been confirmed the scaling law can be applied to the full-scale boiler. However the capability to produce the required steam and therefore the required power for various times of the year is the real obstacle confronted by designers. Correctly identifying the required flux for the required steam output is paramount in the design process. It is easy enough to say that you want 5 kW of power between 11 am and 1 pm during the summer months, however to generate the same power during the winter months will require far greater solar collectors and therefore increase the cost, potentially past the point of economic efficiency.

The initial solar tracking geometry derived from the simple mathematical problem was rudimentary; however a solid understanding was necessary to develop the geometry for application for the latter stages of the thesis.

The heat transfer investigation began with a simplified mathematical model, and progressed into a two-phase heat transfer problem.

It is envisaged that the application of the scaling law will ensure efficient design and manufacture of the boiler and mirror system.

IX. Conclusion

The investigation has proved to be a sizable challenge and has incorporated many elements of the mechanical engineering academic curriculum.

The final results of this investigation provide a start point to developing a detailed final solution to the design problem presented.

A. Further Work

Prior to manufacture of a prototype several elements of the design require further investigation.

Collect data for accurate modeling of the solar flux attainable at a particular geographic location, taking into account the season and expected weather conditions which can be primarily based on historical data.
Further development is required in the solar tracking mechanism of the flat plane solar heliostats.

Identify and incorporate an appropriate steam engine for integration into the system. The Quasiturbine discussed within the text may be a feasible consideration however its was chosen for this thesis as a pure theoretical model to allow further development of the scaling law. Once the engine of choice has been identified the MATLAB model will require altering to accommodate the different output temperatures and pressures.

The verification of the accuracy of the scaling law depends on the construction and testing of models at various sizes. Assuming the scaling law is correct, develop and build a full scale working boiler for a desired output. It is expected that they have the same scaling law and we can verify this by constructing models at various sizes.

X. Recommendations

The following elements derived from the above outlined recommendations could set the basis of several thesis subjects:

a. Development in the geometric component for solar tracking and efficient flux capture.
b. Design and refinement of plane mirror configuration.
c. Detailed design of the boiler.
d. Detailed design of the integration of the boiler, mirror arrangement and all ancillary subsystems.

XI. Acknowledgments

Thanks to members of the engineering faculty that have provided guidance and assisted in this project.

Thanks to Murat Tahtali for his assistance with development of the MATLAB component of the thesis.

Aristotle once said, “Those that know, do. Those that understand, teach.” I am fortunate that Alan Fien has both the knowledge to do and the understanding to teach.

This thesis may only have one name as the author however it was only possible with the support and sacrifice of my loving wife, Belinda.

XII. References


Corradini, Prof M. L., Fundamentals of Multiphase Flow,


