Why do steam power plants have to be very large?

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Abstract

With a lack of investment and limited technological advances, CST power generation has been unable to benefit from industry expertise and consequently failed to appreciate potential cost reductions. Present cost estimates dictate that the minimum economical size for CST plantations is 50MWe. Coincidently, almost 90% of concentrating solar thermal electricity generation is produced by plantations operating with capacities in excess of 50MWe [1]. Currently, Australian pre-existing energy markets are penetrable for smaller capacity CST designs with outputs ranging from 1MWe to 30MWe [2]. The aim of this study is to investigate the motives underpinning the lack of market drive and integration of downscaled CST power generation in Australia.

Research underpinning this investigation focuses primarily on Parabolic Trough and Solar Tower CST plant configurations due to the commercial maturity of each system. As a result, specific process efficiencies were investigated in order to establish prevalent loss mechanisms associated with capacity variations. Examination of the solar field components revealed distinct trends supporting downscaling, notably the collector efficiency of a heliostat field (Solar Tower configuration) shows likely improvement with smaller field size. However, the efficiency of integrated steam turbines was found to be largely dependent on the power block design capacity. Typical turbine efficiency can range between a low of 40% for small capacity, single-stage turbines to a high of up to 90% for large capacity, multi-stage, multi-valve condensing turbine [3].

This led to the conclusion that the major contributor to the deterrence of downscaling CST electricity generation is the reduction in the operating efficiency of the power block with plant size. The impact of this is a relative size increase in the thermal energy input into the system in order to compensate for conversion losses. To achieve a greater thermal input, a larger relative solar field is required; this additional cost presents a less attractive investment option. Furthermore, a review of cost trends highlighted further economic constraints imposed by the economies of scale effect in determining relative plant costs. Identification of these key limitations on downscaling CST power generation may subsequently drive further increases into solar investment within Australia and promote sustainable energy production.
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<tr>
<td>ARENA</td>
<td>Australian Renewable Energy Agency</td>
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<tr>
<td>AUD</td>
<td>Australian Dollars</td>
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<tr>
<td>CAPEX</td>
<td>Capital Expenditure</td>
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<td>CEPCI</td>
<td>Chemical Engineering Plant Cost Index</td>
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<td>CSP</td>
<td>Concentrating Solar Power</td>
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<td>CST</td>
<td>Concentrating Solar Thermal</td>
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<tr>
<td>CO₂</td>
<td>Carbon Dioxide</td>
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<tr>
<td>DNI</td>
<td>Direct Normal Irradiation</td>
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<td>GPSA</td>
<td>Gas Processors Suppliers Association</td>
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<td>GWe</td>
<td>Gigawatt (electric)</td>
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<td>HPT</td>
<td>High Pressure Turbine</td>
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<td>HTF</td>
<td>Heat Transfer Fluid</td>
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<td>IPT</td>
<td>Intermediate Pressure Turbine</td>
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<td>IRENA</td>
<td>International Renewable Energy Agency</td>
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<td>Kilopascal</td>
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<td>Kilowatt Hour (thermal)</td>
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<td>Levelised Cost of Energy</td>
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<td>Low Pressure Turbine</td>
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<td>Solar Photovoltaic</td>
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<td>Revolutions per Minute</td>
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<td>South Australia</td>
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1.0 Introduction

With the continual consumption of non-renewable resources in the energy industry and growing public environmental awareness, Australia is lagging behind the developed world in regard to switching to sustainable energy production [4]. Concentrating Solar Thermal (CST) technologies have the potential to provide a ‘green’ solution to this mounting problem. However with a lack of investment and limited technological advances, CST power generation has been unable to benefit from industry expertise and consequently failed to appreciate potential cost reductions [5].

Australia as a nation presents the perfect candidate for solar technology as it receives on average 58 million Petajoules (PJ) of solar radiation per year, presenting a competitive edge that should not be disregarded. In perspective, this equates to approximately 10,000 times the nation’s total energy consumption [6]. As a point of reference, it has been suggested that implementation of CST technologies in isolated or remote locations could serve to drive an increase into solar investment within Australia. [5] At present, pre-existing potential energy markets are penetrable for designs with output capacities ranging from 1MWe to 30MWe [2]. This form of implementation could serve to replace off-grid communities, private customers, and isolated commercial customers currently relying on expensive and environmentally damaging diesel electricity generators.

Currently, approximately 98% of all current CST power generation arises from the implementation of the Rankine thermodynamic cycle utilising high pressure steam as a ‘working fluid’. In contrast, almost 90% of measured output electricity is generated via plantations operating with capacities in excess of 50MWe, highlighting the apparent market bias in determining plant production size [1]. With further insight into the implications regarding the reduction of electricity generation capacities for CST power plants operating a steam Rankine cycle, solar renewables may present as a viable option to provide a replacement for existing non-renewable practices.
1.1 Purpose

The primary aim of the thesis topic is to investigate the inefficiencies and motives underpinning the lack of market drive and innovation in response to the proposal of downsizing CST steam power generation.

This is to be achieved through the exploration of relevant cost constraints and efficiency parameters concerning down-scaling CST steam power generation. A particular focus has been drawn to the limitations currently imposed on the steam power block and investigations into the effects of downscaling on turbine efficiency.

Furthermore, this document is intended to provide the reader with an analysis of the current cost structures for construction, operation, and maintenance of CST power plants. This information is intended to provide insight into current limitations and prompt recommendations that could act as preliminary concepts for future investigation and design.

1.2 Deliverables

A competent thesis will provide an understanding of the mechanisms limiting CST power stations to operating capacities in excess of 50MWe. It is anticipated that this information will assist in the development of a further insight and understanding into the operation of CST plantations with lower rated capacities. This has the potential to assist in advancing CST technologies into smaller markets and further Australia’s movement into environmentally responsible energy generation.

1.3 Scope

In determining the mechanisms governing CST efficiency and costing this report covers dominant influences of the major effects associated with reducing plant capacity. In order to achieve this, the primary focus will be on estimating the isentropic losses associated with CST technologies operating the Rankine cycle. A brief technology overview is provided, however the report focuses on two specific plant configurations being the Solar Tower and the Parabolic Trough systems due to commercial maturity.
It is assumed that the integration of small scale CST into the current electricity market will place plantations in locations local to the potential market demand. As a result, the associated impacts of electricity transmission have been excluded from the scope.

Specific modelling assumptions are outlined in section 5. The power block model assumes a parasitic loss correction method in order to account for the performance of heat input and extraction processes. This was based on a similar method employed by the System Advisor Model, and resulted in the exclusion of condenser variants (such as air and water cooling) from the study.

1.4 Report Structure

The report begins by introducing the reader to the fundamental technologies underpinning CST energy production. The CST electricity generation process is then subdivided into key processes, where the associated efficiencies are explored in order to predict and detail specific loss mechanisms intensified by a reduction in plantation operating capacity. The relevant cost-capacity relationships are then introduced, prior to the introduction of modelling methodologies. The results are then compiled, introducing estimated changes in both performance and relative cost for a reduced system size. This then allows conclusions to be drawn as to exactly why steam power plants have to be very large.
2.0 Literature Review

CST electricity generation arises from harnessing intermittent solar radiation. Australia as a nation presents as the perfect candidate for solar technology as it receives on average 58 million Petajoules (PJ) of solar radiation per year, equating to approximately 10,000 times the nation’s total energy consumption [7].

This form of renewable energy requires little to no consumption of fuel and as a result, has the potential to provide a ‘green’ alternative to conventional fossil fuelled systems. However with a lack of investment and limited technological advances, CST power generation has been unable to benefit from industry expertise and consequently failed to appreciate potential cost reductions [5].

The intention of the following section is to provide an overview of the potential for concentrating solar power (CSP) integration into Australian markets and the technology fundamentals governing solar thermal electricity production.

2.1 Radiation Profile and Solar Potential

Australia has the highest annual solar radiation per square metre of any continent [8]. A recent assessment by the Australian Department of Resources, Energy and Tourism concluded that solar technologies possess the greatest potential for renewable energy implementation in Australia. Additionally, the report acknowledges that solar resources are currently underutilised with the Australian Clean Energy Council (2015) reporting a mere 0.08% contribution by solar thermal operations to current renewable energy consumption in Australia [9].

Solar irradiation can be segmented into two prominent components, direct irradiation and diffuse irradiation. Solar direct normal irradiation (DNI) describes the component of energy resulting from the uninterrupted Sun’s direct beam measured on a plane normal to the beam trajectory. Diffuse irradiation represents the portion of solar energy that is dispersed by atmospheric particles, effectively reducing the magnitude of the incident sunlight [10]. CST collectors focus DNI beams and consequently do not utilise diffuse radiation.
North-west and central Australia possess the largest annual concentration of DNI resource as shown in figure 2.1; however these areas are typically isolated from the existing electricity grid network [11]. This limits the potential effectiveness of large scale electricity generation without extensive investment into transmission infrastructure. [8]

**Figure 2.1: Australian Annual Solar Radiation Profile, Source: [12]**

### 2.1.1 The Nature of Solar Energy

The amount of solar resource available at a given location is best described as intermittent [8]. Daily sun trajectories, weather patterns, and seasonal variations result in the amount of exposure at a given location to vary quite significantly. Figure 2.2 displays collected global solar exposure data from the Bureau of Meteorology for Nowra, Australia with latitude and longitude coordinates of 34.88°S and 150.6°E respectively [13]. It is worth noting that
the global solar exposure is not equivalent to the available DNI, it is however a strong indicator of DNI fluctuations [10].

![Daily Global Solar Exposure for location: Nowra, Australia](image)

_Figure 2.2 Daily Global Solar Exposure for location: Nowra, Australia, Source: [14]_

Although the available solar resource constantly varies throughout the day, the correlation between peak electricity demand and available resource allows solar thermal electricity production to provide electricity during peak demand times [8]. This timeframe can be lengthened through the implementation of thermal storage systems as discussed in section 2.2.3.

2.1.2 Potential CSP Market

The motivation behind downscaling CSP generation is to develop an economically viable alternative for small off-grid (<10MWe) and medium localised ‘mini-grid’ (>10MWe, <30MWe) locations within Australia.

A 2014 market research paper prepared by AECOM for ARENA, estimates the potential market size for Australian off-grid and mini-grid locations at over 1GWe [15]. This estimate includes a short-term, low penetration market size of approximately 200MWe and an additional high-penetration market size of 850MWe for CSP renewables worth a combined value of AUD $2 billion. A state-by-state market summary has been provided in figure 2.3. Based on this information, Western Australia has been identified as possessing the largest potential CSP market. This paper however does include estimates for the states of NSW, SA, Victoria or Tasmania due to confidentiality reasons.
However, based on findings by the clean energy council in [9], the excluded states are currently dominated by PV and wind energy renewables and hence should not greatly impact these findings. It also be noted that these Australian states, with the exception of SA, do not present with the most suitable solar resource (see figure 2.1).

![Figure 2.3: Estimated Market Size of off-grid renewables, Source: [15]](image)

### 2.2 Concentrating Solar Thermal Technologies

CST systems utilise solar irradiance to produce thermal energy. Lens-based concentrators and mirrored reflectors focus solar irradiance onto thermal receivers. This heat is transferred either directly to a thermodynamic working fluid or to a thermal retention medium for storage and later transferred to an appropriate working fluid. The heated working fluid can then be utilised in conventional power blocks to produce electricity [16].

CST can be categorised in accordance with the general arrangement utilised when harnessing solar radiation. Two fundamental configurations exist; these are classified as linear concentrators (Parabolic Trough and Linear Fresnel configurations) and point concentrators (Solar Tower and Parabolic Dish configurations) [17].
Each of these four configurations has achieved different industry establishment as a result of current operating sites and proof of concept design projects. Parabolic trough and solar tower systems currently possess the highest commercial maturity and as a result, are the focus of this study [11].

2.2.1 Parabolic Trough System

Parabolic trough configurations make use of curved mirrors and single axis rotation to track the sun’s daily trajectory. A parabolic trough solar field is modular, consisting of numerous parallel rows of concentrators aligned on a north-south axis, allowing east-west tracking along a horizontal axis as depicted in figure 2.4 [17]. The mirror design concentrates direct sunlight radiation on the parabola’s focal line where thermally efficient receiver tubes are mounted. In this configuration a heat transfer fluid (HTF), typically synthetic oil, circulates throughout the solar field. The energy contained in this medium is then either stored in a thermal storage facility or fed directly into a heat exchanger to generate steam [17].

![Parabolic Trough Configuration](image1)

Figure 2.4: Parabolic Trough Configuration (Left), Source: [18]. Parabolic Trough System at Jeffco jail, Colorado (Right), Source: [19]

2.2.2 Solar Tower System

Solar tower systems or central receivers; operate utilising an array of mirrors, known as heliostats that independently track the sun’s motion on dual axes providing both azimuth and altitude rotation. These heliostats redirect and concentrate incident sunlight onto a
central receiving tower where the harnessed thermal energy is capable of heating a working fluid to temperatures between 300°C and 1200°C [2].

The HTF is then either; stored, utilised in a heat exchanger to produce steam, or (providing steam is the HTF) directly used in a turbine generator to produce electricity. Figure 2.5 depicts a typical solar tower configuration.

Solar tower plantations have demonstrated the viability of incorporating the utilisation of alternate HTFs, namely molten salt, CO₂ and steam. Operating at a higher temperature than current trough systems, concentrating solar tower configurations offer a distinct advantage in implementing thermal storage systems as a lower quantity of HTF (usually molten salt) is required [20].

![Image](https://example.com/image.png)

*Figure 2.5: Solar Tower Configuration (LEFT), Source: [21], Crescent Dunes Solar Plant (Right), Source: [22]*

### 2.2.3 Thermal Storage

An inherent advantage of CST systems over photovoltaic (PV) electricity generation is the capability of integrating a storage system for collected thermal energy. Thermal energy storage (TES) systems typically work by incorporating a storing facility between the solar receiver and power block. This facility usually consists of two storage tanks which circulate a HTF (usually synthetic oil or molten salt) between a solar thermal receiver and a heat exchanger. Thermal energy absorbed by the HTF in the receiver is initially stored in a *Hot* tank before passing through a steam generator and entering the *Cold* tank. Figure 2.6 depicts an example of a solar tower system incorporating a TES system [11].
TES permits energy to be stored and released in periods of low or no solar activity, allowing the power block turbine(s) to operate for longer periods and at higher capacities. [17] Integrating TES systems into solar plants understandably drives an increase in capital costs as additional investment for tanks, pumps and piping components is required, however appropriately implementing thermal storage in CST power plants also enables surplus energy that exceeds the requirements of the power block to be stored for future consumption [11].

Currently, commercially available systems typically utilise molten salts with a heavily insulated dual storage tank configuration which have shown to be capable of achieving “overnight efficiencies up to 99%” [23]. Meaning energy can be stored for up to 12 hours without substantial losses.

![Figure 2.6: SolarTower Configuration with Molten Salt Thermal Storage Tanks, Source: [24]](image)

**2.3 Rankine Cycle**

Approximately 98% of all current CST power generation arises from the implementation of the Rankine thermodynamic cycle utilising high pressure steam [25]. Fundamentally, the Rankine cycle consists of four processes being; compression, heat addition, expansion (work extraction) and heat rejection. Additional reheat and regeneration processes can be
implemented in order to increase cycle efficiency and maximise output power generation [26].

2.3.1 Simple Ideal Rankine cycle

*Figure 2.7 (a)* demonstrates the typical configuration of a simple power block design for the operation of a Rankine cycle. The ideal Rankine cycle presents four fluid states, numbered 1-to-4 below.

![Simple Rankine cycle schematic](a) ![T-S Diagram for ideal Rankine cycle processes](b) Source: [26]

*Figure 2.7: (a) Simple Rankine cycle schematic and (b) T-S Diagram for ideal Rankine cycle processes. Source: [26]*

Figure 2.7 (b) represents a characteristic Temperature (T) – Entropy (S) diagram, highlighting processes, and fluid state changes in relation to the steam saturation curve (in black). The cycle initiates with saturated water entering the pump (state 1) where it is compressed to the appropriate operating boiler pressure (state 2). Thermal energy added in the boiler causes the water to transition into a superheat vapour (state 3). The superheated vapour is then expanded in a turbine where the steam is used to impart work on rotating shafts connected to an electrical generator (state 4). After expansion, the subsequent working fluid is typically a saturated liquid-vapour mixture. Residual heat is then extracted from the fluid through the condenser, returning it to the initial thermodynamic conditions (state 1) [26]. Thermal efficiency of the Rankine cycle can be determined from equation 2-1.
\[ \eta_{th} = \frac{W_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} \] (2-1)

Where \( \eta_{th} \) is the thermal efficiency, \( W_{net} \) is the net work produced by the cycle, \( q_{out} \) is the heat rejected in the condenser, and \( q_{in} \) represents the heat addition in the boiler.

### 2.3.2 Realisable Rankine Cycle

The actual realisable power generated through the Rankine cycle is not ideal. Processes experience irreversibilities, and hence cyclic energy losses are represented as entropy gains. Frictional effects, steam leakage and heat losses in major components drive the actual operation of the Rankine cycle to conditions portrayed in figure 2.8 (a). Figure 2.8 (b) depicts the deviation due to turbine and compressor irreversibility’s from the ideal Rankine cycle represented at fluid states 2 and 4; denoted by 2s and 4s for isentropic processes and 2a and 4a for actual observed phenomena [26].

![T-S Diagram highlighting (a) Actual power cycle deviation from the ideal Rankine cycle (b) irreversible processes induced by the turbine and pump, Source: [26]](image)

The cycle isentropic efficiency represents the actual work outputted by the cycle in relation to the theoretical isentropic work as represented by equation 2-2.

\[ \eta_s = \frac{W_{net,actual}}{W_{net,isentropic}} = \frac{W_{turbine,actual}-W_{compressor,actual}}{W_{turbine,isentropic}-W_{compressor,isentropic}} \] (2-2)
Where $\eta_s$ is the isentropic efficiency, $W_{net, actual}$ is the actual work outputted by the cycle, accounting for losses, and $W_{net, isentropic}$ is the maximum theoretical work attainable based on operating conditions.

### 2.3.3 Rankine Reheat Cycle

The Rankine reheat cycle operates by completing turbine expansion over two or more stages. High pressure steam is first expanded in a high pressure turbine (HPT) before undergoing a second heat addition process. This steam is then further expanded in a low pressure turbine (LPT) before completing the cycle. Additional stages can result through the inclusion of an intermediate pressure turbine(s) (IPT) and supplementary reheat piping. Figure 2.9 (a) depicts a basic schematic of a Rankine cycle with the introduction of a single reheat process.

Utilising reheat not only provides an efficiency increase, but also assists in limiting the moisture content in the turbine exit steam, improving turbine blade wear and mitigating blade erosion. This is shown clearly in the T-S diagram presented in figure 2.8 (b). [26]

![Figure 2.9: (a) Reheat Rankine cycle schematic, (b) T-S Diagram for ideal Rankine cycle with reheat processes, Source: [26]](image)

Like the simple Rankine cycles, reheat variants also suffer from irreversible processes which induce energy losses. However, by incorporating an additional heating and expansion processes, the average temperature at which thermal energy is supplied to the system increases, thus increasing the thermal efficiency. The number of cycle reheat stages is limited by the cost-benefit conditions surrounding each additional stage.
It has been shown that utilising one reheat stage can improve overall power cycle efficiency by up to 5 percent [26] however, this impact is diminished with each additional reheat added (approaching the relevant Carnot efficiency see section 3.3.1) to an extent that it no longer remains economically viable. Typical large scale power blocks (in the order of 500MWe) implementing a Rankine such as those used in coal-fired power plants cycle utilise two (2) reheating stages [27]. However, through an examination of existing available commercial turbines from providers such as Siemens, General Electric, and Ansaldo Energia it is evident that reheat turbine variants only become available at sizes exceeding approximately 40MWe. At capacities below this, it is inferred that the additional capital and operational costs exceed the benefit of an efficiency gain.

2.4 Steam Turbines

A Steam turbine is a mechanical module that converts the thermal potential energy from a flow of pressurised steam into electrical energy. Kinetic energy from the steam flow is imparted onto rotating blades within the turbine unit which rotates a shaft, providing conversion into mechanical energy [28]. In the case of electricity generation, the turbine shaft interfaces with a generator directly or via gear reductions [29]. Furthermore, turbines utilised to exclusively produce electricity operate at a near constant rotational speed of either 3000RPM or 3600RPM to drive a synchronous generator, as such, variable speed considerations have been excluded from this study.

Mechanical drive steam turbines can be classified according to their construction, number of extraction stages, steam flow conditions, and specific blading design [30]. The following section defines:

- Axial and Radial Turbines
- Single-stage and multi-stage turbines;
- Condensing and non-condensing turbines;
- Impulse and reaction blading; and
- Admission and extraction turbines.

2.4.1 Axial and Radial Turbines

A radial turbine, as the name suggests, is designed so that the steam flow occurs in the radial direction away or towards the central axis. Typically radial turbines are not suited to
applications involving electricity generation as compounding multiple stages for outputs exceeding 2MWe is not easily achieved [30].

Axial turbines operate by expanding steam down the central shaft axis. This application allows multiple stages and cylinders to be coupled, allowing for extraction to occur within practical limits as discussed in section 2.4.2.

2.4.2 Single-stage and Multi-stage Turbines

A single-stage turbine operates by accelerating steam through one cascade of stationary nozzles before being directed to rotating turbine blades to produce shaft work. Due to the large energy associated with the steam flow, the implementation of single stage extraction turbines is limited to approximately 2MWe [30]. At capacities larger than this, the flow velocity increment as a result of a single pressure drop phase forces the rotors to rotate at impractical design speeds (up to 30,000RPM) [31].

Multi-stage turbines allow the total energy extraction to occur over multiple stages as opposed to a single pressure drop, thus reducing the individual stage velocity to practical design limitations [31]. This configuration consists of compounded sets of nozzles and rotors that operate in series along a single shaft as shown in figure 2.10.

![Figure 2.10: General Electric A200 220MWe Multi-stage Axial Steam Turbine, Source: [32]](image)
2.4.3 Condensing and Non-condensing Turbines

The total energy available to a turbine for extraction is dependent on the operating pressure ratio and the temperature on the inlet flow [29]. Condensing turbines are defined as those with exhaust pressures below the surrounding atmospheric pressure, and thus present the highest overall pressure ratio [30]. This means that for a given output capacity, condensing turbines typically require the lowest steam flow rate [31]. It is for this reason that condensing turbines are fundamentally utilised in power generation cycles. As such, this paper will focus on efficiency scaling of condensing turbines only.

Non-condensing or backpressure turbines maintain exhaust conditions above atmospheric pressure. These turbine types are typically utilised in processing plants, where the exhaust steam is utilised in a production process after turbine extraction [30].

2.4.4 Impulse and Reaction Blading

Steam turbines are additionally categorised on the basis of the operational steam path design and can consist of two fundamental blade configurations; impulse and reaction. In an impulse turbine stage, the pressure drop throughout the expansion process takes place in the stationary nozzles only. By comparison, reaction turbines utilise an even pressure drop over both stationary nozzles and rotating rotors [33]. Figure 2.11 illustrates the comparative pressure and velocity components between the design variants. For set operating conditions (ie. steam conditions, power output, and rotational speed) reaction turbines typically require three (3) additional stages compared to the of an impulse design [30].
2.4.5 Admission and Extraction Turbines

Steam turbines can be further identified depending on how the steam is admitted or extracted at a point between the inlet and exhaust flows. This admission or extraction process can be either controlled or uncontrolled depending on the application and power requirement. [30] Figure 2.12 highlights the variation in admission and extraction processes.

Figure 2.11: Impulse and reaction turbine blade comparison, Source: [30]

Figure 2.12: Extraction and admission flow turbines, Source: [30]
3.0 Efficiency Contributions

The process of converting solar radiation to electricity utilising parabolic trough and solar tower configurations incurs significant energy losses due to the nature of the employed technology. Figure 3.1 presents a simplified process flow applicable to the examined technologies, with each sub-process contributing to the overall efficiency of the power plant.

![Diagram of solar energy conversion process](image)

*Figure 3.1: Fundamental CST Electricity Generation Process*

The purpose of this section is to identify and examine the specific loss mechanisms underpinning CSP production in order to identify limiting factors on downscaling plantation capacity.

3.1 Concentrator Efficiency

3.1.1 Parabolic Trough Field

The efficiency of a parabolic trough field can be estimated using separate loss factors defined by methods described in [34]. The subsequent field efficiency is given by equation 3-1.

\[
\eta_{field} = \gamma \eta_{shade} \eta_{end} \eta_{refl}
\]  

(3-1)

Where \( \eta_{shade}, \eta_{end}, \) and \( \eta_{refl} \) are the efficiency components for shade, trough end, and mirror reflectance losses respectively and \( \gamma \) represents the intercept factor. These factors are to be defined in the following section.

3.1.1.1 Field Optical Losses

Intercept Factor

The intercept factor is used to account for the direct solar radiation that does not reach the receiver tube. The intercept factor accounts for manufacturing imperfections in the mirror
reflectors, errors associated with the parabolic shape construction and focal point misalignment. The typical value of the intercept factor of a properly assembled reflector is approximately 0.95 [34].

**End Losses**
End losses exist when the angle of the direct radiation incidence on the collector is not equal to zero. At one end of the trough collector, radiation reflected by the fringes of the mirror deflects away from the receiver, consequently missing it. At the adjacent end of the collector, radiation fails to illuminate the extremities of the receiver tube [34].

**Shading Losses**
Parabolic trough field shading occurs at angles of low solar altitude, whereby the collector rows interfere with adjacent collectors. Shading is dictated by the distance between placed collector rows; at small distances shading losses are at maximum, larger distances between rows increase the land area requirement and imply further thermal losses by increasing the length of HTF piping [34].

**Reflectance Loss**
Reflectance loss occurs as the mirror surface is unable to reflect all of the incident solar flux that reaches its surface area. Factors such as reflector material, age and sediment build-up contribute to reflector losses. New, clean, low-absorption glass can achieve a reflectance as high as 94 percent; however 90 percent is reported as a more realistic value over the lifetime of a system [34].

**3.1.1.2 Field Optimisation**
Parabolic trough technologies present modular collector and receiver configurations. As such, there are minimal limitations imposed on field design and optimal collector arrangements assuming that the field retains a constant exposure to solar flux. Characteristically, efforts are made to ensure the collectors are aligned north to south, allowing for east to west tracking with the sun’s daily trajectory as mentioned in section 2.2.1.

The largest contributor to field optimisation is the perceived piping thermal losses between the collectors and power block (or thermal storage tanks) as discussed in section 3.2. Consequently, trough fields usually utilise a rectangular field configuration with a central power block as depicted in figure 3.2 [35].
3.1.2 Heliostat Field

The efficiency of a heliostat field can similarly be approximated through segmenting loss mechanisms and representing each loss in terms of efficiency. The scale of the individual heliostat should not drive significant changes to available tracking technologies as thus; for the purpose of this report tracking and geometric errors have been neglected [34]. As a result of neglecting tracking and geometric (misalignment) error for simplicity, the efficiency of a heliostat field can be determined by Equation 3-2.

$$\eta_{field} = \eta_{Cos} \eta_{Shadow} \eta_{Block} \eta_{Refl} \eta_{atten}$$ (3-2)

Where $\eta_{Cos}$, $\eta_{Shadow}$, $\eta_{Block}$, $\eta_{Refl}$, and $\eta_{atten}$ are the efficiency components for cosine, shadowing, blocking, mirror reflectance, and atmospheric attenuation losses respectively [34].

3.1.2.1 Field Optical Losses

Cosine Loss

The largest contributor to heliostat field efficiency optimisation is the loss induced by the ‘cosine effect’. This loss has geometric dependence on both the sun’s trajectory and the individual location of a subsequent heliostat relative to the receiving tower. Effective heliostat operation results in the tracking mechanism maintaining a normal surface which bisects the line of irradiance from the sun and a transmittance line between the heliostat and the receiver. As a result, the effective area of reflection is reduced by the cosine of half the angle required to achieve this [34]. Figure 3.3 depicts the heliostat cosine efficiency at various field locations averaged over a 12 month period at a site located in Barstow, CA.
is worth noting that field locations in the northern hemisphere (such as that depicted below) favour a northern orientated heliostat field to minimise cosine losses, while the opposite is true for locations in the southern hemisphere [37].

Figure 3.3: Annual average cosine efficiency at Barstow, CA, Source: [38]

**Shadowing and Blocking Loss**

Heliostats in close proximity interact, thus introducing shadowing and blocking processes which pose a negative influence on concentrator efficiency. Shadowing occurs at low tracking angles of the sun and describes the occurrence of heliostats casting shadows on one another. This reduces the solar flux that can be reflected and thus reduces the reflective area of the mirror. Blocking describes interference on reflected solar flux, preventing it from reaching the receiver. This occurs when a heliostat blocks the line of reflected solar irradiance of another, preventing it from reaching the receiver [34]. Both phenomena are depicted in figure 3.4.

Figure 3.4: Heliostat shadowing and blocking losses, Source: [34]
Atmospheric Attenuation Loss

Atmospheric attenuation losses account for the radiation loss incurred as the reflected solar flux travels a distance from an individual heliostat to the receiver. This places a limitation on heliostat field size and shape [34]. Figure 3.5 presents a model of the atmospheric attenuation at a specific site in Daggett, CA [39]. While the losses reported are site specific, the general trend remains applicable at alternative sites. It can be noted that the loss magnitude is dependent on the visibility; however the overall trend dictates that the attenuation losses increase with distance from the receiver.

![Atmospheric Attenuation Loss Graph](image)

*Figure 3.5: Atmospheric attenuation loss for Daggett, CA at varying visibilities, Source: [39]*

Reflectance Loss

Similar to trough collectors, heliostats also suffer from reflectance loss occurs as the mirror surface is unable to reflect all of the incident solar flux that reaches its surface area. As previously discussed in section 3.1.1.1, 90 percent is also an acceptable approximation for the reflectance of a heliostat mirror [34].

3.1.2.2 Field Optimisation

The optimal heliostat field shape is primarily dependent on the power capacity of the plant. For smaller systems, less than 100 MWth, single or multiple polar orientated fields have been found to be the most economical [40]. However, as thermal requirement increases, heliostats are placed further from the tower, increasing attenuation and cosine losses thus
reducing overall field efficiency; this is evident in figure 3.6. As a result, optimisation places heliostats to the east and west of the tower with the optimal field layout surrounding the tower at system sizes in the vicinity of 500 MWth as shown in figure 3.7 [34].

![Figure 3.6: Annular efficiency for an optimised heliostat field, Source: [41]](image)

![Figure 3.7: Optimal heliostat field shape defined by cosine and attenuation losses, Source: [40]](image)
3.2 Receiver Efficiency

3.2.1 Parabolic Trough

The efficiency of a parabolic trough receiver can be approximated using equation 3-3.

$$
\eta_{receiver} = \eta_{trans}\eta_{Abs}\eta_{Rad}\eta_{Pipe}
$$

(3-3)

Where $\eta_{trans}$, $\eta_{Abs}$, $\eta_{Rad}$, and $\eta_{Pipe}$ are the efficiency components for transmissibility, absorptivity, radiation and piping losses respectively [34]. Trough receiver tubes consist of an internal absorber encased in a vacuum within a glass outer casing. This is shown in Figure 3.8. The vacuum prevents conduction losses between the absorber and the surrounding atmosphere; however minor convective losses occur over the elevated surface temperature of the glass [34]. An overview of the relationship between heat losses and temperature associated with absorber tubing is presented in figure 3.9.

![Figure 3.8: Receiver tube schematic](Image)

**Figure 3.8: Receiver tube schematic, Source: [42]**

3.2.1.1 Absorber Tube Losses

Transmissivity of Glass

Transmissivity losses account for a reduction in solar flux as a result of the reflected beams passing through the outer glass casing. Current receiver tubes with anti-reflective coatings can achieve a transmittance as high as 96.5% [20].

Radiation Losses

Radiation loss occurs as a result of the temperature difference between the receiver tube and the ambient air. Radiation of trough and tower receivers follows identical principles, as such additional information can be found in section 3.2.2.1.
Absorptivity

Absorptivity is a measure of the effectiveness of the internal absorbance of the inner tube. Industry applications have achieved an absorptivity value of 91.5% [34].

Piping Losses

A significant factor in parabolic trough field optimisation is the minimisation of piping thermal losses. This is achieved by reducing the length of piping between the receivers and the power block. Thermal piping losses are further discussed in section 3.2.2.1.

![Figure 3.9: Absorber tube heat loss, Source: [43]](image)

3.2.2 Solar Tower

The similarly to a trough receiver, the efficiency of the solar tower receiver can be approximated using equation 3-4.

\[
\eta_{Receiver} = \eta_{Spill}\eta_{Abs}\eta_{Conv}\eta_{Rad}\eta_{Pipe}
\]  

(3-4)

Where \(\eta_{Spill}\), \(\eta_{Abs}\), \(\eta_{Conv}\), \(\eta_{Rad}\), and \(\eta_{Pipe}\) are the efficiency components for spillage, absorption, convection, radiation and piping losses respectively [34]. The aforementioned loss mechanisms are depicted in figure 3.10.
### 3.2.2.1 Receiver Losses

**Spillage Loss**
Spillage loss describes the portion of focused thermal radiation that does not contact the absorbing receiver surface area. Heliostat tracking errors, beam spread and focus area fundamentally impact the distribution of solar flux as it reaches the receiver and hence are directly attributable to spillage losses [34].

**Absorption Loss**
Absorption loss accounts for the ability of the coating on the absorber surface to transmit incoming concentrated irradiance to the HTF. Specially formulated coatings are capable of achieving an absorbance between 0.95 and 0.98 [34].

**Convection Heat Loss**
Convection loss is one of the primary contributors to overall performance of a tower receiver. Governed by equation 3-5, it can be noted that magnitude of convective losses is proportional to the receiver surface area and the ambient air temperature. Additional factors such as local wind speed and tower orientation can also increase convective losses [34]. Provided that the ambient temperature fluctuation is minimal and assuming the operating receiver temperature can be considered constant, the rate of convective loss should remain relatively constant for a given receiver.

\[
\dot{q}_{\text{conv}} = h_c A (T_s - T_a)
\]  

(3-5)

Where \( \dot{q}_{\text{conv}} \) is the rate of convective heat loss, \( h_c \) is the convective heat transfer coefficient, \( A \) is the surface area of the receiver, \( T_s \) is the receiver surface temperature and \( T_a \) is the ambient temperature [26].

**Radiation Heat Loss**
Along with convection, radiation thermal loss also represents a significant portion of total receiver inefficiency. Similarly, provided operating temperatures are considered constant, the radiation loss can be represented as constant for a given receiver size as illustrated by equation 3-6.

\[
\dot{q}_{\text{rad}} = \varepsilon \sigma T_s^4 A
\]  

(3-6)
Where $\dot{q}_{\text{rad}}$ is the rate of radiation heat loss, $\varepsilon$ is the emissivity constant of the material, $\sigma$ is the Stefan-Boltzmann Constant, $T_s$ is the receiver surface temperature and $A$ is the surface area of the receiver [26].

**Piping Loss**

Piping heat losses are a direct function of internal fluid temperature, ambient temperature and surface area of the piping. This includes losses from the piping in the tower and at ground level. However, with the use of insulation and through minimising length, current piping efficiency in solar tower configurations has been reported to lie between 99% and 99.9% [20].

![Solar tower receiver energy loss modes](image)

*Figure 3.10: Solar tower receiver energy loss modes, Source: [34]*

### 3.3 Power Block Efficiency

The efficiency of the power block depends on multiple factors including the operating conditions governing the thermodynamic cycle efficiency, the turbine efficiency, parasitic cycle loading and the effectiveness of the generator in converting mechanical shaft work into electricity [44]. The equation used to estimate the power block efficiency is provided as equation 3-7.

\[
\eta_{\text{power block}} = \eta_{\text{turb.net}}\eta_{\text{gen}}\eta_{\text{comp}}\eta_{\text{exch}}\eta_{\text{cond}}
\]  

(3-7)

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Where $\eta_{\text{turb, net}}$ is the net turbine efficiency (accounting for the cycle thermal efficiency), $\eta_{\text{gen}}$ is the efficiency of the generator, $\eta_{\text{comp}}$ is the pump compression efficiency, $\eta_{\text{exch}}$ is the efficiency associated with the heat exchanger process and $\eta_{\text{cond}}$ is the condenser efficiency.

### 3.3.1 Rankine Cycle Operating Conditions

Superheating steam to higher temperatures in the boiler or heat exchanger presents opportunities to increase cycle thermal efficiency. At higher boiler temperatures, additional heat is input into the cycle at a higher average temperature, increasing the efficiency [26]. This phenomenon is emphasized by the Carnot cycle efficiency, which is a representation of the highest thermal efficiency achievable by power cycles based on live steam operating temperatures. Equation 3-8 highlights the apparent relationship and effect of increasing the boiler temperature or decreasing the condenser temperature, resulting in an increase in thermal efficiency.

$$\eta_{\text{th, carnot}} = 1 - \frac{T_{\text{min}}}{T_{\text{max}}}$$ \hspace{1cm} (3-8)

Where $\eta_{\text{th, carnot}}$ is the Carnot thermal efficiency, $T_{\text{min}}$ is the minimum operating temperature of the condenser and $T_{\text{max}}$ is the maximum boiler operating temperature.

*Figure 3.11 (a)* presents a graphical representation of the additional work gained through increasing the temperature of steam in the boiler. However this efficiency gain is not without limitations, current metallurgical properties fix the maximum inlet temperature of the turbine to approximately 620°C [26]. Alternatively, efficiency gains can be achieved via increasing the operating pressure of the boiler, which consequently raises the boiling temperature of steam, hence increasing the average temperature at which additional heat is applied. *Figure 3.11 (b)* displays the T-S diagram showing the effects of increasing operating boiler pressure. While the overall net work increases, a consequence of the pressure increase is a resulting increase in the moisture content of steam exiting the turbine resulting in increased turbine blade wear [26].

Lowering the pressure at which condensation occurs can also provide efficiency increases. Lowering the pressure of the saturated mixture in the condenser effectively results in a temperature drop, lowering the temperature at which heat is rejected and hence improving efficiency [26]. Similar to increasing boiler pressure, reducing condenser pressure also
results in an increase in moisture content of the turbine exit steam as shown in figure 3.11 (c). In order to overcome the effects of moisture, reheat processes can be implemented as discussed in section 2.3.3.

![Figure 3.11: Methods of increasing Rankine cycle thermal efficiency. (a) Increasing operating temperature of boiler, (b) Increasing operating pressure of boiler, (c) Reducing operating pressure of condenser. Source: [26]](image)

### 3.3.2 Steam Turbine Efficiency

The efficiency of steam turbines is largely dependent on design capacity. Single-stage turbines as previously discuss are generally limited to a capacity of 2 MWe [45]. Under this threshold, the implementation of a single or multi-stage turbine is typically driven by economic considerations. For a given design capacity, single-stage turbines typically have a lower capital cost however require additional relative operating steam flow than that of a multi-stage turbine. This is as a result of lower efficiency achieved by the single-stage extraction process [3].

The role of the steam turbine is to maximise the use of available energy associated with pressurised steam flow. There are numerous loss mechanism that reduce the isentropic efficiency of operation including, throttling, leakage, friction, and bearing losses [45]. Performance efficiencies can range between a low of 40% for small capacity, single-stage turbines to a high of up to 90% for large capacity, multi-stage, multi-valve condensing turbine [3].
The efficiency of turbine can be represented as a combination of efficiency loss parameters in the form of equation (3-9).

\[ \eta_{turbine,net} = \eta_{isen}\eta_{mech}\eta_{adi}\eta_{th} \]  

(3-9)

Here, \( \eta_{isen} \) is the isentropic efficiency, which accounts for all losses excluding the mechanical losses which are subsequently represented by \( \eta_{mech} \). \( \eta_{adi} \) represents the loss associated with the turbine failing to operate in an adiabatic manner (heat losses not as a result of expansion and condensation). The cycle thermodynamic efficiency, denoted by \( \eta_{th} \), describes the system efficiency of an ideal Rankine cycle and is calculated utilising steam operating conditions by assuming that all components act ideally without incurring any losses.

Bhatt and Kajkumar in [46] report that the average mechanical and adiabatic efficiency for turbines between 30MWe and 500MWe remains almost constant between 99% and 99.9%. As a result, these factors do not warrant further investigation with regard to system performance variation with capacity. And hence equation 3-9 can be further simplified to equation 3-10.

\[ \eta_{net} = \eta_{isen}\eta_{th} \]  

(3-10)

### 3.3.2.1 Turbine Isentropic Losses

The overall efficiency of a steam power plant is primarily impacted by the performance of the turbine in extracting useful mechanical energy from a steam flow. The thermodynamic performance of a steam turbine is predominantly determined by the steam path during extraction and represented as stage losses in figure 3.12. Stage loss factors such as nozzle and bucket profile losses, secondary flow losses and steam leakage attribute to approximately 80-to-90 percent of total stage losses [47]. Figure 3.13 shows the distribution of losses over various stages of a condensing turbine.
Figure 3.12: Loss analysis of a condensing turbine, Source: [48]

Figure 3.13: Loss distribution of a condensing turbine, Source: [48]
**Tip Leakage**

Turbine tip clearance describes the gap between turbine rotors and casing as shown in figure 3.14. This clearance is mechanical necessary to ensure rotating blades have sufficient room to account for thermal expansion and exist as a result of unavoidable manufacturing tolerance. As a result, steam is allowed to escape the extraction path of the rotors, producing losses referred to as “Tip leakage loss” [49].

![Figure 3.14: Basic Schematic highlight tip leakage flow, Source: [50].](image)

The magnitude of the clearance gap varies with turbine design parameters and specific applications, however it has been reported that the gap between unshrouded blade tips and the casing in axial flow turbines is typically 1-2 percent of the blade span [51]. Building upon this, it has been suggested that a tip gap size equal to 1 percent of the blade span results in a stage efficiency reduction of approximate 2 percent [52].

**Profile Loss**

Turbine profile loss is the summation of frictional losses caused by boundary layer development along blade surfaces and induced mixing loss as a result of blade edge thickness [48]. Profile losses also include incidence loss, which occurs as a result of steam flow mismatching the turbine blades. These losses are generally difficult to model and are thus experimentally obtained using cascade tests.

**Secondary Flow Losses**

Secondary flow losses are produced by the unbalance between the pressure gradient as a result of turning of the main stream flow and the centrifugal force inside the end wall boundary layer [48]. Similarly to profile loss, secondary flow loss directly relates to factors such as blade profile, pitch, surface roughness, blade aspect ratio and turning angle. Cascade testing is typically used to obtain the magnitude of secondary flow losses.
**Moisture Loss**

Moisture loss occurs in stages that operate utilising saturated liquid-vapour steam and is comprised of supersaturation losses, condensation shock loss, water droplet acceleration loss, and braking loss due to water droplets interacting with the flow of steam vapour [48]. These typically occur in the low pressure turbine stages as heat is extracted from the steam and subsequent condensation occurs as shown in figure 3.13. Additionally, the presence of moisture accelerates turbine blade erosion and hence can increase profile losses if unmaintained.

**Exhaust Loss**

The exhaust loss describes the kinetic energy of the steam leaving the final turbine stage. This can be reduced by increasing the annulus area of the exhaust hood [45]. However, the increase in exhaust area is restricted by the strength of the moving turbine blade. The blade loading is a direct function of the continual rotation speed and hence practical limitations apply. Figure 3.14 depicts the relationship between the annulus area and the maximum allowable speed of long blade rotors. The solid line represents the existing limitations of current stainless steel materials, however with the development of titanium alloys and maraging steels, a larger annulus area may be achieved in the future [48].

![Figure 3.14: Relation between maximum allowable speed and annulus area, Source: [48]](image-url)
3.3.2.2 Estimating Turbine Isentropic Efficiency

Several methods currently exist in literature for predicting the operational performance of steam turbines in order to assist engineers in optimising turbine design. However, the majority of these are based upon calculating individual stage losses and hence require design based parameters which are typically commercial in confidence. Further to this, employing these methods over a range of turbine capacities would require the individual design and optimisation of multiple turbines with varying ratings. This requires specific industry experience and would further complicate the study; as a result simplistic predictive tools have been investigated in order to identify current technology trends.

A simplistic model by Bahadori & Vuthaluru (2010) for establishing the isentropic efficiency of a multistage, condensing turbine for capacities between 500kWe and 10MWe with varying inlet pressures is presented in figure 3.15. This model was produced utilising data from the GPSA Engineering Data Book [3] and was found to be valid between the examined capacities for turbine inlet pressures between 600kPa and 12100kPa. This image clearly shows the aforementioned efficiency reduction with turbine rating. This model reported an average deviation error of 1.37% when compared to data supplied by the Gas Processors Suppliers Association [45]. However, no information is provided in regard to additional operating parameters assumed.

![Figure 3.15: Basic isentropic efficiency of modelling of multi-valve, multi-stage condensing turbines in comparison to data from GPSA Engineering Data Book, Source: [45]](image-url)

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A report by IT Power (Australia) for the Australian Solar Institute titled ‘Realising the Potential of Concentrating Solar Power’ thoroughly explores the cost deviations with CST power cycle size. In estimating the power block efficiency for deriving plant costs, the report outlines a simple equation estimate (equation 3-11) based upon data provided by Sargent and Lundy (2003) and an investigation by Sun Lab [20].

\[
\text{Relative Efficiency} = (1 - 0.59e^{-0.06 \times \text{Capacity}})
\]  

(3-11)

Here the calculated efficiency is relative to that of a 100MWe rated plant and the capacity is input in MWe.

This estimate assumed a turbine net efficiency of 25% for a 1 MWe plantation, in comparison to a turbine net efficiency of 42.5% for a 100MWe unit [11]. The 1MWe case equates to an isentropic efficiency of approximately 62% for a condensing turbine operating with an inlet pressure of 8MPa and temperature of 540°C. This approximate isentropic efficiency gains further credibility when compared to Bahadori & Vuthaluru’s model which predicts the isentropic efficiency to be in the vicinity of 62% for a realistic inlet pressure of 8.1MPa. The study does however note that the approximations made for plant capacities under 10 MWe contain a high level of uncertainty.

![Figure 3.16: Relative Turbine Efficiency Plot of Equation 3-11](image)
Figure 3.16 shows the result of plotting equation (3-10). It is clear that an expected relative efficiency reduction is inferred as the relative turbine capacity decreases. Since the impact on the net operating efficiency of a power block is directly scalable with the isentropic efficiency for a set of constant steam operating parameters, the plot has been reproduced in figure 3.17. This has assumed a base case for a 100MWe unit operating with an industry accepted value for the isentropic efficiency of 90% [30].

Figure 3.17: Turbine Isentropic Efficiency Derived from Equation 3-11

Data sourced from various turbine manufacturer catalogues for a variety of capacities was then used to estimate the isentropic efficiency of each unit. Information presented in [46] for turbine ratings of 30MWe, 62.5MWe and 110MWe was included in the estimate. It should be noted that this included the design isentropic efficiency of the units examined in the study along with values as a result of testing during service life. Additional averaged approximates provided by Stine and Geyer in [34], and Elliot Group in [53] were included in order to compile the plotted data in figure 3.18. The fitted trend line produced equation 3-12 with a $R^2$ value of 0.8023.

$$\eta_{isen} = 0.3072 \times \text{Capacity}^{0.088} \quad (3-12)$$

Where $\eta_{isen}$ is the turbine isentropic efficiency and Capacity is the turbine rated capacity in kWe. This estimation produces an average deviation of 12.8% when compared to that of equation 3-11. However this estimate utilises a greater number of sources and additionally includes efficiency values of current commercially available designs. This estimate should only be considered to be valid between 1MWe and 120MWe, extrapolating for values
exceeding these limits produces isentropic efficiency values that contradict current technological capabilities.

![Figure 3.18: Turbine Isentropic Efficiency](image)

### 3.3.3 Auxiliary Power Block Efficiencies

The additional components employed in the Rankine cycle power block include the compressor, heat exchanger, and condensing unit with incorporated cooling tower.

**Compressor**

The isentropic efficiency of the compressor implemented in the Rankine bears little impact of the overall cycle performance. This is best highlighted by conducting a sensitivity analysis on a simple Rankine cycle, varying only the assumed value for compressor isentropic efficiency. The results of this analysis are presented in table 3.1, it is clear that the assumed compressor isentropic efficiency bears no significant effect on the overall power cycle efficiency. This is due to the size of the work imparted by the compressor in comparison to the extraction of work by the turbine.

*Table 3.1: Sensitivity Analysis of Compressor Isentropic Efficiency.*

<table>
<thead>
<tr>
<th>Power Block Parameter</th>
<th>( \eta_{isen} = 100% )</th>
<th>( \eta_{isen} = 90% )</th>
<th>( \eta_{isen} = 80% )</th>
<th>( \eta_{isen} = 70% )</th>
<th>( \eta_{isen} = 60% )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max Pressure</td>
<td>8MPa</td>
<td>8MPa</td>
<td>8MPa</td>
<td>8MPa</td>
<td>8MPa</td>
</tr>
<tr>
<td>Max Temp</td>
<td>540°C</td>
<td>540°C</td>
<td>540°C</td>
<td>540°C</td>
<td>540°C</td>
</tr>
<tr>
<td>Min Pressure</td>
<td>0.01MPa</td>
<td>0.01MPa</td>
<td>0.01MPa</td>
<td>0.01MPa</td>
<td>0.01MPa</td>
</tr>
<tr>
<td>Power Block Efficiency</td>
<td>40.03%</td>
<td>40.01%</td>
<td>39.99%</td>
<td>39.96%</td>
<td>39.93%</td>
</tr>
</tbody>
</table>
Generator
The efficiency of the generator unit, which converts the turbine shaft rotation into electricity, remains fairly constant independent of the power block rated capacity. This value was shown by Bhatt and Rajkumar (1999) to be between 97.5% and 98.5% for a range of turbines examined with rated capacities between 30MWe and 500MWe [46]. Assuming a constant value for generator efficiency between this range should serve as an appropriate measure for this study.

Heat Exchanger and Condenser
The heat exchanger and condenser serve to input and reject heat respectively within the operation of the Rankine cycle. Any efficiency loss occurring within these specific processes is compensated by an increase in the parasitic loading. For example, a low condensing efficiency can be compensated for by increasing the cooling medium flow rate within the condensing unit, this increases the power requirement. The same is true for imperfect heat conversion in the exchanger unit. The System Advisor Model (SAM) produced by NREL, utilises a 90% correction factor to account for such effects. In order to simplify the modelling process, the same method is to be employed, that is, the system power block outputs will be calculated based upon a rating scaled according to equation 3-13.

$$\text{Design Capacity} = \frac{\text{Output Capacity}}{0.9}$$  \hspace{1cm} (3-13)
3.4 Efficiency Trend Summary

Table 3.2 presents a summary of the anticipated effects on the performance of a CSP plantation for each of the examined sub-processes as a function of the rated capacity of the system. The cumulative performance change is examined in section 5.0.

Table 3.2: Summary of Expected Efficiency Trend with Reduction in Plantation Capacity.

<table>
<thead>
<tr>
<th>Plant Component</th>
<th>Efficiency Change</th>
<th>Rationale</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Solar Concentrators</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Parabolic Trough</td>
<td>Constant</td>
<td>Trough configurations feature a modular design, in that the removal of trough collectors (size reduction) has no impact on the optical efficiency of the field.</td>
</tr>
<tr>
<td>Solar Tower</td>
<td>Increase</td>
<td>Smaller capacity fields are optimised to reduce cosine and attenuation losses. As a result, the field configuration optimises from a circular field surrounding the central tower (larger scale), to a polar orientated field (small scale), realising an efficiency gain.</td>
</tr>
<tr>
<td><strong>Solar Receiver</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Parabolic Trough</td>
<td>Increase</td>
<td>While the performance increase is not expected to be large, smaller trough fields benefit from a reduction in HTF piping losses and hence should appreciate a slight performance increase.</td>
</tr>
<tr>
<td>Solar Tower</td>
<td>Increase</td>
<td>The governing loss mechanisms for a solar tower act as a function of the receiver area. Thus a smaller receiver area should provide a relative efficiency increase.</td>
</tr>
<tr>
<td><strong>Energy Storage</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TES System</td>
<td>Constant</td>
<td>TES presents an option to increase plant operating capacity factor and prolonging daily production hours. Current technology reports attainable efficiencies as high as 99% independent of system size.</td>
</tr>
<tr>
<td><strong>Power Block</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Turbine</td>
<td>Decrease</td>
<td>As presented in section 3.3.2, there is significant evidence to suggest that the turbine efficiency is heavily dependent on system size. Showing reductions in operating efficiency with plantation capacity.</td>
</tr>
<tr>
<td>Compressor</td>
<td>Constant</td>
<td>Assumed constant as the impact on power block performance is minimal (see 3.3.3).</td>
</tr>
<tr>
<td>Generator</td>
<td>Constant</td>
<td>Literature describing current technologies presents generator efficiencies between 97.5% and 98.5% independent of the examined system size.</td>
</tr>
<tr>
<td>Heat Exchanger &amp; Condenser</td>
<td>Constant</td>
<td>As described in section 3.3.3, the potential variation in heat exchanger and condenser efficiency is compensated for as part of the 90% correction factor, independent of power block rating.</td>
</tr>
</tbody>
</table>
4.0 Economic Cost Relations

In determining the feasibility of a downscaled CST facility, the prime consideration is the requirement for the energy production cost to be competitive with that of existing diesel electric generators. Gurgenci (2013) provides an estimated cost range for diesel-fuelled generators with capacities ranging for 5-10MWe employed within remote mining sites throughout Australia to be AUD 25-50 cents/kWh [5]. The inferred cost of electricity for CST plants is heavily reliant on plant specific factors as discussed in 4.1, however estimates by Fichtner (2010) place achievable electricity costs between AUD 23-31 cents/kWh for a large scale 100MWe parabolic trough system [54]. It can be safely inferred that this cost range does not apply to small scale systems, based solely on the lack of CSP implementation at these sites.

Diesel generation costs are primarily impacted by the current fuel price and have been shown to be quite volatile over the last 10 years. The increase in fuel prices have driven the average operating expense to increase by approximately 38% for fossil generation within the US [55].

CSP renewables benefit from little to no fuel requirements and hence operating expenses remain fixed, requiring consideration for personnel and maintenance only. However, renewables do exhibit large upfront costs and hence estimating the energy cost needs to be achieved by levelising capital expenditures over the operating life of the system.

4.1 Levelised Cost of Energy

A fundamental metric for assessing the economic performance of any energy asset is the Levelised Cost of Energy (LCOE). When implemented as a measure for CST electricity generation, the LCOE is derived from capital cost estimates, available solar resource, plant capacity factor, operation and maintenance (O&M) costs and the cost of financing capital investment [56]. The use of the LCOE allows comparative analysis between facilities with varying locations, capacity, technology and expected operation life. The formula for expressing the LCOE as defined by IRENA (2012) in [57] is given by equation 4.1.
\[ LCOE = \left( \frac{\sum_{t=1}^{n} I_t + M_t + F_t}{\sum_{t=1}^{n} E_t} \right) \frac{1}{(1 + r)^t} \]

Where:

LCOE = Levelised cost of electricity production averaged over the operational lifetime;

\( I_t \) = Investment expenditures in the year \( t \);

\( M_t \) = Operating and Maintenance expenditures in the year \( t \);

\( F_t \) = Fuel expenditures in the year \( t \);

\( E_t \) = Electricity generation in the year \( t \);

\( r \) = discount rate; and

\( n \) = operational life of the system.

Figure 4.1 displays an estimated LCOE breakdown for a 100MWe CSP plantation in South Africa provided by Fichtner (2010). Trough and Tower configurations implementing thermal energy storage were considered in the analysis, where the results clearly indicate that the major driving factor in LCOE estimates is the annualised capital expenditure with 84% of the LCOE derived from these estimates. Operating and maintenance costs including that associated with staffing then make up the remainder with a small influence of 1% associated with operation consumables.
4.2 Capital Expenditure

The capital expenditure component breakdown for parabolic trough and solar tower field configurations is presented in figure 4.2 for a capacity rating of 100MW based on the same model presented in figure 4.1. These plants were shown to have a similar total capital investment cost, with the trough system totalling USD 914 million and the corresponding tower estimate was USD 978 million [54]. However, the cost proportion breakdown for each of the estimated cost categories yielded significant variations respectively.

In considering the trough configuration, the solar field typically reflects the highest proportion of total system installation costs ranging between 35% and 49% [57]. However, the overall cost breakdown for trough technologies varies significantly with dependence on the size of the TES system. The TES cost proportion has been reported to vary between 9% for an implemented capacity of 4.5 hours to approximately 20% for systems featuring a capacity of 13.4 hours [57], [58]. Due to the maturity of the technology and the confidential nature of commercially realised installed costs, there is a lack of data for estimating reliable cost estimates [58].

The key difference in the estimation of the cost proportion between the technologies occurs when considering the TES capacity. Solar tower systems, while reporting similar total costs to that of the parabolic trough configuration, have been shown by Hinkley (2011) in [58] to be achieve a TES installed cost proportion of approximately half that for a trough system. This has been attributed to the larger temperature differential obtainable by tower systems [57].
4.3 Operation and Maintenance Costs

Operational and Maintenance (O&M) costs of CST systems are significantly smaller than that of conventional fossil fuelled power plants and combustion generators, however, as illustrated in figure 4.1, still account for a significant proportion of the LCOE [57].

Three key practices influencing the O&M costs of a CST system were identified by IRENA (2012) being; the replacement of broken concentrator mirrors; the cost of removing sediment build-up from the concentrators, inclusive of the water cost; and the insurance costs associated with the operation of a CST facility.

A report by SunShot (2012) estimates the average O&M to be approximately 2.9 US Cents/kWh with predictions made suggesting a further decline to 1 US Cent/kWh by 2020 [56]. The main drivers behind this expected cost reduction are founded by estimated increases in the plant operating capacity factor and reductions induced by increasing plant sizes.

Labour costs associated with the O&M of a CST plant also see a decline with the increase in plantation capacity, as staffing is shared across multiple units [20]. The opposite of the above relations can be assumed for smaller sized systems. Fichtner (2010) reports that a 50MW plant would see an increase of up to 7% in the total O&M costings when compared to that of a 100MW facility [54] see figure 4.3.

An additional consideration in determining the relevant labour costs is the location of the facility which dictates the available labour force market. IRENA (2012) states: “Personnel
costs for a 100 MW parabolic trough plant in the United States would account for 45% of the total O&M costs, while it is 23% of the total costs in the proposed South African plant“ in [57]. This perfectly illustrates the locational variations inferred on the total O&M costs.

Figure 4.3: Operation and Maintenance cost for parabolic trough and solar tower systems, Source: [57]
5.0 Model of Reduced System Size

A model was created in Microsoft Excel in order to provide an estimate of the efficiency and cost variation when scaling CST solar plants. The effective estimates are calculated based on the findings in literature and detailed investigations presented in chapters 2, 3 and 4 of this document.

The overall plant model itself is constructed around the specific power block inputs as illustrated in section 5.1. From the defined power block system, the average site specific solar resource and optional implemented thermal storage system size is used to approximate the required solar field size in square meters for both solar tower and parabolic trough configurations, see section 5.2.

Cost estimates are then drawn from the modelled plant, enabling the estimation of capital expenditure (CAPEX), operation and maintenance costs (O&M) and hence by assuming an operational life and discount rate, the relative LCOE is then calculated, see section 5.3.

This information can then be compiled for a variety of capacities in order to illustrate the changes as a result of scaling.

The System Advisor Model (SAM) by NREL has been used throughout the model to obtain values relating to the estimation of the solar field area. SAM is a performance based tool that features an extensive library of parabolic trough configurations and hence was used to infer information regarding trough collector efficiencies [59]. Additionally, the Solar Tower module within the SAM program features an optimising tool for calculating the heliostat field area for a given thermal capacity requirement. Utilising this software to infer solar field efficiencies has allowed the model to be simplified and based on industry standard data.
5.1 Power Block

Microsoft Excel was used to generate a user friendly interface for calculating critical power block outputs as a response to specified steam condition and operational inputs for a simple Rankine cycle as shown in figure 5.1.

<table>
<thead>
<tr>
<th>Power Block</th>
<th>Capacity</th>
<th>25000 kWe</th>
<th>User Defined</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reheat Used</td>
<td>No</td>
<td></td>
<td>User Defined</td>
</tr>
<tr>
<td>Required Thermal Input</td>
<td>88293 kWth</td>
<td>Output</td>
<td></td>
</tr>
<tr>
<td>Specific Net Work Output</td>
<td>1043 kJ/Kg</td>
<td>Output</td>
<td></td>
</tr>
<tr>
<td>Required Mass Flow Rate</td>
<td>25.54 Kg/s</td>
<td>Output</td>
<td></td>
</tr>
<tr>
<td>Power Block Efficiency</td>
<td>31.46 %</td>
<td>Output</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Turbine</th>
<th>Inlet Pressure</th>
<th>12000.00 kPa</th>
<th>User Defined</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet Temperature</td>
<td>565.00 °C</td>
<td>User Defined</td>
<td></td>
</tr>
<tr>
<td>Condensing Pressure</td>
<td>10.00 kPa</td>
<td>User Defined</td>
<td></td>
</tr>
<tr>
<td>Isentropic Efficiency</td>
<td>75.59 %</td>
<td>Estimated From Relation</td>
<td></td>
</tr>
<tr>
<td>Specific Isentropic Work Extracted</td>
<td>1397 kJ/Kg</td>
<td>Output</td>
<td></td>
</tr>
<tr>
<td>Specific Net Work Extracted</td>
<td>1036 kJ/Kg</td>
<td>Output</td>
<td></td>
</tr>
<tr>
<td>Turbine Efficiency</td>
<td>31.87 %</td>
<td>Output</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Compressor</th>
<th>Inlet Pressure</th>
<th>10.00 kPa</th>
<th>User Defined</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outlet Pressure</td>
<td>10000.00 kPa</td>
<td>User Defined</td>
<td></td>
</tr>
<tr>
<td>Isentropic Efficiency</td>
<td>90.00 %</td>
<td>User Defined</td>
<td></td>
</tr>
<tr>
<td>Specific Isentropic Work Input</td>
<td>12.11 kJ/Kg</td>
<td>Output</td>
<td></td>
</tr>
<tr>
<td>Specific Net Work Input</td>
<td>13.46 kJ/Kg</td>
<td>Output</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Heat Exchanger/Boiler</th>
<th>Specific Heat Input</th>
<th>3315 kJ/Kg</th>
<th>Output</th>
</tr>
</thead>
</table>

| Cooling Tower/Condensor | Specific Heat Output | 2272 kJ/Kg | Output |

Figure 5.1: Power Block Excel Interface.

The Rankine cycle was first defined in regard to the fluid states between the key cyclic components. The identified states are outlined below, additionally a schematic of the modelled system illustrating basic components and reference fluid states is depicting in figure 5.2.

- State 1 – Fluid state at the condenser outlet / compressor inlet
- State 2 – Fluid state at compressor outlet / heat exchanger inlet
- State 3 – Fluid state at heat exchanger outlet / turbine inlet
- State 4 – Fluid state at turbine outlet / condenser inlet
The fluid enthalpies at each governing state are calculated using compressed liquid water, saturated water, and superheated steam data tables from [26] based on user defined inputs for the cycle maximum boiler pressure, maximum temperature and the condensing pressure. Additionally the following assumptions were defined in order to solve the cycle.

- Compressor inlet is a saturated liquid
- Compressor isentropic efficiency is 90%
- Compressor operates adiabatically
- Heat exchanger is isobaric in operation
- Condenser is isobaric in operation
- Turbine operates adiabatically

The specific net output of turbine is found by calculating the work extracted under isentropic conditions and then multiplying this by the inferred isentropic efficiency calculated using equation 3-12 and the required power block capacity. Equation 5-1 highlights this step, with subscript ‘s’ referring to the isentropic case and subscript ‘a’ referring to the actual value.

\[
\Delta h_{3,4a} = \Delta h_{3,4s} \times \eta_{turbine, s}
\]  

 Fluid state 1 is then determined from the aforementioned assumptions and the condensing pressure. Following this the same process as that described for the turbine is repeated for the condenser to find the fluid properties at state 2. Having the states fully defined, additional cycle outputs are obtained including the enthalpy change in the exchanger and condenser respectively.
The net turbine efficiency can then be calculated using equation 5-2.

\[ \eta_{net,t} = \frac{\Delta h_{3,4a}}{\Delta h_{2,3}} \]  

(5-2)

Where \( \Delta h_{3,4a} \) is the specific enthalpy change between states 3 and 4 and \( \Delta h_{2,3} \) is the specific enthalpy change between states 2 and 3.

The power block cycle efficiency is then deduced from equation 5-3.

\[ \eta_{net} = \frac{\Delta h_{2,3}}{\Delta h_{4,1}} \times \eta_{gen} \]  

(5-3)

Where \( \Delta h_{2,3} \) is the specific enthalpy change between states 2 and 3 and \( \Delta h_{4,1} \) is the specific enthalpy change between states 4 and 1. \( \eta_{gen} \) is the assumed efficiency of the generator as discussed in section 3.3.3.

The required steam mass flow rate and thermal input into the system are then deduced from equations 5-4 and 5-5 respectively.

\[ \dot{m} = \frac{\text{Capacity, MWe}}{\Delta h_{3,4a}} \]  

(5-4)

Where \( \dot{m} \) is the steam mass flow rate, capacity is the required power block capacity in kWe and \( \Delta h_{3,4a} \) is the specific turbine work output in kJ/Kg.

\[ MW_{th} = \frac{\text{Capacity, MWe}}{\eta_{net}} \]  

(5-5)

Where MWth refers to the system thermal requirement form the solar field receivers in order to match the power block rating.

The flow of calculations is presented in figure 5.3. In this diagram orange parallelograms represent user defined inputs, green squares depict the calculations to determine the fluid states, blue parallelograms depict key intermediate calculation outputs and purple stadiums depict crucial power block outputs.
Figure 5.3: Power Block Calculation Flow Chart
5.2 Solar Field

The modelling of the solar field size initially requires that the solar resource available at the desired CST plant location be defined. Specifically, the average annual DNI in kWh/m² and the average hours of sunlight at a given location are then used to define the solar field size for both the Parabolic Trough and SolarTower configurations utilising equations 5-6 and 5-7 respectively. The Excel interface for this has been provided in figure 5.4.

<table>
<thead>
<tr>
<th>Site Location</th>
<th>Average Annual DNI</th>
<th>2800 kWh/m²</th>
<th>User Defined</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average Hours of Sunlight per Day</td>
<td>8 Hrs</td>
<td>User Defined</td>
<td></td>
</tr>
</tbody>
</table>

Figure 5.4: Excel Solar Resource User Input

Field Size (excluding storage):

\[
\text{Field Size (m}^2\text{)} = \frac{\text{Power Block Requirement (kWh)}}{\text{Average Daily DNI (kWh/m}^2\text{)} \times \eta_{\text{net,field}}} \tag{5-6}
\]

Where \( \eta_{\text{net,field}} \) is defined utilising SAM as discussed later in this section.

The average daily DNI is given by equation 5-7.

\[
\text{Average Daily DNI (kWh/m}^2\text{)} = \frac{\text{Average Annual DNI}}{365 \times \text{Average hrs of Sunlight}} \tag{5-7}
\]

In addition to the available resource, the model also allows the user to define an amount of thermal storage for the system, see figure 5.5. This is then used to scale the solar field using equation 5-8.

<table>
<thead>
<tr>
<th>Thermal Storage</th>
<th>Hours of Storage</th>
<th>3 Hrs</th>
<th>User Defined</th>
</tr>
</thead>
<tbody>
<tr>
<td>Storage Capacity</td>
<td>949400 kWh th</td>
<td>Output</td>
<td></td>
</tr>
</tbody>
</table>

Figure 5.5: Excel Thermal Storage User Input

Field Size (with storage):

\[
\text{Field Size (m}^2\text{)} = \text{Field Size w/out storage (m}^2\text{)} \times (1 + \frac{\text{hrs of storage}}{\text{average hrs of sunlight}}) \tag{5-8}
\]

The finalised storage capacity and field thermal capacity are then deduced from equations 5-9 and 5-10 respectively.
Storage Capacity (kWh\textsubscript{th}) = storage hrs \times \text{power block requirement} \quad (5-9)

Hence the total capacity of the solar field is given by:

\[
\text{Field Cap (kW}_\text{th}) = \text{Field Size (m}^2\text{)} \times \text{Average Daily DNI} \left(\frac{kW}_\text{th}}{m^2}\right) \times \eta_{\text{net,field}} \quad (5-10)
\]

**Parabolic Trough**

The trough configuration implemented within the field model consists of a Siemens Sunfield6 collector operating in unison with a Siemens UVAC 2010 solar receiver. As a result the inferred net solar efficiency was defined as 74%. Section 3.4 defines an expected change to the collector field efficiency of a trough system based on HTF piping requirements as the solar field area is varied. This was investigated by running multiple simulations using the SAM program and found to have a negligible effect on the defined net field efficiency; as a result, the simplified Excel model assumes constant net trough field efficiency. The Excel trough system UI however, displayed in figure 5.6 does allow for this value to be varied based upon a different collector and receiver type or user assumed field efficiency.

<table>
<thead>
<tr>
<th>Collector Type</th>
<th>Siemens SunField 6</th>
<th>Siemens UVAC 2010</th>
</tr>
</thead>
<tbody>
<tr>
<td>Receiver Type</td>
<td></td>
<td>SAM Input</td>
</tr>
<tr>
<td>Net Field Efficiency</td>
<td>74 %</td>
<td>From SAM</td>
</tr>
<tr>
<td>Required Thermal Output for Power Block</td>
<td>316467 kW\text{th}</td>
<td>Output</td>
</tr>
<tr>
<td>Required Field Size Inc. Storage Requirements</td>
<td>615226 m\textsuperscript{2}</td>
<td>Output</td>
</tr>
<tr>
<td>Total Field Capacity</td>
<td>435142 kW\text{th}</td>
<td>Output</td>
</tr>
<tr>
<td>Solar to Electric Efficiency</td>
<td>15.26 %</td>
<td></td>
</tr>
</tbody>
</table>

**Figure 5.6: Excel Parabolic Trough Collector Interface.**

**Solar Tower**

The tower configuration could not be modelled utilising the same methods as the trough system. Assuming a constant field efficiency for a tower system would serve to completely invalidate the model as the field efficiency fluctuations due to cosine losses and attenuation losses as described in section 3.1.2 would be unaccounted for. The optimisation of a heliostat field requires computation effort that exceeds Excel’s capabilities, as a result SAM was utilised to estimate the required field size for a given
thermal energy requirement. The assumed annual DNI of 2800kWh/m² was input into SAM to produce the plot shown in figure 5.7.

These results allowed an equation estimate, 5-11, to be inferred for the heliostat field size at a constant DNI.

\[ Field\ Size\ (m^2) = (875.38 \times Thermal\ Req^{0.1374}) \times Thermal\ Req \quad (5-11) \]

Where the required thermal input is in MWth.

This then allowed the process outlined for equations 5-8 to 5-10 to be utilised in establishing the total field capacity. Additionally, an estimate was included in the model for the net field efficiency based upon the field size, available DNI and estimated capacity output. The Excel UI is displayed in figure 5.8.
5.3 Cost Estimate Model

The LCOE model presented in 4.1 was used to estimate the relative cost per kilowatt hour of electricity production for the given system parameters. This required the estimation of the expected capital expenditure incurred in producing the defined system, as well as the annual O&M costs and the expected yearly energy production.

Available references for the capital and O&M costs implemented within the model were provided in USD (US Dollars); as a result, a conversion method established by Turton (2009) in [60] was implemented. Turton utilises the Chemical Engineering Plant Cost Index (CEPCI) to appropriately scale historically estimated plant costs in order to account for industry specific inflation and technology trends. Additionally, a currency conversion based upon averaged yearly exchange rates is incorporated to approximate costs to AUD (Australian Dollars). The relationship is defined by equation 5-12.

\[
C = C_o \times \left( \frac{CEPCI_{2016}}{CEPCI_{ref}} \right) \times \left( \frac{AUD}{USD} \right)_{2016}
\]  

(5-12)

Here \(C_o\) is the selected historical cost provided by literature and \(C\) is the adjusted cost implemented into the model.

5.3.1 Energy Production Estimate

The yearly energy produced for a given power block capacity and thermal storage system was estimated utilising equation 5-13.

\[
Yearly \text{ Production} = \text{Cap}_{\text{power block}} \times \text{Cap Factor} \times 365 \times 24
\]  

(5-13)

The capacity factor was assumed to be 20% for a system operating with 0 hours of thermal storage and hence was scaled according to the amount of thermal energy storage hours input by the user using equation 5-14. The Excel model interface is highlighted in figure 5.9.

\[
\text{Cap Factor} = 0.2 \times \left(1 - \frac{\text{Hours of Storage}}{6} \right)
\]  

(5-14)

<table>
<thead>
<tr>
<th>Plant Specific Outputs</th>
<th></th>
<th>Output</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capacity Factor</td>
<td>30 %</td>
<td>Output</td>
</tr>
<tr>
<td>Estimated Yearly Generation</td>
<td>262.8 GWh</td>
<td>Output</td>
</tr>
</tbody>
</table>

*Figure 5.9: Excel Capacity Factor and Yearly Generation Outputs.*
5.3.2 Capital and O&M Estimates

*Parabolic Trough*

Parabolic trough direct capital costs are estimated for six (6) identified key parameters based on information presented in [61]. The model scales the relative cost for the site preparation, solar field, and HTF systems based upon the calculated solar field size in m². Storage estimates are produced according to the required tank storage capacity in $kWh_{th}$ which is given by multiplying the hours of system storage by the required thermal input into the power block in the model. Power block and balance of plant cost estimates are then derived from the operating capacity of the power block in kWe.

Similarly to the estimate provided in [61], the indirect system cost and a contingency allowance is then derived from the subtotal of the direct costs at an assumed rate of 25% and 10% respectively, this method has also been applied to the solar tower costings. A summary of the unit cost and conversion to local currency and current year for deriving the capital expenditure of a system is given in table 5.1.

**Table 5.1: Capital Expenditure Unit Costs for Parabolic Trough System**

<table>
<thead>
<tr>
<th>Reference Cost (USD) per unit</th>
<th>CEPCI Ratio</th>
<th>Currency Conversion (AUD/USD)</th>
<th>Cost (AUD) per unit</th>
<th>Units</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>Site Preparation</td>
<td>28</td>
<td>0.9820</td>
<td>1.31</td>
<td>36.0</td>
<td>$/m²</td>
</tr>
<tr>
<td>Solar Field</td>
<td>295</td>
<td>0.9820</td>
<td>1.31</td>
<td>379.5</td>
<td>$/m²</td>
</tr>
<tr>
<td>HTF System</td>
<td>90</td>
<td>0.9820</td>
<td>1.31</td>
<td>115.8</td>
<td>$/m²</td>
</tr>
<tr>
<td>Storage</td>
<td>81</td>
<td>0.9820</td>
<td>1.31</td>
<td>104.2</td>
<td>$/kWh_{th}</td>
</tr>
<tr>
<td>Power Block</td>
<td>946</td>
<td>0.9820</td>
<td>1.31</td>
<td>1217.0</td>
<td>$/kWe</td>
</tr>
<tr>
<td>BOP</td>
<td>120</td>
<td>0.9820</td>
<td>1.31</td>
<td>154.4</td>
<td>$/kWe</td>
</tr>
<tr>
<td>Contingency</td>
<td></td>
<td></td>
<td></td>
<td>10</td>
<td>%</td>
</tr>
<tr>
<td>Indirect Costs</td>
<td></td>
<td></td>
<td></td>
<td>25</td>
<td>%</td>
</tr>
</tbody>
</table>

Operating and Maintenance costs are then estimated using fixed cost by capacity (in kWe) and variable cost by yearly generation (MWh) criteria based on the process outlined in [61]. Table 5.2 displays the assumed values utilised in the Excel model in addition to the conversion process previously mentioned.
Table 5.2: Operating and Maintenance Unit Costs for Parabolic Trough System

<table>
<thead>
<tr>
<th>Reference</th>
<th>CEPCI</th>
<th>Currency Conversion</th>
<th>Cost (AUD) per</th>
<th>Units</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cost</td>
<td>Ratio</td>
<td>(AUD/USD)</td>
<td>per unit</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fixed Cost by Capacity</td>
<td>69</td>
<td>0.9820</td>
<td>1.31</td>
<td>88.77</td>
<td>$/kWe</td>
</tr>
<tr>
<td>Variable Cost by Generation</td>
<td>2.5</td>
<td>0.9820</td>
<td>1.31</td>
<td>3.22</td>
<td>$/MWh</td>
</tr>
</tbody>
</table>

**Solar Tower**

Direct costs for the solar tower configuration are estimated in a similar fashion to that described for the Parabolic Trough configuration. Estimates provided by Turchi (2013) have accordingly been scaled utilising the CEPCI methodology previously described. Site preparation and heliostat field costs are determined from the estimated size of the heliostat array in m². The tower and reviercer costs include the relevant cost estimates pertaining to the HTF transfer system along with construction of the tower itself. The magnitude of this category is scaled according to the thermal energy requirement of the field, accounting for oversizing for storage in kWth. The storage system is costed utilising the same capacity rating as described for the trough system (in kWhth), notably the estimation of the unit cost for solar tower configurations is significantly smaller ($33.7/kWhth vs $104.2/kWhth). This is explained by the higher operating temperature of the tower configuration, which consequently requires a lower quantity of thermal medium in the HTF system. Again BOP and the power block are costed according to the plant capacity in kWe.

Table 5.3: Capital Expenditure Unit Costs for Solar Tower System

<table>
<thead>
<tr>
<th>Reference</th>
<th>CEPCI</th>
<th>Currency Conversion</th>
<th>Cost (AUD) per</th>
<th>Units</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cost</td>
<td>Ratio</td>
<td>(AUD/USD)</td>
<td>per unit</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Site Preparation</td>
<td>17</td>
<td>0.9535</td>
<td>1.31</td>
<td>21.2</td>
<td>$/m²</td>
</tr>
<tr>
<td>Heliostat Field</td>
<td>181</td>
<td>0.9535</td>
<td>1.31</td>
<td>226.1</td>
<td>$/m²</td>
</tr>
<tr>
<td>Tower &amp; Receiver</td>
<td>127</td>
<td>0.9535</td>
<td>1.31</td>
<td>158.6</td>
<td>$/kWth</td>
</tr>
<tr>
<td>Storage</td>
<td>27</td>
<td>0.9535</td>
<td>1.31</td>
<td>33.7</td>
<td>$/kWhth</td>
</tr>
<tr>
<td>Power Block</td>
<td>1200</td>
<td>0.9535</td>
<td>1.31</td>
<td>1498.8</td>
<td>$/kWe</td>
</tr>
<tr>
<td>BOP</td>
<td>355</td>
<td>0.9535</td>
<td>1.31</td>
<td>443.4</td>
<td>$/kWe</td>
</tr>
<tr>
<td>Contingency</td>
<td>10</td>
<td>%</td>
<td></td>
<td></td>
<td>[62]</td>
</tr>
<tr>
<td>Indirect Costs</td>
<td>25</td>
<td>%</td>
<td></td>
<td></td>
<td>[62]</td>
</tr>
</tbody>
</table>
O&M costs for the tower configuration were obtained utilising the same method as described for the parabolic trough. A slight variation in the magnitude between the categories is observed when implementing tower costings using information from [62] as shown in table 5.4. This is best explained by noting the variation in complexities of the solar fields. Heliostats operate using dual axis tracking and thus it makes sense that maintaining the additional field hardware would place an onus on the O&M cost estimate.

<table>
<thead>
<tr>
<th>Reference Cost (USD) per unit</th>
<th>CEPCI Ratio</th>
<th>Currency Conversion (AUD/USD)</th>
<th>Cost (AUD) per unit</th>
<th>Units</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fixed Cost by Capacity</td>
<td>72</td>
<td>0.9535</td>
<td>1.31</td>
<td>89.931</td>
<td>$/kWe</td>
</tr>
<tr>
<td>Variable Cost by Generation</td>
<td>4</td>
<td>0.9535</td>
<td>1.31</td>
<td>4.996</td>
<td>$/MWh</td>
</tr>
</tbody>
</table>

### 5.3.3 LCOE Assumptions

The implemented LCOE estimate for the two examined solar field configurations functions under the following assumptions.

- Discount Rate, $r = 12\%$
- CPI = 0\%, meaning fixed O&M costs over plant lifetime
- Capital investment costs are incurred over one construction year, (year 0)
- Plant operational lifetime of 30 years
- Yearly electricity generation is determined from assumed scaling of capacity factor with thermal storage
- Yearly electricity generation is fixed for a given system; anomalies in weather patterns and system degradation causing this to be untrue are ignored.
- System fuel costs are negligible
- The influence of economies of scale in determining capital costings is ignored.

The discount rate of 12\% has been chosen to represent the relative risk involved in introducing CSP into the Australian market. A sensitivity analysis examining the effects of this is presented in section 6.3.
5.4 Model Limitations
The designed model outlined in this chapter serves as a basis to examine the efficiency and cost trends associated with downsizing concentrating solar power generation. The cost estimates output by the model serve to explain the variation as model inputs change and should not be taken as conclusive accurate estimates.

The model was tested and shown to give reasonable results for maximum pressure ratings between 2000kPa and 20000kPa, outside of these limitations modification to the Rankine cycle model would be required. Additionally, as previously mentioned in section 3.3.2.2, the turbine isentropic efficiency estimation should be limited to capacities between 1MWe and 120MWe.

The power block model fails to consider the additional efficiency increase achievable by implementing reheat processes into the operating cycle. Research of industry applications has shown this to be implemented in cycles operating at capacities in excess of 50MWe. The impact of this would further reduce energy production costs for plant capacities greater then this 50MWe limit.

The accuracy of the solar tower solar field model is additionally questionable. SAM estimates utilised to obtain the solar field sizing trend were conducted for a heliostat with constant reflective surface area. It is believed that the estimation provides field size data within the correct ballpark; however it would be incorrect to assume these values as optimised and inaccuracies in costing flow on effects are assumed.

The LCOE estimates are limited by the assumptions outlined in section 5.3.3. The most notable assumption is the exclusion of economies of scale effects. Again, this was done in order to not over complicate the model and the inclusion would serve only to provide additional bias to larger scale systems. The assumption to maintain O&M costs over the lifetime of the project is expected to also incorporate a certain level of error into modelled outputs. This assumption means that no consideration for inflation or improvement in operating processes is taken into account.

Overall however, there is sufficient evidence to believe that the modelled outputs identify the general trends associated with plantation scaling and assist in determining why CST plantations have a minimum capacity requirement in order to be competitive in the energy market.
6.0 Modelling Results

6.1 Parabolic Trough Configuration

The parabolic trough configuration was modelled using input parameters outlined in table 6.1. The temperature was set at 400°C as this represents the maximum achievable temperature of current trough systems [54]. The DNI was chosen to correspond to that of Australian sites most suitable to CST integration in states such as WA, SA, and NT as evident in figure 2.1.

![Table 6.1: Constant Input Parameters for Parabolic Trough Analysis](image)

Inputs were then modelled for thermal storage hours of 0, 3, 6, 9, and 12 for power block capacity ratings between 1MWe and 100MWe as illustrated in table 6.2. The resulting data was plotted and has been presented as figures 6.1 to 6.5.

![Table 6.2: Variable Input Parameters for Parabolic Trough Analysis](image)

The power block net efficiency output is independent of the hours of thermal storage integrated into the system; however it shows a positive non-linear trend with net plant capacity. This evaluation clearly highlights the idiosyncrasies involved in the operating steam cycle, showing an approximate decrease in operating efficiency of 37.2% between a
50MWe system and a 1MWe system. While a decrease of just 6.16% is present between a power block rating of 100MWe and 50MWe.

![Power Block Efficiency - Parabolic Trough](image)

*Figure 6.1: Parabolic Trough Power Block Efficiency Output*

The annual solar to electricity efficiency is often quoted as a basic measure of plant performance. This indicator involves calculating the ratio between the annual quantity of solar irradiance that contacts the solar collectors to the annual energy produced by the plant as shown in equation 6-1.

\[
\eta_{\text{solar-elec}} = \frac{\text{Reflective Area (m}^2\text{)} \times \text{Annual DNI (kWh/m}^2\text{)}}{\text{Yearly Production (see eq. 5-13)}} \tag{6-1}
\]

Fichtner (2010) in [54] reports an annual solar-to-electricity range of 14-16% for realised trough systems between 50MWe and 300MWe with a thermal storage capacity of 7hrs. When compared to the model output present in figure 6.2 for a variety of TES hours, this shows that output estimated through excel modelling is within this bound, further strengthening the validity of the model.
The capital cost estimate based on the procedure outlined in section 5.3.2 produced the plot illustrated in figure 6.3. Here it is clear that the implementation of storage dramatically increases the capital cost per kWe of plant capacity. With nonlinear deviations observed between approximately 1MWe and 50MWe for systems with integrated TES.

The specific cost contributions per kWe over a range of plant capacities showing the cost category contributions is presented in figure 6.4 for a parabolic trough system with 6hrs of implemented storage.
The LCOE results based on the method provided in section 4.1 and the assumptions stated in section 5.3.3 are presented in figure 6.5. This plot clearly realises why stakeholders are reluctant to invest in smaller scale CST trough systems. Regardless of level of storage implemented, the LCOE for small scale plantations grows dramatically with smaller capacity systems. It is also important to note that due to the LCOE taking into consideration annual generation capacities, systems with implemented thermal storage achieve a lower LCOE. The difference between 6 hours of storage and 12 hours of storage for a 100MWe plant is 0.61cents/kWh.
6.2 Solar Tower Configuration

The solar tower configuration was modelled using input parameters outlined in table 6.3. The temperature was set at 565°C as this represents the maximum achievable temperature of current tower systems [54]. The DNI was kept consistent with the aforementioned trough model as was the turbine inlet and condensing pressures.

Table 6.3: Constant Input Parameters for Solar Tower Analysis

<table>
<thead>
<tr>
<th>Input Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Site Location</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Average Annual DNI</td>
<td>2800</td>
<td>kWh/m²/year</td>
</tr>
<tr>
<td>Average Sunlight Hours</td>
<td>8</td>
<td>hrs</td>
</tr>
<tr>
<td>Power Block</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Turbine Inlet Pressure</td>
<td>10000</td>
<td>kPa</td>
</tr>
<tr>
<td>Turbine Inlet Temperature</td>
<td>565</td>
<td>°C</td>
</tr>
<tr>
<td>Condensing Pressure</td>
<td>10</td>
<td>kPa</td>
</tr>
<tr>
<td>Compressor Isentropic</td>
<td>90</td>
<td>%</td>
</tr>
</tbody>
</table>

Inputs were then modelled for thermal storage hours of 0, 3, 6, 9, and 12 for power block capacity ratings between 1MWe and 100MWe as illustrated in table 6.4. The resulting data was plotted and has been presented as figures 6.6 to 6.10.

Table 6.4: Variable Input Parameters for Parabolic Trough Analysis

<table>
<thead>
<tr>
<th>Input Parameter</th>
<th>Values</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal Energy Storage</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Hours of Storage</td>
<td>0, 3, 6, 9, 12</td>
<td>hrs</td>
</tr>
<tr>
<td>Power Block</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Output Capacity</td>
<td>1000-100000</td>
<td>kWe</td>
</tr>
</tbody>
</table>

The power block net efficiency output is independent of the hours of thermal storage integrated into the system; however it shows a positive non-linear trend with net plant capacity. This evaluation, shown in figure 6.6, demonstrates a similar trend as found for the trough system, however the maximum power block efficiency occurring at a capacity of 100MWe was found to be 35.11% in comparison to 32.97% in the trough field case. This is a result of the higher cycle operating temperature of the power block, which in turn increases the net system efficiency as found in section 3.3.1.
Figure 6.6: Solar Tower Power Block Efficiency Output

The solar-to-electricity efficiency for solar tower systems is presented in figure 6.7. It is observable that comparatively to the trough configuration, the solar tower results appear to depend more on the level of implemented TES. It is also noted that the rate as which this measure increases is more reactive to total plant capacity than that of the trough system.

Figure 6.7: Solar Tower Annual Solar to Electric Efficiency
The capital cost estimate base on the procedure outline in section 5.3.2 produced the plot illustrated in figure 6.3. Here it is clear that the implementation of storage dramatically increases the capital cost per kWe of plant capacity. With nonlinear deviations observed between approximately 1MWe and 50MWe for systems with integrated TES and a linearization of the curves between 50MWe and 100MWe.

![Capital Expenditure - Solar Tower](image)

*Figure 6.8: Solar Tower Capital Expenditure*

The specific cost contributions per kWe over a range of plant capacities showing the cost category contributions is presented in figure 6.9 for a solar tower system with 6hrs of implemented storage.

![Cost Contributions - Solar Tower 6hrs Storage](image)

*Figure 6.9: Solar Tower with 6hrs Storage Cost Contributions*
The LCOE results for a solar tower system are presented in figure 6.5. This plot, unlike that of the parabolic trough, presents immediate cost reductions through the implementation of TES at small plant capacities. Evidence presented in section 3, suggests that this is a result of the higher field efficiency attainable with smaller implemented heliostat field sizes. Specifically, attenuation and cosines losses are proven to be reduced at smaller field capacities.

![Figure 6.10: Solar Tower LCOE](image-url)
6.3 Sensitivity Analysis

Discount Rate

The assumed discount rate is critical in determining accurate levelised cost of electricity estimates for a given CST system. The modelled results in sections 6.1 and 6.2 utilise an estimated value of 0.12 for the discount rate. This was deemed appropriate as the level of risk involved in penetrating the current fossil-fuel dominated industry would be high for potential investors and financers. Figure 6.11 displays the results of conducting a sensitivity analysis for a specified 20MWe trough system with 6 hrs of implemented storage. The discount rate inferred in the model is highlighted with the red marker, corresponding to a LCOE of $0.40475/kWh. It is clear from the figure that as the applied discount rate is reduced, so too is the LCOE with an implied rate of 0.04 corresponding to a LCOE of $0.21478/kWh.

![LCOE Sensitivity to Assumed Discount Rate](image_url)
7.0 Conclusions

The primary aim of the thesis topic was to investigate the inefficiencies and motives underpinning the lack of market drive and innovation in response to the proposal of downsizing CST steam power generation. This was achieved through an extensive review of literature, which then developed into the formulation of simple predictive tool utilising Microsoft Excel. While the modelling was conducted using fundamental assumptions, the results support expectations derived from literature.

Examination of the solar field components revealed distinct trends supporting downscaling, notably the efficiency associated with a heliostat field (Solar Tower configuration) shows likely improvement with smaller field size. In contrast the Parabolic Trough field exhibits a fairly constant efficiency, regardless of field size. However, the efficiency of integrated steam turbines was found to be largely dependent on the turbine rating and hence, the power block design capacity.

The relation developed for the turbine isentropic efficiency, based upon literature estimates and commercially available designs shows a clear reduction in the isentropic efficiency at smaller ratings. This relation causes a flow-on effect to the power block, effectively reducing the net operating efficiency. The impact of this is a relative size increase in the thermal energy input into the system in order to compensate for additional conversion losses. To achieve a greater thermal input, a larger relative solar field is required; which in turn was shown to increase the relative capital expenditure costs, driving the LCOE estimate upwards and further presenting a less attractive investment option.

The Excel model outputs estimations for both parabolic trough and solar tower collector configurations. The trends presented in section 6 signify the model validity for the simpler trough system. The solar tower configuration results suggest that the heliostat field size estimation outlined in section 5.2 may have introduced inaccuracies into the model. This presents as an area for future potential modification and improvement of results.

The incorporation of reheat processes with in the operating Rankine cycle was found through literature to improve the net cycle efficiency. The drawback associated with the implementation of such processes is the added costs involved in not only the acquisition of the turbine, but in the additional complexity introduced into plant operations. In addition, current commercially available turbine units appear to have an associated cost-benefit
relation in regards to whether steam reheat is included ‘as standard’. For turbine unit ratings in the vicinity of 50MW (or larger), unit models emerge with this capability. However, below this capacity, commercially available products with this technology were found to be uncommon.

For this reason, the developed model, which was constructed with the goal of simulating cost relations and efficiency estimates for small capacity cycles, fails to consider the effects of reheat processes on the relevant outputs. This presents an interesting potential avenue for future consideration if an appropriate turbine unit costing model can be quantified.

The estimated LCOE was found to largely depend on the assumed discount rate within the model. The rate utilised of 0.12 or 12% was based upon the level of risk associated with the development of CSP in an Australian market that is heavily dominated by competitive, conventional coal-fired and fossil-fueled forms of electricity generation. The discount rate can be likened to the required return that an investor would find sufficient in undertaking the risk of the investment. With that in mind, it is proposed that small scale CST generation using current available technologies could provide LCOE estimates within the competitive range bracket if a source of lending can be obtained at a reduced discount rate.

While this seems illogical for a private investor, for the Australian Government this presents an opportunity to further advances in reaching the renewable energy generation target along with producing employment opportunities for the labor force.

The completion of this thesis study signifies the achievement of the set forth goals, in providing insight into the operational idiosyncrasies associated with downscaling CST electricity production. It is expected that the study has provided some clarity with regard to CST plantation scaling, both qualitatively and quantitatively.
Bibliography


[33] “Chapter 14: Compressors and Turbines,” in Energy Management Series, Ottawa,

[34] w. Stine and M. Geyer, Power From The Sun, 2001.


[54] Fichtner, “Technology Assessment of CSP Technologies for a Site Specific Project in


