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Parabolic trough CSP modelling and its application to a domestic scale solar air Brayton engine

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PARABOLIC TROUGH CSP MODELLING AND ITS APPLICATION TO A DOMESTIC SCALE SOLAR AIR BRAYTON ENGINE

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A dissertation submitted to the Faculty of Engineering and the Built Environment, University of Witwatersrand, in fulfilment of the requirements for the degree of Masters of Science in Engineering.

Johannesburg, 2021

Supervisor: Dr. Shehzaad Kauchali

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I declare that this dissertation is my own unaided work. I was under the part-time employ of Scipio Technologies in Boksburg for the duration experimental stage of this research from 2016 to 2017. Assistance was provided in the form of fabrication of experimental equipment, however, all designs, workings and operation thereof were of my own unassisted endeavour.

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Abstract

Linear focus concentrated solar power collectors such as parabolic troughs and linear Fresnel collectors are usually relegated to relatively low temperature operation. The large surface area of the receiver suggests a propensity for significant heat losses to occur. Conventional linear focus receivers operate at approximately 350 °C. At these temperatures, it is common to employ the use of Steam Turbines or Organic Rankine Cycles for electricity generation.

In this work it is shown that a contemporary parabolic trough linear focus receiver is capable of operating with an outlet temperature of 700 °C – much greater than conventional operation at 350 °C – at a cumulative receiver efficiency of 57 to 62 %.

A numerically intensive linear receiver model was successfully developed to model linear focus receivers for practically any combination of dimensions, materials of construction, Heat Transfer Fluids and atmospheric conditions. In this work it was applied to parabolic trough receivers, however, the model may be easily adapted to linear Fresnel reflector collectors or any other newer designs of linear focus collectors.

Modifying vehicle turbochargers to operate as small, inexpensive and relatively efficient gas turbines have been discussed in a multitude of papers. Such engines produce power at a domestic or small commercial scale. In this work, a simulation of using a linear focus receiver together with such an engine was investigated.

An apparatus was built as a first iteration proof of concept in hybrid operation of a modified turbocharger air gas turbine Brayton Cycle Engine. An off-the-shelf Garrett GT0632SZ turbocharger was used in conjunction with fabricated parabolic trough collectors to preheat air prior to combustion with LPG fuel. Maximum power extraction was 30 W at a calculated air flow rate of 2.63 kg/min, with a single uncovered bare copper receiver parabolic trough providing a temperature gain of up to 39 °C, at a maximum outlet temperature of 133 °C.

Complications were experienced due to the diminutive size of the turbocharger and its implied low operating efficiencies, however, it was indeed shown to be possible to extract electricity from a modified vehicle turbocharger.

This work challenges the heuristic that linear focus collectors be relegated to low temperature operation. The intensive linear receiver model predicts reasonably efficient operation of contemporary linear receivers at moderate to high temperatures. The cost benefit of linear focus receivers may be leveraged together with the use of more efficient engine designs, such as supercritical CO₂, or alternate heat cycles such as gas turbines such that thermal and cost efficiencies, power production and profitability may be maximised.

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The help and patience of François, René and Andrés in the fabrication department of Scipio Technologies who help me build all of the apparatus according to my specifications. Being done this way such that I may maintain the requirement of unaided work required for this dissertation, despite the inefficiencies and various difficulties introduced by me in the designs due to lack of experience.

Rhod and Rodney in the Chemical Engineering workshop for the tens if not hundreds of hours spent there discussing and fabricating shafts, wheels and covers for the compressor and turbine. A special thanks to Rodney for teaching me how to use the lathe that I may do some of the metal shafts' fabrication myself without taking up much more of their time than I already had.

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List of Symbols

α	Absorbance	
α_T	Thermal diffusivity	$[\text{m}^2 \text{s}^{-1}]$
γ	Heat capacity ratio (c_p/c_v)	
γ_{SEC}	Solar Tracking Intercept Efficiency Factor	
η	Efficiency	
ϵ	Emmisivity	
μ	Dynamic Viscosity	$[\text{Pa.s}]$
ν	Kinematic Viscosity	$[\text{m}^2 \text{s}^{-1}]$
λ	Gaseous fluid heat conductivity	$[\text{W m}^{-1} \text{K}^{-1}]$
σ	Stephan-Boltzmann Constant	$5.670373 \times 10^{-8} [\text{W m}^{-2} \text{K}^{-4}]$
ρ	Density	$[\text{kg m}^{-3}]$
a	Aperture width	$[\text{m}]$
c	Gas velocity	$[\text{m s}^{-1}]$
c_p	Heat Capacity at Constant Pressure	$[\text{J kg}^{-1} \text{K}^{-1}]$
c_v	Heat Capacity at Constant Volume	$[\text{J kg}^{-1} \text{K}^{-1}]$
g	Acceleration due to gravity	$9.81 [\text{m s}^{-2}]$
h	Convective heat conductivity	$[\text{W m}^{-2} \text{K}^{-1}]$
k	Heat conductivity	$[\text{W m}^{-1} \text{K}^{-1}]$
$\dot{m}$	Mass flow rate	$[\text{kg s}^{-1}]$
$\dot{m}_c$	Dimensionless Corrected Mass flow rate	
p	Pressure	$[\text{Pa}]$
p_c	Corrected Pressure	$[\text{Pa}]$
r	Radius	$[\text{m}]$
ρ_{mirror}	Reflectance of mirror surface	
τ	Transmittance	
v	Wind velocity	$[\text{m s}^{-1}]$
w	Width of parabolic receiver normal to active insolation	$[\text{m}]$
A	Area	$[\text{m}^2]$
AR	Aspect Ratio (receiver length to outer diameter)	
CR	Concentration Ratio	
D	Diameter	$[\text{m}]$

F	Focal Length		[m]
H	Enthalpy		[J]
I	Irradiance		[W m ⁻²]
K	Incidence angle losses factor		
Nu	Nusselt Number		
N	Rotational velocity		[rpm]
P	Power		[W]
Pr	Prandtl Number		
Q	Heat		[J]
$\mathcal{R}$	Ratio		
R	Ideal Gas Constant	8.31446	[J mol ⁻¹ K ⁻¹]
Re	Reynolds Number		
S	Entropy		[J]
T	Temperature		[K]
T_c	Corrected Temperature		[K]
W	Work		[W]

Nomenclature

AR	Aspect Ratio
BC(H)(E)	Brayton Cycle (Heat) (Engine)
CR	Concentration Ratio
CSP	Concentrated Solar Power
DC	Duty Cycle
DI	Diffuse Irradiance
DNI	Direct Normal Irradiance
DoE	(South African) Department of Energy
GT	Gas Turbine
HTF	Heat Transfer Fluid
IPP	Independent Power Producer
LCOE	Levelized Cost of Energy
ORC	Organic Rankine Cycle
PPA	Power Purchase Agreement
PV	Photovoltaic(s)
REIPPPP	Renewable Energy Independent Power Producer Procurement Programme
sCO ₂	Supercritical Carbon Dioxide
SEC	Solar Energy Collector
STE	Solar Thermal Energy
UCG	Underground Coal Gasification

Absorber	The surface of the receiver where photons are absorbed and converted into heat. The absorber is typically coated with a solar selective material.
Aperture	The opening through which sunlight is collected by the SEC. For a parabolic trough collector, it is the distance between the edges of the concentrator above the curved profile (i.e. radially from the receiver) multiplied by the length of the concentrator.
Collector	The entire solar energy collector structure; including the concentrator, receiver and frame.

Co-generation	Heat recovery in another unit usually for the purpose of direct electricity production. Conventionally a lower temperature Rankine Cycle steam or ORC heat engine.
Concentrator	The mirrored section of the collector on which the sun shines.
Heat Recovery	The re-use of waste heat energy from one unit given to another unit (such as exhaust from the heat engine unit used in a separate heat engine or water heater).
Heat Recuperation	Another term for Heat Recycling in Journal Articles written in British English.
Heat Recycling	The re-use of heat energy within that operating unit to boost that unit's performance, efficiency and/or effectiveness by means of a heat exchanger (such as heat recycling within a heat engine unit).
Hybrid CSP	Operation of a process where heat energy is obtained from both CSP and the combustion of hydrocarbon fuels.
Insolation	Incoming solar radiation, also known as solar exposure, is the integral of solar irradiance over a period of time adjusted for projection (that is the total amount of solar energy reaching a surface horizontal to the ground while accounting for the apparent transit of the sun in the sky over the area in question).
Irradiance	Radiant flux (power) received by a surface perpendicular to direction of sunlight. Also known as beam radiation, DNI or direct insolation.
Receiver	The point at which the solar rays are received. The absorber of the receiver is the surface on which the solar energy is absorbed and converted to heat energy. The centre of the receiver is usually placed at the focal point of the collector. A cover may be placed around or in front of an absorber to limit convective and/or radiative losses.
Recuperator	The heat exchanger unit(s) used for heat recycling.
Turbojet	A Brayton Cycle Heat Engine gas turbine where the compressor and turbine are connected together mechanically. Turbojets propel a significant mass of air and are often used in conjunction with a nozzle to generate thrust.
Turboshaft	A gas turbine designed to produce mechanical power. The primary shaft of the engine is extended and connected mechanically to the load.

1. Introduction

Concentrated Solar Power (CSP) heat engines may be separated into three temperature groups: Low temperature Rankine Cycle Engines which operate from 80°C until about 600°C, Sterling Cycle Engines which operate from about 550°C to 800°C and Brayton Cycle Engines (BCEs) which operate at temperatures of approximately 750°C and above (Stine & Harrigan, 1986, p. 536).

The complexity and capital outlay of a solar collector's design generally increases with increasing operating temperatures together with increasing thermal efficiencies. Simple flat plate designs are associated with very low temperature heat engines. Moderately complicated linear focus receivers traditionally operate at low to medium temperatures. Complicated point focus collectors target operation at high temperatures (Fletcher, 2000, pp. 1-12).

Extensive research has gone into the modelling of the different collector types within their respective conventional Heat Engine standard operating temperature ranges (Chen, et al., 2007, pp. 512-525; Lloyd & Moran, 1974, p. 443). Conventional Rankine Cycle Engines are suitable for application with linear receivers at temperatures between 300°C and 550°C. Brayton and Sterling Cycles are reserved almost exclusively for high temperature point focus collectors at temperatures between 800°C and 1000°C (Duffie & Beckman, 2013, p. 629).

The thermal performance of linear focus receivers are primarily modelled with the assumption that convective losses are the driving heat loss transport mechanism given their typically low operating temperatures (Kalogirou, 2009, p. 200). Overall heat loss functions of linear receivers are usually empirically modelled as 2nd degree polynomial expressions of temperature (Goswami, 2015, p. 181) where radiative heat losses are generally neglected when modelling them.

A series of greatly detailed thermodynamic simulations of linear receivers have been performed by the U.S. Department of Energy Laboratory. It was found that optical losses for linear receivers account for by far the majority solar energy losses. Optical losses are typically 2.5-4 times that of combined convective and radiative losses for temperatures of 400°C and below (National Renewable Energy Laboratory, 2003, pp. 44, 54). These losses are primarily a function of mirror reflectivity, glass envelope transmittance, receiver surface material characteristics and the angle of solar incidence relative to the collector's aperture (National Renewable Energy Laboratory, 2003, pp. 17, 18).

The U.S. Department of Energy Laboratory's investigation was performed up to a temperature of 550°C as the maximum stable temperature of the heat transfer fluids (HTF) modelled (National Renewable Energy Laboratory, 2003, pp. 82,83). The investigation strictly focused on the thermal energy transfer performance of a linear receiver and not the potential performance of an attached heat engine.

A somewhat recent innovation in the Scientific Journals *Energy* and *Energy Procedia* detail the use of modified vehicle turbochargers modified as Concentrated Solar Power Brayton Cycle Heat Engines (CSP-BCHEs). This achieved by adding a solar heat addition stage between the turbocharger's compressor outlet and turbine inlet (Le Roux, et al., 2011; Le Roux, et al., 2012; Mariscal-Hay & Leon-Rovira, 2014)

The author has been unable to find any cases of an attempt to physically build or test such a solar powered device.

Altogether this presents an opportunity for research: The methodology used by the U.S. Department of Energy Laboratory's investigation may be expanded to mathematically investigate a greater spectrum of HTFs particularly at higher temperatures. This may further include newer and more effective receiver surface materials as well as the implied heat engine performance of an attached Carnot Engine. A practical example of a solar-assisted linear receiver CSP-BCHE may also be fabricated for initial experimentation and serve as a proof-of-concept of the idea.

At these implied higher temperatures for linear receivers, radiative heat losses may not be neglected. The temperature profile along the receiver and relative proportions of each mechanism of heat loss may be modelled. A parametric analysis and optimization of an attached Carnot Engine may help to better understand the efficacy of a high temperature linear receiver CSP system as a whole.

BCHEs are agnostic to the source of heat used and therefore lend to being used in a hybrid configuration. Preheating the working gas may be done by CSP prior to fuel injection and combustion. The solar energy thereby increases the exergy of the fuel. The relatively small size and high efficiency of commercial vehicle turbochargers implies potential use as domestic-to-commercial scale electrical power generation with easily recoverable heat.

1.1 Justification for this Line of Research

Conventional research in the field of CSP electricity generation can be generalised into two categories – low to moderate temperature Rankine Cycles that range with production capacities from a few kW_e to 1000s of MW_e, and high temperature Brayton and Sterling Cycles that range from 10s of kW_e to 100s of MW_e (Zhang, et al., 2013). High temperature CSP operations generally disregard the use of linear receivers due to the associated increases in surface area and therefore potential for convective and radiative heat losses.

A major benefit of linear focus receivers lies with their simplicity of construction, straightforward control and automation, fundamental robustness and relatively low fabrication costs compared to point focus collectors (Kalogirou, 2009, pp. 135-136). Further research may be incentivised by showing the theoretical potential benefits of operating contemporary linear receivers at modestly high temperatures by means of modern materials of construction.

The SunShot Initiative is a project started in 2011 by the United States Department of Energy with the ultimate purpose of reducing the cost of solar power to US\$1 per watt or US\$0.06 per kWh by 2030. SunShot is the largest initiative of its kind in the world, and represents the forefront of research in the field of CSP (Solar Energy Technologies Office, 2017, pp. 1-2).

Research funded by the SunShot Initiative is currently placed on the use of Supercritical Carbon Dioxide (sCO₂) Rankine Cycles at temperatures of about 700 °C as the primary heat engine design for the next generation of CSP heat engines (Bauer, 2016; Obrey, et al., 2016). Research is targeted at increasing the cost effectiveness of both linear and point focus receiver technologies at these temperatures (Solar Energy Technologies Office, 2019).

Current methods for modelling low temperature linear receivers make use of various simplifications and assumptions. In doing so they tend to overestimate convective heat losses while disregarding proportionately negligible radiative losses (Duffie & Beckman, 2013, pp. 328-329). However, only about 20% of the receiver's thermal input is lost due to convection and radiation at these temperatures (550K).

Empirical methods for modelling linear receivers generally make use of 2nd degree polynomials to approximate heat loss dynamics (Goswami, 2015, pp. 181,182). The function constants are derived from least squares minimization of lab test data. These models are fairly accurate within their respective interpolative temperature ranges that correspond to the respective boiling points of different HTFs. However, these models are only applicable to temperatures up to approximately 300°C since these temperatures are approximately the thermal stability limit of the heat transfer oils. Temperatures above this point are extrapolations of empirical data (Goswami, 2015, p. 182).

Operating at higher temperatures implies a gain in thermodynamic efficiency. There may therefore be a case for operating linear receivers at higher temperatures – the increase in radiative and convective heat losses may be offset by the gain in heat engine efficiency.

The methodology used by the U.S. Department of Energy Laboratory's theoretical study of linear receivers may be expanded for investigation at higher temperatures. The previous study was limited by the maximum permissible temperatures of the liquid oil HTFs.

Gas phase HTFs do not have the same temperature limitations, but their use introduces new constraints. Fluid thermal conductivities and dynamics, receiver structural and surface material aspects, and changing heat loss dynamics must all be modelled to better understand thermal performance of linear receivers at these higher temperatures.

Modelling an arbitrary linear receiver over a wide range of physical dimensions, materials and temperatures will allow for a quantitative parametric analysis and optimisation to be performed by means of connecting the heated HTF to a Carnot Engine.

Performing an optimisation in this fashion will allow for the determination of ideal design specifications and operating conditions for given constraints such as concentration ratio, HTF, receiver shielding and receiver surface materials.

The programming required for performing the simulations will be written in MATLAB. Emphasis will be placed on a function and class-based programming style in order to simplify the process of modifying and adding stages to the model. The intent will not necessarily be for the code to processes quickly, but rather to be relatively human legible and simple to adapt, expand and build upon for future research.

The turbocharger-to-engine simulation methodology used by Le Roux et al. (2011) may be used to simulate operating a modified commercial vehicle turbocharger as a BCHE in conjunction with a high temperature linear receiver. This may be used as a rough order-of-magnitude viability investigation in using such a setup for the generation of domestic or commercial scale electricity.

An important factor to be considered is the BCHE's agnosticism to the source of heat. Commercial implementations of CSP-BCHEs focus primarily on hybrid designs utilizing natural gas as the main high temperature heat source, with a solar pre-heat prior to fuel combustion (Dhanireddy, 2010; European Commission for Research, 2005). This gives the flexibility to operate the engine at night or during times of inclement weather.

Hybrid operation of BCHEs may lend toward the use of CSP to act as a preheating stage for the engine. Limitations exist on the maximum combustion chamber inlet temperature so as to inhibit harmful emissions. If it is possible to reach these temperatures with a linear focus receiver, it may be more cost competitive to use a linear focus receiver over a more expensive point focus receiver design.

It may also be advantageous to use a linear focus receiver in series with a point focus receiver. The majority of low to medium temperature heat may be added to the HTF with a less expensive per-thermal-watt linear focus receiver. Only the more expensive per-thermal-watt high temperature heat may be delivered by a smaller point focus receiver.

The relatively high surface area of linear focus receivers helps to mitigate issues of low thermal conductivities for gas phase HTFs which are a challenge for point focus receivers (Duffie & Beckman, 2013, p. 331).

Bulk heat storage may be used in conjunction with BCHEs. Salomoni et al. (2014) discussed various methods of solid-state heat storage with minimal pressure drop in the form of counter-current flow rock beds and large perforated concrete cubes for temperatures up to 900K (Salomoni, et al., 2014). This would be suitable for use as medium temperature heat storage for the linear focus receiver based BCHE.

Finally, a proof of concept a micro scale experimental linear receiver hybrid CSP-BCHE setup will be constructed using a small motorcycle turbocharger. This will provide a foundation for the demonstration of the idea as well as a platform for future investigation and research on the topic. The fundamental engineering

challenges of solar tracking, engine control, and linear collector's fabrication will be addressed.

The information obtained from the experimental setup may also be used as a basis for feasibility and viability studies to compare the technology to a conventional photovoltaic setup of the same duty or expense. Factors such as heat recycling and heat storage may be considered further in these studies too.

1.2 Problem Statement and Aim

Conventional models describing linear receivers are inadequate for use with temperatures beyond those empirically quantified (Goswami, 2015, p. 181). The use of linear receivers for CSP heat engines are usually relegated to lower temperature operation due to their large relative surface areas available for heat losses (Kalogirou, 2009, p. 200). However, modern materials for receiver surfaces and shieldings may mitigate heat losses to an extent where operating such a linear receiver at higher temperatures may in fact provide an overall boost in overall heat engine efficiency (Muñoz-Anton, et al., 2014).

A robust numerical modelling of a linear focus receiver will help to understand these dynamics. A multitude of liquid and gas-phase HTFs may be simulated together with modern receiver surface materials and shieldings. The hypothetical receiver may be attached to a Carnot Engine to quantify the upper bound of the feasibility of this approach. A similar methodology may be used by attaching a linear receiver to a turbocharger based BCHE.

Parametric studies may then be performed on these variables to optimise the design and operating conditions of such a linear receiver for high temperature applications subject to the constraints of different receiver materials, shieldings or other physical restrictions.

The ultimate aim of this research therefore is threefold:

The first overarching aim is to develop a robust and highly detailed model of a linear focus receiver for its use within a MATLAB simulation. This will allow for a parametric study of the receiver and optimization of an attached heat engine (Carnot, Brayton or otherwise). The model has been built to simulate a variety of construction and shielding materials and several liquid and gas phase HTFs. Various environmental simulation factors of the such as irradiation intensity and wind velocities have been included. Additional materials and fluids may be easily added to the model for future work.

The second overarching aim is to show how this simulation may be used as tool to optimise the design and operating conditions of a heat engine using a linear focus collector for a given set of design constraints. This may used to argue the feasibility of operating a linear focus receiver with modern materials of construction at temperatures far higher than conventional.

Finally, the third overarching aim is to determine the practicality in using a modified vehicle turbocharger together with linear collectors in a hybrid CSP-BCHE configuration for domestic and larger scale electricity production applications.

An experimental test rig of a hybrid CSP-BCHE using a motorcycle turbocharger has been built. This apparatus was intended as a proof of concept using a linear receiver in a hybrid CSP modified turbocharger BCHE configuration.

1.3 Research Questions

The following questions were used as a guideline and for the choices of direction taken during the research:

- How do the dynamics of heat transfer and losses change across the surface of the receiver, especially at higher temperatures and with modern shieldings and receiver surface materials?
 - For an arbitrary receiver of contemporary materials: what relative fractions of the losses are due to radiative and convective heat loss mechanisms? At what temperature does radiative heat losses become driving over convective losses? How important are shielding and surface materials to this?
- May any conclusions be drawn on best practice for linear collector design?
- Linear focus collectors are usually operated together with liquid phase HTFs. To what extent does using a gas phase HTF such as air affect heat transfer specifically in a linear receiver?
- Is there any validity to the heuristic of relegating linear receivers to low temperature operation when taking into account modern shieldings and receiver surface materials?
 - Is there a significant enough thermodynamic or other benefits to justify operating linear collectors at elevated temperatures?
 - To what extent do factors such as wind velocity change this?
- Is the operation of a modified vehicle turbocharger in a hybrid CSP-BCHE configuration feasible? Is the use of a linear collector feasible over the use of a normal point collector?
- Does a heat recycling stage provide substantial benefit to the turbocharger based BCHE?
- What is the environmental impact for the fabrication, installation, operation, maintenance and disposal of a modified vehicle turbocharger linear collector hybrid CSP-BCHE?
- What opportunities exist for heat recovery of such a system?
- How difficult is it to fabricate and control such a system?
- When optimised for materials and working components, what sort of efficiencies, sizes and duties should be expected for a modified vehicle turbocharger linear collector hybrid CSP-BCHE?

- Does it work at a domestic or larger scale? Is it applicable to rural application – especially somewhere with excess insolation and combustible gas reserves such as the Karoo?
- How does a rough estimate of total system cost compare with a similarly sized commercially bought photovoltaic (PV) system? Are the costs for each system within the same order of magnitude?

1.4 Research Objectives

The following objectives outlined the direction and fundamental approaches taken during the research:

- A robust model describing the heat transfer dynamics present for a linear receiver is required. The approach has been based off of that used by the 2003 National Renewable Energy Laboratory report on linear receivers.
 - Various additional factors in the model will be added. This includes a wider range of HTFs with both gas and liquid phases, receiver shieldings and surface materials, wind velocities, sky and ambient temperatures for radiative heat transfer etc.
- Use the model to write a MATLAB simulation of the linear receiver. Emphasis will be placed on human legibility and ease of future modification and expansion.
- Use the robust model and MATLAB code to perform parametric analyses and optimizations for a heat engine attached to an arbitrary receiver.
 - First for a Carnot, then a modified vehicle turbocharger linear collector hybrid CSP-BCHE.
 - Determine the extent to which heat recycling affect these heat engines.
- Use the model and optimization to compare to real world linear CSP receiver designs.
 - Compare real world plant dimensions to those suggested by the optimizations.
 - Simulate real world designs and compare simulated to measured performance results.
- A positive outcome of the high temperature simulations may be used to argue intentional operation of modern linear receivers at much higher temperatures than conventionally used.
- Determine the feasibility and viability of using a best case optimized modified vehicle turbocharger linear focus collector hybrid CSP-BCHE in a domestic or larger scale implementation.
 - Perform a rough order of magnitude economic viability comparison to a commercial PV system of similar size.

- Demonstrate a proof-of-concept in the use of a small modified motorcycle turbocharger in a linear collector hybrid CSP-BCHE setup.
 - Try to compare operating results to those simulated.

1.5 Scope of Research and Targeted Outcomes

The research focuses primarily on the following areas:

A section has been dedicated to researching CSP collectors with particular attention on linear focus type collectors and their models of performance. Conventional applications of CSP technologies have been discussed in detail along with the theoretical thermodynamic limits of devices.

From this point a series of mathematical models and functions were derived to robustly model the performance of linear focus receivers. The differential functions were then programmed into MATLAB to be used in simulations. This yielded a complete model for a linear focus receiver capable of incorporating any specified HTF, shielding and surface material, for a wide range of dimensions as well as atmospheric conditions. A Carnot Engine was connected to the HTF outlet to quantify the upper bound of the performance of an arbitrary linear receiver. Parametric analyses and optimizations were then performed on the dimensions of the collector.

This allowed for performance and efficiency curves to be generated describing all aspects of an arbitrary linear focus receiver and connected heat engine. A Carnot Engine sets the upper bound on the performance of the linear receiver. The receiver may also be connected to a BCHE for a more realistic estimation of real-world performance taking into account various environmental factors.

Finally, a test apparatus was built as a proof-of-concept of the idea for operating a linear receiver at a moderate temperature in conjunction with a hybrid BCHE. This apparatus provides a starting point for future research.

Therefore, the following targeted outcomes may be clearly defined:

- A robust simulation model for linear receivers must be built. It must operate over a wide range of temperatures for any HTF of any phase, shielding, surface material(s), physical dimensions and atmospheric conditions.
 - The simulation model may then be compared to the performance of real world CSP linear receivers to provide some validation of its authenticity of a model.
- The linear receiver simulation model may be used in conjunction with various types of heat engines to perform parametric analyses and

optimizations by iterating the dimensions of the receiver(s) and collector(s), HTF types and HTF flow rates.

- Real world linear CSP fields may then have their actual design dimensions compared to those predicted by the model.
- The viability of a domestic or larger scale linear receiver hybrid CSP-BCHE may be examined. This may be done by selecting an appropriate commercially available vehicle turbocharger and optimising for its operating condition. Performance and estimated costs may then be compared to a PV setup of similar duty.
- The intricacies of building, operating, controlling and metering a proof-of-concept of the modified vehicle turbocharger linear receiver hybrid CSP-BCHE need to be worked out. With the use of a Raspberry Pi, as much as possible of the metering and operation of the engine may be automated.

A conclusion must be reached on whether or not the heuristic of relegating linear receivers to low temperature operations is justified. The viability of operating a hybrid BCHE in a domestic application may be assessed.

2. Literature review

Worldwide demand for electricity continues to grow at an unprecedented rate. The World Energy Council estimates global electricity demand will be between 26 000 and 28 000 TWh/y by 2020, and that at least 21% of this demand will be satisfied by renewable sources (World Energy Council, 2013). For South Africa specifically, it is estimated that annual electricity demand by 2020 will be between 275 000 and 290 000 GWh/y (Council of Scientific & Industrial Research (CSIR), 2010). Currently about 86% of South Africa's energy generation is derived from coal, 9% from hydropower and the remaining 5% from nuclear and other renewables (KPMG, 2014).

The South African Government has shown a keen interest in implementing the Integrated Resource Plan (IRP) which aims to change South Africa's energy mix to less than 50% from coal by the year 2030 (Department of Energy, 2019, p. 86). The 2019 March Draft of The IRP2019 report targets an energy mix of 49% coal, 2.4% nuclear, 3.8% hydro, 7.2% instant dispatchable storage (batteries and pumped storage), 12% PV, 16% wind, 9.2% gas (natural and syngas) and 1% CSP by the year 2030 (Department of Energy, 2019, p. 39).

A potentially important decision affecting distributed power production and particularly power generation within the Western Cape and Mpumalanga is the decision to extend nuclear power production from Koeberg past the original planned decommissioning date of 2024 (Department of Energy, 2019, pp. 15,42). Decommissioning Koeberg would produce challenges and expenses in the otherwise necessary transmission network upgrade between Mpumalanga and the Western Cape.

CSP technologies have for now had all future government investment and investment removed due to its high relative cost to PV alternatives (Department of Energy, 2019, pp. 49, 57, 60)

The IRP attempts to achieve its goals by having government focus on funding construction of large-scale renewable projects, as well as incentivising Independent Power Producers (IPPs) with tax credits, favourable CPI linked purchasing price agreements and various tax subsidies (Department of Energy, 2013).

Electricity is a crucial part of everyday life. It is a vital factor in determining an individual's quality of life where both an ample and stable electricity supply are key factors of a country's potential for economic growth (Morimoto & Hope, 2001). While most developed nations have an abundant supply of electricity, less developed countries tend to struggle to either keep up with demand or capacity to supply it without interruption on an ongoing basis (United Nations Economic Commission for Africa, 1963).

Electricity supply issues are especially exacerbated in rural areas often lacking the infrastructure to supply it to the end users or the lack of proper maintenance thereof (Dziubinski & Chipman, 1999).

It is estimated that a small four-person energy-efficient dwelling requires approximately 6 kWh of electricity per day (Dzioubinski & Chipman, 1999). A solar powered electricity source of 1kW operating on average for about six hours of sunlight per day (Duffie & Beckman, 2013, p. 64) will produce enough electrical energy to sustain a single household, excluding electricity storage considerations. Additionally, the average heat required for hot water for a family of four is approximately 5.02 kWh/day (Kalogirou, 2009, p. 303)

South Africa has fallen victim to an unstable supply with the continued return of 'Load-Shedding' affecting not only individuals but having measurable impacts on the economy as a whole (Lindeque, 2019; Yelland, 2015).

27 Independent Power Producer (IPP) projects from the Renewable Energy Independent Power Producer Procurement Programme (REIPPPP) 2015 Bid Window were finalized in 2018 when Cyril Ramaphosa succeeded Jacob Zuma as President (Creamer, 2019).

Notwithstanding the contentious issue of nuclear power procurement, Eskom has shown a keen interest in developing IPPs with over R18-billion earmarked for grid integration of conventional and renewable IPPs over the next ten years with a further R119-billion set aside for network expansion and refining grid stability (Engineering News, 2017). Eskom therefore recognises the potential of the grid being used in a distributed power generation fashion.

Africa receives some of the highest levels of insolation on the planet throughout the year making it a prime candidate for solar based energy generation applications (African Development Bank Group, 2014).

Solar energy is conventionally converted into electricity in one of two ways: through Photovoltaics (PV) or using the Solar Thermal Energy (STE) to run heat engines in schemes collectively known as Concentrated Solar Power (CSP) (Stanford Global Climate and Energy Project, 2006).

The most common heat engine used for the generation of electricity is the Rankine Cycle Engine (Werner, 2010). This setup is characterized by relatively low operating temperatures and therefore relatively low thermal efficiencies (Zhang, et al., 2013). Steam turbines are the most common form of Rankine Cycle Engines.

Stirling Cycle Engines tend to operate at moderate temperatures and benefit from the associated gain in thermodynamic efficiency when operating at higher temperatures (Zhang, et al., 2013). The design and fabrication of Solar Stirling Cycle Engines is more complicated (and therefore more expensive) since point collectors and dual axes solar tracking mechanisms are required to maintain the higher temperatures needed for operation (Steinfeld & Palumbo, 2001).

For higher temperature applications, the BCHE tends to be used (Werner, 2010). While BCHEs are most commonly associated with the operation of aircraft jet engines, the principal of single axel compressor/turbine engines are collectively known as Gas Turbines (GT). The term Gas Turbine is primarily associated with axial flow GTs (such as jet engines) which burn hydrocarbons as the primary heat source (Dhanireddy, 2010). The high operating temperatures and pressure ratios

associated with BCHEs correspond to high thermodynamic efficiencies (Zhang, et al., 2013).

While there have been some successful attempts at using solar energy as the heat source for pilot commercial-scale BCHE electricity generation facilities, there has of yet been very little corporate interest in small scale sub-100kW Brayton Cycle implementations (European Commision for Research, 2005; Le Roux, et al., 2011).

2.1 Solar and Renewable Energies

Most planetary sources of energy are derived from our sun (Daniels, 1964). Stored solar energy in the form of biomass and fossil fuels are derivatives of photosynthesis processes. Other indirect energy sources from the Sun manifest as wind and hydro currents. Only nuclear (and by extension geothermal) energy is technically unrenueable. The materials used for nuclear energy originally coalesced together during the formation of our planet (Daniels, 1964).

The Earth receives an intense quantity of irradiance throughout the year. If 1% of the average irradiance passing through the atmosphere and hitting the surface of the planet were captured and converted to electricity at 10% efficiency, 105 TW of electricity may be produced (Kalogirou, 2009, p. 18). This would account for 3-4 times the total projected global power requirements for the year 2050 (Kalogirou, 2009, p. 18).

Not all places on the Earth are equally suitable for the use of ground based solar energy. The greatest potential energy available lies in bands between 20-30 degrees of latitude either side of the equator and decreases both toward the equator (due to tendency of cloud cover) and the poles (due to the relative direction of solar projection) (Goswami, 2015, p. 13). Earth's continental deserts tend to be found in these regions too (Goswami, 2015, p. 13).

Solar energy systems require the use of land surface area. That land cannot be used concurrently for other purposes that require sunlight – most notably agricultural activities (Kalogirou, 2009, p. 532). This implies that non-fertile desert areas are ideal for the use of solar applications.

Two factors need to be addressed when operating a solar process in a desert area. The first is the inherent lack of water available for cooling or as a feed. The second is the issue of power transmission from sparsely populated remote areas (Goswami, 2015, p. 14).

It has been estimated that a desert area of 65 000 km² – about 1% of the Sahara, 7% of the Kalahari or 16% of the Karoo – has the potential to produce electricity equal to the worldwide electrical consumption from the year 2000 when using conventional CSP technologies (Geyer & Quaschnig, 2000).

Depending on the season and geographical location, solar energy is only available for a few hours a day and is generally unusable during periods of cloud cover or poor weather (Goswami, 2015, p. 14). Energy produced by means of solar therefore

introduces the need for energy storage or supplementary sources of energy (Goswami, 2015, p. 14)

Intermittent renewables such as wind and solar could provide a significant proportion of the total energy mix on a country-wide scale without the need for heat or energy storage by feeding the electricity generated directly into the grid. Capital inexpensive gas-fired turbines which are able to quickly adjust power output may help to substitute the fundamental intermittency of such renewable energy sources (Verdolini, et al., 2016). Such turbines may be fuelled by natural gas or renewable sources such as methane, methanol and other biofuels (Johanson, et al., 1993).

Of particular importance in a South African context is the ability for renewable technologies to generate jobs. New technologies require new production, construction, marketing and operational activities to be performed. Installation and maintenance of such devices need to be considered too, especially in decentralized setups (Johanson, et al., 1993).

Arguably the most important benefit of renewable energy technologies is the positive impact on the environment manifested as a decrease in pollution. By offsetting conventional fossil-fuel based energy production, greenhouse gasses, particulate matter and other poisonous emissions are reduced (Kalogirou, 2009, p. 16)

Additional advantages of the adoption of renewable energy systems over conventional energy systems include (Edinger & Kaul, 2000; Bürger, et al., 2008; Johanson, et al., 1993; Mathews, 2007; Mohtasham, 2015):

- Land restoration
 - Growing biomass provides incentive for the restoration and development of degraded lands and soils. This may produce jobs in rural locations and provide a better habitat for wildlife than otherwise present.
- Fuel and energy supply diversity, security and stability
 - Diversifying fuel and energy sources for transport reduces the monopoly fossil fuels have over other energy sources. This reduces the risks of supply disruptions and encourages market competition between the different technologies. Alternatives to fossil fuels for transport include biofuels, hydrogen fuel cells and the adoption of electric vehicles.
- Reducing the threat of nuclear weapons proliferation
 - With less incentive to develop nuclear energy infrastructure, the ability to relatively easily source, produce, refine, transport and store nuclear materials for other purposes is otherwise hindered.
- Decentralization of power production
 - Decreasing the capital expenditure barrier to entry of the energy market encourages market participation and competition between businesses. Even individuals have the choice of providing for their own electricity requirements if necessary or convenient such as in under-serviced rural locations. Transmission lines for the electrical

grid will also undergo a reduction in required duty especially on national trunk lines.

- Accelerated rural development
 - Renewable technologies can be dispatched to rural locations much easier than connecting these locations to the grid without having to worry about the construction/upgrading and maintenance of grid infrastructure.

The ability to extract renewable energy depends heavily on the location of extraction. In general, renewable energy is captured directly or indirectly by means of solar, wind, hydro or water wave and tidal sources (Michaelides, 2012).

Wind power is an established, well understood and affordable technology. Global wind kinetic power available in the lower atmosphere has been estimated to be about 55 TW (Soerensen, 1979). Only 59% of this energy may be converted into mechanical power due to physical limitations of fluid kinetic energy transfer (Beurskens & Garrad, 1996). This maximum theoretical power rating is insufficient to meet current global energy demand (Kalogirou, 2009, p. 33). Other disadvantages for wind turbines include the large geographical area requirement, inadvertent harm to bird populations and power output of wind-based energy may not correlate to times of demand (Kalogirou, 2009, p. 37).

Hybrid designs incorporating solar and wind co-generation have shown some promise in combating the fundamental power supply intermittency of each approach (Ingole & Rakhonde, 2015). Solar systems tend to operate well during the day with clear weather, while wind turbines tend to perform well during periods of elevated wind speeds such as at night or during “poor” weather (Mohtasham, 2015). The combination of wind and solar power generation tend to produce a more or less consistent and stable supply of electricity with limited periods of absence or overlap (Ingole & Rakhonde, 2015).

Hydro energy sources convert the gravitational potential of water between two heights as a result of the water cycle. Like wind power, hydro power is an indirect form of solar energy (Kalogirou, 2009, pp. 43-45). Water and ocean tidal and wave energy on the other hand are conversions of the Earth’s rotational kinetic energy as a result of viscous forces and the Coriolis Effect (Stephenson, 2003).

Solar energy consists of the irradiance emitted by the sun reaching the planet. Photons are selectively absorbed by the atmosphere then scattered and reflected by clouds (Fluri, 2009). The photons that are not scattered are termed Direct Normal Irradiation (DNI) while those which have been scattered are termed Diffuse Irradiation (DI) (Fluri, 2009).

Converting solar energy into useful energy is generally done by one of two methods (Michaelides, 2012): The first is to convert the photons directly into electricity by means of the Photovoltaic Effect with the technology collectively titled Photovoltaics (PV). Alternatively, Solar Thermal Energy (STE) is the absorption of the photons to generate heat. The heat is then either used directly or further converted into shaft work by means of a heat engine (Stanford Global Climate and Energy Project, 2006).

2.2 Solar Energy Collectors

A Solar Energy Collector (SEC) is mechanism that collects solar irradiance which is then absorbed and converted to heat energy and deposited into a fluid (Kalogirou, 2009, p. 121). Downstream from the SEC, the heated fluid may be used directly or it may be used as a Heat Transfer Fluid (HTF) (Kalogirou, 2009, p. 121).

SECs are available in a variety of shapes, sizes and formats. Collectors may be classified as either non-concentrating or concentrating. Those of the concentrating type may be imaging (that is there is a definite focus point or line in that an image of the sun is formed) or non-imaging (an area of concentration with no specific focus) (Kalogirou, 2009, pp. 121-207).

Various types of commercial implementations of SECs have been summarised in Table 2.2 1 (Kalogirou, 2009, pp. 121-150; Kalogirou, 2003; Mills, 2001; Tabor, 1996; Zhang, et al., 2013). In general, as the concentration ratio of the receiver increases, the operating temperature of the SEC increases.

Table 2.2 1: Types of Commercial SECs (Kalogirou, 2009, pp. 121-150; Kalogirou, 2003; Mills, 2001; Tabor, 1996; Zhang, et al., 2013)

Concentration Type	Collector Type Name	Motion Tracking	Receiver Type	Concentration Ratio	Conventional Indicative Operating Temperature Range (°C)
Non-concentrating	Flat-Plate Collector (FPC)	None	Flat	1	30-80
	Evacuated Tube Collector (ETC)		Tubular	1	50-200
Non-imaging	Concentrating Flat-Plate Collector (CFPC)		Flat	1-5	60-240
	Compound Parabolic Collector (CPC)	None or single-axis	Tubular	1-15	60-240
	Cylindrical Trough Collector (CTC)	Single-axis	Tubular	1-10	60-200
Imaging	Linear Fresnel Reflector (LFR)		Tubular / Linear Focus	10-40	60-250
	Parabolic Trough Collector (PTC)			10-85	60-400
	Linear Fresnel Lens Collector (FLC)	5-100		60-400	
	Parabolic Dish Reflector (PDR)	Two-axis	Flat, tubular or concave / Point Focus	600-2000	100-1500
	Point Fresnel Lens Collector (FLC)		10-1500	100-2000	
	Heliostat Field Collector (HFC)		300-1500	150-2000	

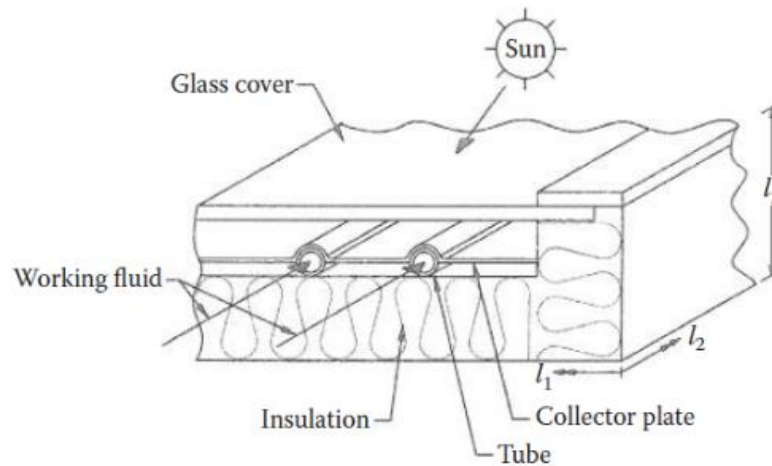


Figure 2.2 1: Flat Plate Collector cross-section (Goswami, 2015, p. 135)

Flat-Plate Collectors (FPC) generally consist of a plate covered with absorber substrate enclosed within a glass covered box. The plate may be used to heat a fluid directly or heat exchanging tubes may be placed within the enclosed area avoiding direct contact between the heated fluid and the absorber (Goswami, 2015, p. 129). An example of an FPC may be seen in Figure 2.2.1.

FPCs are most convenient to use for heating fluids up to approximately 75 °C above ambient conditions. They are characterised by their very low cost of production and simple maintenance. They are generally used for water heating, air and building heating, air-conditioning (specifically lowering the relative humidity of air to assist in drying processes) and process heat recovery (Goswami, 2015, pp. 129-132).

Evacuated Tube Collectors (ETCs) consist of a finned tube or a pipe coated with an absorber substrate enclosed within a vacuum-sealed cover (usually glass). The vacuum between the absorber and cover eliminates virtually all heat losses due to convection (Goswami, 2015, p. 155). The cylindrical shape of the receiver allows this collector design to accept a wide range of incidence angles ensuring performance throughout the day without solar tracking (Kalogirou, 2009, p. 132).

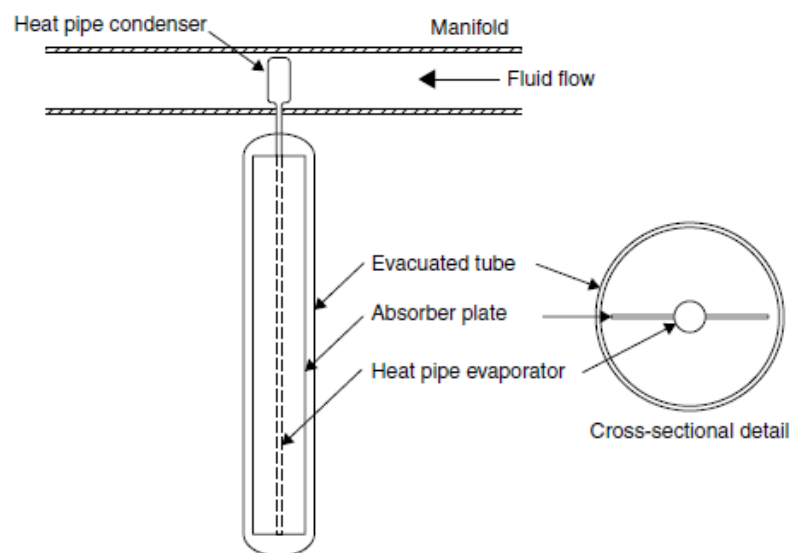


Figure 2.2 2: Evacuated Tube Collector (Kalogirou, 2009, p. 132)

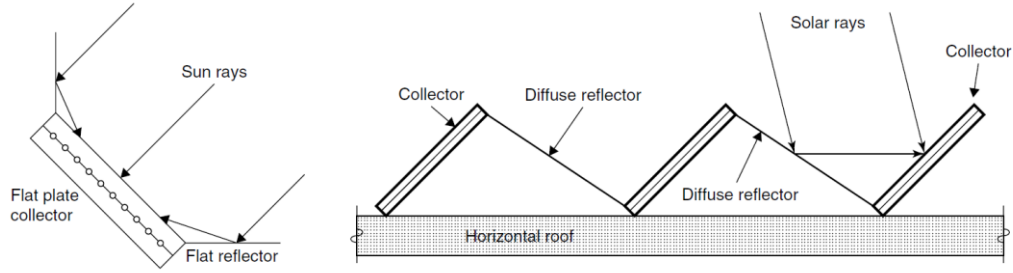


Figure 2.2.3: Concentrating Flat-Plate Collector (CFPC) (Kalogirou, 2009, pp. 136, 137)

ETCs usually make use of a liquid-vapour phase change fluid for heat transfer between the absorber “heat pipe” and a condensing chamber housed within a manifold above. ETCs are often used for water heating and are capable of operating at temperatures well above 100°C (Kalogirou, 2009, p. 132). An example of an ETC may be seen in Figure 2.2.2.

Concentrating Flat-Plate Collectors (CFPC) are essentially FPCs which include a reflector along one or more of the edges of the container, thereby increasing the aperture of the collector (Kalogirou, 2009, pp. 136, 137). An example of a CFPC may be seen in Figure 2.2.3.

Compound Parabolic Concentrators (CPC) are designed to accept irradiation over a wide range of incidence angles by allowing for multiple internal reflections (Duffie & Beckman, 2013, p. 338). The absorber at the back of the CPC may come in a variety of shapes. CPCs may be stationary or make use of approximate solar tracking – the wide acceptance angle and low concentration ratio do not justify the expense of an accurate tracking mechanism (Kalogirou, 2009, p. 130). Figure 2.2.4 shows various designs of CPC absorber shapes.

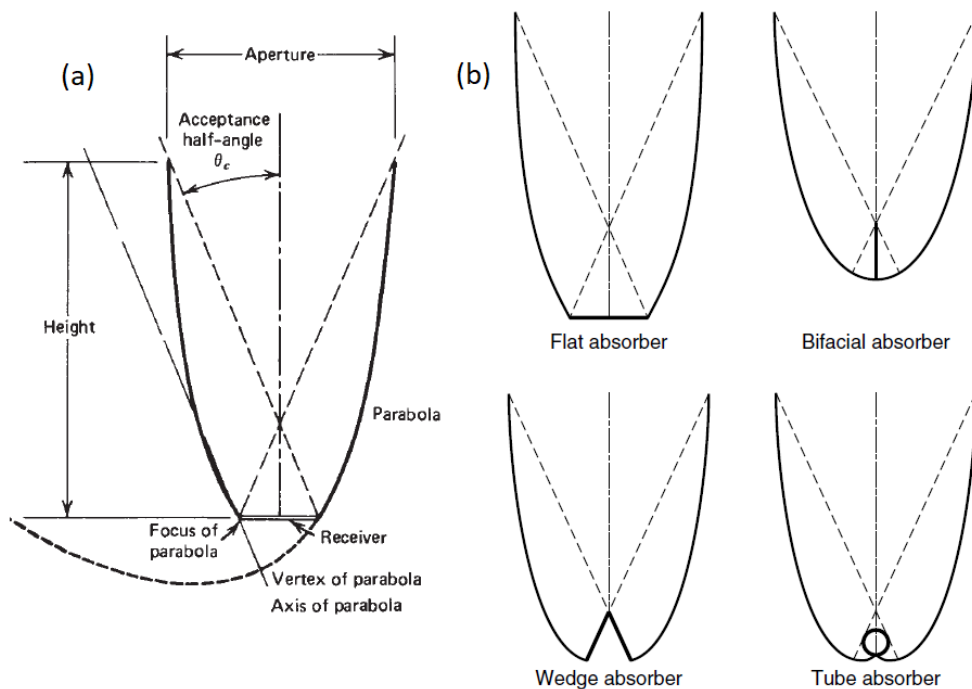


Figure 2.2.4: Compound Parabolic Concentrators (CPC). (a) Parabola Section (Duffie & Beckman, 2013, p. 338). (b) CPC Receiver Types (Kalogirou, 2009, p. 130)

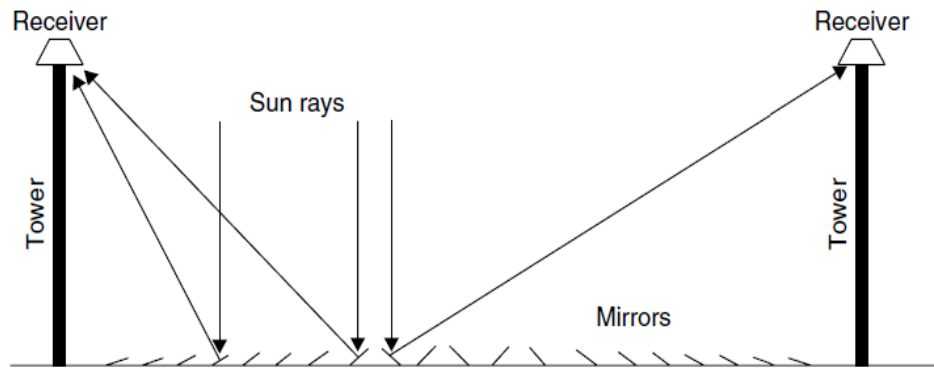


Figure 2.2 5: Linear Fresnel Reflector fields with downward facing receivers in an interleaved pattern (Kalogirou, 2009, p. 147)

Linear Fresnel Reflectors (LFR) and Fresnel Lens Collectors (FLC) are an approximation of a parabola in two or three dimensions (Kalogirou, 2009, pp. 144-146). The approximation is performed by using a series of reflective (in the case of LFRs as in Figure 2.2.5) or transparent (in the case of FLCs as in Figure 2.2.6) flat sheets.

Plastic and acrylic based LFRs and FLCs are significantly cheaper to manufacture than glass based parabolic type concentrators to manufacture and may be made to have a linear or point focus (Kalogirou, 2009, p. 144).

A design issue with LFRs made from flat strips placed on the same plane is the shading that occurs between the reflectors at low solar altitudes during winter months (Kalogirou, 2009, p. 146). Therefore, a spacing between the strips is introduced to avoid this, which leads to a lower effective ground surface area usage efficiency. An alternative design to maintain as high ground usage as possible and avoid cross-shading is to interleave the reflectors to concentrate along multiple absorbers in a format known as Compact Linear Fresnel Reflector (CLFR) (Kalogirou, 2009, pp. 146,147) as in Figure 2.2.5.

Parabolic Dish Reflectors (PDR) and Heliostat Field Collectors (HFC) are capable of attaining extremely intense concentration ratios in excess of 1500 (Kalogirou, 2009, p. 149). The high temperatures involved promote effective heat transfer and a high thermodynamic efficiency for the heat engine attached (Goswami, 2015, p. 449). An example of PDR and HFC designs are depicted in Figure 2.2 7.

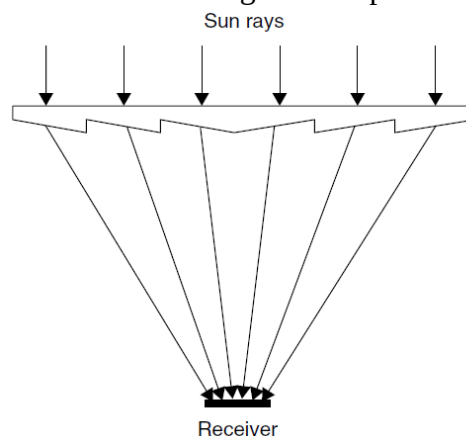


Figure 2.2 6: Transparent Fresnel Lens Collector (Kalogirou, 2009, p. 145)

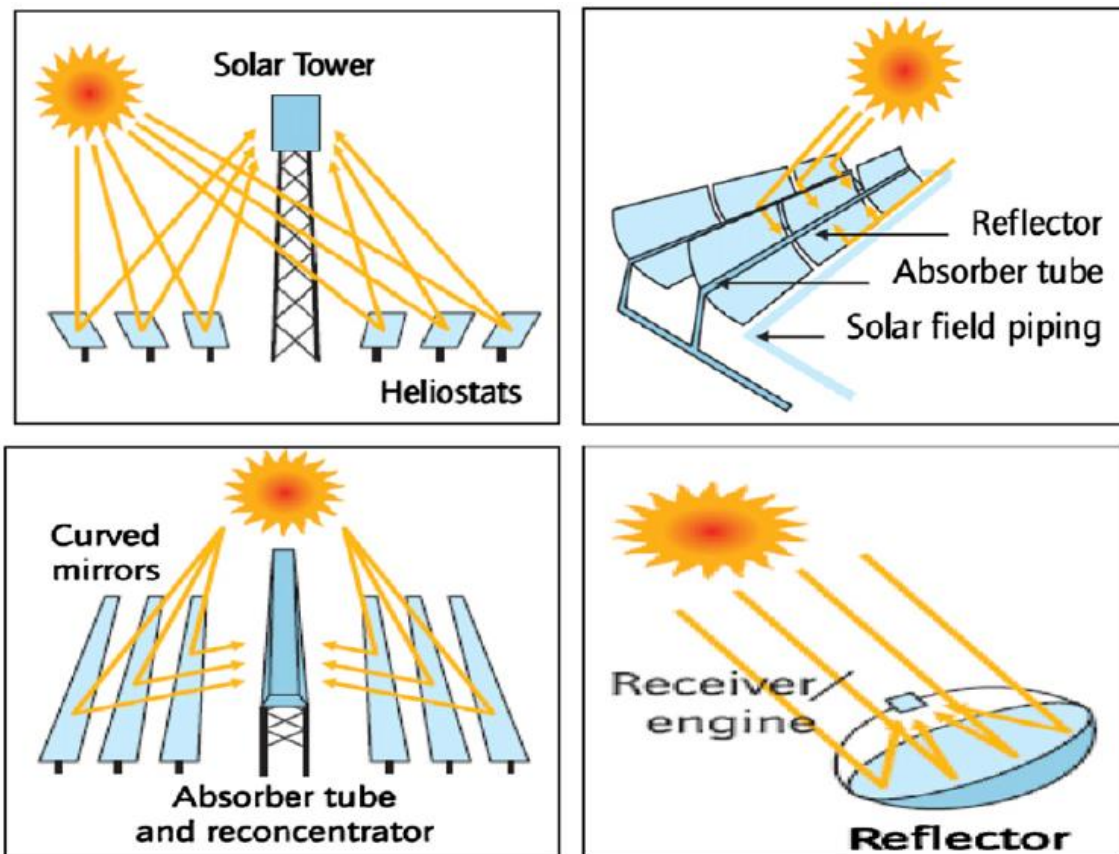


Figure 2.2 7: Commercial CSP SEC Implementations. HFC, PTC, LFR, and PDR (Zhang, et al., 2013, p. 469)

A primary advantage of PDRs is that each collector directly faces the sun such that there are no incidence angle losses. This in combination with two-axis tracking and a point focus make PDRs the most optically efficient of all SEC types (De Laquil, et al., 1993, p. 220). If there is no shading between nearby dishes, power output is consistent throughout the day from soon after sunrise until just before sunset.

HFCs (also known as Central Receiver Collectors or Power Towers) are typically constructed for turbine duties above 10 MW_e and therefore benefit from economies of scale for projects that large (De Laquil, et al., 1993, p. 219). A single point of heat reception simplifies issues of thermal energy transport found in other SEC designs.

The effective aperture of HFC, PTC and LFR collector arrays change throughout the day. The magnitude of variance of the values for both the altitude and the azimuth of the sun depend primarily on the latitude of the proposed site of the SEC (De Laquil, et al., 1993, p. 218). This is especially exacerbated in locations very far from the equator during winter when the sun may only reach a maximum altitude of a few degrees at noon – if at all (Goswami, 2015, p. 13). Therefore, HFC and LFR arrays are best suited for locations reasonably near the equator (Duffie & Beckman, 2013, p. 236; Goswami, 2015, pp. 13,74-76).

PTCs are permitted to rotate about the axis of the linear receiver and therefore have incidence angle losses from either the sun's azimuth or altitude, or both, depending on whether the collector is oriented N-S, E-W or mounted with the receiver not

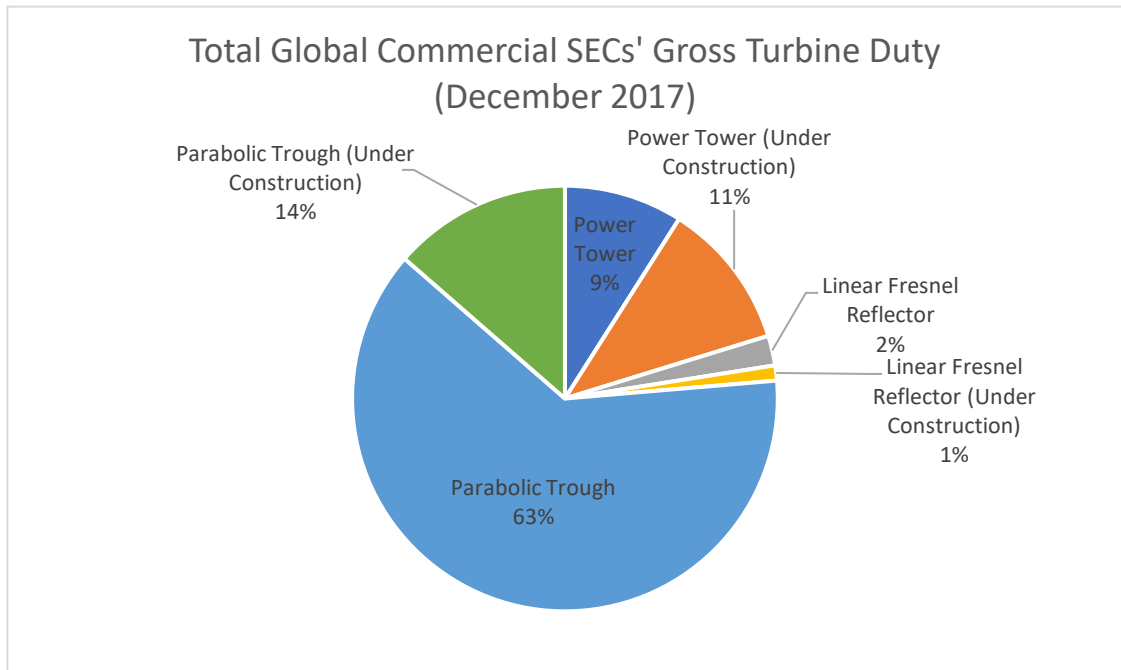


Figure 2.2 8: Combined Total Global Commercially Active CSP SEC Turbine Duty per technology (SolarPACES, 2017)

parallel with the ground (De Laquil, et al., 1993, pp. 217,218; Duffie & Beckman, 2013, p. 20).

The International Energy Agency maintains a database of all CSP projects around the world through the SolarPACES (Solar Power and Chemical Energy Systems) program (SolarPACES, 2017). The database includes project details from all 19 of SolarPACES's member countries: Australia, Austria, Brazil, Chile, China, European Commission, France, Germany, Greece, Israel, Italy, Mexico, Morocco, Republic of Korea, South Africa, Spain, Switzerland, United Arab Emirates, and the United States of America.

Table 2.2 2: South African Commercial CSP Projects (SolarPACES, 2017)

Name	Technology	Status	Gross Turbine Duty (MW)	Location	Production Start	Thermodynamic Cycle
Bokpoort	Parabolic Trough	Operational	55	Groblershoop, NC	March 14, 2016	Steam Rankine
Ilanga 1	Parabolic Trough	Under Development	100	Upington, NC	2020	Steam Rankine
Kathu Solar Park	Parabolic Trough	Under Construction	100	Kathu Solar Park, NC	Early 2019	Steam Rankine
KaXu Solar One	Parabolic Trough	Operational	100	Poffader, NC	March 2, 2015	Steam Rankine
Khi Solar One	Power Tower	Operational	50	Upington, NC	February 5, 2016	Steam Rankine
Redstone Solar Thermal Power Plant	Power Tower	Under Development	100	Postmasburg, NC	2018	Steam Rankine
Xina Solar One	Parabolic Trough	Operational	100	Poffader, NC	August 16, 2017	Steam Rankine

Figure 2.2 8 and Table 2.2 3 collate the combined total gross turbine duty for all commercial CSP projects and plants in the world per technology either in operation or under construction.

Table 2.2 2 provides an overview of all CSP projects in South Africa. In South Africa, all commercial CSP projects are located in the Northern Cape and are Steam Rankine based.

PTCs account for 77% of global CSP turbine duty. All commercially operational SECs and those under construction utilize a Rankine Cycle as the heat engine (SolarPACES, 2017). Approximately 9 out of every 10 projects use a steam-based Rankine Cycle with the remainder being Organic Rankine Cycles.

2.2.1 Solar Hybrid BCEs

An alternate approach to harness STE is by means of a Brayton Cycle based heat engine (European Commission for Research, 2005, p. 2). This technology along with its commercial applications have garnered significant research, experimentation and pilot demonstrations in the last 15 years (European Commission for Research, 2005; Korzynietz, et al., 2016; Schwarzbözl, et al., 2006). Particular emphasis has been placed on the concept of a solar hybrid BCE air-based GTs. The principal of the idea is to operate a standard GT engine but include an additional heat addition stage prior to fuel combustion – where this heat is sourced from STE.

Benefits associated with this approach include (Korzynietz, et al., 2016, pp. 578-589; Rovensea, et al., 2017, pp. 675-682; Schwarzbözl, et al., 2006, pp. 1231-1240):

- Fast start-up of the Gas Turbine with instantaneous dispatch for grid stability.
- Wide range of operating output power possible which may be augmented by burning more fuel as required.
- Continued electricity generation during inclement weather.
- Straightforward control and operation compared to other CSP technologies.
- Air as the HTF is free, non-hazardous and doesn't suffer from overheating and freezing issues.
- Little or no water usage for cooling; Rankine Cycles - especially steam based – require substantial amounts of water to operate condensers.
- Adaptability in design allows for configuration specific to region of use.

Table 2.2 3: Total Global Active Commercial CSP Turbine Duties by Technology (SolarPACES, 2017)

Technology	Gross Turbine Duty (MW)
Parabolic Trough	4 366
Parabolic Trough (Under Construction)	945
Linear Fresnel Reflector	158
Linear Fresnel Reflector (Under Construction)	80
Power Tower	627
Power Tower (Under Construction)	781

- May be used with or without heat storage.
- High turbine operating temperatures imply high thermodynamic efficiencies.

A series of demonstration projects of solar hybrid Brayton Cycle air-based gas turbines represent the state-of-the-art application of the technology (Bryner, et al., 2016). The projects were conducted near Seville, Spain (37.2° latitude) and were titled: the SOLGATE project in 2003, the follow up project SOLHYCO performed in 2008, and the most recent SOLUGAS project performed in 2012.

SOLGATE was the first practical demonstration of a solar hybrid Brayton Cycle air-based Gas Turbine (European Commission for Research, 2005, p. 2). A HFC was chosen as the SEC method to preheat air fed to a modified Allison Model 250 helicopter engine. A modified combustor for the helicopter's turbine was built to accept air at an inlet temperature of up to 800 °C (European Commission for Research, 2005, pp. 3-5). Three hexagonal cavity type receivers were built and connected in series atop the central tower. It was successfully shown that the receivers were capable of producing air outlet temperatures of up to 1000 °C (European Commission for Research, 2005, p. 12). As the project was a proof of concept, the apparatus omitted the use of a recuperator as well as a co-generation system. A nominal electrical power output of 230 kW_e was demonstrated at a measured receiver thermal efficiency of 77% ± 5% (European Commission for Research, 2005, p. 18).

SOLHYCO continued from SOLGATE with various improvements and targeted three outcomes: operation on completely renewable resources, a more cost-effective design for the receiver and receiver cavity, and the use of a bottoming co-generation heat recovery Rankine Cycle steam generator (European Commission for Research, 2011, pp. 1-3).

The first objective was met through turbine operation with biodiesel as the fuel instead of kerosene as previously used. Modifications to the combustor unit of the SOLGATE's Allison M250 Gas Turbine setup were made and it was successfully demonstrated to generate electricity completely from renewable resources. The nominal 220 kW_e produced was only 4.3% less duty than the kerosene-based combustor under similar solar conditions (European Commission for Research, 2011, p. 30).

During operation of SOLGATE it was noted that the volumetric cavity receiver design presented two challenges: the first was the cost of producing the receivers with the refractory materials as designed, the second was the large thermal inertia of the cavities meant start-up and preheating stages took a few hours (Amsbeck, et al., 2008, pp. 1-4). SOLHYCO solved these problems by the use of a series of multi-layered metallic absorber tubes through which the air flows, with the tubes arranged in a conic shape about a single receiver cavity (Amsbeck, et al., 2008, pp. 5,6). This design proved to be far more economically viable to produce along with a lower receiver pressure drop and a more even temperature distribution within the receiver cavity (Amsbeck, et al., 2008, pp. 7,8).

With the new receiver design and biodiesel shown to be an effective fuel, the core SOLHYCO microturbine based apparatus was assembled. The receivers in the

central tower used in the SOLGATE project were replaced with the new single receiver in conjunction with the HFC used during SOLGATE (European Commission for Research, 2011, pp. 23-40). A more efficient purpose built 100 kW_e Turbec T100 microturbine was used in place of the Allison M250 helicopter engine, along with a recuperator and a final co-generation steam turbine unit.

During testing significant issues were experienced involving the combustion unit and receiver cavity lining (European Commission for Research, 2011, p. 28). The turbine was however able to operate at 70 kW_e stably. Due to a crack in the refractory lining of the receiver cavity which formed within the first few hours of operation, measured receiver thermal efficiency was found to be between 37.8% - 45.6%. Nevertheless, the project was deemed to be a successful proof-of-concept (European Commission for Research, 2011, pp. 85-86).

Following from SOLHYCO, SOLUGAS was the first MW scale demonstration of a solar hybrid BC air-based GT (Korzynietz, et al., 2016, p. 579). A new HFC and demonstration plant was designed and built. The plant operated over a period of more than one and a half years in a variety of weather conditions with hundreds of cold start-ups and over 1000 hours of turbine operation (Korzynietz, et al., 2016, p. 588).

In operation the SOLUGAS setup demonstrated a receiver duty of 2.9 MW_{th} heating air at approximately 8 bar_{gauge} and 5.6 kg/s with receiver outlet temperatures of up to 800 °C and stable turbine output power of 3.2 MW_e (Korzynietz, et al., 2016, p. 586). Measured thermal efficiency for the receiver ranged between 71.3 and 78.1%, with cold start-ups demonstrated in less than 30 minutes (Korzynietz, et al., 2016, p. 587).

A significant problem with solar hybrid BC air-based GT lies in the design of the combustor unit: in order to limit emissions in conventional GTs, fuel and air are usually premixed before combustion within the combustor unit (Bryner, et al., 2016, p. 4). With a lean fuel mixture in a homogenous state, NO_x formations are inhibited (Bryner, et al., 2016, p. 10).

In a solar application with combustor inlet temperatures above 650 °C, auto-ignition and flashback of the fuel mixture becomes a concern (Bryner, et al., 2016, p. 10). If the fuel mixture is ignited before sufficient mixing, localized regions with fuel-air ratios that are stoichiometric or rich favour the formation of CO and NO_x gasses – thus rendering the renewable energy source “unclean” (Bryner, et al., 2016, p. 10).

Fortunately, a novel design for a high temperature combustor unit built specifically for solar Brayton Cycle applications has been demonstrated by the Southwest Research Institute by using multiple banks of tens to hundreds to thousands of low volume “micro-mix” injectors (Bryner, et al., 2016, p. 4). In this fashion a homogeneous fuel-air mixture is achieved in a very short period of time before spontaneous ignition of the mixture thus inhibiting the formation of harmful emissions (Bryner, et al., 2016, pp. 4,9,10).

2.3 Concentrated Solar Power

2.3.1 Linear and nonlinear SEC Characteristics

The operational aspects of different SECs need to be considered in commercial implementations of CSP installations. Conventional choices for SECs include HFCs, PDRs, LFRs and PTCs which may be organized as nonlinear (PDR, HFC) and linear collector types (LFR, PTC).

Concentrating imaging nonlinear collectors (i.e., point focus receivers such as HFCs and PDRs) tend to operate at high concentration ratios (CR) and therefore have relatively low surface area receivers (Goswami, 2015, p. 185). As a result; the thermal performance of the collector is much more sensitive to optical properties such as incidence angles and imperfections on the reflective surface(s) than to thermal losses due to convection and radiation away from the receiver (Goswami, 2015, p. 185).

For the purpose of achieving high temperatures (1000K and above) it is generally recommended to use a point receiver with the absorber section positioned within a cavity having an aperture no larger than necessary. Excessive heat losses due to air convection and radiation are avoided by virtue of limiting the surface area of the receiver section (Reddy & Sendhil-Kumar, 2008, pp. 812-819).

The individual heliostats of HFCs and the dishes of PDR arrays must overcome the issue of shading each other at times of low solar altitude especially during winter. Individual heliostats and PDR dishes are usually placed far enough away from one other as to prevent this (Duffie & Beckman, 2013, p. 368). Therefore, only about 30-50% of the ground space available is typically used effectively by the individual concentrators. As such, HFCs and PDR arrays are only viable in areas with inexpensive land (Duffie & Beckman, 2013, p. 368).

The complexity of operating a HFC requires SECs of this design to be built to a large enough scale so as to benefit from an economy of scale. HFCs therefore tend to operate at very high concentration ratios and high temperatures (typically 800-1000 °C) with electrical power output duties in the order of 1-1000 MW_e (Goswami, 2015, pp. 193, 460; Kalogirou, 2009, p. 535).

Individual PDR concentrators are limited to practical sizes due to materials limitations in performing 2-axis tracking by moving the entire dish structure (Kalogirou, 2009, p.

Table 2.3.1 1: Commercial Performance Characteristics of Various SEC Technologies (Müller-Steinhagen & Trieb, 2004, pp. 43-50)

Technology	Duty (MW _e)	Concentration Ratio	Peak Solar-Electric Efficiency (%)	Standard Solar-Electric Efficiency (%)	Land Requirement (m ² /MWh.a)
PTC	10-200	70-80	21	10-15	6-8
LFR	10-200	25-100	20	9-11	4-6
HFC	10-150	300-1000	20	8-10	8-12
PDR	0.01-0.4	1000-3000	29	16-18	8-12

535). PDRs therefore tend to be produced in a modular fashion with duties of 10-100kW_{th} per dish with arrays of dishes deployed to provide the duty as required (Goswami, 2015, pp. 193, 460). CRs may range from 60-2000 with operating temperatures above 1500 °C, however, most commercial PDRs tend to operate at about 700 °C.

Table 2.3.1 1 compares the performance of commercial implementations of the most common types of SECs. Linear focus collectors such as PTCs and LFRs operate at approximately the same solar-to-electric efficiency as point focus HFCs, however, PTCs and LFRs require roughly double the land surface area as HFCs for the same annual energy output to limit shading between units (Müller-Steinhagen & Trieb, 2004, pp. 43-50)

HFCs are more expensive to build per kWh_a than PTCs and LFRs, but are more spatially efficient. Therefore, the price and availability of land is a primary factor in deciding between linear focus and point focus collector technologies (Duffie & Beckman, 2013, p. 368).

Point focus PDRs are the most expensive type of SEC technology and commercial utility scale applications of PDRs are not competitive with other collector designs (Barlev, et al., 2011, pp. 2703-2725). PDRs are therefore best suited for niche and military uses where cost isn't the primary concern.

While some PDR implementations utilize an inexpensive and straightforward steam-based Rankine Cycle, the relatively high cost of producing each concentrator permit the use of more efficient (and more expensive) Sterling Cycle Heat Engines at the focal point of each concentrator (Goswami, 2015, pp. 449-457; Kalogirou, 2009, pp. 523, 537). Sterling based PDRs operate at efficiencies of up to 30-40%, however, gas phase heat transfer issues mean that these engines are usually constructed from exotic materials with hydrogen as the HTF.

Concentrating linear collectors (specifically PTCs and LFRs) operate at low to moderate concentration ratios with much greater receiver surface areas than competing point focus collectors (Kalogirou, 2009, p. 135).

The primary benefit of linear collectors over point focus collectors is the overall lower cost per kWh of thermal energy collected as a result of greatly simplified collector manufacture with lower tolerances, and the simplified manufacture and control of single axis heliostat systems (Kalogirou, 2009, pp. 135-136).

The conventional use of linear collectors at low to moderate temperatures negate the need to model the relatively low radiative losses along the receiver (Kalogirou, 2009, p. 200). Heat losses from PTCs are usually modelled as second-degree polynomial equations where the exponents are determined in an empirical fashion experimentally (Goswami, 2015, p. 181).

The orientation of a linear collector's receiver section is an important factor in determining the performance of the collector throughout the year (Duffie & Beckman, 2013, p. 20). For a flat ground surface, the receiver may be oriented either along an E-W axis or N-S axis so as to minimise the solar incidence angle and

thereby maximise the effective aperture of the concentrator(s) (Duffie & Beckman, 2013, p. 20).

E-W orientation of the receiver provides peak heat collection rate at noon zenith when the solar incidence is normal to the concentrator's aperture (Duffie & Beckman, 2013, p. 20). N-S orientation provides a lower peak heat collection rate throughout the day (as there is always incidence) but the overall daily quantity of heat collected is greater than that of an E-W orientation (Duffie & Beckman, 2013, p. 20).

PTCs are conventionally operated almost exclusively in conjunction with Rankine type Heat Engines as an indirect Rankine Cycle with the working turbine fluid separated from the HTF by means of a heat exchanger (Goswami, 2015, p. 438). It is common for one or more additional bottoming Rankine Cycle Engines (either steam or an ORC) to operate as a heat recovery unit for SECs of all types (Goswami, 2015, p. 438).

Commercial implementations of Rankine type PTC SECs are typically designed with CRs of between 70-80, operating temperatures between 40-400°C and peak solar-electrical conversion efficiencies of about 21% (Kalogirou, 2009, pp. 138, 524; Müller-Steinhagen & Trieb, 2004). While there are thermodynamic benefits of operating at higher temperatures, reliable high temperature fluid pumps for the HTFs and working turbine fluids impose significant engineering, materials and costs challenges (Kalogirou, 2009, p. 529).

2.3.2 Conventional Operational and Commercial Aspects

An important factor to recognise in CSP installations is the necessity of periodic washing of the mirrored surfaces especially in dusty and sandy desert environments (Kalogirou, 2009, p. 531). Significant research has taken place to find optimal methods for reflector washing using a variety of methods. Consensus stands that the best overall method is a deluge flush followed by a high-pressure direct pulsating spray carried out at night with demineralized water using either strategically positioned overhead pipes or a mobile washing unit (Kalogirou, 2009, p. 531).

CSP installations may be used for more than simply heating a HTF. The energy contained by the photons may be used to affect a chemical change rather than only a thermal change. Such a process is known as photolysis; which when used in conjunction with a catalyst is known as photocatalysis (Goswami, 2015, p. 559).

Photolysis and photocatalysis have found application mostly in disinfection and detoxification procedures (Glaztmaier & Bohn, 1993). Straightforward processes may involve concentrating sunlight using the UV spectrum for the sterilization of bacteria, fungus and other microbes in liquid and gas streams.

Using concentrated light for photolysis and photocatalysis may improve the cost effectiveness of certain processes (Blake, 1994; Elizardo, 1991). For example, using

Table 2.3.2 1: Properties of Various Non-Thermal Energy Storage Technologies (Barton & Infield, 2004)

Technology	Energy Cost €/kWh/Year	Power Cost €/kW/Year	Independence of power rating to energy rating?	Characteristic Time, Hours	Round Trip Efficiency %
Hydrogen	6.8	128	Y	12 +	32
Vanadium Redox Flow Cells	12.5	144	Y	2 to 12	75
Regenesys Redox Flow Cells	9.4	70	Y	6 to 12	75
Nickel Metal Hydride	24	96	N	1 to 4	64
Sodium Sulphur	34	132	N	4 to 8	87
Nickel Cadmium	117	351	N	1 to 10	72
Zinc Bromide	79	158	N	2 to 5	70
Lead Acid	55	73	N	1 to 5	63
High Speed Flywheel	77	0.88	N	10^{-4} to 10^{-2}	89
Super Capacitor	570	4.8	N	10^{-4} to 10^{-2}	86
SMES	3×10^5	47	N	10^{-3} to 10^{-4}	21

TiO₂ as a catalyst in a water stream undergoes the photoelectric effect to produce hydroxyl radicals, which have twice the relative oxidation power as chlorine while being able to break down halogenated organics, herbicides, pesticides and surfactants.

A major advantage for CSP systems over PVs other renewable technologies is the relatively straightforward management and storage of the primary collected energy component – heat (Barton & Infield, 2004). While electrical and even kinetic energy can be collected, buffered and stored, making use of the stored potential electrical or kinetic energy by conventional methods is either inefficient or prohibitively expensive, especially at smaller scales (Barton & Infield, 2004).

Table 2.3.2 1 compares some commercially available technologies used for non-thermal energy storage.

In contrast, heat energy can be stored cost-effectively with high energy retrieval efficiencies for multiple days (Duffie & Beckman, 2013, p. 373). Heat storage is used in cases where power load requirements are not necessarily present at the same heat supply is available. Such uses include smoothing daytime power output, electricity demand price arbitrage and mitigation against inclement weather.

Heat energy may be stored as sensible heat, latent heat and/or a reversible chemical reaction (Barton & Infield, 2004). The chosen method of heat storage depends on the energy source, heat flux density, operating temperatures, temperature stratification, volumetric heat capacity, storage and retrieval efficiency, cost effectiveness and the purpose of the stored heat (Barton & Infield, 2004; Duffie & Beckman, 2013, pp. 373,374).

Packed beds store energy as sensible heat within large piles of rock and/or concrete (Duffie & Beckman, 2013, pp. 384,385). Depending on the packing, packed beds are capable of operating over a very large temperature range and are normally the most cost-effective method for heat storage. However, packed beds are usually associated with low volumetric heat capacities and heat may not be simultaneously added and extracted (Duffie & Beckman, 2013, p. 385).

Latent heat storage may be done in tanks with the material choice linked to the operating temperature of the application. Water/steam is inexpensive and commonly used, whereas molten salts, eutectic mixtures and even molten metals may be used at higher temperatures (Duffie & Beckman, 2013, p. 397; Kauffman & Grunfest, 1973; Morrison & Abdel-Khalik, 1978).

While reversible thermochemical energy storage mechanisms have been studied in detail, very few practical or commercial demonstrations have been produced (Duffie & Beckman, 2013, p. 401). High temperature metal and non-metal oxides show promise with potential temperature operating ranges between 300-900 °C with materials such as KO_2 having a heat of decomposition of 2.1 MJ/kg (Duffie & Beckman, 2013, p. 401; Offenhartz, 1976).

An alternate method of storing heat energy in a concentrated form is known as fuel-reforming (Goswami, 2015, p. 211). This process involves mixing methane and water at high temperatures to produce CO and H_2 syngas. The syngas may be reversibly converted back into water and heat, or further refined into other fuels or have the hydrogen separated and used in fuel cells (Goswami, 2015, p. 211; Kalogirou, 2009, p. 401).

In domestic, industrial or utility scale operations with a conducive geographic layout, SECs may have their waste heat harnessed to produce hot water (Duffie & Beckman, 2013, p. 376). In some cases, this may be more economically advantageous than adding a separate heat recovery unit such as an ORC to produce electricity (Kalogirou, 2009, p. 533).

2.3.3 Brayton and Rankine Cycles

Linear collector CSP installations are almost exclusively associated with Rankine Cycle Heat Engines (Chen, et al., 2007, pp. 512-525; Lloyd & Moran, 1974, p. 443). Point focus receivers have been routinely experimented, piloted and commercially used (with varying degrees of success) in combination with Rankine, Brayton and Sterling Type Heat Engines (Duffie & Beckman, 2013, p. 629).

A significant problem with Brayton and Sterling Cycle Heat Engines are physical limitations of the relatively low heat conductivity of the gaseous fluids used (Duffie & Beckman, 2013, p. 629). By the nature of a point focus receiver, there is usually very little relative surface area for heat to be conducted to the HTF or working fluid (Duffie & Beckman, 2013, p. 629).

It has been found that it is often the case that for a particular given SEC, operation at a lower temperature with a Rankine Cycle instead of a higher temperature Brayton or Sterling cycle results in a higher overall effective operational efficiency or cost-effectiveness – despite the lower thermodynamic efficiency of the lower temperature operation (Duffie & Beckman, 2013, p. 629).

Brayton and Rankine cycle engines are heat agnostic and may be run in a hybrid fashion where heat is accepted into the HTF by separate solar and fuel combustion units (Schwarzbözl, et al., 2006, pp. 1231-1240).

If the fuel source is vast enough, it may be more efficient overall to instead primarily use the heat from the combustion process to run a high temperature GT, and reheat the exhaust gases by means of solar for use in an ORC or similar heat recovery unit (Goswami, 2015, p. 483). This type of setup is known as an Optimized Hybrid Integrated Solar Combined-cycle system (ISCCS) ISCCSs typically operate at about 58% overall thermal efficiency with capital costs in the order of 1000-2000 USD/kW (Kalogirou, 2009, p. 528).

The vast majority of commercial CSP installations produce superheated steam (either directly or by means of another HTF) at 400°C driving Rankine Cycle Turbines ranging from 10s to 100s of MW_e (Zhang, et al., 2013). Another common use for the steam is to drive desalination processes (Kalogirou, 2009, p. 26). ORC installations are more often used for relatively small installations of up to a few MW_e with operating temperatures between 70-300°C (Goswami, 2015, p. 415).

Rankine Cycles (especially steam based) usually require a substantial amount of water for use in conventional cooling towers to reject the heat and entropy from condensers (Goswami, 2015, p. 426; Kalogirou, 2009, p. 531). This therefore limits the use of the technology in particularly arid environments such as deserts where solar energy is otherwise plentiful.

A Rankine Cycle may be operated in a process known as a Supercritical Rankine Cycle (SRC) (Goswami, 2015, p. 424). In this configuration, the working fluid is pressurized up to a supercritical state such that the isothermal isobaric vaporization stage present in a standard Rankine Cycle is skipped entirely greatly increasing the efficiency of the heat exchange process (Goswami, 2015, p. 424). While steam may be used in this configuration, the temperatures and pressures that are necessary to do so make this configuration expensive and hazardous. Organic fluids are therefore often used for their lower critical temperatures and pressures as SRC ORCs which results in higher efficiencies than standard ORCs (Goswami, 2015, p. 424).

A relatively novel idea in the field of Brayton Cycle based CSP is the use of a radial flow compressor/turbine mechanism instead of an axial flow mechanism such as those used in conventional GTs (Jansen, et al., 2015; Le Roux, et al., 2011; Mariscal-

Hay & Leon-Rovira, 2014). Particular emphasis has been placed on the use of commercial radial flow vehicle turbochargers for this purpose where their lower operational efficiencies is offset by their smaller power ratings, low capital and operational costs and high reliability.

A series of theoretical studies were performed by Le Roux et al. in 2011 and 2012 investigating the feasibility of using a modified vehicle turbocharger to act as the compressor and turbine in an air-based BCHE in conjunction with a large PDR fitted with a cavity-type receiver (Le Roux, et al., 2011; Le Roux, et al., 2012). For the chosen turbocharger and SEC, it was calculated that the engine is theoretically capable of operating at about 30% thermal efficiency to produce 60kW of shaft work from 201 kW_{th} of collected solar energy.

This work was further iterated upon the above design through the addition of a second heat regenerative turbine-compressor stage with an accompanying recycler (Jansen, et al., 2015). It was found that the system could perform at up to 41% overall solar-to-work thermal efficiency.

Similar studies have been performed analysing the performance of commercial turbochargers as a power generating unit. It has been shown that a straightforward single pass open-air design is capable of operating at about 20% thermal-to-work efficiency without the use of heat recycling or recovery units (Mariscal-Hay & Leon-Rovira, 2014).

2.4 Fundamentals of Concentrated Solar Power

Theoretical CSP Heat Engine and Operating Temperature Efficiencies

The efficiency of a solar powered heat engine can be modelled as a function of the solar collector's efficiency used in conjunction with a Carnot engine (Fletcher, 2000). If it is assumed that the temperature of the receiver's surface is the same as the fluid passing through the receiver where that fluid is the source of heat for the Carnot Engine's heat reservoir; then it stands that (Fletcher, 2000, p. 66):

$$\eta_{thermal} \equiv \frac{P_{work}}{P_{thermal}} = \eta_{receiver} \cdot \eta_{Carnot} \quad (2.4 \ 1)$$

Where

$$\eta_{Carnot} = 1 - \frac{T_C}{T_H} \quad (2.4 \ 2)$$

And

$$\eta_{receiver} = \frac{Q_{absorbed} - Q_{lost}}{Q_{solar}} \quad (2.4 \ 3)$$

Where Q_{solar} is the insolation radiant flux, $Q_{absorbed}$ is the heat absorbed by the receiver surface and Q_{lost} is the heat lost by the collector due to conduction, convection and radiation.

For a receiver with a black body surface enclosed within a vacuum (i.e. no convective or conductive heat losses) (Kalogirou, 2009, p. 181):

$$Q_{solar} = \eta_{optics} I C A_{receiver} \quad (2.4.4)$$

$$Q_{absorbed} = \alpha Q_{solar} \quad (2.4.5)$$

$$Q_{lost} = A_{receiver} \epsilon \sigma T_H^4 \quad (2.4.6)$$

Explicitly, $A_{receiver}$ is the ‘active’ surface area to which the insolation is concentrated and C is the Concentration Ratio of the concentrator’s insolation collection area to the active area of the receiver. η_{optics} is the efficiency of the reflector(s) surface(s) of the concentrator(s) (i.e. the percentage of captured light successfully reflected and concentrated onto the receiver(s)). I is the standard non-concentrated solar insolation, α is the absorbance of the receiver surface, ϵ is the surface emissivity and σ is the Stephan-Boltzmann Constant.

For high temperature receivers it may be assumed that losses associated with the receiver are essentially only radiative (Steinfeld & Palumbo, 2001, p. 6). In these cases, losses due to convection are negligible especially with the receiver enclosed in a vacuum, while the structure of the collector may be practically designed to limit conduction losses.

For a black radiative body to be used as the receiver’s surface, and assuming perfect reflector operation: $\alpha = \epsilon = \eta_{optics} = 1$, which reduces Equation 2.4.1 to (Fletcher, 2000, p. 66):

$$\eta_{thermal, Black\ Body} = \left(1 - \frac{\sigma T_H^4}{I C}\right) \cdot \left(1 - \frac{T_C}{T_H}\right) \quad (2.4.7)$$

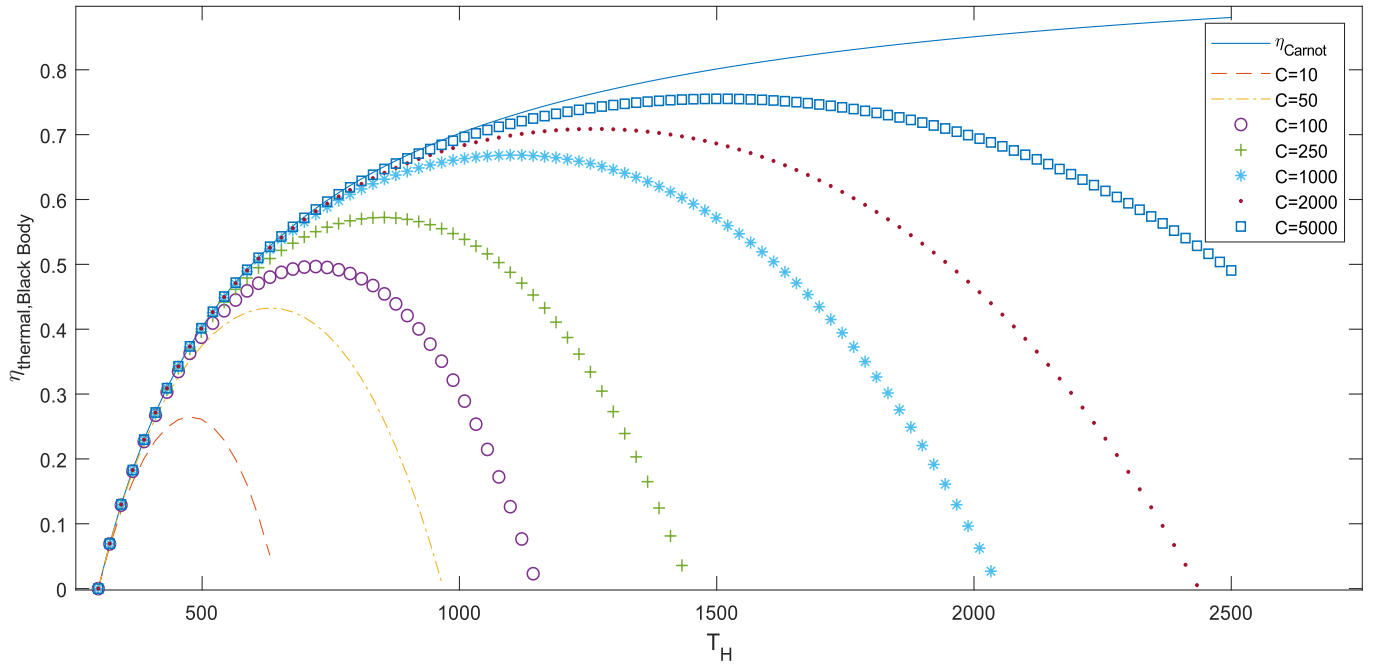


Figure 2.4.1: Thermal Efficiencies of a Black Body receiver surface CSP Carnot Engine

Plotting Equation 2.4 7 for a range of T_H and C values produces Figure 2.4 1. It can be seen that for a given concentration ratio, there exists an optimal operating temperature, as well as a maximum obtainable temperature of the receiver's surface.

The maximum operating temperature T_{max} may be calculated where (Fletcher, 2000, p. 66):

$$0 = \left(1 - \frac{\sigma T_{max}^4}{IC}\right) \cdot \left(1 - \frac{T_C}{T_{max}}\right) \quad (2.4 8)$$

$$\therefore 0 = \left(1 - \frac{\sigma T_{max}^4}{IC}\right) \quad (2.4 9)$$

$$\therefore T_{max} = \sqrt[4]{\frac{IC}{\sigma}} \quad (2.4 10)$$

Similarly, the optimum temperature T_{opt} may be calculated as (Fletcher, 2000, p. 66):

$$\begin{aligned} \frac{d}{dT_H}(\eta_{Thermal, Black Body}) &= 0 \\ \therefore 0 &= T_{opt}^5 - \frac{3}{4} T_{opt}^4 T_C - \frac{T_C IC}{4\sigma} \end{aligned} \quad (2.4 11)$$

Figure 2.4 2 has been generated from Equations 2.4 10 and 2.4 11. From Figures 2.4 1 and 2 it can be seen that increasing the concentration ratio C leads to diminishing returns on the maximum temperature and overall optimal system

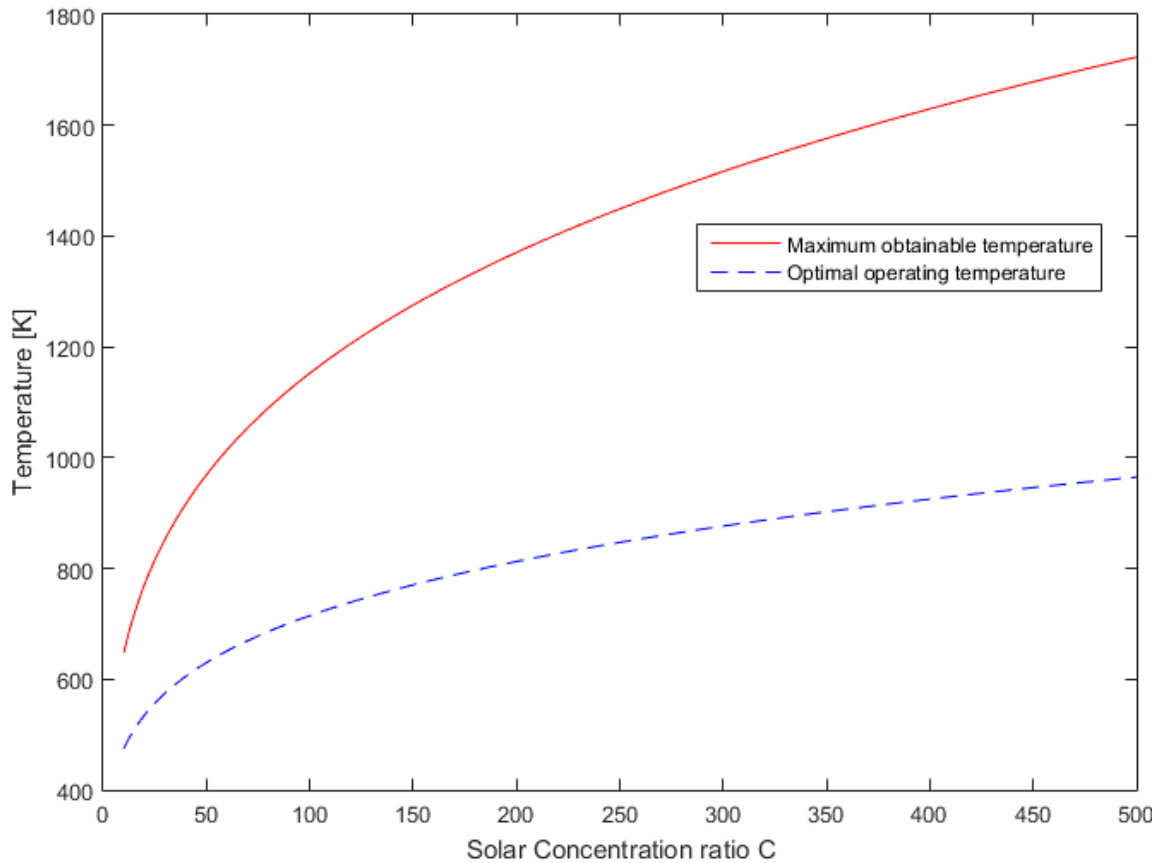


Figure 2.4 2: Maximum and Optimal Temperatures of a Black Body receiver surface CSP Carnot Engine

efficiency for a CSP Engine, irrespective of the SEC technology used (Fletcher, 2000, p. 66).

An alternative approach to find the optimal operating temperature of a SEC connected to a Carnot Engine for a given concentration ratio is to instead optimize for the recovered exergy of the supplied heat (Kalogirou, 2009, pp. 208-210). This approach is characterized by minimizing the entropy of the collector attached to the engine. If it is assumed that heat losses are a linear function of receiver temperature in an ambient surrounding, it has been shown that: (Kalogirou, 2009, pp. 209,210)

$$T_{opt} = \sqrt{T_{max} \cdot T_{ambient}} \quad (2.4\ 12)$$

Both approaches of estimating optimal operating temperature suggest operating the CSP Engine at higher flowrates than otherwise possible (Kalogirou, 2009, p. 210). In this fashion the operating temperature is to be intentionally kept at less than that maximally obtainable.

The performance of SECs is intrinsically linked to the total irradiance – and by extension insolation - available (Duffie & Beckman, 2013, p. 236). The value for Direct Normal Irradiance (DNI) flux density is dependent on factors such as the weather, season, sky clearness, distance from the equator and the time of day. The measured value for DNI on the surface of the Earth is generally in the region of about 1000 W/m² (Goswami, 2015, pp. 74-76).

From the perspective of the surface of the Earth, the Sun appears as a disc which subtends an angle of about 32 arc minutes on average throughout the year (Goswami, 2015, p. 168). This places a practical limit on the maximum possible concentration ratio possible for a single imaging collector. The maximum CR possible in air for tracking the sun in one axis (i.e. concentrating in one dimension such as a PTC) is about 216, while two axis tracking (and concentrating in two dimensions by means of a PDR) is limited to a CR of about 46'700 (Kalogirou, 2009, pp. 181-183).

2.5 Physical and Modelling Characteristics of Linear Receivers

SECs of the single axis tracking variety are subject to losses from the angle of incidence that exists between the plane of the collector's aperture and that of the incident solar rays, as well as shadow overlap on the ends of the receiver (Goswami, 2015, pp. 176-178). For example, a PTC in an E-W configuration will only have an incidence angle of zero at noon zenith, whereas in a N-S configuration there will always be some incidence losses.

Losses as a result of the angle of incidence may therefore be included in a function describing the optical efficiency ($\eta_{optical}$) of a linear receiver (Goswami, 2015, p. 178):

$$\eta_{optical} = \rho_{mirror} \tau \alpha_{absorber} \gamma_{SEC} F_{soiling} F_{end\ losses}(i) K(i) \quad (2.5\ 1)$$

Where ρ_{mirror} is the reflectance of the collector's mirror when clean (typically about 0.93); τ is the transmittance factor of any transparent anti-convection losses shielding (around 0.93-0.96 for glass); $\alpha_{absorber}$ is the absorptance of the absorber surface (typically 0.94-0.95); γ_{SEC} is an intercept factor which handles tracking error, support structure shading and mirror imperfections (typically 0.92-0.94); $F_{soiling}$ represents phenomena such as dust coverage on the mirror(s) (typically 0.97); and finally $F_{end losses}$ and K represent end losses and incidence angle modifiers which are functions of the SEC construction and the angle of incidence at a certain time of day (Goswami, 2015, p. 178).

Functions for K and $F_{end losses}$ may be found for various popular PTC designs in literature (Vasquez-Padilla, 2011).

For a PTC collector and an apparent solar disc angle of 32 arc minutes (for an optical half acceptance angle of 16 arc minutes, or 16/60 radians), the minimum diameter of the receiver tube necessary to collect all of the light captured by the reflector may be calculated as a result of the concentrated projected image of the Sun as (Duffie & Beckman, 2013, p. 353):

$$D_{min} = 2r_{rim} \sin\left(\frac{16}{60}\right) = \frac{a \cdot \sin\left(\frac{16}{60}\right)}{\sin(\phi_{rim})} \quad (2.5.2)$$

Where D_{min} is the minimum receiver diameter, r_{rim} is the radius from the centre of the receiver to the rim of the concentrator; a is the aperture of the concentrator (the distance from one rim to the other); and ϕ_{rim} is the rim angle (half the total angle of the arc formed by the parabola).

In order to model linear receivers that undergo a large temperature gradient along the length of the receiver, it is necessary to divide the receiver into smaller sections such that each section may have its heat loss variables calculated individually (National Renewable Energy Laboratory, 2003, pp. 22-25). Convective and radiative heat losses are functions of the temperature of the receiver at each point and therefore heat loss dynamics change significantly along the length of the receiver (Stuetzle, et al., 2004, pp. 187-193).

Real world surfaces have values for emissivity and absorptivity which are functions of the surface's temperature (Duffie & Beckman, 2013, pp. 188-199). It is possible to select a surface that displays the characteristics of a high absorptivity (that is a low reflectivity for photons specifically within the solar spectrum) and a low emissivity (as compared to the rate of energy radiated by a black body at that temperature).

Such a surface readily absorbs solar energy where it is converted into heat while at the same time limits the rate of energy radiated away from the surface (Duffie & Beckman, 2013, p. 188). This is known as a solar selective surface, and is ideal for use in SEC receivers – and is especially important for receivers of large surface areas.

Usmani and Harinipriya (2015) have compiled an extensive summary of modern technologies and approaches in the development of solar selective surfaces and

coatings (Usmani & Harinipriya, 2015). Recent developments of high temperature solar selective surfaces employ one or more of the following materials and techniques: refractory oxide coatings with interspersed noble metal nanoparticles; ceramic-metal “cermet” compounds made from metal and dielectric composites; and transition metal nitride matrices and substrates.

Pt-Al₂O₃ has shown promise as a contemporary solar selective absorber cermet (Usmani & Harinipriya, 2015, p. 187). It has been shown to have excellent solar selectivity ($\alpha = 0.92$, $\epsilon = 0.14$ at 300 °C) as well as a high thermal stability; being stable in air at 650 °C (Nuru, et al., 2011). However, a primary problem preventing its wide-scale commercial adoption as a selective coating is the high cost of the platinum used in its production (Usmani & Harinipriya, 2015, p. 185).

Rubbia et al. (2004) have developed a multi-layered Mo, Mo-SiO₂, SO₂ cermet series claiming superb solar selectivity at temperatures up to 580°C ($\alpha = 0.93$, $\epsilon = 0.065$ at 580 °C) (Rubbia, et al., 2004).

Hernández-Pinilla et al. (2016) investigated MoSi₂-Si₃N₄ composite and reported thermal stability at 600 °C in a vacuum and no evidence of degradation when operating at 650 °C ($\alpha = 0.88$, $\epsilon = 0.11$ at 600 °C) (Hernández-Pinilla, et al., 2016).

Escobar-Galindo et al. (2018) investigated Al_yTi_{1-y}(O_xN_{1-x}) compounds which showed thermal stability in air at 600 °C and resistance to degradation in air up to 800 °C ($\alpha = 0.91$, $\epsilon = 0.14$ at 600 °C) (Escobar-Galindo, et al., 2018).

Elam et al. (2017) have reported a unique cermet coating of W:Al₂O₃ whereupon a silicate substrate with monodisperse polystyrene and polymethylmethacrylate nanoparticles are used as sacrificial materials in an Atomic Layer Deposition reactor. The result is cermet which is stable up to 700 °C ($\alpha \geq 0.9$, $\epsilon \leq 0.1$ at 700 °C) (Elam, et al., 2017).

An exhaustive list of functions detailing phenomena related to heat transfer to the HTF and losses to the atmosphere which are used in the rest of the dissertation are detailed in Appendix A.

2.6 High Temperature Linear Receivers and High Temperature Turbocharger and Brayton Cycle Thermodynamics

In February 2011 the United States Department of Energy launched the SunShot Initiative with the objective of drastically reducing the cost of solar energy (Solar Energy Technologies Office, 2017, pp. 1-3). The goal for SunShot is set at reducing the levelized cost of energy production (LCOE) to 0.06 US\$ per kWh by 2020, and to 0.03 US\$ per kWh by 2030. In September 2017 announced the goal of obtaining a LCOE of 0.06 US\$ per kWh was reached ahead of schedule (Solar Energy Technologies Office, 2017, p. 1).

The Solar Energies Technologies Office provides funding to universities and businesses for related research (Solar Energy Technologies Office, 2017). In 2012,

US \$56M was awarded as part of the CSP SunShot R&D programme. In 2015, a further US \$29M was awarded to the CSP SunShot National Laboratory Multiyear Partnership (SuNLaMP) programme, with an additional US \$32M to the Advanced Projects Offering Low LCOE Opportunities (APOLLO) programme. In May 2018, US \$62M was awarded to the Generation 3 Concentrating Solar Power Systems (Gen3 CSP) programme (Solar Energy Technologies Office, 2018).

One of the projects under SuNLaMP is dedicated to the development and testing of a high temperature linear receiver named the SunTrap (Stettenheim, 2016; Obrey, et al., 2016). SunTrap is an innovative design for the receiver of a parabolic trough designed to operate at a temperature of 750 °C while targeting a low cost of production (Stettenheim, 2016, p. 6).

The SunTrap design involves mechanically connecting the receiver to the trough such that the receiver cavity and the mirrors tilt simultaneously to follow the sun. The cavity itself does not undergo a vacuum evacuation process opting rather for a thick solar selective glass cover, further increasing operational reliability and reducing production costs (Obrey, et al., 2016, pp. 11-14). Figure 2.6 1 depicts the design of the SunTrap Receiver cavity.

SunTrap makes further use of air-stable cermet coatings for the absorber and includes an anti-reflective coating on the outside of the receiver to reduce incidence losses, and an IR reflective coating for use on the inside of the cavity to reduce radiative losses (Obrey, et al., 2016, p. 13).

One of the directives of the SunShot, SuNLaMP, APOLLO and Gen3 CSP programmes is the focus on the development and testing of Supercritical Carbon Dioxide Brayton Cycles (sCO₂) (Solar Energy Technologies Office, 2018). A sCO₂ power cycle operating at a temperature of about 700 °C represents an improvement of about 21% thermal efficiency over a steam Rankine cycle operating at 400 °C (Obrey, et al., 2016, p. 18).

sCO₂ represents a viable technology which may replace conventional steam Rankine based power cycles, principally for the next generation of nuclear reactors, Generation IV (Ahn, et al., 2015, p. 1). While there is significant interest from the nuclear power sector, sCO₂ promises similar gains in performance for coal, waste heat and solar thermal energy sources (Ahn, et al., 2015, pp. 1,2).

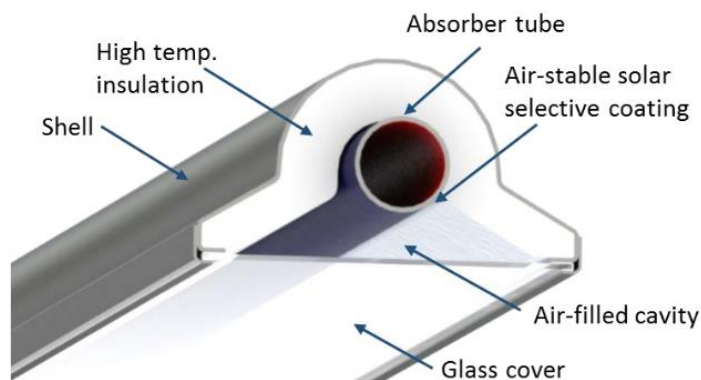


Figure 2.6 1: SunTrap Receiver Concept Art (Obrey, et al., 2016)

The primary benefit of sCO₂ is operation at higher temperatures (approximately 500-900°C) than that of conventional technologies (up to about 550°C) translating to higher thermal efficiencies (Ahn, et al., 2015, p. 2). Using higher temperatures in steam Rankine systems requires the use of the ultra-supercritical steam cycle, which accelerates material degradation in the turbine (Ahn, et al., 2015, p. 2). sCO₂ turbines may be built out of conventional materials which show reduced signs of strain and wear in the absence of water (Ahn, et al., 2015, p. 2).

By operating near the critical point, the compression stage in sCO₂ requires far less relative work than standard Brayton and Rankine cycles (Ahn, et al., 2015, p. 3). As sCO₂ operates at a fairly low pressure ratio compared to steam Rankine, the turbine outlet temperature is relatively high thus necessitating a heat recycling stage (Ahn, et al., 2015, p. 3).

A particular challenge for sCO₂ is the large change in CO₂'s heat capacity of two to three fold between the hot side flow and the cold side flow, therefore complicating the design of the heat recycling unit (Ahn, et al., 2015, p. 3).

There are many permutations of proposed plant and flow layouts for sCO₂ Brayton Cycles (Ahn, et al., 2015, p. 3; Crespi, et al., 2017). Each of these permutations aim to reduce heat wastage in the heat recycling stage by different methods, but usually by splitting and recombining the CO₂ stream between different radiator and recompression stages. Over 80 layouts have been formally proposed of varying degrees of complexity and thermal efficiencies. sCO₂ cycles tend to operate at about 40% thermal efficiency, while combined sCO₂-Rankine cycles have been calculated to obtain 50-60% thermal efficiency (Crespi, et al., 2017).

One advantage of sCO₂ over conventional steam Rankine in particular is the possibility of operating the power plant without cooling water; instead opting for an air-cooling mechanism (Ahn, et al., 2015, p. 4). This would allow for sCO₂ operation in arid environments which is particularly suitable for solar based applications in desert areas – assuming the capital costs could be justified for the surface area required of the air-cooler.

Muñoz-Anton et al. (2014) set about outlining the theoretical basis and subsequent experimental testing of a gas based high temperature parabolic trough (Muñoz-Anton, et al., 2014, pp. 373-378). The primary motivation for this endeavour was testing the viability of obtaining a high operating temperature (and thus a higher 2nd Law efficiency for a greater potential thermal efficiency for an attached engine) using a low-cost (relative to point receivers) PTC setup (Muñoz-Anton, et al., 2014, pp. 374, 375).

Additionally, by using a gas as the main circulating fluid, operation at high pressure is possible (Muñoz-Anton, et al., 2014, p. 375). This significantly reduces pumping losses, as the required pumping power of a compressible fluid is proportional to the inverse of the pressure squared (Muñoz-Anton, et al., 2014, p. 375).

It has been estimated that sCO₂ power cycles produce 5-10% net greater power output than similar heat duty steam Rankine cycles due to the decrease in parasitic pumping losses alone (Obrey, et al., 2016, p. 18).

Gasses such as air, N₂, CO₂ and He are all non-flammable, non-toxic, and do not have maximum operating temperatures such as the oils used as conventional HTFs (Muñoz-Anton, et al., 2014, p. 375).

The high temperature linear receiver approach was shown to be experimentally validated in the test performed by Muñoz-Anton et al. (2014) at the Plataforma Solar de Almería solar energy testing site in Spain (Muñoz-Anton, et al., 2014, pp. 377-380).

In the test, CO₂ gas was heated to 500°C at 65 bar by means of a parabolic trough as the SEC. It was identified, however, that high temperature linear receivers have a propensity to form leaks due to differences in thermal expansion between joints, bearings, and different receiver and shielding materials of construction (Muñoz-Anton, et al., 2014, p. 380).

Grena and Tarquini (2011) performed a series of simulations of a cermet based high temperature LFR utilizing molten salt nitrates as the HTF (Grena & Tarquini, 2011, pp. 1048,1049). It was calculated that at an operating temperature of 550°C, the LFR setup would yield an estimated 10-20% lower thermal transfer efficiency than a PTC of similar aperture (Grena & Tarquini, 2011, p. 1054). However, the LFR approach would incur a 32% relative cost savings per square meter of mirror area over the PTC approach, for an estimated net cost saving of 25% per unit of thermal energy (Grena & Tarquini, 2011, p. 1054).

2.7 Turbocharger and Brayton Cycle Thermodynamics

The Brayton Cycle is a type of heat engine where the working fluid (normally air or CO₂) is compressed, heated isobarically, then decompressed with work extracted (Dhanireddy, 2010, p. 2).

In the case of air as the working fluid, combustion of hydrocarbons may be used as the primary heat source, where such a BCHE is known as a Gas Turbine (Dhanireddy, 2010, p. 1).

Most designs for both axial and radial flow BCHEs have the compressor and turbine share a common shaft (Dhanireddy, 2010, p. 1). More advanced designs may include a transmission and gearing system between the two stages, or even have the compressor and turbine exist mechanically separated from one another (Dhanireddy, 2010, p. 5).

According to the First Law the maximum amount of energy that can be extracted as shaft work is the difference between the total energy in and total energy out (Dixon & Hall, 2014, p. 7):

$$W_{net} = Q_{in} - Q_{out} \quad (2.7 \ 1)$$

Assuming the heat addition and removal processes are isobaric, ideal gas law holds and that the compression and decompression are adiabatic reversible, it can be

shown that: (Boyce, 2006, p. 59; Dhanireddy, 2010, p. 3; Saravanamuttoo, et al., 2017, p. 46)

$$\therefore \eta_{Brayton,thermal,max} = \frac{Q_{in} - Q_{out}}{Q_{in}} = \frac{W_{net}}{Q_{in}} = 1 - \frac{T_{compressor,inlet}}{T_{compressor,outlet}} \quad (2.7\ 2)$$

And if compression is assumed to be adiabatic:

$$\eta_{Brayton,thermal,max} = 1 - \frac{1}{\left(\frac{p_{compressor,outlet}}{p_{compressor,inlet}}\right)^{\frac{\gamma-1}{\gamma}}} \quad (2.7\ 3)$$

Real-world large-scale implementations of BCHEs such as those used in aircraft and open cycle GTs operate at pressure ratios of 14-20, corresponding to overall cycle thermal efficiencies of approximately 45-55% (Boyce, 2006, pp. 59,60).

A turbocharger is a type of BCHE primarily used with internal combustion engines. Waste heat is harnessed from the engine's exhaust as it is run through a turbine to generate shaft work (Nguyen-Schäfer, 2015). This turns an air compressor positioned prior to the internal combustion engine's air inlet. By increasing the pressure of fresh air available to the internal combustion engine's intake, a greater mass of oxygen is available to be burnt per stroke, thereby increasing the power the engine is capable of producing (Nguyen-Schäfer, 2015).

For an ideal turbocharger made up from an ideal compressor and ideal turbine, following from the First Law and assuming adiabatic operation (Dixon & Hall, 2014, pp. 14-17; Smith, et al., 2018, p. 182):

$$\Delta H = Q + W_s = W_s \quad (2.7\ 4)$$

$$dH = C_p dT + V \left(1 - \frac{1}{V} \left(\frac{\partial V}{\partial T} \right)_p T \right) dT \quad (2.7\ 5)$$

Assuming an ideal gas:

$$\Delta H = \int_{T_{in}}^{T_{out}} C_p dT = W_s \quad (2.7\ 6)$$

Similarly, from the Second Law:

$$dS = \frac{C_p}{T} dT - \frac{R}{P} dP \quad (2.7\ 7)$$

$$\therefore \Delta S = \int_{T_{in}}^{T_{out}} \frac{C_p}{T} dT - R \ln \left(\frac{P_{out}}{P_{in}} \right) \quad (2.7\ 8)$$

For the turbine section of the Brayton Cycle, the maximum amount of energy that can be converted into shaft work occurs at the point where no entropy is generated, i.e. $\Delta S = 0$ (Smith, et al., 2018, p. 190). Similarly, the least amount of energy required for compressing air in the compressor also occurs when the process is isentropic (and therefore reversible).

$$\Delta S = \int_{T_{in}}^{T_{out}} \frac{C_p}{T} dT - R \ln \left(\frac{P_{out}}{P_{in}} \right) = 0 \quad (2.7 9)$$

For a turbocharger, the efficiency of the compressor is defined as (Nguyen-Schäfer, 2015, pp. 22,23):

$$\eta_{compressor} = \frac{\Delta H_{isentropic\ compression}}{\Delta H_{polytropic\ compression}} \quad (2.7 10)$$

Where $\Delta H_{polytropic\ compression}$ is the enthalpy change over the compressor that includes friction and other losses, and is a representation of the actual shaft work delivered to the compressor housing. In other words, the enthalpy change across a real compressor is greater than that of an ideal compressor since a real compressor requires more work to do the compression than an ideal compressor (Nguyen-Schäfer, 2015, p. 23).

Similarly for the turbine section (Nguyen-Schäfer, 2015, p. 24):

$$\eta_{turbine} = \frac{\Delta H_{polytropic\ expansion}}{\Delta H_{isentropic\ expansion}} \quad (2.7 11)$$

Where $\Delta H_{polytropic\ expansion}$ is a representation of the actual work delivered to the shaft of the turbine and is less than the value obtained through an isentropic expansion due to friction and other losses. In other words, less work is extracted from a real turbine than an ideal turbine for a given inlet temperature and pressure drop.

Due to friction losses in the central bearing system of the Centre Housing Rotating Assembly (CHRA) of the turbocharger, the turbine efficiency term is usually combined with a mechanical efficiency term η_{mech} . Manufacturers' curves may be interpreted as the following (Nguyen-Schäfer, 2015, p. 23):

$$\eta_C = \eta_{compressor} \quad (2.7 12)$$

$$\eta_T = \eta_{mech} \cdot \eta_{turbine} \quad (2.7 13)$$

Definitions used in manufacturer's curves for the compressor and turbine maps are as follows (Nguyen-Schäfer, 2015, pp. 21,27):

$$\dot{m}_c = \frac{\dot{m} \sqrt{\frac{T_t}{T_{STP}}}}{\frac{p_t}{p_{STP}}} \bigg|_{inlet} \quad (2.7 14)$$

$$T_t = T_s + \frac{c^2}{2C_p(T_s)} \quad (2.7 15)$$

$$p_t = p_s + \left(\frac{T_t}{T_s} \right)^{\frac{\gamma}{\gamma-1}} \quad (2.7 16)$$

Where:

- $\dot{m}_c$ is a dimensionless value called the corrected mass flow rate
- T_t is the total temperature
- p_t is the total pressure
- c is the velocity of the gas
- T_s is the static temperature of the gas (that is the temperature of the gas along the wall of the associated pipe)
- p_t and p_s are the total and static (i.e. measured at the wall of the pipe) pressures of the gas.

The overall performance of a BCHE is dependent on the performance characteristics of the compressor and turbine stages (Nguyen-Schäfer, 2015, p. 24). Large axial flow BCEs such as open cycle GTs used for electricity production or in aircraft are designed for optimal efficiency or effectiveness for a narrow range of operating conditions (i.e., electricity production at maximum duty, or aircraft maximum thrust and/or efficiency at cruising speed) (Boyce, 2006, p. 59).

Turbochargers on the other hand are designed to operate across a wide range of operating conditions given the transient nature of vehicle operation (Moustapha, et al., 2003; Nguyen-Schäfer, 2015, pp. 3,4).

Figures 2.7 1-3 depict typical operating characteristics of turbocharger turbines and compressors. As turbochargers experience a wide range of operating conditions during normal use, the phenomena of choke and surge to become apparent as they define the limits of the turbocharger's operation (Nguyen-Schäfer, 2015, pp. 27,153).

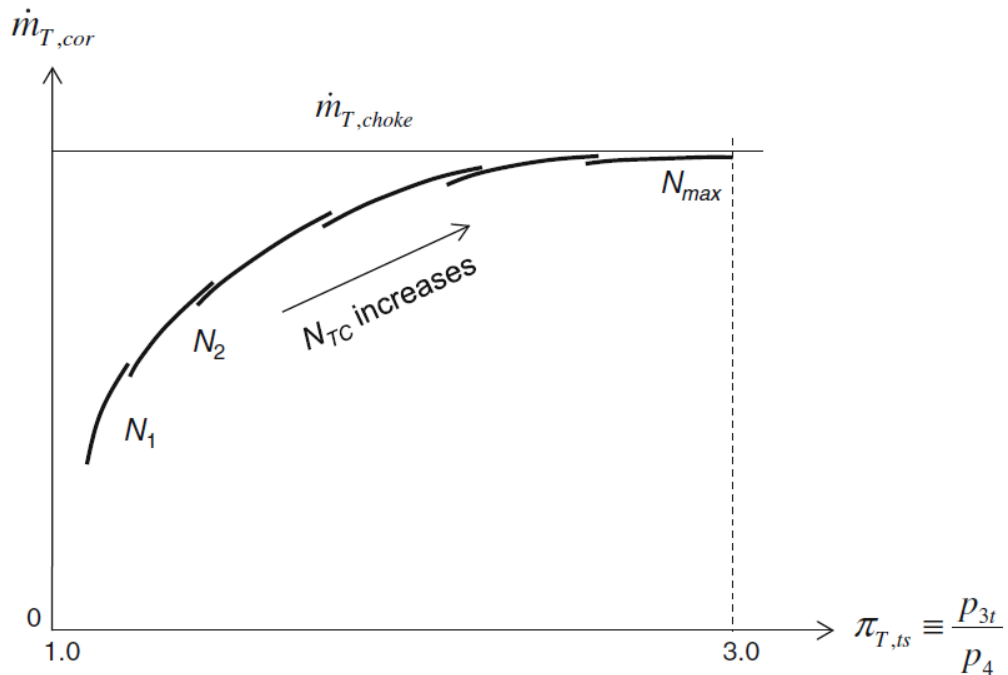


Figure 2.7 1: Typical Turbocharger Turbine Performance (Nguyen-Schäfer, 2015, p. 27)

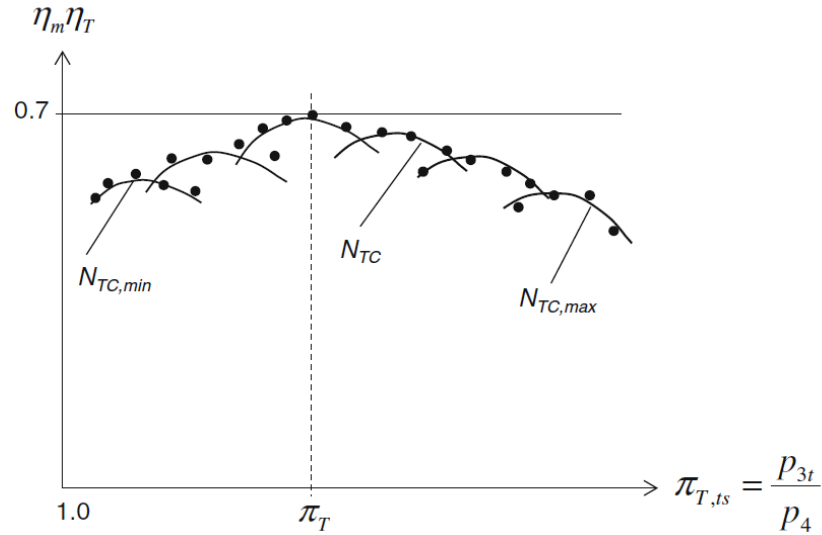


Figure 2.7 2: Typical Turbocharger Turbine Efficiency Performance (Nguyen-Schäfer, 2015, p. 28)

Choke occurs in either the compressor or the turbine sections at the point where the mass flow rate of air is greater than the capacity of the compressor or turbine to handle (Nguyen-Schäfer, 2015, pp. 27,28). When in choke, the rotational speed of the CHRA increases rapidly and the turbine and compressor efficiencies quickly deteriorate. This may lead to damage of the turbocharger's bearings as well as a high compressor outlet temperature.

Choke may be mitigated against in the turbine with the installation of a waste-gate (Nguyen-Schäfer, 2015, p. 14). The waste-gate provides a path for air to divert around the turbine and reconnect to the main exhaust stream at an exhaust manifold.

Operating within a turbocharger's choke region suggests the turbocharger is too small for the current application (Garrett, 2016, pp. 8,9).

Surge occurs in the compressor when there is insufficient mass flow to maintain the compressor's outlet pressure (Nguyen-Schäfer, 2015, p. 153). This leads to a stall of the compressor's inducer. Surge may lead to temporary reverse flow of gas backwards through the compressor until the stalling of the compressor impeller subsides.

If the mass flow is still insufficient for the pressure downstream of the compressor, the inducer stalls again (Nguyen-Schäfer, 2015, pp. 153,237). This manifests as the distinctive sound of a turbocharger surging. Surging may lead to turbocharger vibration with bearing and impeller damage.

A blow-off valve may be used to inhibit surging for turbochargers connected to internal combustion engines when the throttle is closed rapidly (Nguyen-Schäfer, 2015, pp. 7,8). The blow-off valve allows for the gas downstream of the compressor to vent if the pressure is too high for the given flowrate.

Operating within the stall region of a turbocharger suggests the use of a smaller turbocharger for the current application (Garrett, 2016, pp. 8,9).

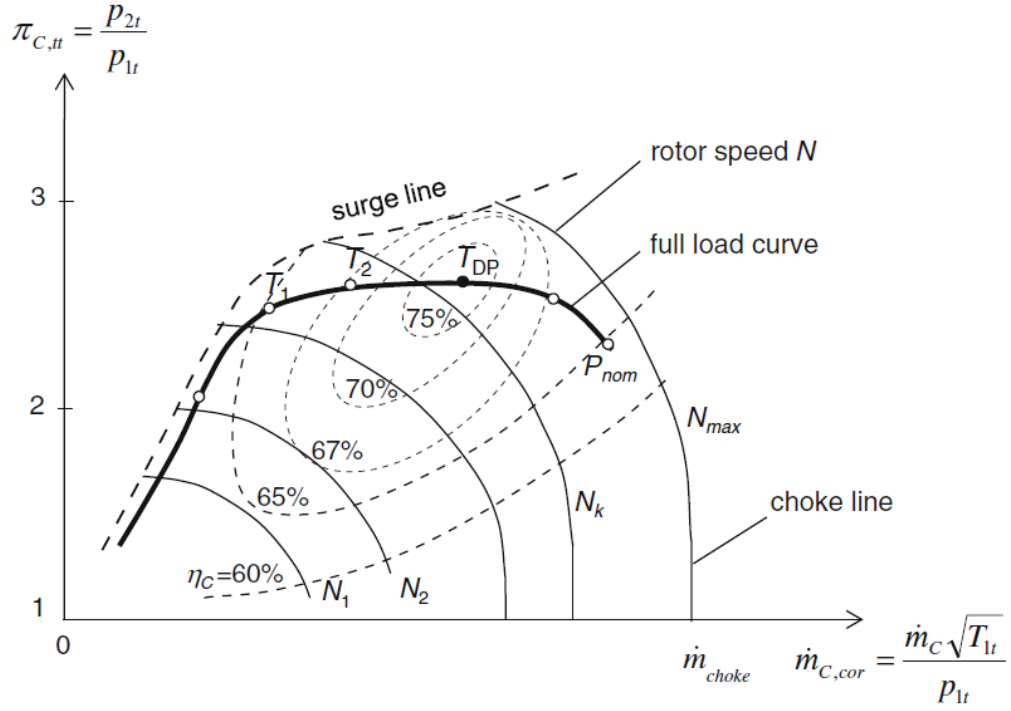


Figure 2.7 3: Typical Turbocharger Compressor Performance Map (Nguyen-Schäfer, 2015, p. 30)

The typical performance of the turbine section of a turbocharger is outlined in Figures 2.7 1 and 2.7 2.

As depicted in Figure 2.7 1, a mass flow rate generally corresponds to a certain expansion ratio ($\pi_{turbine,ts} \equiv p_{t,turbine,inlet}/p_{s,turbine,outlet}$) along with an associated small range of rotational velocities of the CHRA ($N_{turbocharger}$) (Nguyen-Schäfer, 2015, p. 27). A greater expansion ratio (i.e. higher turbine inlet pressure) corresponds to a greater flowrate and higher rotational velocity of the CHRA.

The maximum permissible expansion ratio of the turbine corresponds to the maximum mass flow rate of the turbine, as well as the maximum rotational velocity of the CHRA (Nguyen-Schäfer, 2015, pp. 27,28). A turbine inlet pressure greater than this will put the turbine into choke.

Figure 2.7 2 depicts the efficiency of the turbine section of a turbocharger as it relates to the turbine expansion ratio and CHRA rotational velocity.

There exists a certain expansion ratio which corresponds to a maximal efficiency of the turbine at a certain CHRA rotational velocity (Nguyen-Schäfer, 2015, p. 28). Operating at an expansion ratio higher or lower than this point corresponds to a decrease in turbine efficiency.

Therefore, there exists a certain operating condition where the expansion ratio, CHRA rotational velocity and mass flowrate correspond to the maximal efficiency possible for that turbine (Nguyen-Schäfer, 2015, p. 28).

Figure 2.7 3 depicts a typical turbocharger compressor performance map. The full load curve represents the acceleration of a vehicle engine when attached to a turbocharger (Nguyen-Schäfer, 2015, p. 30).

The surge line in the upper left portion of the compressor map defines the minimum mass flow rate required of a certain compression ratio to prevent the compressor from going into surge (Nguyen-Schäfer, 2015, p. 30).

Similarly, the lower right outline of the map defines the choke line as the maximum permissible mass flow rate for a given pressure ratio without the compressor operating in choke (Nguyen-Schäfer, 2015, p. 30).

In general, an increase in CHRA rotational velocity leads to an increase in mass flowrate and an increase in the compression ratio for the compressor (Nguyen-Schäfer, 2015, p. 29). For a certain CHRA rotational velocity, a higher pressure-ratio corresponds to a lower mass flowrate, and a lower pressure ratio to a higher mass flow rate.

Efficiency islands may be drawn on the compressor map which define the efficiency of the compressor within a region of pressure ratio and mass flow rate, along with a corresponding range of CHRA rotational velocities (Garrett, 2016, p. 7). The region of maximum compressor efficiency exists close to the surge and choke lines, at near the maximum compression ratio for the compressor.

Operating at a pressure ratio, mass flow rate or CHRA rotational velocity slightly greater than those at the optimal point leads to a sharp decrease in compressor efficiency (Nguyen-Schäfer, 2015, pp. 26-30).

2.8 DIY Turbocharger Turbojet Modification

The conversion of vehicle turbochargers to gas turbine engines is a fairly well documented topic by DIY turbojet enthusiasts. A plethora of videos and guides are available online on YouTube as well as dedicated internet forums. Turbocharger turbojets have been successfully used to propel scooters, motorcycles, go-karts, cars and trucks (Furze, 2018; Giandomenico, 2014; JATO, 2016; Popular Science, 2010).

A gas turbine consists of three operating components: the compressor, the combustion chamber, and the turbine (Nguyen-Schäfer, 2015, p. 3). A turbocharger conveniently provides a relatively well-matched combination of compressor and turbine in a single package.

In theory, therefore, all that is needed to convert a turbocharger to a gas turbine is a combustion chamber (Tech Ingredients, 2019, p. 11:14).

The most complicated task in the conversion of a turbocharger to a gas turbine is the fabrication of the combustion chamber (Tech Ingredients, 2019, p. 11:20). A liquid or gaseous fuel flame-front needs to be able to maintain itself without being blown out within a high velocity air stream.

The design of the combustion chamber is also complicated by the fact that combustion needs to be maintained for a relatively large variance of fuel and air

flow rates within normal start-up and operation parameters of the gas turbine (Tech Ingredients, 2019, p. 15:10).

Ideally all of the fuel injected to the combustion chamber is burnt such that all of the heat from the fuel is extracted. It is not possible to target a stoichiometric fuel-air ratio since flame temperatures would easily exceed 2000 K and readily melt most metals especially within localized regions (Tech Ingredients, 2019, p. 14:40).

A design known as the “flame tube” has proven popular with DIY turbocharger turbojet enthusiasts to address these concerns with a relatively simple fabrication procedure (Susante & Akker, 2004). A typical flame tube combustion chamber design is depicted in Figures 2.8 1 and 2.

Two concentric tubes of different diameters are welded to a bulkhead on one side and capped with a ring on the other side. An annular volume exists in the space between the two tubes (Susante & Akker, 2004).

The outer tube defines volume of the combustion chamber. The inner tube is known as the flame tube. The gas turbine’s air flow is introduced to the combustion chamber tangentially near the bulkhead. A vortex of air is formed in the space between the two tubes which travels in an axial direction away from the bulkhead toward the turbine (Tech Ingredients, 2019, p. 11:31).

Fuel is injected at the bulkhead inside of the flame tube (Tech Ingredients, 2019, p. 15:20). An automotive spark plug is usually mounted on the bulkhead to perform initial ignition of the flame.

Holes are drilled along the flame tube in bands such that air may enter into the flame tube in a controlled fashion (Tech Ingredients, 2019, p. 15:20). The positions and total areas of the holes are designed to promote complete combustion of the fuel for the entire range of air and fuel flowrates.

A small fraction of the air enters the flame tube initially and is used for fuel combustion (primary and secondary zones) (Boyce, 2006, p. 33). Further down the tube, the remaining air is mixed into the flame tube (dilution zone) thereby preventing melting of the tube (Boyce, 2006, p. 34). The highly turbulent flow ensures effective bulk heat transfer from the combustion process to the air, which is in turn fed to the turbine.

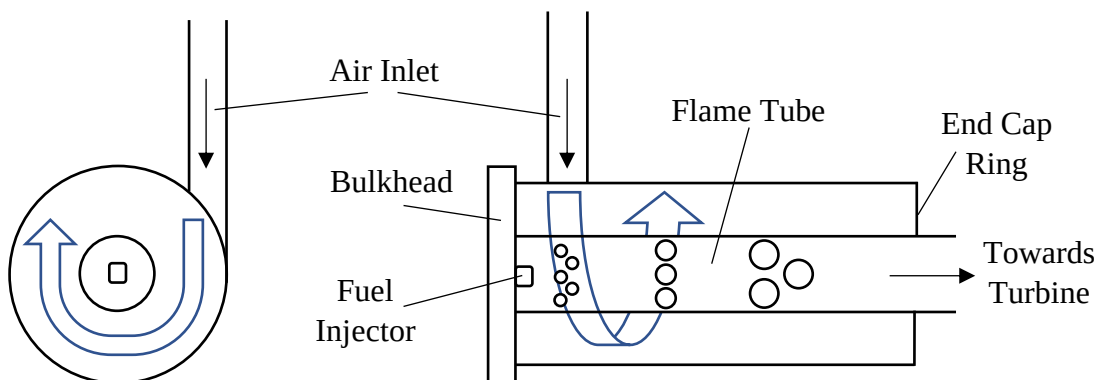


Figure 2.8 1: Flame Tube Combustion Chamber Rear and Side View

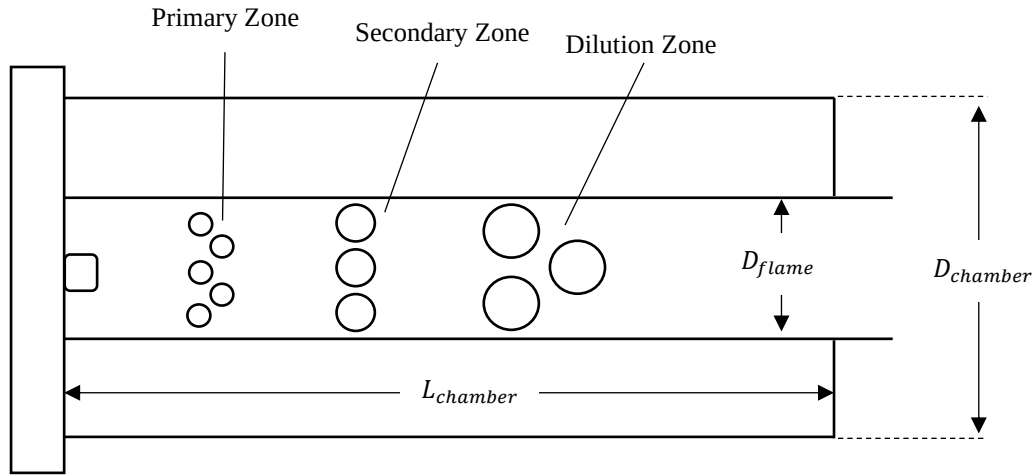


Figure 2.8 2: Flame Tube Dimensions and Zones

The primary zone is designed such that enough air enters the flame tube to mix with the fuel at a rich fuel ratio while maintaining flammability (Tech Ingredients, 2019, p. 15:36). Its purpose is to combust most of the fuel without raising the temperature high enough to melt the tube itself.

The secondary zone mixes air at a lean fuel ratio at a rate that is still within the flammability limit of the flame front to complete the combustion (Tech Ingredients, 2019, p. 16:01).

Finally, the rest of the air enters the flame tube at the dilution zone where it is homogeneously mixed (Tech Ingredients, 2019, p. 16:14).

General consensus has been formed among DIY turbocharger turbojet enthusiasts through experimentation as to suggested ideal dimensions and layout of the flame tube and combustion chamber. Heuristic relationships have been based on the diameter of the compressor's inducer (Furze, 2013; JATO, 2016; Jesse, 2003; Susante & Akker, 2004; Tech Ingredients, 2019)

The turbocharger's compressor inducer $D_{inducer}$ is used to size the flame tube and combustion chamber (Furze, 2013). Furze (2013) recommends the following:

The inner diameter of the flame tube D_{flame} is suggested to be 1.5 to 3 times $D_{inducer}$. Smaller turbochargers are recommended to have a larger relative flame tube diameter. Turbochargers with $D_{inducer} < 45$ mm are considered small and are suggested to have a $D_{flame} = 3 \cdot D_{inducer}$. Moderate size turbochargers with a $D_{inducer}$ between 45 and 80 mm have shown to be reliable with $D_{flame} = 2 \cdot D_{inducer}$. Very large turbochargers have shown to run effectively with $D_{flame} = 1.5 \cdot D_{inducer}$ while saving weight by not unnecessarily oversizing the combustion chamber.

The length of the flame tube chamber $L_{chamber}$ for all size turbochargers is recommended to be set as 6 times that of $D_{inducer}$ (Tech Ingredients, 2019, p. 18:13). The outer tube inner diameter is recommended as $D_{chamber} = D_{flame} + 0.5 \cdot D_{inducer}$.

The diameter of the holes used in the secondary band is generally recommended to be twice that of those used in the primary band (Jesse, 2003). The diameter of the holes used in the dilution band is usually set at twice that of the holes used in the secondary band (Furze, 2013; Jesse, 2003).

The total area of all the holes are made to be approximately equal to the area of the inducer (Tech Ingredients, 2019, p. 16:35). The primary holes constitute 30% of the total hole area, the secondary holes 20% of the total hole area, and the dilution holes 50% of the total hole area.

Generally, the number of holes used in the dilution band is equal to that of the secondary band, and the number of holes used in the primary band is three to five times that of the other bands (Jesse, 2003).

Furze (2013) recommends that the number of holes used for the primary, secondary dilution zones to be 26-5-5 respectively.

The centre of the primary band is generally located at a length 20-30% that of $L_{chamber}$ (Furze, 2013; JATO, 2016; Jesse, 2003). The centre of the dilution band is located approximately halfway along the flame tube at about 40-50% the length of $L_{chamber}$. The centre of the secondary band is located half-way between centres of the primary and dilution bands.

The heuristics outlined above have shown to be reliable dimensions for combustion chambers for turbocharger-based turbojets for inducer diameters from about 20 mm to 100 mm (Furze, 2013; JATO, 2016).

3. The Intensive Linear Receiver Model

The purpose of the intensive linear receiver model is to provide a robust mechanism for simulating a linear receiver across a wide range of temperatures, atmospheric conditions, HTFs, materials of construction and dimensions of the receiver tube itself. This would in turn provide a means for evaluating the performance of linear receivers at high temperatures and with various HTFs beyond those operating conditions used in conventional linear receiver applications.

All of the source code developed is available in the Appendix F.

A similar methodology to that used by Forristall (2003) in *Modelling of a Parabolic Trough Solar Receiver Implemented in Engineering Equation Solver* has been used to develop the intensive linear receiver model (National Renewable Energy Laboratory, 2003).

The intensive linear receiver model has been programmed for use in MATLAB. MATLAB provides a convenient mechanism for performing the calculations necessary and displaying the data graphically. In adopting a modular approach to the code, it allows for further additions and modifications to the model to be made part-wise easily in the future.

The intensive linear receiver model is subject to a series of limitations and assumptions. The following assumptions have been made in the derivation of the intensive linear receiver model:

- Heat flux density into the receiver is uniform across and around the receiver surface. At any length along the receiver, the temperature about the circumference at that length is uniform (Forristall, 2003, p. 23).
 - This greatly simplifies calculations of heat convection between the receiver/absorber's inner surface and the HTF, as well as uniform air convection and radiative losses in all directions from the receiver surface.
 - In reality a greater concentration of solar rays is focused nearer to the sides of the absorber relative to the collector's aperture due to the imaging of the parabolic mirror. Incoming and outgoing heat flux densities are in reality functions of both receiver length and angle of rotation about the receiver's axis.
 - In justification of this assumption; if the heat conductivity of the receiver surface and/or substrate is high, then the temperature within a local region is approximately equal.
- Radiative losses not in the direction of the sky are assumed to behave the same as the sky (Forristall, 2003, pp. 26, 27).
 - The ground visible from the receiver surface as well as the collector's mirror itself are assumed to have absorptivities and emissivities equal to that of the sky.
 - Radiation to the environment can therefore be calculated as a single term.

- Absorber inner temperature is equal to the absorber surface temperature (Forristall, 2003, pp. 63, 64).
 - If the receiver tube material of construction is highly heat conductive (for example copper or even stainless steel), then this is a very close approximation of reality
- The absorber surface, receiver cover and sky behave as greybodies, where their absorptivities, emissivities and transmissivities are not functions of temperature and are equal for all frequencies and frequency distributions of radiation (Forristall, 2003, pp. 11, 14, 16, 27).
 - Such information for temperature-based functions is difficult to obtain for cermet coatings especially at low temperatures and for long wavelengths.
 - It will be assumed that the impact this has - especially for low temperatures - is fundamentally negligible, as radiative losses are proportional to the fourth power of temperature.
- Incidence losses will be assumed to behave similarly to that of conventional linear receivers (Forristall, 2003, pp. 18, 26, 27).
 - Anti-glazing coatings on conventional receiver covers and microscopic pitting on absorber surfaces help to minimize incidence losses compared to $\cos(\theta_{incidence})$ as the sun transits the sky during the day.
 - Designs for concentrators and receivers of new dimensions and materials will be assumed to perform at least as well as conventional collectors with respect to incidence losses.
- The absorber glass cover, if present, is opaque to longwave radiation (Forristall, 2003, p. 18).
 - This is an approximation of the behaviour of glass.
- Pressure drop throughout the receiver is neglected (Forristall, 2003, pp. 25, 73).
 - This is not the focus of the model for its current purpose; however, pressure drop functions may be added to the model in the future.
- Heat losses by conduction to the collector frame, as well as convection and radiative losses in additional connecting piping and manifolds to and from the receiver are not accounted for (Forristall, 2003, pp. 20, 28).
 - With appropriate insulation, this should be negligible compared to losses from the receiver itself.
- The intensive linear receiver model works only for a single receiver, and does not take into account operating many receivers in either series or parallel configurations (Forristall, 2003, p. 26).
 - The intensive linear receiver model may be applied to each receiver recursively if necessary. If this is the case, additional considerations must be made for collector frame shadowing and other pressure and heat losses in connecting piping.
- Only Direct Normal Irradiance will be considered (Forristall, 2003, pp. 16, 17).
 - Additional heat flux provided by diffuse radiation is negligible compared to the heat flux density present on the absorber surface.

3.1 Derivation of the Intensive Linear Receiver Model

The following subsection details the derivation of the functions implemented in the Intensive Linear Receiver Model.

The segmentation of each unit and point of interest is inspired by the work performed by Forestall (2003), with the exception that all function and function variables are purposefully not simplified or approximated owing to the greater memory availability and processing speeds in modern programming languages such as MATLAB.

By leaving all functions in their full form, both human legibility of the code as well as iterative calculation accuracy is maintained.

MATLAB R2019 was configured such that each function call for each and every calculation produced a double precision 64-bit depth output. An 8 core AMD Ryzen 1700 at 3.8 GHz was used for the processing of the dissertation (by means of running the MATLAB script DOALL.m with CoolProp functioning). Theoretically the CPU was able to perform processing the dissertation at about 243 FP64 GFLOPS.

Using the above setup, it took about 2 hours to process the entire script (at a peak of about 240 billion calculations per second when certain function calculations could be parallelized), and at certain points required in the order of 11GB of memory while performing different function calls. Hence the name “Intensive Linear Receiver Model”.

Appendix A contains the full derivation and adaptation of certain long-form functions and formulae from literature which are used in the model.

The functions described below are those used in the intensive linear receiver model itself. The names of function variables, the symbols used as well as call order are the same as those as defined in Appendix F.

Special care has been taken in Appendix F to comment each function header and sub-section of the code. This in-situ documentation will aid in future development and adaptation of the code.

Performing an energy balance for a section of the receiver tubing results in:

$$Q_{solar, supplied} = Q_{HTF, absorbed} + Q_{losses} \quad (3.1 \ 1)$$

And:

$$Q_{solar, supplied} = A_{concentrator} I \eta_{optical} = a L I \eta_{optical} \quad (3.1 \ 2)$$

Where a is the width of the aperture of the concentrator, L is the length of the section of tubing in question, and $\eta_{optical}$ is that defined in Equation 2.5 1.

Where:

$$Q_{solar} = a L I = Q_{solar, supplied} + Q_{solar, optical \ losses} \quad (3.1 \ 3)$$

Such that:

$$Q_{solar,optical\ losses} = aLI(1 - \eta_{optical}) \quad (3.1\ 4)$$

Further:

$$Q_{HTF,absorbed} = h_{T_{HTF,bulk}} D_{absorber,inner} \pi L (T_{absorber} - T_{HTF,bulk})$$

Where $h_{T_{HTF,bulk}}$ is that defined by Equations A.10-14.

Performing an energy balance between the receiver and surroundings:

$$Q_{losses} = Q_{convection,receiver-surroundings} + Q_{radiation,receiver-surroundings} \quad (3.1\ 5)$$

$$\begin{aligned} Q_{radiation,receiver-surroundings} \\ = \sigma \pi L D_{receiver,\epsilon} \pi \epsilon_{receiver,outer} (T_{receiver,\epsilon}^4 - T_{sky,blackbody}^4) \end{aligned} \quad (3.1\ 6)$$

Where $T_{sky,blackbody}$ may be found from Equations A.30 and 31.

Furthermore:

$$Q_{convection,receiver-surroundings} = \pi L D_{receiver,outer} h_{wind} (T_{receiver,outer} - T_{air}) \quad (3.1\ 7)$$

$$h_{wind} = \frac{Nu_{wind} k_{air}}{D_{receiver,outer}} \quad (3.1\ 8)$$

Where h_{wind} may be found according to Equations A.3 and 4.

A set of constraints exist if it is the case that there is a cover over the receiver. Performing an energy balance between the receiver's absorber and cover:

$$Q_{vacuum,convection} + Q_{radiation,absorber-cover} = Q_{cover,conduction} = Q_{losses} \quad (3.1\ 9)$$

$$Q_{cover,conduction} = \frac{2\pi L k_{cover} (T_{cover,inner} - T_{cover,outer})}{\ln\left(\frac{D_{cover,outer}}{D_{cover,inner}}\right)} \quad (3.1\ 10)$$

Where $Q_{vacuum,convection}$ is as that defined by Equations A.21-27, $Q_{radiation,absorber-cover}$ is as that defined by Equation A.28, and k_{cover} is the heat conductivity of the cover material.

The heat flow model of the receiver may be simulated by dividing the length of the receiver into pieces of δL . Each piece of δL may be considered individually by selecting temperatures of the absorber surface such that Equation 3.1 1 is satisfied.

This is done numerically in practise by performing Newton's Bisection Method upon the function of Equation 3.1 1.

An absorber surface temperature too low would result in the sum of HTF heat absorption and overall losses less than that of the irradiance available. An absorber surface temperature too high would result in heat absorption and losses greater than

the supplied power. As such, the absorber temperature may be iterated upon until a temperature value of sufficient approximation of the true value is found.

The tolerance of this approximation is set in the program, and by default each δL section's temperature is found to within 4 decimal places.

To simply numerical analysis of air as the HTF, various tables of air properties have been adapted into polynomial equations by means of least-squares minimization. Table 3.1 1 contains the result of these adaptations (Çengel, 2009).

Within each iteration of Equation 3.1 1, the presence of a cover will dictate whether or not a nested iteration step is required, since in this case there are effectively two unknown temperatures at every δL .

If there is a cover, then the inner surface temperature of the cover must be iterated for within each iteration of absorber surface temperature to satisfy the energy balance.

In this case, steady state is assumed such that the conduction and radiation losses from the absorber surface to the cover must equal the heat absorbed by the inner side of the cover, the conduction through the cover, and the convective and radiative losses to the ambient from the cover outer surface – i.e., Equation 3.1.9 must be satisfied.

The iterative logic of the intensive linear receiver model is depicted in Figure 3.1 1. In the program, the iteration tolerance for each iteration cycle is adjustable and set to 4 decimal places by default.

Table 3.1 1: Polynomial Functions for Various Air Properties; Adapted from Çengel (2009)

Air property	Equation [T in °C from -150°C to 2000°C]	R ²	Units	Pressure Dependence
Density ρ	$1/(T * 0.002833191 + 0.774043)$	0.99999	kg/m ³	$\rho * \frac{p}{p_{(1 atm)}}$
Heat Capacity c_p	$(4.73252 \cdot 10^{-11})T^4 + (-2.07769 \cdot 10^{-7})T^3 + (0.000238738)T^2 + (0.106809446)T + 999.9268821$	0.99263	J/kg.K	Independent
Heat conductivity k	$(4.63986 \cdot 10^{-12})T^3 + (-2.5217 \cdot 10^{-8})T^2 + (7.56219 \cdot 10^{-5})T + 0.023635444$	0.99999	W/m.K	Independent
Thermal diffusivity α_T	$(-1.50263 \cdot 10^{-14})T^3 + (9.71026 \cdot 10^{-11})T^2 + (1.39146 \cdot 10^{-7})T + 1.84666 \cdot 10^{-5}$	0.99986	m ² /s	$\alpha_T * \frac{p_{(1 atm)}}{p}$
Dynamic viscosity μ	$(-3.28024 \cdot 10^{-18})T^4 + (1.56331 \cdot 10^{-14})T^3 + (-3.02471 \cdot 10^{-11})T^2 + (4.87271 \cdot 10^{-8})T + 1.72057 \cdot 10^{-5}$	0.99988	kg/m.s	Independent
Kinematic viscosity ν	$(4.18424 \cdot 10^{-18})T^4 + (-2.64477 \cdot 10^{-14})T^3 + (9.60235 \cdot 10^{-11})T^2 + (8.70674 \cdot 10^{-8})T + 1.34434 \cdot 10^{-5}$	0.99998	m ² /s	$\nu * \frac{p_{(1 atm)}}{p}$

The following functions will be used to define and assess the performance of the linear receiver.

$$\eta_{receiver} = \frac{Q_{HTF,absorbed}}{Q_{solar}} \quad (3.1 \ 11)$$

$$\mathcal{R}_{losses} = \frac{Q_{losses} + Q_{solar,optical \ losses}}{Q_{solar}} = 1 - \eta_{receiver} \quad (3.1 \ 12)$$

$$\mathcal{R}_{convection,losses} = \frac{Q_{convection,receiver-surroundings}}{Q_{losses} + Q_{solar,optical \ losses}} \quad (3.1 \ 13)$$

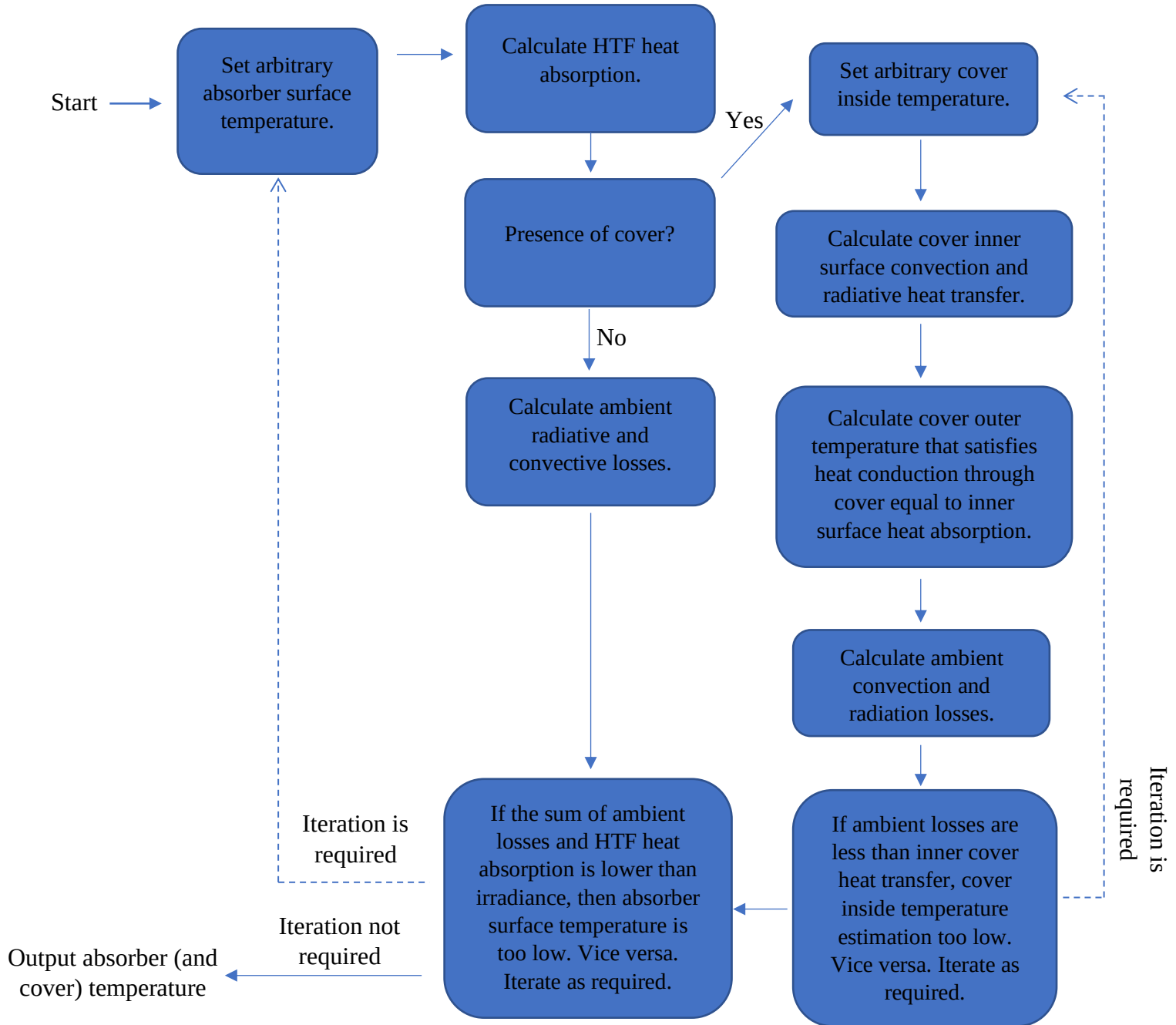


Figure 3.1 1: Intensive Linear Receiver Model Section Temperature Calculation Iteration Logic

$$\mathcal{R}_{radiation,losses} = \frac{Q_{radiation,receiver-surroundings}}{Q_{losses} + Q_{solar,optical losses}} \quad (3.1 14)$$

$$\mathcal{R}_{optical,losses} = \frac{Q_{solar,optical losses}}{Q_{losses} + Q_{solar,optical losses}} \quad (3.1 15)$$

Where:

- $\eta_{receiver}$ is the receiver's efficiency – what fraction of the total original solar energy available finds it way absorbed into the HTF
- $\mathcal{R}_{losses}$ is the losses ratio – what fraction of the total original solar energy is eventually lost to the environment
- $\mathcal{R}_{convection,losses}$ is the ratio of convection losses – what fraction of losses which are lost to the environment by means of air convection is compared to the sum of all losses; what fraction of total losses for which the convection component is responsible
- $\mathcal{R}_{radiation,losses}$ is the ratio of radiation losses – what fraction of total losses for which the radiation component is responsible
- $\mathcal{R}_{optical,losses}$ is the ratio of optical losses – what fraction of total losses for which the optical component is responsible

Knowing how $\eta_{receiver}$, $\mathcal{R}_{convection,losses}$ and $\mathcal{R}_{radiation,losses}$ change along the length of the receiver will allow for the holistic evaluation of the receiver's operating temperature range, HTF, absorber surface and shielding effectiveness.

$\eta_{receiver}$, $\mathcal{R}_{convection,losses}$, $\mathcal{R}_{radiation,losses}$ and $\mathcal{R}_{optical,losses}$ may be calculated either cumulatively over the entire length of the receiver or instantaneously at each δL , such that how each type of energy loss changes along the length – and temperature – of the receiver may be evaluated.

3.2 Modelling an Arbitrary Linear Receiver using the Numerically Intensive Receiver Model

For the purpose of demonstration, the intensive linear receiver model may be used in a series of parametric analyses.

A suitable default number of slices along the length of the receiver to be used for numerical integration may be determined such that sufficient accuracy is achieved at an acceptable processing time in comparison to an excessive number of slices.

An arbitrary 5 m long Pyrex glass vacuum covered W:Al₂O₃ cermet absorber surface receiver transporting pressurized preheated air as the HTF was modelled with the following physical characteristics at an incidence angle of 0 (i.e. noon zenith for an E-W orientated PTC):

$$\begin{aligned}
 L &= 5 \text{ [m]}, Q_{solar} = 5000 \text{ [W]}, T_{HTF_{bulk,inlet}} = 353 \text{ [K]} \\
 p_{HTF} &= 2 \text{ [atm]}, T_{air} = 298 \text{ [K]}, D_{absorber} = 0.045 \text{ [m]}, \\
 D_{cover,inner} &= 0.057 \text{ [m]}, D_{cover,outer} = 0.077 \text{ [m]}, p_{air} = 1 \text{ [atm]}, \\
 p_{vacuum} &= 0.013 \text{ [Pa]}, \epsilon_{absorber} = 0.1, k_{cover} = 1.005 \left[\frac{W}{m \cdot K} \right], \\
 \epsilon_{cover} &= 0.86, v_{wind} = 2 \left[\frac{m}{s} \right], T_{dewpoint,air} = 287 \text{ [K]}, \\
 RH &= 50, \dot{m}_{HTF} = 0.01 \left[\frac{kg}{s} \right], \alpha_{absorber} = 0.9, \rho_{mirror} = 0.93, \tau = 0.96, \\
 \gamma_{SEC} &= 0.94, F_{soiling} = 0.97, F_{end\ losses}(i) = 1, K(i) = 1
 \end{aligned}$$

Values of the variables were set to match those measured in literature, and may be referenced from Section 2.5 and Appendix A.

Figure 3.2 1 depicts the calculation error of this covered receiver against the number of δL pieces used along the length of the receiver. The difference error in calculation considering the whole receiver as one slice at steady state or considering 5000 points along its length is 28.67%.

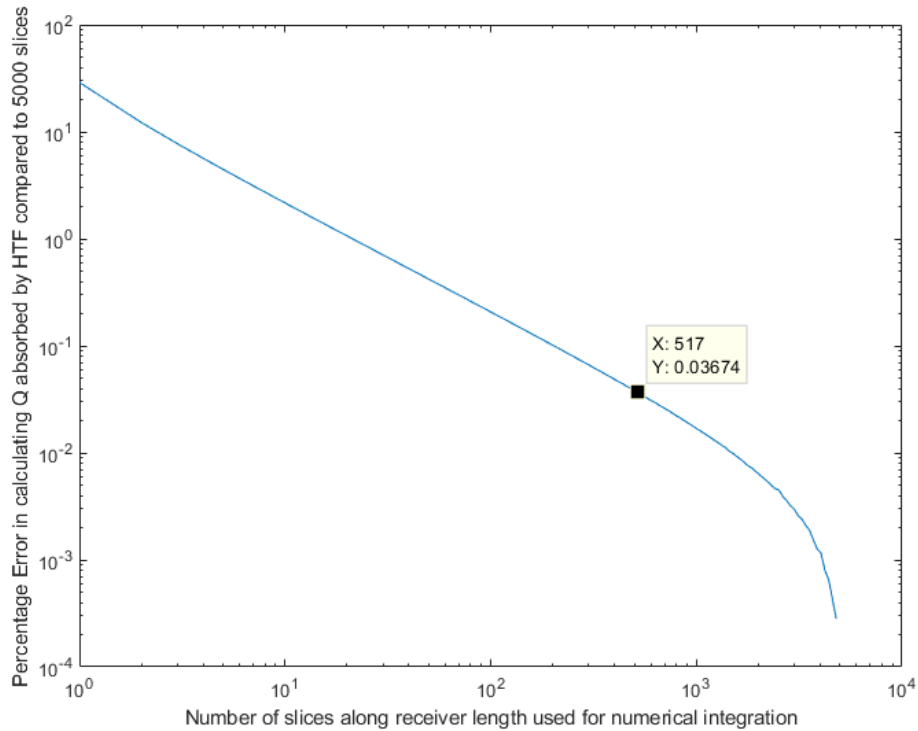


Figure 3.2 1: Comparison of Calculation Error for number of slices used in Numerical Integration of the Receiver

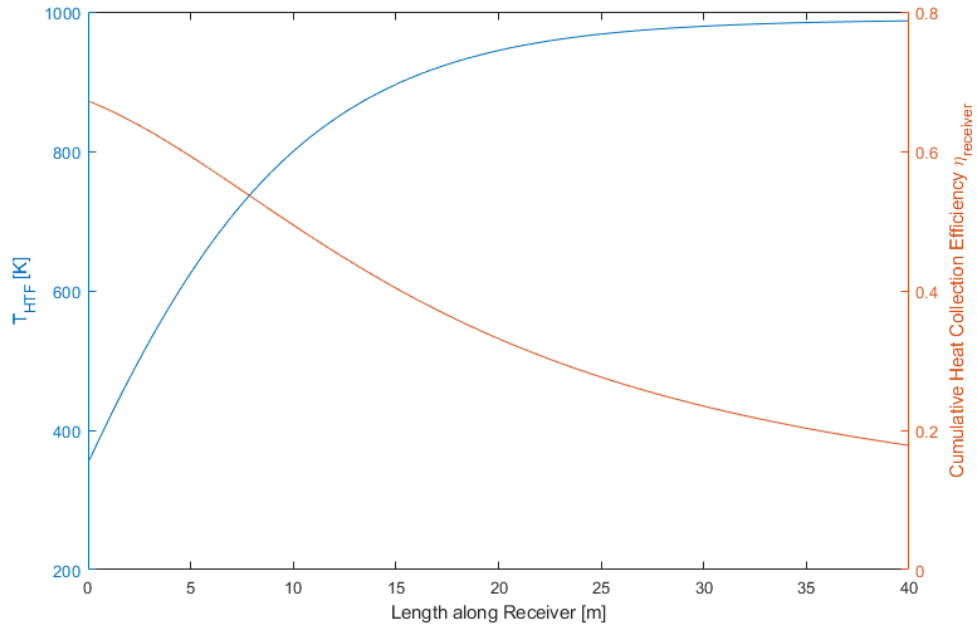


Figure 3.2 3: Arbitrary Receiver HTF Temperature and Cumulative Efficiency along its length

At 5000 slices, the calculation took 13.72 seconds to complete. At 517 slices, the program took 1.46 seconds to complete. The error of the 517-slice calculation is only 0.037% compared to the 5000-slice calculation while only costing 10% the processing time.

As such 500 slices was chosen as the default number of receiver sections to be used in future calculations. (Such that the full MATLAB script for this dissertation required approximately 2 hours to process on the machine used, instead of weeks.)

Figure 3.2 2 depicts the HTF Temperature and cumulative receiver efficiency along the length of the same arbitrary receiver variables as in Figure 3.2 1, with the

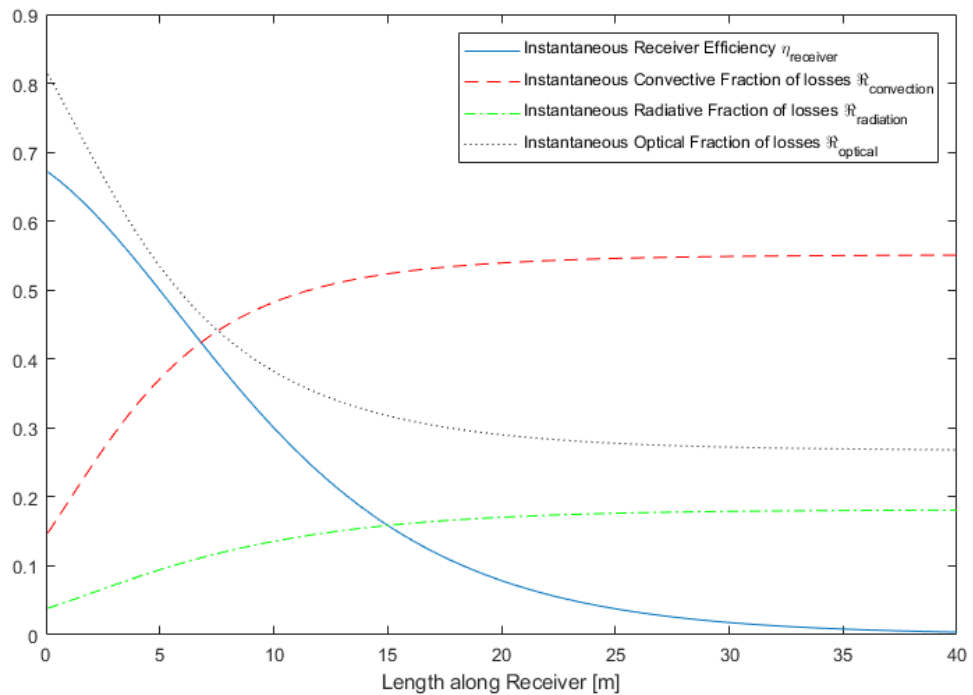


Figure 3.2 2: Arbitrary Receiver Instantaneous Receiver Efficiency and Ratio of Losses along its length

exception that the receiver now has a length of 40 m with insolation set at 1000 W/m of receiver length.

Figure 3.2 3 depicts the instantaneous receiver efficiency as well as the fraction of radiative and convective losses for the same arbitrary receiver as Figure 3.2 2.

From Figure 3.2 2 it can be observed for this arbitrary receiver that at about 30 m in length, the air HTF reaches its maximum temperature of about 980 K. Lengthening the receiver from this point does not heat the HTF appreciably.

From Figure 3.2 2 and 3, it can be seen that at low temperatures (0 to 4 m for 80-305 °C), the receiver efficiency is high (61-67%) and optical losses account for the vast majority of total losses. As the temperature increases along the length of the receiver, the receiver efficiency decreases. At about 7.5 m, convective losses become driving for the rest of the length of the receiver.

Figure 3.2 4 depicts the simulation of the same arbitrary receiver as that in Figures 3.2 2 and 3 but without a Pyrex cover. In this case, convective losses are driving for the entire length of the receiver.

The intensive linear receiver model also provides a mechanism for comparison of different absorber surface coatings. Figure 3.2 5 compares the performance of the arbitrary receiver with various absorber surface coatings along as well as a coverless case.

Figure 3.2 5 was generated using the same variable inputs as Figures 3.2 2 and 3, with the exception that $\text{Al}_y\text{Ti}_{1-y}(\text{O}_x\text{N}_{1-x})$ was set with $\alpha_{\text{absorber}} = 0.91$ and $\epsilon_{\text{absorber}} = 0.14$ (as per Section 2.5) and the Blackbody was set with $\alpha_{\text{absorber}} = \epsilon_{\text{absorber}} = 1$.

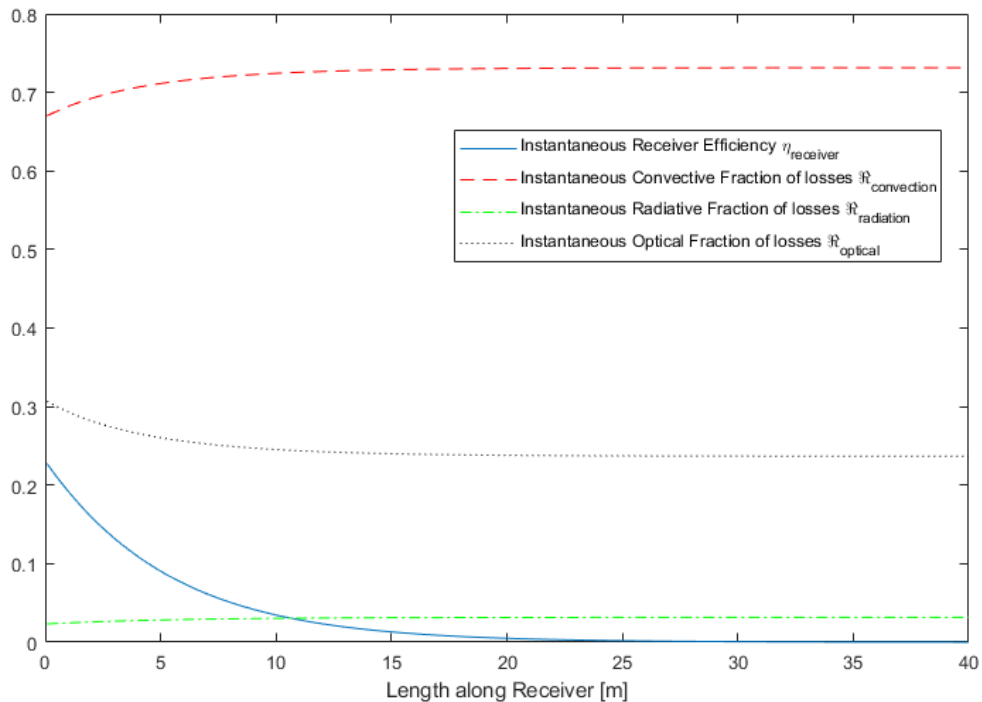


Figure 3.2 4: Arbitrary Receiver Performance without a cover

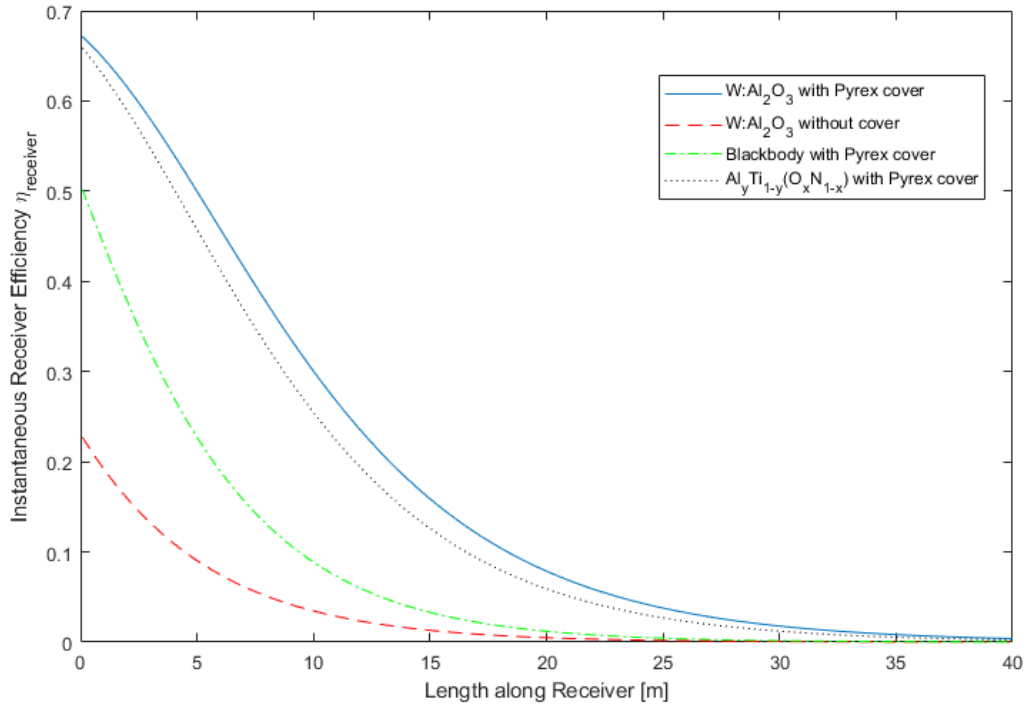


Figure 3.2 6: Arbitrary Receiver Efficiencies for Various Absorber Coatings

For these arbitrary receivers in Figure 3.2 5 it can be seen that $W:Al_2O_3$ is a more effective solar-selective absorber surface coating than $Al_yTi_{1-y}(O_xN_{1-x})$ under these conditions. This is a result of the lower emittance of $W:Al_2O_3$ ($\epsilon_{absorber} = 0.1$ for $W:Al_2O_3$ vs 0.14 for $Al_yTi_{1-y}(O_xN_{1-x})$), which offsets the higher value of absorptance of $Al_yTi_{1-y}(O_xN_{1-x})$ ($\alpha_{absorber} = 0.91$ for $Al_yTi_{1-y}(O_xN_{1-x})$ vs 0.9 for $W:Al_2O_3$).

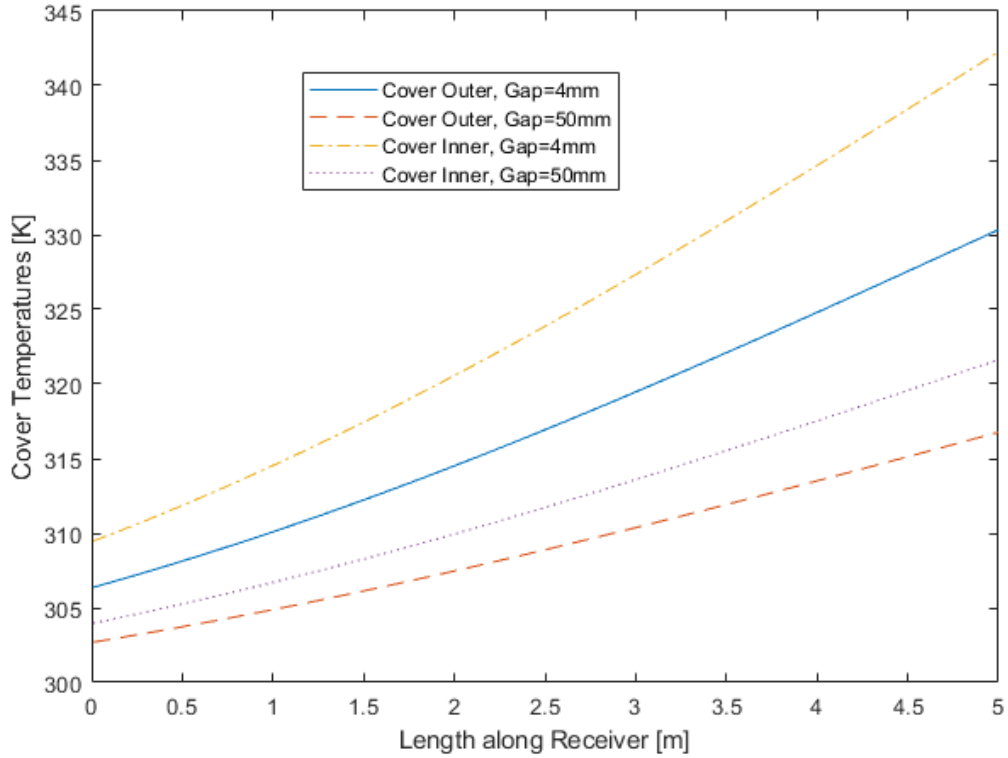


Figure 3.2 5: Cover Temperature Dependency on Vacuum Gap Width on the Arbitrary Receiver

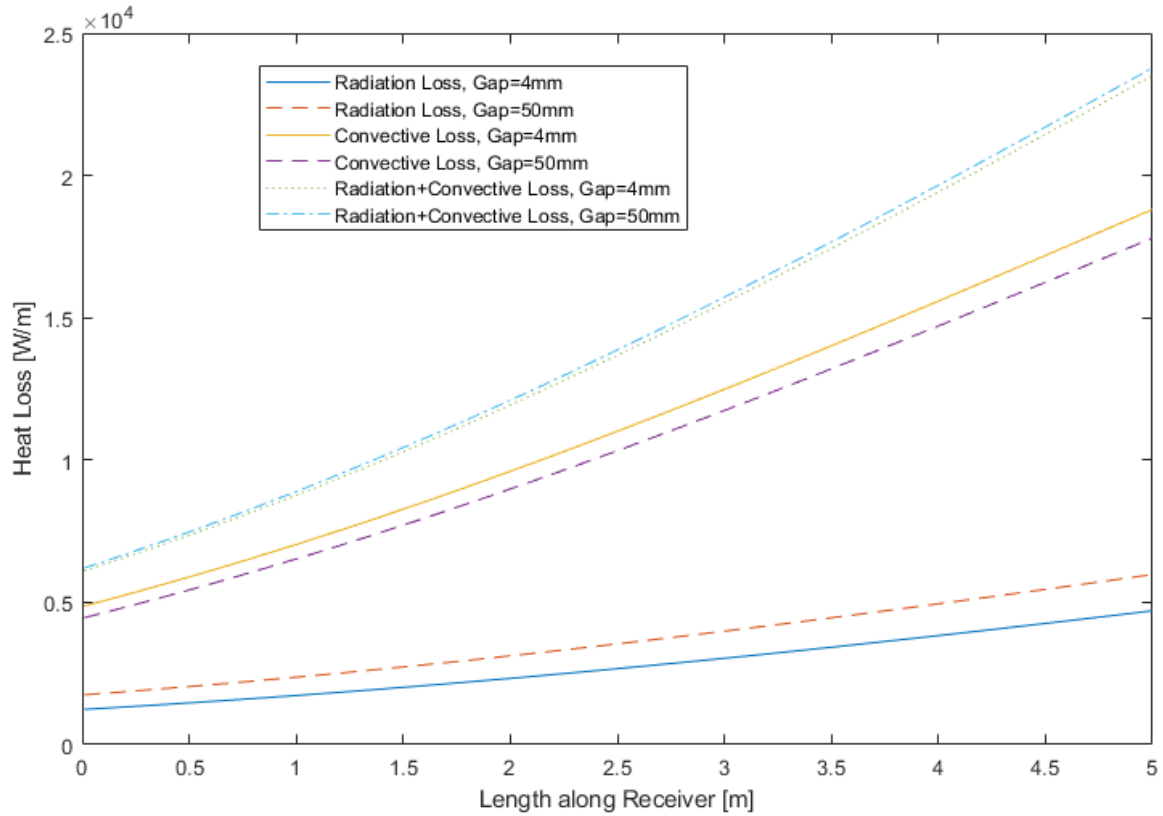


Figure 3.2 7: Heat Losses Dependencies on Vacuum Gap Width on the Arbitrary Receiver

The much higher emittance of a blackbody surface coating in Figure 3.2 5 severely hampers overall performance of the receiver despite the perfect absorptance when compared to a selective surface coating.

This is a non-intuitive result which indicates that heat transfer between the absorber outer surface and inner cover surface is radiative driven when the cover annulus pressure is close to vacuum. In other words; for a receiver of these dimensions, total heat losses to the atmosphere is driven primarily by the absorber's emittance.

The effect of the width of the gap between the absorber surface and cover inner may also be modelled. Figures 3.2 6, 7 and 8 use the same arbitrary receiver variable settings as Figures 3.2 2 and 3 but with an absorber-cover gap of between 4 mm and 50 mm (Figures 3.2 2 and 3 had a gap of 6 mm)

This was performed with the same variable settings as Figures 3.2 2 and 3 where D_{absorber} was kept at 0.045 m, but $D_{\text{cover,inner}}$ was set at D_{absorber} + twice the gap length (function input is diameter not radius) and $D_{\text{cover,outer}}$ was set as $D_{\text{cover,inner}}$ + 0.02 m to maintain a cover thickness (i.e. radial length) of 1 cm.

Figure 3.2 6 shows that for a lower vacuum gap size (and therefore lower cover outer and inner diameter), cover outer and inner temperatures increases. For a greater gap width (and therefore larger outer diameter cover), a greater surface area of the outer cover is available to transmit losses through convection and radiation to the surroundings. A question arises as to whether there is benefit to be gained in using a larger gap width to lower the cover's outer temperature and if this reduces overall losses given the increase of surface area of the cover.

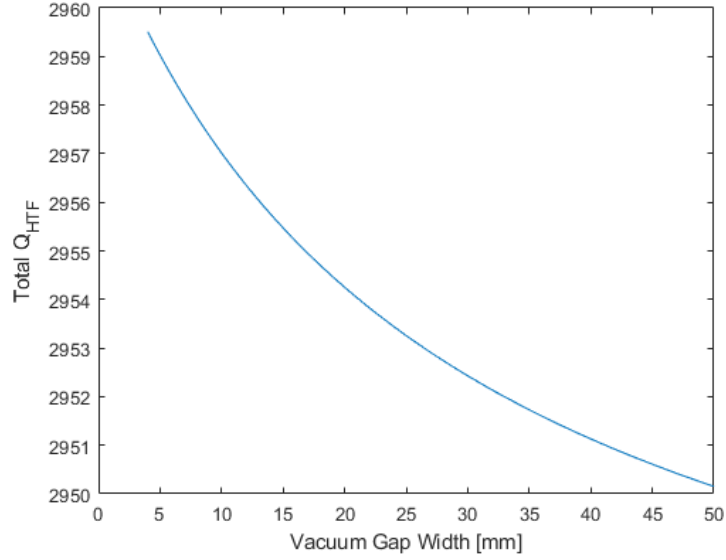


Figure 3.2 8: Total Heat Transfer to HTF Dependency on Vacuum Gap Width on the Arbitrary Receiver

From Figure 3.2 7 it may be seen that increasing vacuum gap width in fact decreases overall heat losses to the surroundings due to convection, while increasing overall heat losses to the surroundings due to radiation. This implies that there are benefits to be gained by selecting a cover material with lower emissivity over a higher emissivity cover material, even if the lower emissivity material necessitates a larger vacuum gap width (and greater cover outer surface area) for manufacturing reasons.

Figure 3.2 8 shows that for a greater vacuum gap width, total losses to the surroundings are increased, however, the increase in total losses is arguably negligible for the large range of sensible vacuum gap widths of 4 mm to 50 mm – a difference of only 9 W or 0.18% of Q_{solar} . This suggests gap width selection in real-world manufacture may target a gap as small as possible, but is heavily subject to manufacturing costs when working with such tight tolerances.

It is possible to model the receiver's performance over an entire day by simulating the performance of the arbitrary receiver at various times of day. For any location on the surface of the Earth at a certain time of day, the relative position of the sun may be calculated. Since the orientation of the receiver is known, the angle of incidence between the sun and the receiver may be calculated.

It will be assumed that newer cover anti-glazing coatings perform at least as well as conventional commercial cover coatings. Trough orientation is also an important factor, which depends on the location of the SEC's use.

For this arbitrary receiver, it was assumed that the function for incidence factor behaves similarly to (i.e., at least as well as) that of the Industrial Solar Technology Corporation product (IST-PTC) PTC Pyrex cover and coating (Goswami, 2015, p. 176):

$$K(\theta_{deg}) = 1 + 0.0003718 \left(\frac{\theta_{deg}}{\cos(\theta_{deg})} \right) - 0.00003985 \left(\frac{\theta_{deg}^2}{\cos(\theta_{deg})} \right)$$

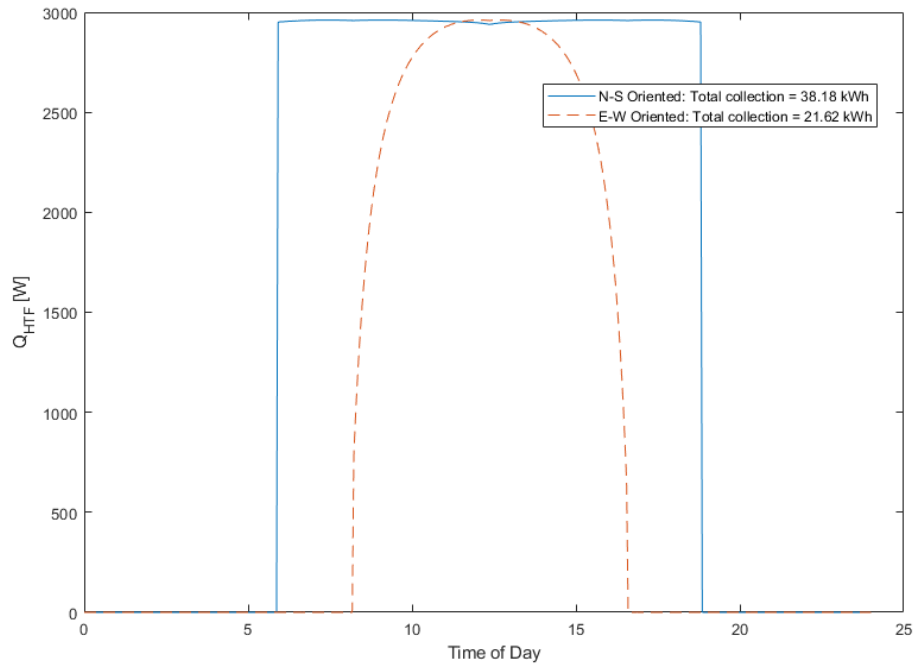


Figure 3.2 10: Daily Rates of Heat Collection for an Arbitrary Receiver of N-S and E-W Orientations, Simulated for The University of Witwatersrand on a midsummer day, February 15th 2019

Solar altitude and azimuth have been modelled using the functions published by the United States National Oceanic & Atmospheric Administration Earth System Research Laboratory (NOAA ESRL, 2019).

Figures 3.2 9 and 10 depict the performance of an arbitrary receiver with the same variables as those in Figures 3.2 2 and 3 located at The University of Witwatersrand for a midsummer's day and on Winter Solstice. The simulation did not take into

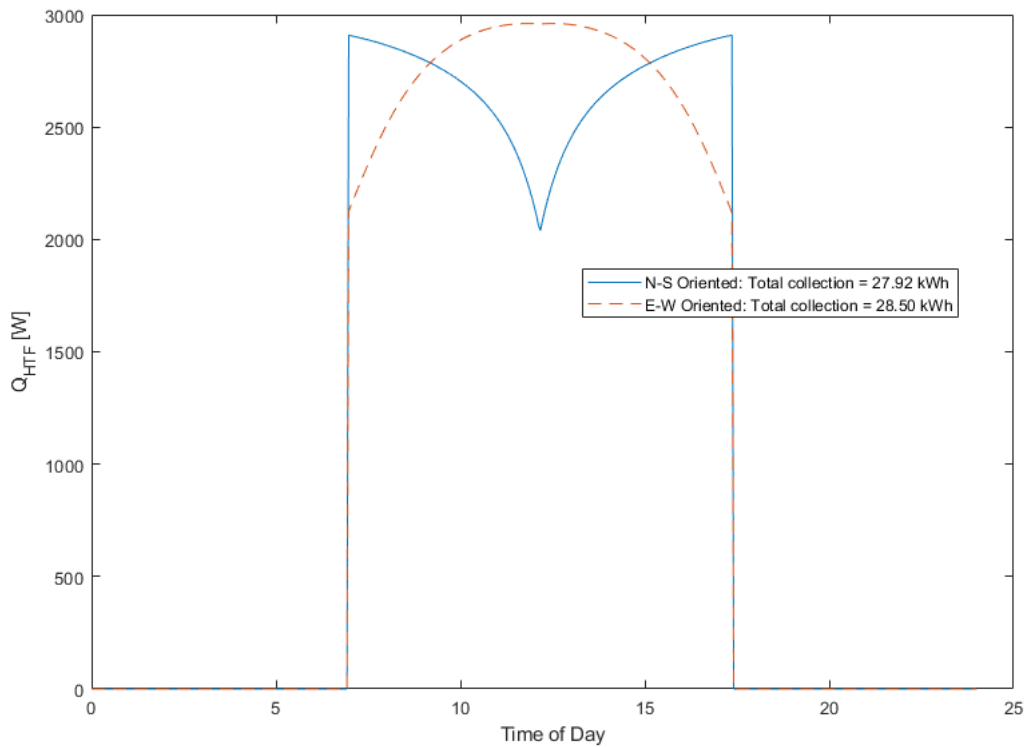


Figure 3.2 9: Daily Rates of Heat Collection for an Arbitrary Receiver, Simulated for The University of Witwatersrand on Winter Solstice, June 21st 2019

account average weather conditions for those days, but rather assumes perfectly clear skies with an irradiance of 1000 W/m^2 .

For the location at The University of Witwatersrand ($26^\circ 11' 30'' \text{S}$ and $28^\circ 01' 38.8'' \text{E}$), a N-S orientation of the trough is greatly preferred over an E-W orientation. Summer energy collection for N-S orientation is approximately twice that of E-W orientation (Figure 3.2 9), while Winter Solstice energy collection for E-W orientation is only 2% greater than a N-S orientation (Figure 3.2 10).

The intensive linear receiver model may be used to simulate the overall thermal performance of a linear SEC. This may be done by connecting the output of the SEC to a heat engine.

In this case, similarly to Equation 2.4 7:

$$\eta_{thermal} = \frac{W_{engine}}{Q_{solar}} = \frac{Q_{engine} \cdot \eta_{engine}}{Q_{solar}} = \eta_{receiver,cumulative} \cdot \eta_{engine} \quad (3.2 \ 2)$$

For the purpose of a numerical analysis and optimization, a Carnot Engine may be used as the mechanism to extract work from the SEC. As an example, a hypothetical task may be to find the optimal length for the arbitrary receiver to maximise overall thermal efficiency for the condition where $T_{HTF,bulk,inlet} = T_{air}$.

Figure 3.2 11 depicts the overall thermal efficiencies of an arbitrary receiver of various lengths with the same variables as Figures 3.2 2 and 3 in which the HTF outlet is attached to a Carnot Engine. The receiver length of greatest thermal efficiency lies at a length of 8.18 m, with a thermal efficiency of 32.8% and a HTF outlet temperature of 718 K.

In a similar fashion, a parametric study may be performed on the arbitrary receiver by changing the width of the reflector's aperture as well in addition to the overall

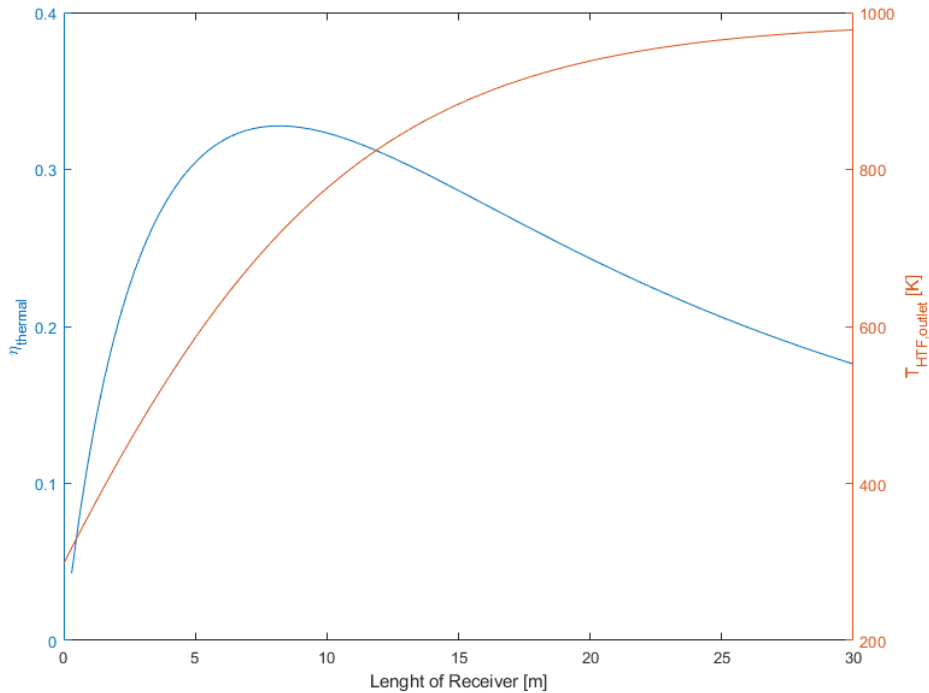


Figure 3.2 11: Thermal Efficiency Optimization of an Arbitrary Receiver's Length when attached to a Carnot Engine

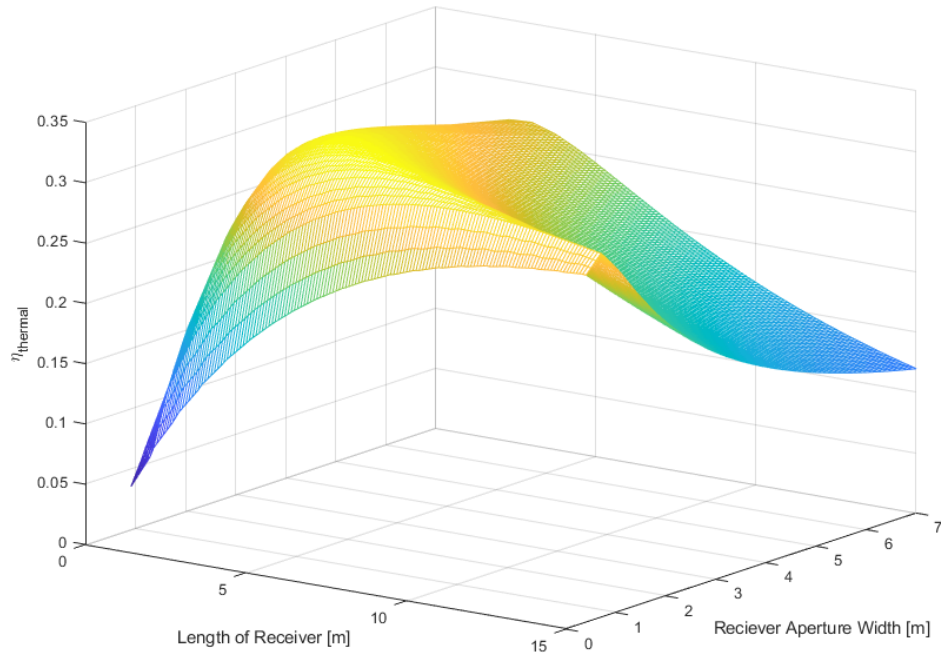


Figure 3.2 13: Arbitrary Receiver's Thermal Efficiency as a function of Receiver Length and Aperture Width

length of the receiver. These changes may then be quantified in terms of the SEC's thermal efficiency by connecting the outlet of the HTF to a Carnot Engine.

Figures 3.2 12 and 13 depict the thermal efficiency of the arbitrary receiver with the same variables as Figures 3.2 2 and 3 where it has been attached to a Carnot Engine for various receiver lengths and aperture widths.

Figure 3.2 11 exists as a slice of Figures 3.2 12 and 13 along the line where receiver aperture width is 1 m.

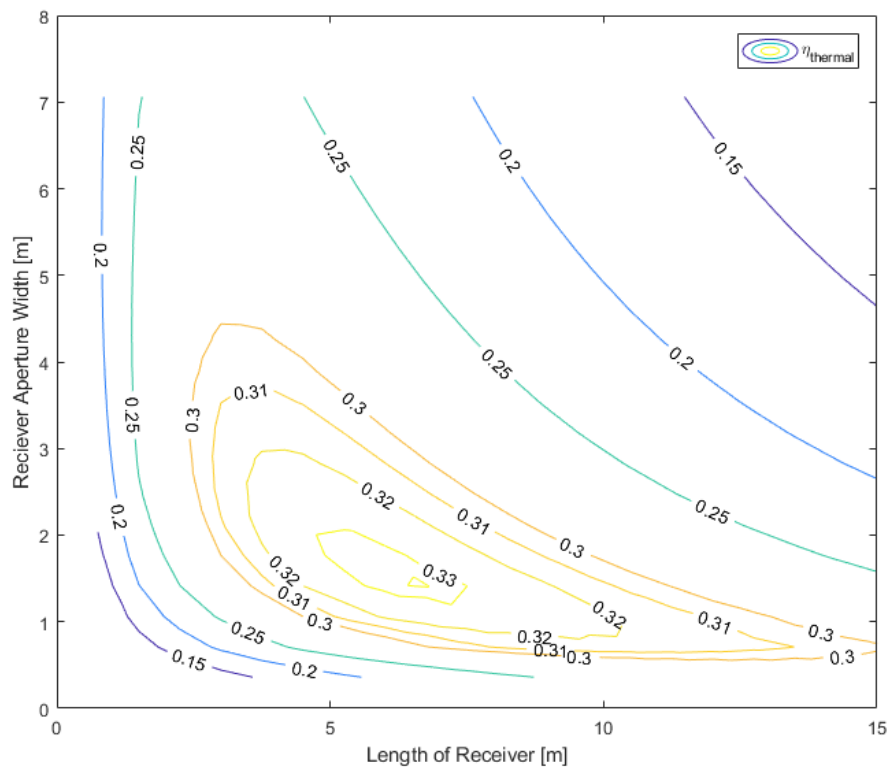


Figure 3.2 12: Contours of an Arbitrary Receiver's Thermal Efficiency as a function of Receiver Length and Aperture Width

To increase the available irradiation to be collected by a linear SEC, either the receiver aperture width may be increased (i.e. increase the CR) or the receiver may be lengthened (i.e. increase the Aspect Ratio (AR)).

Increasing the AR increases the available surface area on the receiver for convective and radiative losses to occur, since the receiver grows in length.

Increasing the CR does not change the surface area of the receiver; since there is a greater aperture for the reflector which is the same length as the receiver.

From Figures 3.2 12 and 13, it is suggested that for any given receiver aperture width there exists a receiver length that results in the highest thermal efficiency, and where either too long or too short a receiver length results in a decrease in thermal efficiency.

For example, drawing a horizontal line on Figure 3.2 13 at an aperture width of 2 m and reading left to right; thermal efficiency at 1 m of receiver length is about 15%, maxima is reached at about 6 m length for about 33% thermal efficiency, and by 15 m length thermal efficiency is approximately 25%.

This phenomenon may be interpreted as too great a receiver length leads to diminishing returns on adding heat to the HTF as the absorber nears its maximum temperature.

Similarly, from Figures 3.2 12 and 13 it is suggested that for a given receiver length (i.e., a vertical line on Figure 3.2 13), too small a receiver aperture provides a large surface area for losses to occur relative to insolation collected, while too large an aperture leads to diminishing returns for the rate of heat absorption by the HTF.

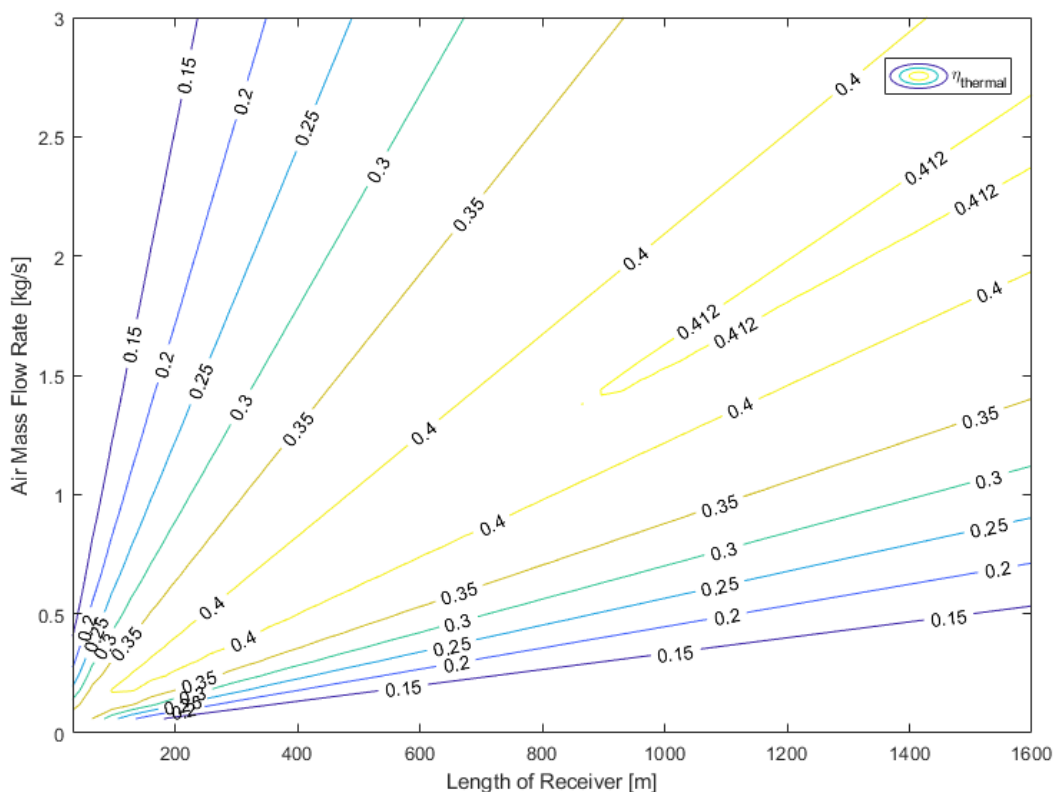


Figure 3.2 14: Thermal Efficiency of an Arbitrary Linear Receiver for various HTF Flow Rates and Receiver Lengths

This implies that there exists a certain aperture width and receiver length that optimizes the thermal efficiency of an arbitrary receiver attached to a Carnot Heat Engine. Specifically, it is implied that there exists an optimal design for aperture width and receiver length for a given HTF, mass flowrate and absorber diameter, for a collector attached to a Carnot Engine.

This is somewhat non-intuitive that it is not always the case to simply target as high a CR as possible when designing a SEC. Simply sizing a collector as wide as possible may in fact decrease overall operational thermal efficiency.

A similar parametric analysis may be performed on an arbitrary receiver upon its mass flow rate and length when keeping its CR (i.e. its aperture width) constant.

Figure 3.2 14 depicts the optimization of mass flow rate to receiver length for the arbitrary receiver with the same variables as Figures 3.2 2 and 3 but an aperture width of 1.414 m as per Figure 3.2 13's maxima. This is meant to emulate the case of real CSP plant design where a unit piece of the SEC is commercially available such as an Abengoa 8 m SpaceTube PTC (Abengoa Solar, 2013). In this case, the CR and receiver outer diameter is set, but the mass flow rate and length (i.e., the number of series PTC pieces) needs to be optimised.

From Figure 3.2 14 for this arbitrary receiver, it can be seen that a linear line (i.e., contour spine) represents an optimal ratio of HTF mass flow rate to receiver length. For this case, an ideal ratio of receiver length to mass flow rate would be about 640 m.s/kg. Thermal efficiency increases as receiver length and mass flow rate increases about this ideal ratio, approaching an asymptote of about 41.5% as the length extends toward infinity (but outside of Re numbers permitted by Equation A.9). For

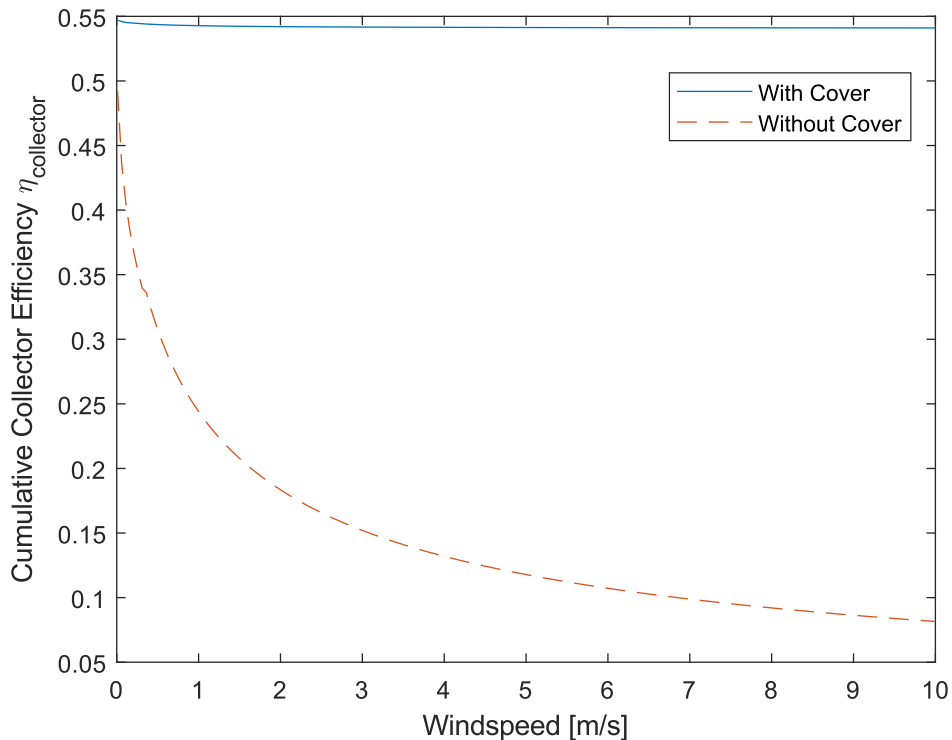


Figure 3.2 15: Windspeed vs. Cumulative Collector Efficiency of an Arbitrary Receiver

a length of 1.6 km and mass flow rate of 2.5 kg/s, thermal efficiency is found to be 41.26%. However, pressure drop at this length is likely to be non-negligible.

The intensive linear receiver model may also be used to analyse the effect of windspeed on a covered and coverless receiver. Figure 3.2 15 depicts the cumulative collector efficiencies for an arbitrary receiver with and without a cover. This arbitrary receiver uses the same variables as Figures 3.2 2 and 3 but with an aperture width of 1.414 m and a length of 6.75 m as found as maxima in Figure 3.2 13

From Figure 3.2 15 when there is no forced convection (i.e., a windspeed of 0 m/s), the collector efficiency of a non-covered receiver is not substantially worse than that of a covered receiver – 55% collector efficiency when covered compared to 50% uncovered. However, even slight forced convection significantly increases convective losses; at just 2 m/s, the collector efficiency of the non-covered receiver is only 34% that of the covered receiver. At greater windspeeds, this gap increases diminishingly.

Up to this point in the analyses, air has been used as the HTF for the arbitrary receiver. The intensive linear receiver model is capable of simulating any fluid if the flow is sufficiently turbulent enough along with a reasonable Prandtl number.

Marlotherm SH is a premier high temperature oil HTF manufactured by Sasol that is designed for closed loop operation and the heating of reactors and heat engines at an operational temperature of up to 350 °C (Sasol, 2015). Table 3.2 1 has been generated from Sasol’s Marlotherm product data sheet and converted into second degree polynomials to be used in the MATLAB code.

Figure 3.2 16 depicts the parametric optimization of the arbitrary receiver attached to a Carnot Engine with the same variables as in Figure 3.2 2 and 3 with Marlotherm SH as the HTF at 3.5 kg/s (any lower flow rate result in a Reynolds number below 2300 and out of bounds of the function described by Equation A.9). The data points that produced a HTF outlet temperature above 623 K have been removed as they are above Marlotherm SH’s stability limit of 623 K (350 °C).

Table 3.2 1: Thermodynamic Properties of Marlotherm SH, adapted from Sasol (2015)

Marlotherm SH property	Equation [T in °C from -5°C to 350°C]	R ²	Units
Heat Capacity c_p	$1.14256 \cdot 10^{-7}T^2 + 0.003685184 \cdot T + 1.476857143$	0.9999471	kJ/kg.K
Density ρ	$-4.60711 \cdot 10^{-6}T^2 + -0.713166003 \cdot T + 1058.311278$	0.9999868	kg/m ³
Thermal Conductivity k	$-2.76426 \cdot 10^{-9}T^2 + -0.000130847 \cdot T + 0.133201504$	0.9996099	W/m.K
Kinematic Viscosity ν	$\frac{1}{2.03452 \cdot 10^{-5}T^2 + 0.001417023 \cdot T - 0.02232397}$	0.9993702	mm ² /s

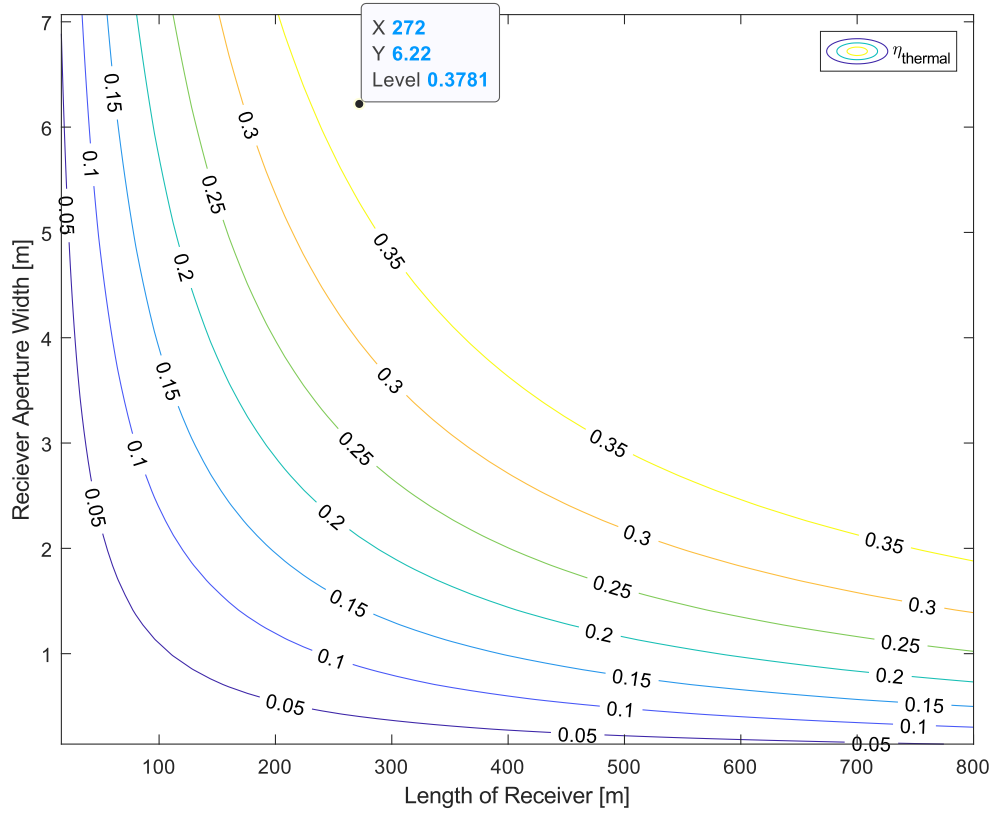


Figure 3.2 16: Thermal Efficiency of an Arbitrary Receiver with liquid phase Marlotherm HTF attached to a Carnot Engine

From Figure 3.2 16, the point of greatest thermal efficiency for this arbitrary receiver occurs at an aperture width of 6.22 m and a receiver length of 272.0 m for a thermal efficiency of 37.81% and HTF outlet temperature of 622 K.

Both the gas and liquid phase HTFs used in the arbitrary receiver were able to optimise receiver dimensions such that optimal thermal efficiency was in the region of 35%. This agrees with that as expected performance of real-world PTC CSP plants in Table 2.3.1 1 when adding a 50% penalty for thermal performance difference between Carnot and real heat engines.

Two prominent differences between the gas and liquid phase HTFs used in the arbitrary receiver is the large difference in mass flow rate (higher for gas phase) and the lower temperature of the liquid phase HTF. However, both approaches produced a comparable resulting thermal efficiency when attached to Carnot engines.

The higher flow rate necessary for the gas phase may be explained by the necessity of turbulent flow for the HTF to ensure heat transfer between the bulk HTF and that in contact with absorber wall. By virtue of being a liquid, the viscosity of the liquid HTF is orders of magnitude greater than that of the gas HTF.

From Equation 3.2 3, it can be seen that the mass flow rate for gas phase must be much larger in order to maintain a Reynolds number above 2300 (i.e, since μ is much smaller, $\dot{m}$ must be larger to maintain the value for Re).

$$\begin{aligned}
Re &\equiv \frac{\rho \dot{v} D}{\mu} \\
\dot{m} &= \dot{V} \rho; \dot{V} = \frac{\dot{m}}{\rho} \\
\dot{V} &= \dot{v} A; \dot{v} = \frac{4\dot{V}}{\pi D^2} \\
\dot{v} &= \frac{4\dot{m}}{\pi \rho D^2} \\
\therefore Re &= \frac{4\dot{m}}{\mu \pi D} \tag{3.2 3}
\end{aligned}$$

The liquid phase HTF yields a lower outlet temperature than that of the gas phase HTF of 622 K and 770 K respectively. This is a limitation of the stability temperature of the HTF oil used. Despite a lower outlet temperature, the thermal efficiency when using the liquid phase HTF is slightly greater than that of the gas phase HTF. This is due to the greater collector efficiency of the liquid phase HTF as the resulting value of the coefficient of bulk HTF heat transfer $h_{T_{HTF}bulk}$ in Equation A.9 is much greater for the oil than air.

3.3 Discussion

In this section the framework, algorithms and MATLAB code were developed for an intensive linear receiver model. The purpose of this model is to allow for the simulation, parametric analysis and optimization of linear focus receivers of various construction materials, covers, HTFs, physical dimensions and atmospheric conditions outside of that for which conventional models can accommodate.

A parabolic trough design was used in the application of the model in this section. The intensive linear receiver model may be easily adapted to other collector designs by making the necessary changes.

For instance; a linear Fresnel reflector collector may be modelled by changing the model's outer surface of the receiver cover to a flat plate only facing the ground, or the SunTrap may be modelled by setting the absorber reception surface area to the lower arc of a tube while accounting for direct heat conduction through the absorber's insulation casing.

This is a powerful tool which may be used in the design and optimisation of linear focus collectors and associated power plants. The model is free from the oversimplified and/or proprietary collector models used by manufacturers to protect intellectual property and revenue streams in the modelling of their receivers and collectors on behalf of clients.

Conventional linear focus collectors are usually relegated to relatively low temperature operation. The large surface area of the receiver suggests a propensity for significant heat losses to occur even at moderate temperatures. However, it has been shown that the use of contemporary receiver materials in conjunction with a shielded vacuum cover permit the collector to obtain moderately high outlet temperatures in the order of 700 °C at an acceptable cumulative efficiency.

The purpose of this approach would be to leverage the cost effectiveness of linear focus collectors over the more complicated and expensive point focus collectors. Operating at these higher temperatures would permit the use of higher efficiency heat engines and heat engine cycles. This would help to maximise power production, minimise the cost of production, and maximise the profitability of energy production.

The intensive linear receiver model permits the analysis of linear focus receivers at absorber temperatures much higher than those produced by conventional PTC and LFR SECs. This allows for the next generation of high temperature linear receivers powering different engine designs to be simulated. Higher temperature engines such as supercritical CO₂ or even different heat engine cycles such as hybrid air Brayton Cycles may be computationally analysed.

The design philosophy of the MATLAB code is for each component to be modular. In this fashion, the addition of new materials, substrates and HTFs etc. may be done so easily with minimal adaptation to the code. While the runtime of the code is performed interpretively within a MATLAB instance, the code may be compiled as a C, C++ or an appropriate binary with caching to substantially increase the speed of processing the intensive linear receiver model. (Processing of Sections 3 and 4 within MATLAB took approximately 2 hours on a modern desktop.). This will provide the opportunity to run the model on larger scale optimizations or even commercial applications.

The intensive linear receiver model includes a function to simulate the sun's azimuth and altitude for any time of day at any location on the planet. This allows for the angle of incidence to be calculated between a linear SEC's aperture and the sun at different times during the day. In this fashion, the overall cumulative daily performance of a linear receiver may be accurately modelled.

An arbitrary receiver was used as the basis for simulations performed using the intensive linear receiver model. The receiver dimensions, absorber substrate and cover materials were chosen to be an approximate analogue of modern linear SEC technologies. The majority of simulations were performed with air as the HTF, with an additional parametric analysis performed with Sasol's oil based Marlotherm SH as the HTF.

The model was used to make observations about the arbitrary receiver in order to identify trends that may be present in real world applications.

It was observed that for the arbitrary receiver, the use of a cover and vacuum decreased overall losses by 1/3rd at low receiver temperatures. At higher receiver

temperatures losses were reduced to $1/15^{\text{th}}$ to that of not using a cover during a period of a moderate wind.

A receiver cover is therefore essential for any application of moderate or higher temperatures.

Wind velocity was shown to have marginal effects on performance for an optimally sized and covered receiver, and in this case optical losses account for the majority of losses along the receiver length. For an uncovered receiver, however, even relatively low wind velocities of a few m/s contribute substantially to losses and significantly decrease thermal efficiency.

The intensive linear receiver model was used to demonstrate the optimisation of the physical dimensions of the arbitrary receiver by targeting thermal efficiency with the outlet of the receiver attached to a Carnot Engine at a fixed mass flow rate.

It was found that it is not always the case to have as high a CR as possible; rather that a maximised work thermal efficiency exists for the combination of an optimal AR and CR for a given HTF, mass flow rate and receiver materials and dimensions.

It was demonstrated that for a contemporary receiver design such as that emulated by the arbitrary receiver, optical losses account for the majority of losses at a thermal efficiency optimized AR and CR. Increasing the length (i.e. increasing the AR) beyond this point lowers the thermal efficiency of the receiver where higher output temperatures are associated with a greater proportion of thermal losses. As AR increases, eventually convective losses become larger than optical losses.

The intensive receiver model was used to optimise the HTF flow rate and receiver field length for a given CR and receiver dimensions. This is meant to emulate the real-world design process of a CSP SEC where the dimensions of a unit piece of the PTC are known, and the primary decisions for the design of the SEC is the mass flow rate of the HTF and the number of trough sections to be placed in series with each other.

When the dimensions of a collector section were set, it was found that there is an optimal efficiency ratio of AR (i.e., the number of series trough sections) to mass flow rate.

It was found that radiative heat transfer between the absorber surface and the cover is the primary heat loss transport mechanism to the cover when the cover annulus is close to a vacuum.

In other words, the primary driving function of losses to the atmosphere is in fact the emissivity of the absorber surface, since this is the primary transport mechanism of heat from the absorber to the cover. A major reduction in overall heat losses may therefore be obtained by a slightly lower emissivity of the absorber surface.

Liquid or gas phase fluids may be used as the HTF in a linear receiver. Liquid oils and molten salts have been favoured historically for their stability, heat conductivity and relatively high boiling points compared to water (Goswami, 2015, p. 486; Parzen, 2017, pp. 28, 29). Oils tend to have maximum temperatures in the region of about 350-400°C, limiting potential thermodynamic efficiency of the heat engine.

Molten salts may be able to operate at higher temperatures, but have other operational requirements such as minimum acceptable temperatures to avoid freezing inside of operational equipment as in the case of unexpected shutdown (Parzen, 2017, p. 32).

Gas phase HTFs tend to have significantly lower thermal conductivities and heat capacities than oils and molten salts, but tend to remain stable even at very high temperatures over 1000°C (Parzen, 2017, pp. 35, 58).

By using the intensive linear receiver model to find an optimised AR, CR and flowrate, a very high collector and thermal efficiency may be obtained for the SEC – where the primary limiting factor is the stability of the materials of construction of the absorber or stability limits of the HTF itself.

The intensive linear receiver model has been designed to have a wide utility and easy adaptability to new materials and functional models.

It is necessary to validate the model by comparing simulations with real-world performance data of receivers made from contemporary materials and operating at high temperatures.

The author made requests for access to real-world performance data of South African commercial linear receiver CSP plants: Bokpoort, Ilanga I, Kathu Solar Park, KaXu Solar One and Xina Solar One. Unfortunately, representatives from each plant denied these information requests due to each of the firms' concerns over IP security in the distribution of operational information.

For future work it is suggested that an official contract may be drawn up between the University and the operational South African CSP plants such that their operational information may be kept secure while only exposing the necessary information to help validate the intensive linear receiver model.

A representative from Kathu Solar Park – Mr. David McDougall – expressed interest in sharing data in the future once their operation has moved out of its testing phase at the end of August 2020. Kathu Solar Park's relatively high HTF outlet temperature of about 400 °C would be useful in validating the intensive receiver model.

A representative from Bokpoort – Mr. JC Nel – also expressed interest in using the intensive receiver model to help understand the extent to which cracks in covers would affect STE collector field performance in return for plant performance data at an unspecified future date.

Obtaining high temperature linear receiver performance data from these and other sources would be useful in validating the intensive receiver model for future research.

3.4 Conclusion

In this section the framework, algorithms and MATLAB code were developed for an intensive linear receiver model. The purpose of this model is to allow for the simulation, parametric analysis and optimization of linear focus receivers of various construction materials, covers, HTFs, physical dimensions and atmospheric conditions outside of that for which conventional models can accommodate.

An arbitrary parabolic trough receiver was used as the basis for simulations performed using the intensive linear receiver model. The arbitrary receiver's dimensions, absorber substrate and cover materials were chosen to be an approximate analogue of modern linear SEC technologies.

A few observations were made by means of a series of parametric analyses using the arbitrary receiver. Parametric variables included varying physical dimensions, construction materials, HTF flow rates and types of HTF used, and connecting the outlet to a theoretical Carnot engine.

While these observations were based on the arbitrary receiver itself, the observations should apply to real-world linear receivers. Some of the more important observations include:

- A linear receiver made from modern materials of construction should be able to achieve high outlet temperatures in the order of 700 °C at an acceptable cumulative receiver efficiency.
- Wind velocity was shown to have marginal effects on performance for an optimally sized and covered receiver, such that optical losses account for the majority of losses along the receiver length.
- A vacuum cover over the absorber is absolutely essential to limit heat losses for anything above moderate temperatures (300 °C).
- Collector dimensions may be optimised for a given HTF, such that both gas and liquid phase HTFs may achieve acceptable thermal efficiency of approximately 35% when connected to a Carnot engine.
- In a vacuum covered linear receiver, the emissivity of the absorber's surface is the primary factor which determines total heat loss from the collector.
- For a given HTF and flowrate, there exists an optimal combination of receiver length (i.e., the number of series trough unit sections) and reflector aperture (i.e., the construction width of the parabolic reflector's arc) that maximises overall thermal efficiency of the SEC and heat engine. To high or low either aperture width or receiver length will lead to a decrease in thermal efficiency.
 - Simply sizing a collector as wide as possible to obtain a CR as high as possible may in fact decrease overall operational thermal efficiency.
- When the dimensions of each collector trough unit and the HTF used is known (as is the case when designing a SEC power plant using commercial PTC sections), there exists an approximately linear ratio of HTF flow rate

to the number of trough unit sections that optimises overall thermal efficiency.

If the intensive linear receiver model is to be developed in further research, it will first be necessary for it to be validated by comparing simulations with real-world performance data of receivers made from contemporary materials and operating at high temperatures.

4. Simulations of a Turbocharger Linear Receiver Air Hybrid Brayton Cycle CSP Heat Engine

In Section 3 the numerically intensive linear receiver model was developed and used in a series of parametric analyses by connecting its outlet to a Carnot engine. In this section, the intensive linear receiver model will instead be connected to a “more realistic” turbocharger-based Brayton Cycle Engine.

This thought experiment is essentially to provide the basis of feasibility in intentionally operating a linear receiver at temperatures far greater than conventional implementations. This may be done by means of modern materials of construction, new absorber substrates, and the use of gas and supercritical fluid HTFs.

A high outlet temperature obtained by the receiver implies a thermodynamic efficiency gain for the heat engine high enough to offset the low thermal conductivities and heat capacities present in air and similar fluid HTFs.

In leveraging the cost benefit of PTCs and other linear SEC designs over two axis SEC designs, together with operation at high temperatures (and high efficiencies), the ultimate intent is to minimise overall lifetime cost per kW and kWh.

In a similar fashion to the work done by various authors previously, the core of this investigation will be done on the operation of a modified vehicle turbocharger (Jansen, et al., 2015; Le Roux, et al., 2011; Le Roux, et al., 2012). The impetus for the use of radial flow turbochargers over axial flow purpose-built GTs are the low cost, high reliability, easy maintenance and acceptable thermal efficiencies of the compressor and turbine stages of commercial turbochargers.

In deviation to the HFC SEC investigated by Le Roux et al. (2011 and 2012) and Jansen et al. (2015), a PTC will be modelled by means of the intensive linear receiver model.

STE collected by the receiver may either be configured to be the only source of heat used by the engine, or otherwise used to preheat air for a combustion stage. This hybrid operation would have the STE act as an augmentation to the thermal energy available in the fuel to run the attached heat engine (Schwarzbözl, et al., 2006, pp. 1231-1240).

Hybrid operation allows for the engine to be capable of full power output at night and during periods of inclement weather. The compressed and preheated air is also available to combust practically any renewable and non-renewable fuel sources such as natural gas, biofuels, hydrogen, syngas, diesel, petrol, kerosene and UCG products etc. The only limitation is an appropriate combustion unit.

In an effort to limit harmful emissions of the engine by avoiding pre-ignition, flashback, and non-homogenous mixtures, the target outlet temperature of air will be set at 650 °C (Bryner, et al., 2016, p. 10).

It will be assumed that the use of a contemporary fuel injection method together with the inlet temperature of 650 °C will inhibit CO and NO_x formation.

4.1 Using a Turbocharger as the Compressor and Turbine stages of a pure STE Brayton Cycle Heat Engine

A turbocharger consists of two housings containing the compressor and turbine stages. The housings are connected together by means of a common shaft. In standard operation of the turbocharger, a stream of air is compressed and heated and fed to the vehicle's internal combustion engine. The high temperature exhaust from the engine is fed to the turbine, where the extracted work is used to power the compressor. Figure 4.1 1 shows a simplified connection of a turbocharger to a combustion engine.

Operating the turbocharger as a BCHE involves replacing the internal combustion engine with an isobaric heat addition stage. This may be done through heat exchange and/or by direct combustion. Assuming that the mass flow rate of fuel is negligible compared to the total air flowrate, the mass flow rate in the turbine is equal to that in the compressor, and ideally the pressure of the turbine outlet is equal to the pressure of the compressor inlet.

For a turbocharger operating as a Brayton Cycle Engine, a point may be chosen on the manufacturer's performance curves such that the operating flowrate, shaft angular velocity and compression ratio exists on an efficiency island which corresponds to the highest operating efficiency for the compressor (Le Roux, et al., 2011, p. 6030).

When operating a turbocharger at a flowrate, shaft angular velocity and pressure ratio corresponding to a maximum efficiency island for the compressor, it will be assumed that the turbine is also operating at maximum efficiency. This assumption is necessary because turbocharger manufacturers do not usually include efficiency curves in turbine performance maps as it is assumed there is an excess of energy

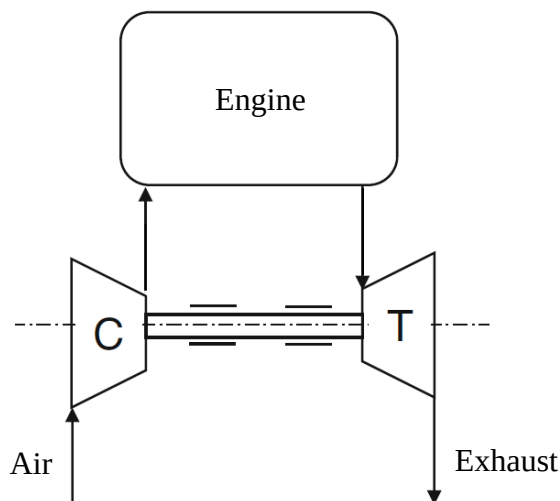


Figure 4.1 1: Typical Turbocharger Configuration in a Vehicle

available from combustion engine exhaust gasses to drive the compressor (Garrett, 2016).

To convert the units used in manufacturers curves for turbochargers, the following adaptations have been made to Equations 2.7 14-16:

$$\dot{m} = \dot{V}\rho = \left(\frac{\pi D^2}{4} \cdot c\right) \rho(p, T) \quad (4.1 \ 1)$$

$$\therefore c = \frac{4\dot{m}}{\pi D^2 \rho(p, T)} \quad (4.1 \ 2)$$

$$\therefore T_t = T_s + \frac{\left(\frac{4\dot{m}}{\pi D^2 \rho(p, T)}\right)^2}{2C_p(T)} \quad (4.1 \ 3)$$

$$\therefore p_t = p_s + \left(\frac{T_s + \frac{\left(\frac{4\dot{m}}{\pi D^2 \rho(p, T)}\right)^2}{2C_p(T)}}{T_s} \right)^{\frac{\gamma(T)}{\gamma(T)-1}} \quad (4.1 \ 4)$$

Therefore, to calculate the corrected mass flow rates for the compressor and turbine maps:

$$\therefore \dot{m}_{corrected} = \frac{\dot{m} \sqrt{\frac{T_s + \frac{\left(\frac{4\dot{m}}{\pi D^2 \rho(p, T)}\right)^2}{2C_p(T)}}{T_{STP}}}}{\left(\frac{p_s + \frac{T_s + \frac{\left(\frac{4\dot{m}}{\pi D^2 \rho(p, T)}\right)^2}{2C_p(T)}}{T_s}}{p_{STP}} \right)^{\frac{\gamma(T)}{\gamma(T)-1}}} \quad \text{turbine or compressor inlet} \quad (4.1 \ 5)$$

It is conventional for turbocharger maps to be issued in imperial units. Some conversion factors are therefore necessary while assuming Ideal Gas Law:

$$1 \left[\frac{kg}{s} \right] = \frac{60}{0.45359237} \left[\frac{lb}{min} \right] \quad (4.1 \ 6)$$

$$C_v = C_p - R_{specific}$$

$$C_{v,air} = C_{p,air} - 287.058 \left[\frac{J}{kg \cdot K} \right]$$

$$\therefore \gamma_{air} = \frac{C_{p,air}(T)}{C_{p,air}(T) - 287.058} \quad (4.1 \ 7)$$

With STP conditions at sea level:

$$T_{STP} = 288.15 [K], p_{STP} = 101325 [Pa]$$

Performing an energy balance on the shaft connecting the turbine and compressor for a Brayton Cycle Heat Engine:

Energy into the shaft = Energy out of the shaft

$$\therefore \Delta H_{\text{polytropic expansion}} = \Delta H_{\text{polytropic compression}} + W_{\text{electrical}}$$

$$\therefore W_{\text{electrical}} = \Delta H_{\text{polytropic expansion}} - \Delta H_{\text{polytropic compression}} \quad (4.18)$$

For the purpose of a parametric analysis, it will be assumed that the electrical generator operates at 100% efficiency at converting shaft work to electricity.

From this point, enough information is available to parametrically analyse the performance of a commercial vehicle turbocharger as the compressor and turbine stages of a CSP BCHE.

A Garrett GT0632 was chosen as one of the turbochargers to be analysed as a real one was readily available to the author for experimentation purposes, as detailed in Section 5. An aperture width of 6 m was chosen for the aperture of the arbitrary receiver to reflect commercially available PTCs (Abengoa Solar, 2013, p. 15).

Using the intensive linear receiver model, a Garrett GT0632 turbocharger was attached to an arbitrary receiver with the same variables as Figure 3.2 2 and 3 from Section 2. The aperture width was set at 6 m and windspeed was set as 5 m/s. The receiver inner diameter was set to 32 mm to match the turbine inducer diameter so as to minimise pressure drop otherwise experienced across couplings, flanges and

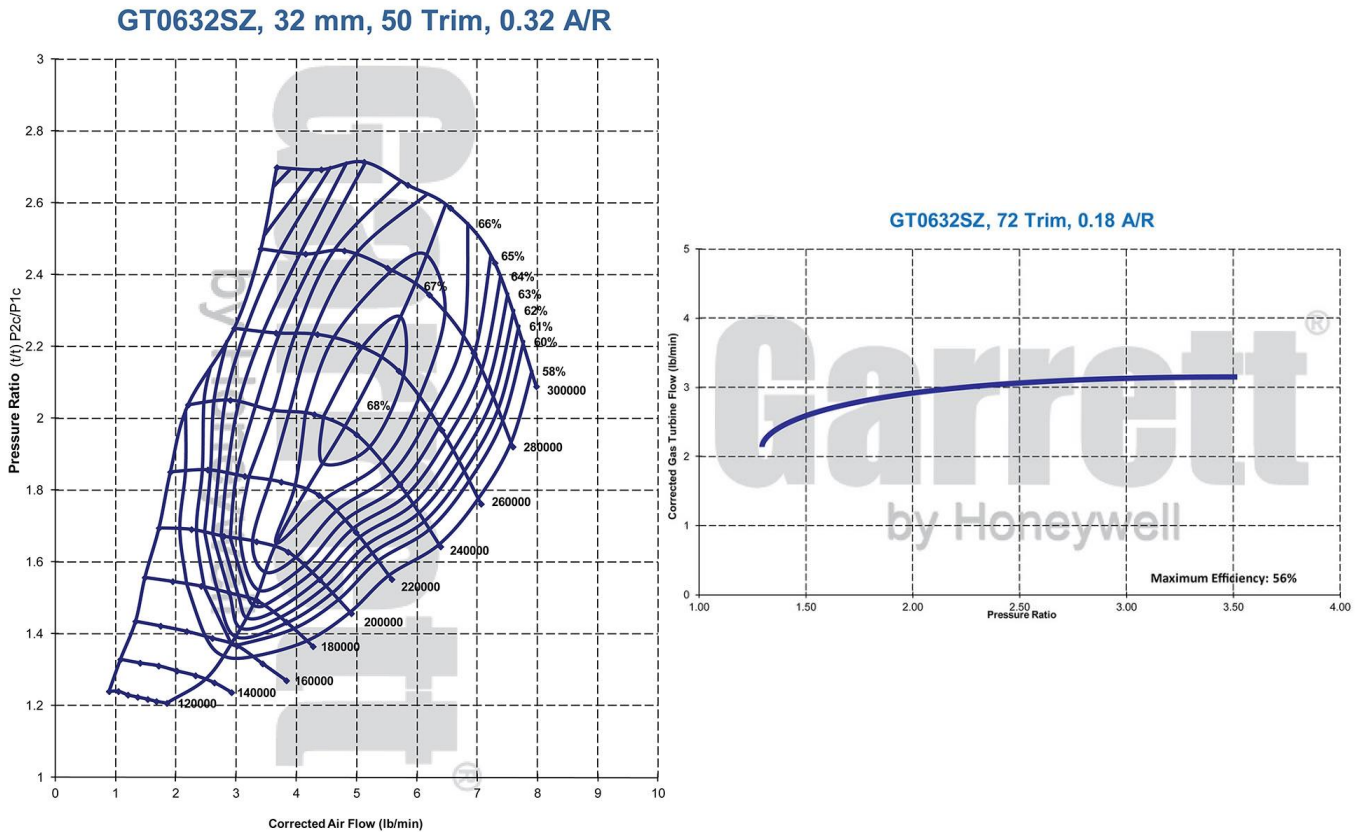


Figure 4.1 2: Garrett GT0632 Compressor and Turbine Maps (Garrett, 2016)

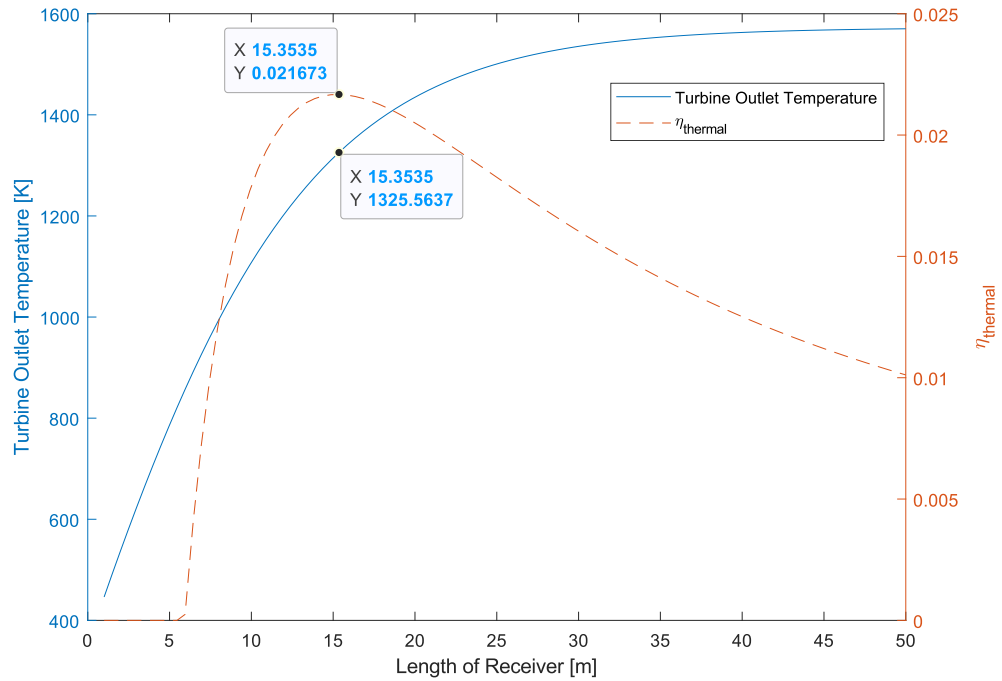


Figure 4.1 3: Turbine Outlet Temperature and Overall Heat Engine Thermal Efficiency of an Arbitrary Receiver attached to a Garrett GT0632 Turbocharger

manifolds. The cover gap width and glass thickness were kept the same as in Figure 3.2 2 and 3.

As per Figure 4.1 2, the smaller turbocharger was simulated operating in the middle of its compressor efficiency island at a static/static compressor pressure ratio of 2 and a mass flow rate of 5 lb/min; for a compressor efficiency of 68% and a turbine efficiency of 56%.

Figure 4.1 3 depicts the turbine side performance of the setup above with an increasing length of the receiver. As the length of the receiver increases, the outlet temperature of the turbine increases. At a certain point (about 5.95 m with a compressor outlet temperature of 389 K, a turbine inlet temperature of 939 K and a turbine outlet temperature of 857 K), the inlet temperature to the turbine is high enough to completely power the compressor stage. Any receiver length longer than this (and resultingly higher turbine inlet temperature) results in excess power being available to run the attached electrical generator.

That is to say, that the minimum turbine inlet temperature of air required to run a Garrett GT0632 turbocharger as an electrical generator is 939 K when operating at an ideal turbine static/static pressure ratio of 2 and real mass flow rate of 5 lb/min.

From Figure 4.1 3 it may be observed that operating the GT0632 turbocharger in conjunction with this arbitrary linear receiver as the only heat source results in a poor overall thermal efficiency. A maximum STE to electrical work thermal efficiency of 2.17% is located at a receiver length of 15.4 m. At this optimal receiver length; a compressor outlet temperature of 389 K, turbine inlet temperature of 1447 K, turbine outlet temperature of 1326 K, and an electrical power output of 1.997 kW is achieved.

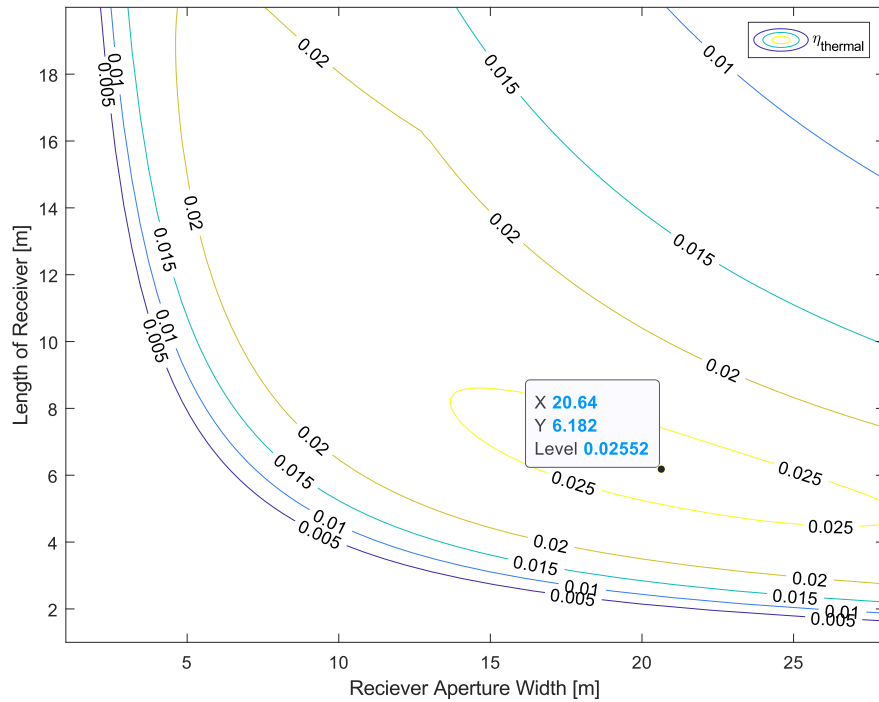


Figure 4.1 4: Thermal Efficiency of an Arbitrary Receiver of varying Aperture Widths and Receiver Lengths attached to a Garrett GT0632 Turbocharger

It is also possible to parametrically analyse the performance of the heat engine if it is the case that both the length and width of the linear receiver field are variable. This is unlikely to be the case in a real world implementation owing to the costs, mechanical control, mechanical stability and other design implications of fabricating a concentrator of arbitrary width. The overall most effective LCOE per kWh linear receiver aperture width is within the region of about 6-10 m (Abengoa Solar, 2013, p. 15).

Figure 4.1 4 shows the thermal efficiency of operating the GT0632 turbocharger in conjunction with the arbitrary receiver across a range of receiver lengths and aperture widths. A global optimal of 2.55 % is found at an aperture width of 20.6 m and receiver length of 6.2 m. This is not a substantial gain over the previous case of a more commercially viable aperture width of 6 m for an optimal thermal efficiency of 2.17% as in Figure 4.1 3.

A similar exercise may be performed for a more ideal choice of turbocharger and receiver aperture width.

The Garrett GTX3584 represents one of the best combinations of maximum turbine and compressor efficiencies in Garrett's catalogue (Garrett, 2016, p. 21). From Figure 4.1 5, an efficiency island for the GTX3584 exists at a compressor efficiency of 76 %, a turbine efficiency of 78 % at a static/static compressor pressure ratio of 2.25 and a mass flow rate of 50 lb/min.

The 8 m SpaceTube PTC represents one of the widest aperture and most cost effective PTC design manufactured by Abengoa (Abengoa Solar, 2013).

Figure 4.1 6 depicts the turbine performance of a GTX3584 connected to an arbitrary receiver with the same variables as Figure 4.1 3 but with a absorber

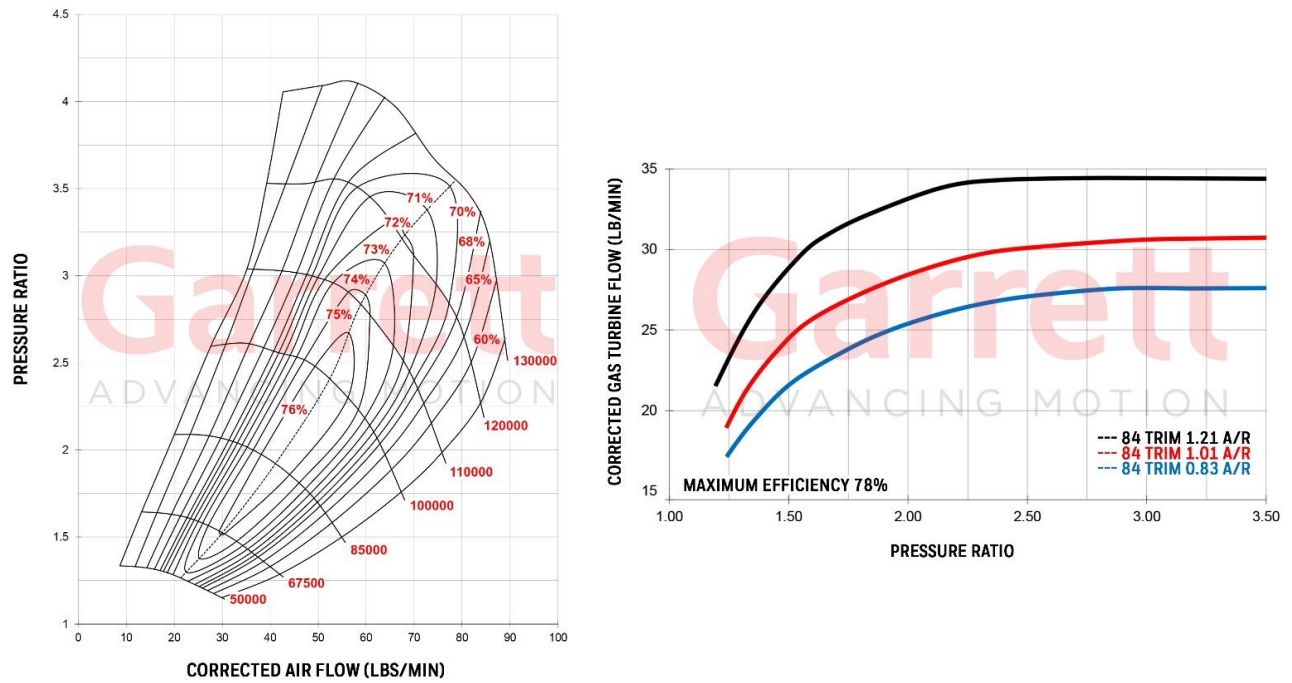


Figure 4.1 5: Garrett GTX3584 Compressor and Turbine Maps (Garrett, 2016)

diameter of 68 mm, aperture width of 8 m, real mass flow of 50 lb/min and static/static pressure ratio of 2.25 to match the efficiency island of Figure 4.1 5. A maximum thermal efficiency of 6.2% is achieved at a receiver length of about 63 m.

The parametric analyses and hypothetical optimisations above were performed for a single operating condition of mass flow rate and operating pressure ratio per turbocharger. These conditions were arbitrarily chosen as a point approximating the

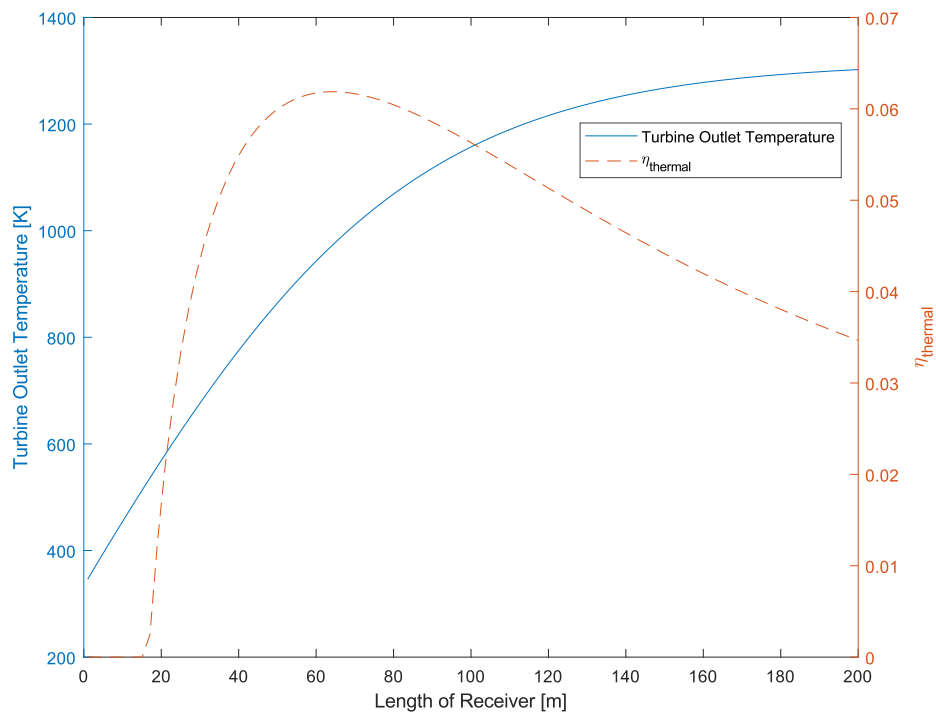


Figure 4.1 6: Turbine Outlet Temperature and Overall Heat Engine Thermal Efficiency of an 8 m Arbitrary Receiver attached to a Garrett GTX3584 Turbocharger

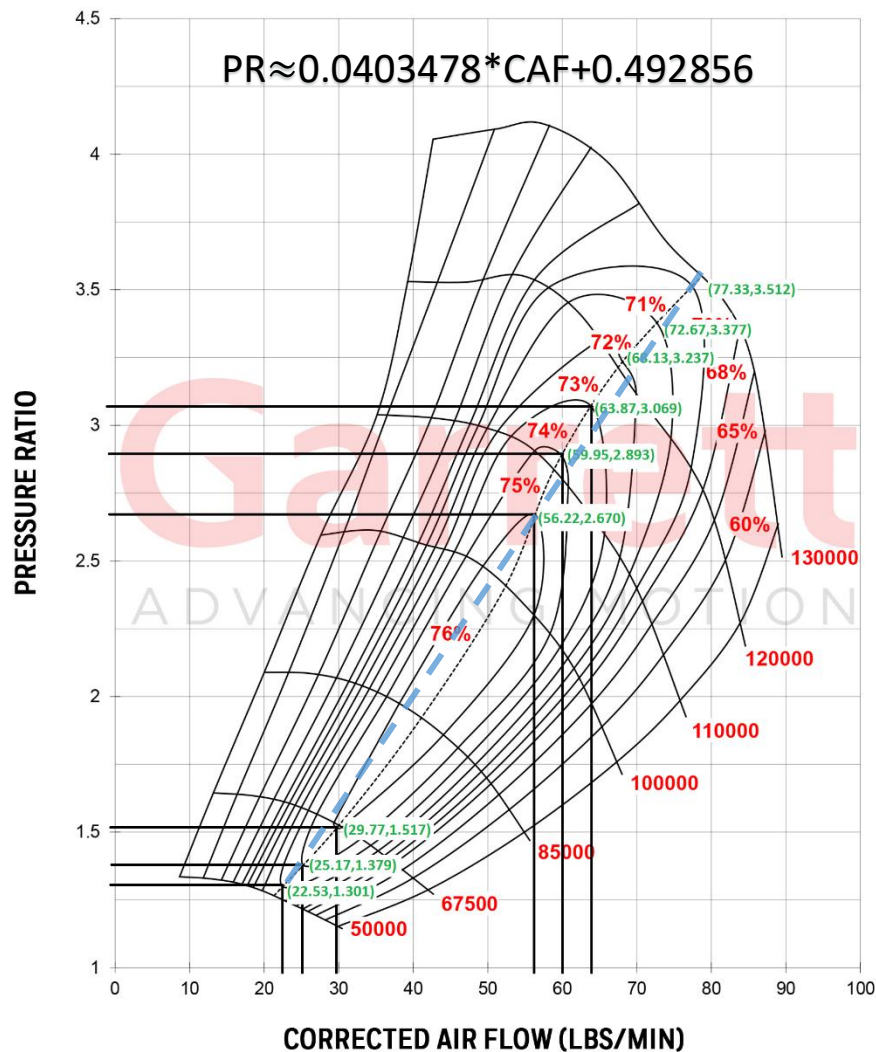


Figure 4.1 7: Building the Function of Corrected Mass Flow to Pressure Ratio for a Garrett GTX3584

compressor efficiency island peak on each of the GTX3584 and GT0632 compressor maps.

Mass flow rate and operating pressure is intrinsically linked to heat transfer dynamics within the SEC as well as the governing thermodynamics of the heat engine as a whole. What may be most efficient for the operation of the compressor as per manufacturers' charts may not necessarily be the most efficient for the CSP engine overall. Therefore, a hypothetical receiver length optimization may be performed for a range of mass flow rates and operating pressure ratios for different receiver lengths.

The efficiency spine which exists on a turbocharger compressor map may be approximated as function of corrected air mass flow rate to total/total compression ratio. This function has been approximated for a Garrett GTX3584 on Figure 4.1 7 as a straight-line function. Doing so allows for an operating compression ratio to be chosen for a wide range of mass flow rates; allowing for a more in-depth parametric analysis of such an engine.

In Figure 4.1 7 the values which may be simulated corresponds to values between 22.5 and 77.3 lb/min corrected mass flow rate and operating pressure ratios between 1.3 and 3.5 with compressor efficiencies between 71 and 76%.

The function for the linear line of best fit and the points along the spine which were used to calculate that line are given on Figure 4.1 7. Further, since it is known where on the line the compressor would be operating, the appropriate compressor efficiency may be set inside of the simulation code.

For example, from Figure 4.1 7; operation at a pressure ratio of 2 would yield a mass flow rate of 40 lb/min and compressor efficiency of 76 %, while operation at a pressure ratio of 3.5 would yield a mass flow rate of 70 lb/min and a compressor efficiency of 70 %

To simplify calculations in this respect, it will be assumed that the compressor total/total pressure ratio is equal to static pressure ratio. This is because $p_{t,compressor\ outlet}$ is equal to $p_{s,compressor\ outlet}$ to the 4th decimal figure for temperatures between ambient and 700 K and pressure ratios of 1 to 5 for the GTX3584. This avoids doing unnecessary iterative converging calculations to find the static turbine outlet temperature and pressure to calculate $p_{t,compressor\ outlet}$. However, values of $\dot{m}_c$ read from the compressor map must be converted to $\dot{m}$ using Equation 2.7 14.

Figure 4.1 8 shows the optimal receiver length and corresponding thermal efficiency for a range of mass flow rates with the GTX3584 connected to an arbitrary receiver with the same variables as Figure 4.1 6. As the operating mass flow rate is increased, a corresponding higher operating pressure ratio and lower

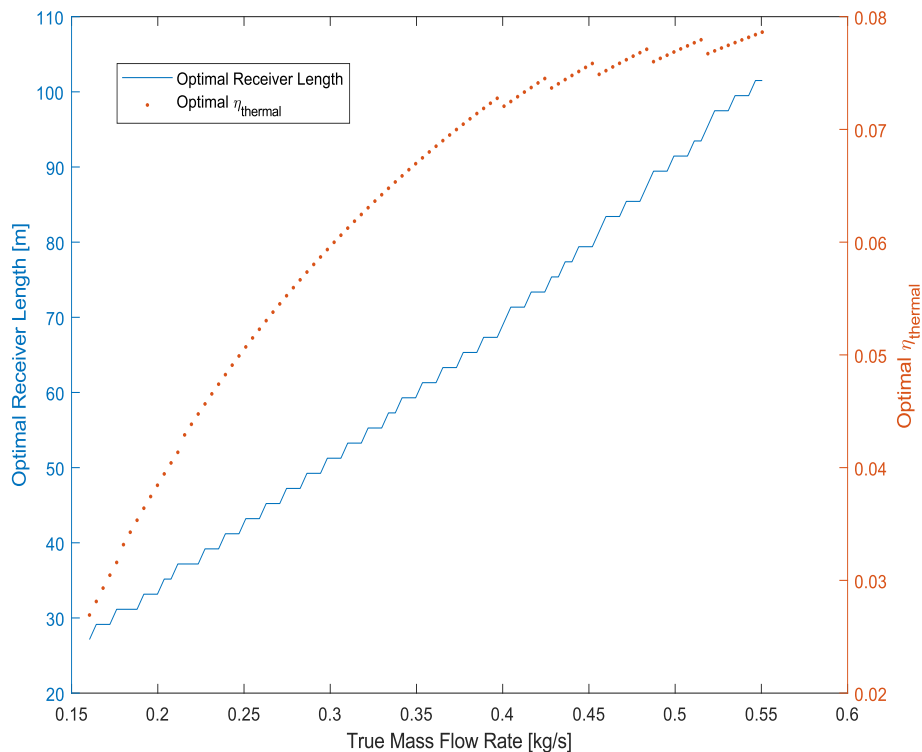


Figure 4.1 8: GTX3584 True Mass Flow Rate to Optimal Thermal Efficiency and Receiver Length

compressor efficiency is used as per the compressor map efficiency island spine as seen in Figure 4.1 7.

For Figure 4.1 8, operating at a higher compression ratio leads to a higher overall engine thermal efficiency. This is despite the fact that an increase in flow rate results in lower compressor efficiency for mass flow rates and pressure ratios higher than the efficiency peak on the compressor map as from Figure 4.1 7.

The shape of the optimal efficiency curve follows that of a negative power function with a negative exponent. This is the same as what is expected by the function of the thermodynamic limit of a Brayton Cycle's efficiency – $\eta_{Brayton,thermal,max}$ – as defined in Equation 2.7 3. Specifically, there are diminishing returns on efficiency gain of the engine as the pressure ratio is increased.

Therefore, for relatively low pressure ratios in the region of about 2-6, the thermodynamic benefit of operating the turbocharger-based engine at higher mass flow rates and pressure ratios is greater than the corresponding decrease in compressor efficiency as pressure ratio increases.

It follows that it is advisable to rather operate the turbocharger-based heat engine at a point as near to as high a pressure ratio as possible. The fundamental issue with this operational strategy is that operating a turbocharger at a high mass flow rate and pressure ratio inevitably means that the compressor will be operating at a point close to its choke line and surge lines too.

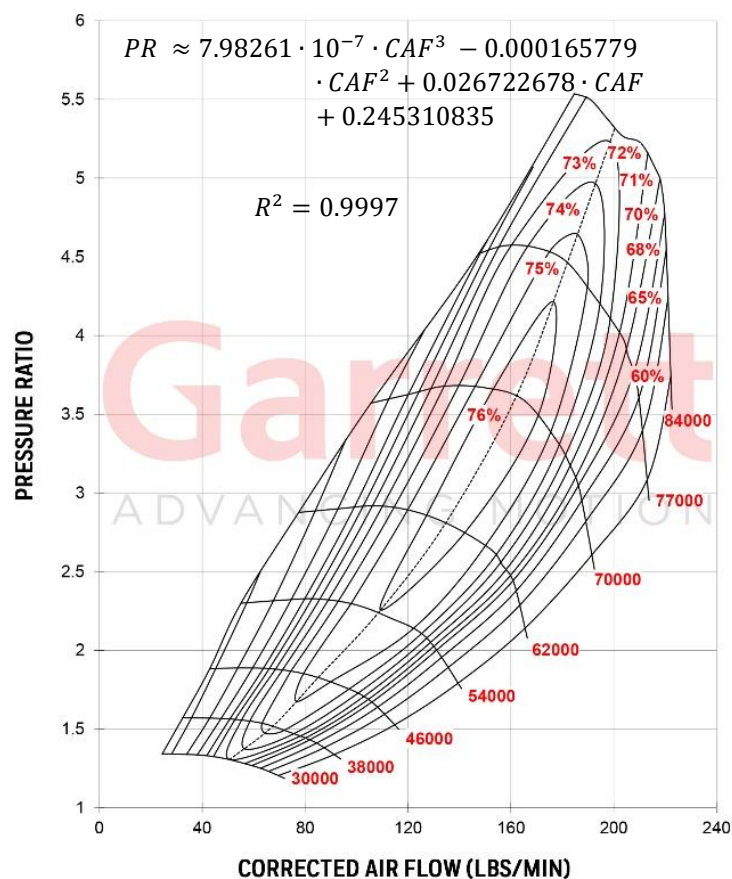


Figure 4.1 9: Garrett GTX5533R GEN II 98mm Compressor Map (Garrett, 2016) with Function of Corrected Mass Flow to Pressure Ratio

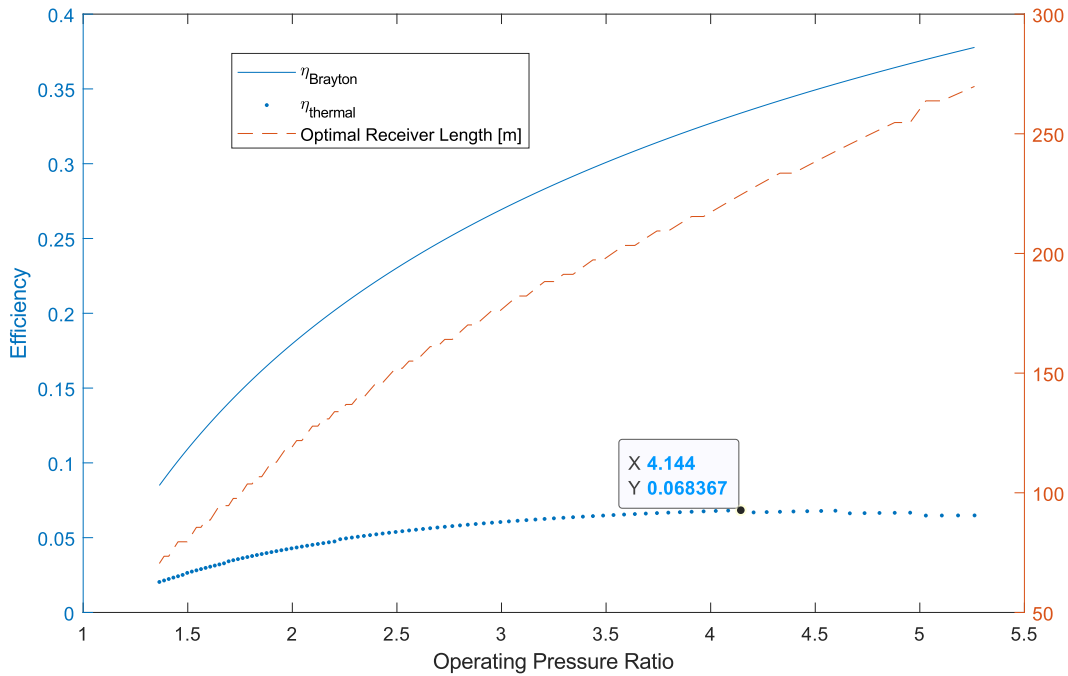


Figure 4.1 10: GTX5533R GEN II 98 mm Operating Pressure Ratio to Thermal and Brayton Efficiencies at Ideal Arbitrary Receiver Lengths

Operating a turbocharger-based CSP BCHE at a mass flow and pressure ratio as high as possible will make the compressor susceptible to going into choke or stall if there is an unexpected spike in the heat addition stage, or a sudden flow restriction or pressure change anywhere in the engine.

A similar exercise as Figures 4.1 7 and 8 may be performed on a turbocharger targeting as high an operating compression ratio as possible. The Garrett GTX5533R GEN II with the 98 mm compressor inducer represents the greatest available pressure ratio of Garrett's catalogue of approximately 5.25 (Garrett, 2016).

For Figure 4.1 9, instead of a straight-line function being chosen to approximate the maximum efficiency spine on the compressor map such as Figure 4.1 7, a polynomial function of third degree was used to better characterise the design line function.

To generate the function for Figure 4.1 9, a high-resolution image (1775×2075) of the compressor map was downloaded from Garrett's website (Garrett, 2018). Each point of intersection of the dotted efficiency line with an island edge of isenthalpic efficiency had its pixel co-ordinates recorded and compared to the whole scale of the graph providing a list of x-y coordinates which was then used to calculate the polynomial line of best fit.

Figure 4.1 10 shows the performance of a GTX5533R connected to an arbitrary receiver with the same variables as Figure 4.1 6 with an absorber diameter of 112 mm and mass flows and pressure ratios corresponding to the derived function on Figure 4.1 9. The theoretical maximum Brayton Cycle thermal efficiency from Equation 2.7 3 for each pressure ratio is also plotted to provide an efficiency upper limit comparison to the calculated CSP heat engine.

For Figure 4.1 10, the GTX5533R based heat engine achieves a maximum thermal efficiency of 6.84% at a pressure ratio of 4.14 and receiver length of 224.5 m. The maximum Brayton Cycle thermal efficiency for this pressure ratio is 33.4%.

Up to this point, an appropriate arbitrary receiver has been connected to three different turbochargers; a micro GT0632, an efficient GTX3584 and a high flow and pressure ratio GTX5533R. The configuration of the arbitrary receiver was such that STE was the only source of energy, and the turbocharger engines were operated without any form of heat recuperation or heat recycling. The GT0632 was found to have optimal receiver dimensions for a thermal efficiency of 2.17 % at a pressure ratio of 2 ($\eta_{Brayton,thermal,max}$ of approximately 18 %), the GTX3584 obtained 7.80 % thermal efficiency at a pressure ratio of 3.41 ($\eta_{Brayton,thermal,max}$ of approximately 29 %), and the GTX5533R 6.84 % thermal efficiency at a pressure ratio of 4.14 ($\eta_{Brayton,thermal,max}$ of approximately 33 %),.

From Equation 2.7 3, an increase in pressure ratio at low pressure ratios (i.e., less than 10) would infer a substantial gain in engine performance, however, the turbochargers capable of the higher pressure ratios tend to be much larger and require significantly higher flow rates and larger diameter SEC receiver tubes. This means that the receiver tube for the larger turbines must itself be wider and much longer to capture the insolation necessary to heat the HTF. This substantially increases the surface area available for losses along the receiver.

With thermal efficiencies in the region of 7 %, it is clear that a that operating a commercial turbocharger as the compressor and turbine of a BCHE with heat addition solely from STE using a linear concentrator without any form of heat recycling is not viable.

4.2 Hybrid Operation of a Turbocharger Based Linear Receiver CSP BCHE

In Section 4.1 it was shown that it would be technically feasible to operate a modified commercial turbocharger as a Brayton Cycle Engine solely from the heat from a linear receiver SEC made from contemporary materials. However, doing so would not be economically viable for thermal efficiencies in the region of 7% when compared to conventional Rankine Cycle CSP heat engines or even PV panels.

An alternative approach is to rather use solar energy to preheat air prior to the mixing and combustion of an added hydrocarbon-based fuel.

Low temperature heat is added to the engine in the form of solar energy, reducing the potential for losses at or along the collector's receiver. Heat generated through combustion may be done so in a single combustion chamber unit. By hybridizing the two heat sources, inexpensive heat from the linear receiver augments the heat released by burning the fuel (Korzynietz, et al., 2016, pp. 578-589; Rovensea, et al., 2017, pp. 675-682; Schwarzbözl, et al., 2006, pp. 1231-1240). This upgrades the exergy of the fuel and increases the fuel's economy (Merchán, et al., 2018).

Keeping surface areas visible to ambient at a minimum helps to reduce ambient heat losses throughout the engine.

A range of other operational benefits are also present when operating in a hybrid fashion which is detailed in Section 2.2.1.

One fuel of particular interest in the South African context would be the combustion of the products of Underground Coal Gasification. Conventionally the syngas products of UCG would simply be burnt in a gas turbine to generate electricity (Thopila & Pourisb, 2016, p. 30). A solar-UCG hybrid would effectively mean that the solar energy is used to upgrade the exergy of the UCG syngas before it is burnt in a gas turbine.

In Section 4.1 it was found that operating the turbocharger at as high a pressure ratio as possible implies the greatest overall thermal efficiency of the turbocharger-SEC engine. For the purpose of this parametric analysis, an air flowrate for the engine will be chosen that corresponds to 90 % of the point where the compressor efficiency spine meets the choke line. This provides a reasonably high operating pressure ratio for the turbocharger as well as providing some operational padding to prevent the compressor from stalling or going into choke.

Each turbocharger will be modelled with its own appropriate arbitrary receiver. The variables used in modelling each arbitrary receiver will be the same as that detailed for in Figures 3.2 2 and 3 unless otherwise stated.

The absorber diameter for each arbitrary receiver will be set equal to the turbine inducer diameter to limit pressure drop within the engine. A 6 mm gap will be used between the absorber and the cover's inner. The cover will be set at 10 mm thick. The absorber surface will be set as a W:Al₂O₃ cermet with the cover made from Pyrex. The aperture width of the collector will be set at 8 m. Finally, the windspeed will be assumed to be 5 m/s.

The length of each arbitrary receiver attached to each turbocharger setup may be found for a combustion chamber inlet temperature of 650 °C. Temperatures higher than this lead to complications in fuel mixing, preignition and emissions which somewhat defeats the purpose of a renewables approach (Bryner, et al., 2016). Research is currently being conducted on designing better high inlet temperature combustors; but for the purposes of this analysis a 650 °C combustion chamber inlet temperature will be targeted.

Table 4.2 1: 90% of Choke Air Flow Rate Operating Conditions for Various Turbochargers

Turbocharger	Corrected Air Flow Rate [lb/min]	Pressure Ratio	Compressor Efficiency	Turbine Efficiency	Turbine Inducer Diameter [mm]
GT0632SZ	5.833	2.323	0.67	0.56	32
GTX3584RS	70.51	3.315	0.72	0.78	68
GTX5533 Gen II 98mm	180.4	4.358	0.75	0.74	112

Table 4.2 2: Arbitrary Linear Receiver Operating Conditions for Various Turbochargers

	Compressor outlet Temperature [K]	$\Delta H_{compressor}$ [kW]	Receiver Length [m]	Q_{HTF} [kW]	Q_{solar} [kW]	$\eta_{receiver}$	Receiver outlet Temperature [K]
GT0632SZ	412.18	3.11	2.98	14.77	23.84	0.6197	923.18
GTX3584RS	456.20	84.50	49.41	264.87	395.29	0.6701	923.66
GTX5533 Gen II 98mm	491.31	275.15	128.20	652.80	1025.62	0.6365	923.78

Table 4.2 1 lists the 90% choke flow rate conditions for each of the three turbochargers as used in Section 4.1; a micro GT0632, an efficient GTX3584 and a high flow and pressure ratio GTX5533R.

Table 4.2 2 lists the calculated arbitrary receiver's operating conditions such that a receiver outlet temperature of 650°C is obtained for each turbocharger at the engine operating conditions set in Table 4.2 1.

From Table 4.2 2, all three arbitrary receivers operate at receiver thermal efficiencies in the region of about 64 %. The SOLGATE and SOLUGAS pilot tests of a point focus receiver heating air and operating at about 650 °C with measured receiver thermal efficiencies of about 75% (European Commission for Research, 2005; Korzynietz, et al., 2016).

Point focus receivers have a LCOE between 20-30% more than parabolic troughs (International Renewable Energy Agency, 2012), however, the measured efficiency of the SOLGATE/SOLUGAS point focus receiver is only 12-21% greater than that calculated for an arbitrary linear receiver of contemporary materials (Korzynietz, et al., 2016). Assuming all other factors being equal, there is an argument to be made for the application of contemporary linear PTCs for heating HTFs up to about 650 °C over conventional point focus HFC methods.

To perform a parametric analysis on the rate of heat added to the turbocharger-based engines, the following terms have been defined:

$$\eta_{thermal,Solar+HC} = \frac{W_{electrical}}{Q_{solar} + Q_{HC}} \quad (4.2\ 1)$$

$$\eta_{thermal,net\ to\ HTF} = \frac{W_{electrical}}{Q_{HTF,absorbed} + Q_{HC}} \quad (4.2\ 2)$$

Where Q_{HC} is the rate of heat added to the engine by means of combustion of hydrocarbons to the air stream within the combustion chamber.

$\eta_{thermal,Solar+HC}$ represents the overall thermal efficiency of the engine for all heat sources supplied to the collector and combustion chamber. $\eta_{thermal,net\ to\ HTF}$ is the thermal efficiency of the heat that is successfully transferred to the HTF from all sources.

Figure 4.2 1 shows the performance for various turbochargers operating in a hybrid fashion each attached to appropriate arbitrary receivers to produce a combustion

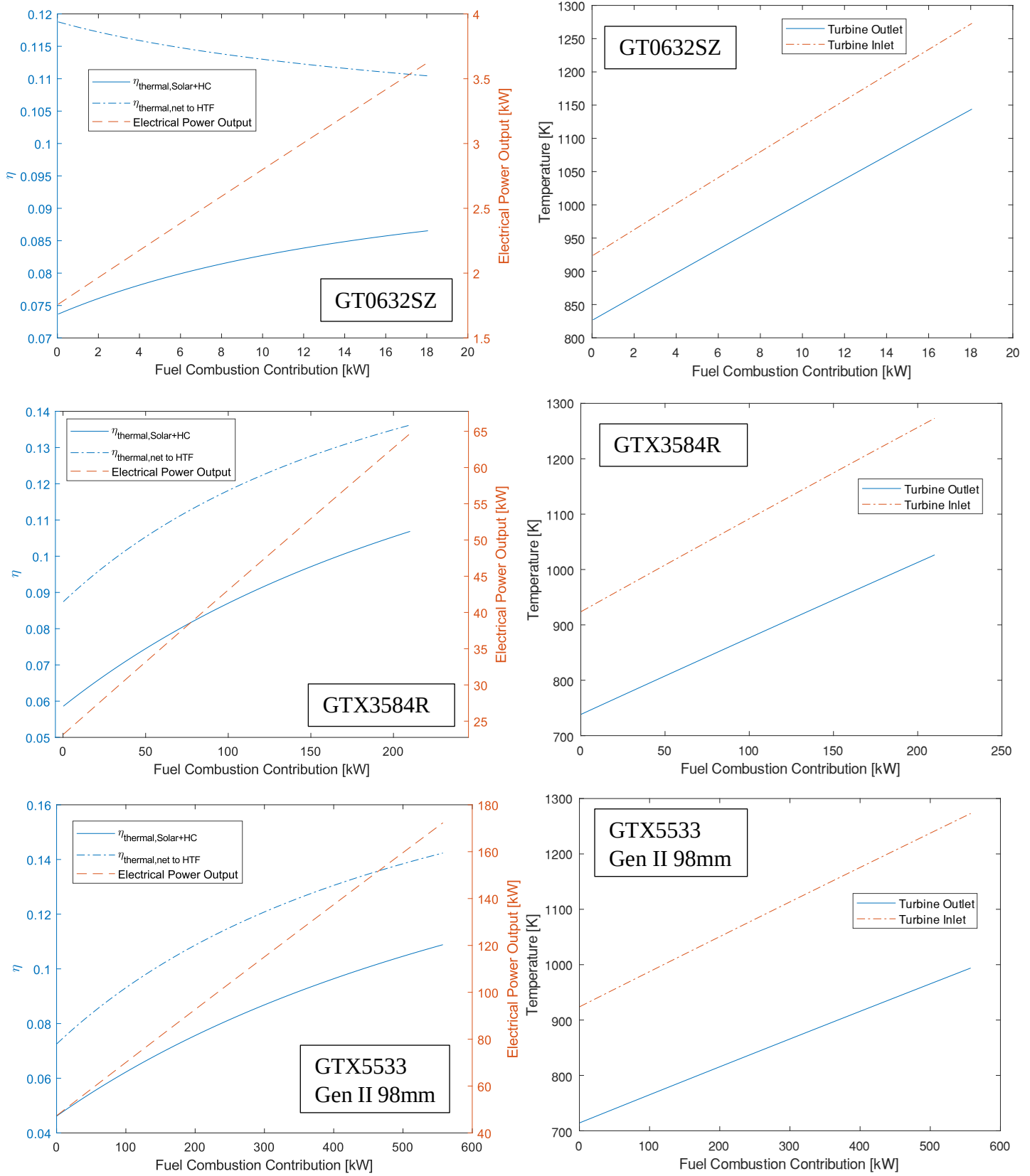


Figure 4.2 1:Hybrid Operation of Various Size Turbochargers With Appropriate Arbitrary Recievers Without Heat Recycling

chamber inlet temperature of 650 °C. Turbine inlet temperatures up to 1000 °C are considered as that is about the maximum design turbine inlet temperatures for commercial turbochargers' materials of construction (BorgWarner, 2016).

For all of the turbochargers modelled in Figure 4.2 1, as the rate of fuel burnt is increased, engine power output increases in an approximately linear fashion. As the rate of fuel burnt is increased, thermal efficiency for heat successfully delivered to the HTF increases with diminishing returns.

From Figure 4.2 1, all three of the turbochargers are capable of producing electrical power without burning any hydrocarbons at a turbine inlet temperature of 923 K (650 °C). The GT0632 produces a base of 1.76 kW of electricity without burning any fuel, and produces an extra 104 W of electricity per kW of fuel consumed to a maximum of 3.63 kW of electricity. The GTX3584 produces a base of 23.2 kW of electricity plus an additional 198 W per kW of fuel consumed up to 64.3 kW of electricity. The GTX5533R produces a base of 47.4 kW of electricity plus an additional 224 W per kW of fuel consumed up to 172 kW of electricity.

From Figure 4.2 1, all of the solar hybrid turbocharger-based engines performed at relatively poor thermal efficiencies. Overall engine thermal efficiencies ($\eta_{thermal,Solar+HC}$) were low at turbine inlet temperatures of 923 K when no hydrocarbons were burnt: the GT0632 7.4 %, the GTX3584 5.8 % and the GTX5533R 4.6 %. These values are about half that of standard solar to electric thermal efficiencies of conventional CSP technologies which lie in the region of about 11 %, but where conventional CSP technologies occasionally reach peak solar to electric thermal efficiencies of 20 % (Müller-Steinhagen & Trieb, 2004, pp. 43-50).

Peak overall thermal efficiencies for the hybrid turbocharger engines were also low at the maximum turbine inlet temperatures of 1273 K (1000 °C): the GT0632 8.6 %, the GTX3584 10.7 % and the GTX5533R 10.9 %. The most “fuel efficient” of the three turbochargers was the GTX5533R which produced 224 W of electricity per kW of fuel consumed – for a strictly hydrocarbon thermal efficiency of only 22.4 %, which itself too is below standard simple-cycle Gas Turbine operational fuel efficiencies of 30 to 40 % (Brooks, 2000, p. 7).

The cases analysed have been for a configuration without any heat recycling. When operating in a solar preheating hybrid fashion for the turbocharger-based engines, a turbine inlet temperature of 1000 °C leads to turbine outlet temperatures well above 650 °C. Since operating at an optimised thermal efficiency is desired, the turbine inlet temperature would be kept as high as possible. From Figure 4.2 1, in the most efficient turbine case (and therefore lowest possible turbine outlet temperature), the GTX5533 obtains a turbine outlet temperature of 727 °C.

In this case with an appropriate heat recycling unit attached to the compressor outlet, a combustion chamber inlet temperature of 650 °C may be easily reached without the need for a CSP pre heating field.

This invalidates the purpose of the CSP preheating stage with solar preheating hybrid operation of modified vehicle turbochargers. Such a heat recycling unit

would be far less expensive to fabricate, maintain and operate than a whole SEC field. Importantly, operation would not depend on the weather or time of day. This observation is perhaps why in literature concerning modified turbochargers, emphasis is placed on optimising the design of heat recycling units and not necessarily the SECs themselves (Le Roux, et al., 2012; Jansen, et al., 2015; Mariscal-Hay & Leon-Rovira, 2014)

Therefore, it is not viable to operate modified vehicle turbochargers in a solar preheating hybrid fashion without any form of heat recycling by means of direct combustion in conjunction with CSP of any design.

4.3 Operation with Heat Recycling

In the Section 4.2, it was shown that using CSP to preheat air to 650 °C for use in direct combustion to run a turbocharger-based Brayton Cycle Heat Engine is technically feasible but wholly unviable.

Since as high a turbine inlet temperature is desired for maximal power output and thermal efficiency, high turbine outlet temperatures invalidate the purpose of a solar based preheating stage prior to a direct combustion chamber. This is because the heat from the turbine outlet may be recycled to the compressor outlet stream and relatively easily be brought up to 650 °C. This unit would be far more compact, less expensive and more reliable than an entire SEC field with the added benefit of 100% uptime without relying on the weather or time of day.

As turbine inlet temperatures are kept as high as possible, high turbine outlet temperatures suppose the reuse of energy as much as possible for a real-world application. Therefore, operation with a heat recycle unit or at least a co-generation heat recovery unit is implied.

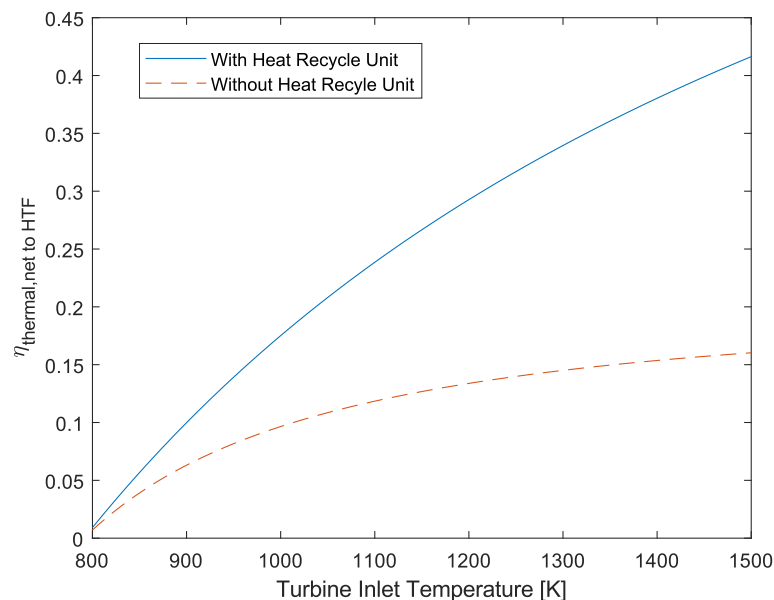


Figure 4.3 1: GTX5533 Based BCHE Net Thermal Efficiency with and without a Heat Recycling Unit

Figure 4.3 1 shows the net heat to HTF thermal efficiency for a GTX5533 Gen II. 98 mm turbocharger based BCHE calculated in the same fashion as Figure 4.2 6 considering the net thermal energy to the HTF both with and without a heat recycle unit. It has been assumed that the counter-current heat exchanger performing the heat recycling is capable of obtaining a temperature delta of 50 °C between the high temperature stream inlet (from the turbine outlet) and the low temperature stream outlet (to the SEC and/or combustion chamber) and assumes a negligible pressure drop.

From Figure 4.3 1, there is an extraordinary benefit to thermal efficiency when operating with a heat recycling unit. For a modified GTX5533 based BCHE, there is approximately a doubling of thermal efficiency for any temperature by the use of a heat recycling unit.

From Figure 4.3 1, for a turbine inlet temperature of about 1000 °C (1273 K), the net heat absorbed thermal efficiency with a heat recycling unit is approximately 33%, and nearly 15% without one. This is very close to the net heat absorbed thermal efficiency of approximately 30% found by Le Roux et al. (2011) and their HFC based modified turbocharger CSP BCHE with a heat recycling unit and turbine inlet temperature of 1000 °C (Le Roux, et al., 2011).

As a further comparison, a GTX5533 may be operated in conjunction with a W:Al₂O₃ cermet based arbitrary linear receiver in a CSP only fashion with its performance calculated in the same fashion as Figure 4.3 1. The surface temperature stability limit of the W:Al₂O₃ cermet is about 700 °C.

Table 4.3 1 summarises operating a GTX5533 in conjunction with an appropriate arbitrary receiver to achieve a turbine inlet temperature of 700 °C for the cases with and without a heat recycling unit. At these operating conditions, the turbocharger engine is capable of outputting about 65 kW of work or electricity. A receiver length of about 90 m is required to achieve this with a heat recycle unit, or a receiver length about 60% greater without a heat recycle unit. The overall thermal efficiency of the engine with a heat recycle unit is about 9%, which is comparable with conventional commercial CSP technologies (Table 2.3.1 1).

Such an installation would be suited for a domestic or small commercial application, particularly in remote and arid areas without the access to cooling water for a similar Rankine Cycle setup. Coupling the CHRA of the turbocharger to the rotor of a generator means the entire assembly contains only one moving part implying high reliability and low maintenance of the setup.

Table 4.3 1: CSP-Only Linear Receiver Based GTX5533 BCHE at a Turbine Inlet Temperature of 700 °C

	Receiver Length [m]	$\eta_{receiver}$	$\eta_{thermal\ to\ electricity}$	$\eta_{thermal,net\ to\ HTF}$	P_{elec} [kW]
With Recycle	90.1	57.7%	9.00%	15.6%	64.9
Without Recycle	147	62.0%	5.51%	8.89%	64.9

At this scale, a linear receiver CSP turbocharger based BCHE competes with PV based electrical systems in terms of solar efficiency and implied reliability. The utility of a hybrid setup may need to be assessed in this case when choosing between the two technologies. It may be more economical to store and burn fuel in this scenario compared to the large capital investment necessary for battery storage.

4.4 Discussion

This section consisted of a series of simulation based parametric analyses on a CSP linear focus receiver attached to a modified vehicle turbocharger Brayton Cycle Heat Engine. Three commercial Garrett turbochargers were used in the analyses; the small GT0632SZ motorcycle turbocharger – the same that was used for experimentation that will be detailed in Section 5 – the highly efficient mid-sized GTX3584RS, and the large high pressure-ratio and high air flow rate GTX5533 Gen II turbocharger outfitted with a 98 mm inducer.

Modifying vehicle turbochargers to operate as small, inexpensive and relatively efficient gas turbines have been discussed in a multitude of papers (Le Roux, et al., 2012; Jansen, et al., 2015; Mariscal-Hay & Leon-Rovira, 2014). Such engines produce power at a domestic or small commercial scale. The mass production of these turbochargers implies a degree of reliability and low cost which may be adapted to produce electricity in a cost-effective fashion.

In this work, a simulation of using a linear focus receiver together with an attached turbocharger-based engine was investigated. Two approaches were modelled: pure solar operation which is analogous to PV technologies; and hybrid operation where CSP was used to preheat air prior to combustion to upgrade the exergy of the fuel.

In parametrically optimizing the dimensions of an arbitrary receiver made from contemporary materials attached to a Garrett GT0632SZ turbocharger without any form of heat recycling, it was found that the maximum operational thermal efficiency of the setup would be in the region of about 2.3%. This very low efficiency was attributed to a low operating pressure ratio and relatively low efficiencies of the compressor and turbine stages. This implies that the experiment in Chapter 5 using this turbocharger should expect very low thermal efficiencies.

It was shown that an arbitrary parabolic trough linear focus collector made from contemporary receiver materials should be capable of operating with an outlet temperature of 700 °C at a cumulative receiver efficiency of 57 to 62 %. It was also shown that the linear focus receiver should be capable of outlet temperatures in the excess of 1000 °C but at very low cumulative efficiency when assuming the absorber would be stable in this state.

For the turbocharger engines analysed it was found that in general it was beneficial to operate at as high a pressure ratio as possible. This produced an overall gain in thermal efficiency, despite the fact that operating at these higher pressure-ratios decreases compressor efficiency. Explicitly, the best operational point of the

turbocharger as an engine is not at the point of highest efficiency on the compressor map.

Subsequent analyses of the turbocharger engines were operated at a mass flow rate at 90% of the intersection between the compressor choke and surge lines. Doing so maximises the operating pressure ratio (hence maximising engine thermal efficiency) but leaves an operational buffer to prevent the compressor going into surge or choke for small unexpected changes in operating conditions.

Larger turbochargers capable of higher pressure-ratios themselves require larger piping for their higher air mass flow rates. Since it is only mechanically practical to build linear collectors with an aperture of about 6-10 m, much longer receiver lengths are required to heat the air. The greater circumference receiver tubes along with the greatly lengthened receiver loops leads to a substantial increase in surface area of the receiver available for heat losses.

It was found that a GTX3584RS attached to an appropriately optimized arbitrary linear focus receiver operated at a higher overall thermal efficiency at a lower pressure ratio than a physically larger GTX5533 Gen II based engine. This may be attributed to a greater proportion of heat losses from the SEC associated with the GTX5533 despite the GTX3584RS operating at a lower pressure ratio and mechanical efficiency.

Larger turbochargers are associated with higher compressor and turbine efficiencies, but their larger flow rates necessitate larger diameter piping. A balance must be struck in optimising between the use of larger more efficient turbochargers, and the associated increase in surface area of the attached collectors, since the turbochargers capable of the higher-pressure ratios tend to be much larger and require significantly higher flow rates and larger diameter SEC receiver tubes. This means that the receiver tube for the larger turbines must itself be wider and much longer to capture the insolation necessary to heat the HTF. This substantially increases the surface area available for losses along the receiver.

Next, an investigation was made to the operation of the engine in a hybrid fashion by using solar energy to preheat the air charge isobarically prior to combustion, thereby upgrading the exergy of the hydrocarbon fuel.

A physical limitation of this preheating process is the maximum combustion chamber inlet temperature of 650 °C with conventional fuel jet nozzle designs. Temperatures higher than this lead to complications in fuel mixing, preignition and emissions that somewhat defeat the purpose of a renewables approach. Therefore, a linear receiver outlet temperature of 650 °C was targeted for this investigation.

It was shown though the intensive linear receiver model that the preheating air to a temperature of 650 °C should be easily possible using contemporary materials of construction for the collector's receiver. This is contrasted with the SOLGATE/SOLUGAS project as well as the investigations done by Le Roux et al. (2011) where a more expensive point focus receiver were used for preheating or as primary heat in similar air-Brayton gas turbine engines but achieving the same effect.

Point focus receivers have a LCOE between 20-30% more than parabolic troughs (International Renewable Energy Agency, 2012). The experimentally measured efficiency of the SOLGATE/SOLUGAS point focus receiver hybrid CSP BCHE obtained an overall receiver efficiency about 12-21% greater than that of a simulated arbitrary linear receiver of contemporary materials (Korzynietz, et al., 2016).

Assuming all other factors being equal, there is an economic argument to be made in the application of contemporary linear PTCs for heating HTFs up to about 650 °C instead of conventional point focus HFC methods. The HTF may be used directly or indirectly in any heat engine cycle. Supercritical CO₂ Rankine Cycle Engines are particularly well matched to this temperature range (Ahn, et al., 2015; Bauer, 2016).

During solar preheating hybrid operation, the turbocharger-based engine output power produced is approximately linearly correlated to the rate of fuel burnt. As the rate of fuel burnt increases, thermal efficiency increases with diminishing returns.

Simulating a GTX5533 Gen II BCHE with an appropriate arbitrary linear receiver SEC to obtain a combustion chamber inlet temperature of 650 °C yielded a receiver efficiency of 64%. An overall thermal efficiency $\eta_{thermal,Solar+HC}$ of 11% is obtained by burning enough fuel to produce a turbine inlet temperature of 1000 °C. This is an unsatisfactorily low overall thermal efficiency.

At an optimal operational turbine inlet temperature of 1000 °C for the GTX5533, a turbine outlet temperature of about 725 °C is obtained in the best case of operating conditions for all of the turbochargers analysed. This is much higher than the combustion inlet temperature of 650 °C. This means that the entire SEC preheating stage may be replaced by a single much more reliable – and far less expensive – counter current heat exchanger.

Therefore, due to the limitation of a combustion chamber inlet temperature of 650 °C, using CSP as a preheating stage for direct combustion as a modified turbocharger-based hybrid BCHE is not viable. This is true for any form of CSP preheating intended for fuel combustion using modified commercial turbochargers when there is no heat recycling stage, not just linear focus collectors.

A GTX5533 based BCHE was simulated to quantify the value of a heat recycling unit. At a turbine inlet temperature of 1000 °C, the net heat absorbed thermal efficiency with a heat recycling unit is approximately 33% and about 15% without one. This is very close to the net heat absorbed thermal efficiency of approximately 30% found by Le Roux et al. (2011) and their HFC based modified turbocharger CSP BCHE with a heat recycling unit.

This implies that linear focus receivers could be competitive with point focus HFCs in terms of receiver efficiency at a temperature range of about 650 °C. An economic argument may be made in support linear receivers in this case due to their lower LCOE.

Finally, a parametric optimization was performed on a GTX5533 Gen II BCHE and associated linear receiver operating in a CSP-only fashion with a heat recycling unit. This setup would perform functionally identical to a PV installation.

Using W:Al₂O₃ cermet as the absorber surface substrate provides a thermal stability limit of about 700 °C. By calculating appropriate receiver lengths to achieve this absorber surface temperature, the overall thermal efficiency of the engine with a heat recycle unit is about 9% generating 65 kW of electricity.

This efficiency and scale is somewhat comparable to that of communal domestic and small industrial PV systems which achieve solar-electric efficiencies of 14 % (Goodrich, et al., 2012, p. 13). The utility of a hybrid setup especially at night or during inclement weather may need to be assessed in this case when choosing between the two technologies, particularly for rural applications. In these scenarios it may be more economical to store and burn fuel compared to the large capital investment and security risk of battery-based electricity storage.

Alternatively, the multitude of moving parts, unitary scalability in approximately 65 kW_e sections, regular required maintenance and oil consumption potentially make PV a more compelling alternative.

Since it has been shown that achieving HTF temperatures of 700 °C by means of a linear receiver may be possible, it is worth investigating the use of a sCO₂ Rankine Cycle over that of a modified turbocharger Brayton Cycle.

4.5 Conclusion

In this section a series of simulations were performed using the intensive linear receiver model developed in Section 3 by connecting appropriately sized arbitrary linear receivers to commercial vehicle turbochargers. These turbocharger-based engines were connected to linear receivers such that heat addition was performed isobarically, effectively operating the turbochargers as Brayton Cycle Heat Engines.

Simulating the operation of such a turbocharger-based engine at an appropriate compression ratio and turbine inlet temperature of 1000 °C in conjunction with a heat recycling stage produced a net heat absorbed thermal efficiency of approximately 33%. This is close to the net heat absorbed thermal efficiency of approximately 30% found by Le Roux et al. (2011) and their HFC based modified turbocharger CSP BCHE with a heat recycling unit and turbine inlet temperature of 1000 °C.

Three different arbitrary receivers were modelled for different turbochargers in a solar preheating hybrid layout. This solar hybrid layout used solar energy from the linear receiver to act as a preheating stage for air prior to a combustion chamber which would be located immediately before the turbine inlet. When targeting a combustion chamber inlet temperature of 650 °C to inhibit emissions, all three arbitrary receivers operated at receiver thermal efficiencies in the region of about 64 %. This is not a significant performance departure to that measured at the SOLGATE and SOLUGAS pilot test plants using a point focus receiver and heating similarly compressed air to 650°C with measured receiver thermal efficiencies of about 75% (European Commisison for Research, 2011; Korzynietz, et al., 2016).

Parabolic troughs have a LCOE benefit of 17 to 23 % compared to point focus receivers (International Renewable Energy Agency, 2012). If these simulations are accurate in that linear focus receivers made using modern materials of construction are capable of similar thermal performance as point focus receivers at temperatures up to 700 °C for low thermal conductive fluids such as air, then there is an argument to be made for the application of contemporary linear PTCs for heating all manner HTFs up to 700 °C.

Since it has been shown that achieving HTF temperatures of 700 °C by means of a linear receiver may be possible, it is worth investigating the use of a sCO₂ Rankine Cycle over that of a modified turbocharger Brayton Cycle.

When using a turbocharger-based Brayton Cycle Heat Engine and given the condition of restricting combustion chamber inlet temperature to 650 °C to inhibit combustion emissions, operation with any form of solar preheating is not viable. This is due to the fact that temperatures from the turbine outlet far exceed 650 °C, and therefore if hydrocarbons are being consumed anyway, the entire solar preheating stage may be replaced by a single heat exchanging unit.

Using a heat-recycling unit in conjunction with a turbocharger-based engine typically doubles the engine's thermal efficiency. Such a turbocharger-based engine with heat recycling may be connected to an appropriately sized arbitrary receiver

without an additional combustion chamber, such that operation is analogous to PV electrical generation. In this case, it was found that the turbocharger-based engine could operate at 700 °C at a solar-electrical efficiency in the region of about 9 %; compared to typical PV operation of about 14 % (Goodrich, et al., 2012, p. 13).

5. Experimenting upon a Hybrid Concentrated Solar Power Air Brayton Cycle Engine with a Linear Receiver

This section details the procedure that was followed in an attempt to experiment upon a Garrett GT0632SZ miniature motorcycle turbocharger that was available to the author.

A set of small linear receivers were fabricated to be used together with the turbocharger to act as a preheating stage, and a combustion chamber was built for the purpose of burning propane from a portable tank.

The purpose of the experiment was in essence a proof of concept of a turbocharger-based engine. Explicitly, the objectives of the experiment were to:

- Demonstrate that electricity may indeed be extracted from a modified commercial vehicle turbocharger through operation as a gas turbine;
- Validate the use of a linear focus receiver to preheat air for a solar hybrid BCHE in place of a more expensive point focus receiver;
- To measure the real-world performance of such an engine.

During the initial stages of fabrication, it was presumed that the small motorcycle turbocharger would not produce a substantial amount of output power. It was assumed that the low operating pressure ratio and relatively low efficiencies of the turbine and compressor stages as per the manufacturer's curves would lead to a very low overall measured thermal efficiency for the engine. Sections 3 and 4 were still being developed, and a true estimate of just how low the efficiency could be expected to be was unknown.

The underlying purpose of the experiment was to simply see if the miniature turbocharger could be made to run in a hybrid fashion and generate a small amount of useful work in the form of electricity. This information may be used to make an assessment on the viability of the technology to domestic sized applications.

The experiment was plagued with complications due to the very small size and low efficiencies of the turbocharger. The majority of difficulties were experienced in extracting power from the diminutive setup. A number of unsuccessful approaches were taken to try to extract power from the turbocharger.

Eventually a method was found to successfully deliver power from the engine by means of extending the Centre House Rotating Assembly (CHRA) out from the compressor wheel. Magnets were attached to this shaft extension, and a stator housing was machined and mounted onto the compressor inducer. In this fashion, the compressor was adapted into an in-runner brushless DC motor/generator. Unfortunately, heat conduction through the CHRA shaft extension would heat the magnets beyond their Curie point and hamper power output a few minutes into turbine operation. Thus, the use of a GT0632SZ proved to be unsustainable as a means to generate output power.

The GT0632SZ based engine was successfully run and it produced some power. The measured power output and thermal efficiency of the engine was ultimately unsatisfactory. It is not recommended by the author to use this turbocharger for further experiments owing to the diminutive scale of the engine and the related engineering challenges involved.

The linear concentrator solar hybrid turbocharger-based engine was indeed shown to work, albeit barely. The experiment itself may be considered to be inconclusive.

5.1 Fabricating the Apparatus

During the period of time for the experiment from June 2016 to December 2016, the author was working part-time at Scipio Technologies in Boksburg. The use of the metal fabrication facilities was generously permitted to the author. This included a 5 m x 5 m laser cutter, general workshop tools, welding equipment (TIG, MIG, stick, arc and brazing) as well as free access to any workshop off-cuts destined for recycling.

While assistance was provided with the operation of the machinery and tools at both Scipio Technologies and the various Engineering Workshops at the University of Witwatersrand, all of the designs, plans and blueprints were created solely by the author.

The apparatus may be thought of as four distinct sections: the SEC(s), the combustion chamber, the turbocharger with attached generator, and the electronic

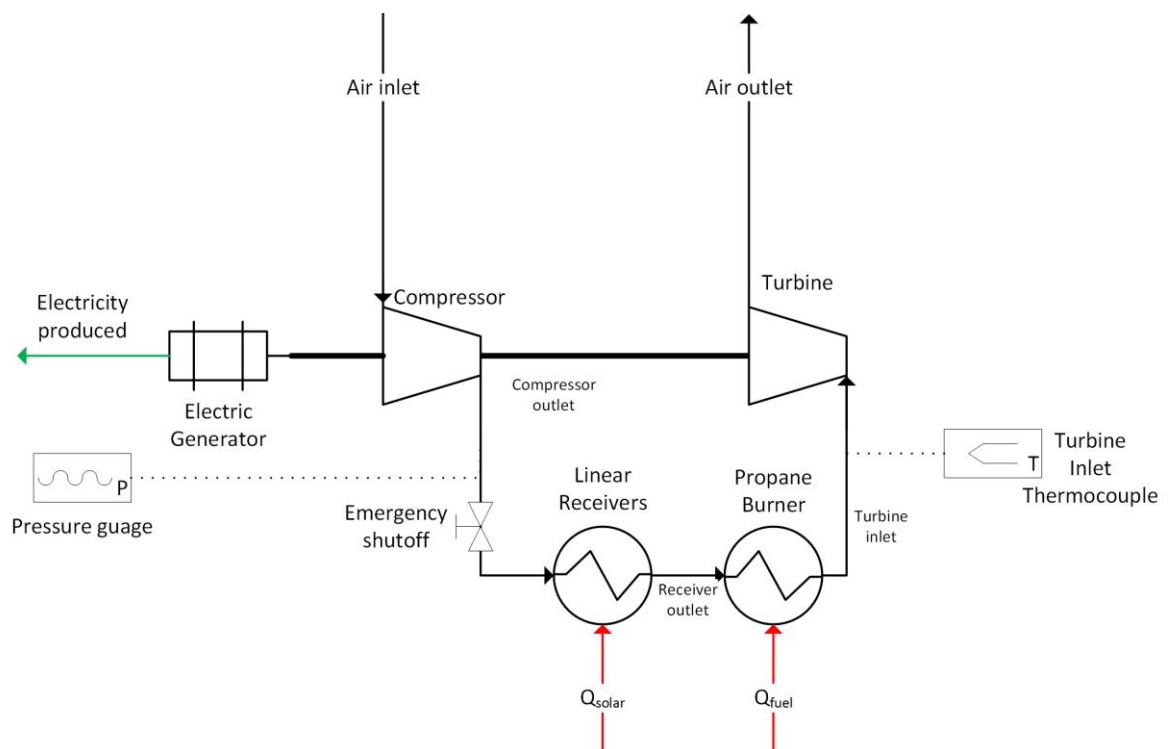


Figure 5.1 1: Simplified Apparatus Setup

control and monitoring system. Figure 5.1 1 outlines a simplified setup of the apparatus.

The intended operation of the apparatus is as follows:

- Fresh air is taken in and compressed by the compressor stage of the turbocharger and fed to the SEC array.
- Air is preheated by the SEC and fed into a combustion chamber where it is mixed with propane and burnt at constant pressure.
- The high temperature air is fed to the turbocharger's turbine and exhausted to the atmosphere.

Power is extracted from the engine by coupling an electric generator to the turbocharger's CHRA. Thermocouples were placed before and after the combustion chamber. Gauge pressure was measured immediately after the compressor and an emergency shutoff was located at an easily accessible location before the SEC.

The parabolic trough's solar tracking mechanism was operated electronically and the engine start-up procedure was automated. A low-cost high-power Raspberry Pi single board computer was used for these purposes. For safety it was decided to operate the propane flow rate manually instead of automating it by means of the Raspberry Pi. In this way an operator may quickly close both the propane and emergency shutoff valve from one location.

5.1.1 Design and Fabrication of the Linear Collectors

The first step in the fabrication of the apparatus was to design the parabolic collectors. As large an aperture as possible was targeted such that the greatest amount of heating may be obtained.

Two constraints existed for this. The first was the physical limitation of the size of the laser cutters' beds. The second was the available budget for the project. A simple-as-possible design for the parabolic trough was chosen to simplify both fabrication and operation.

Appendix D contains the derivation of the design and fabrication of the linear collector units. Each parabolic trough unit came to an effective width of approximately 2.8 m and a length of 2 m. Each parabolic trough therefore had an effective solar aperture of 5.6 m². Figure 5.1.1 1 shows one of the completed mirror brace assemblies.

The next task was to decide on the receiver tube itself. Ideally the receiver would consist of a cermet substrate applied to a stainless steel or copper pipe inside of a



Figure 5.1.1 1: A Completed Frame and Mirror Brace Assembly

vacuum glass cover. Fabricating such a receiver was beyond the scope of this project. Solar water heater vacuum tubes (commercially labelled “evacuated tubes” to avoid confusion with electronic vacuum tubes) were considered, however, the highest stable inner-surface temperature found to be available was about 120°C.

Since this was the first attempt at fabricating the apparatus, a standard domestic copper tube was used for the receiver. A 42 mm OD Class 0 (non-ductile) copper pipe was used without a cover and therefore subject to relatively high convective heat losses from the wind. Section 3.2 and Figure 3.2 4 detail the performance impact of such a setup.

The use of unshielded copper pipe as the receiver was essentially an effort to provide a basis for future iterative improvements on the apparatus. The receiver presents an area for substantial improvement in future experiments.

For this receiver outer diameter of 42 mm and reflector aperture width of approximately 2.8 m (depending on the acrylic mirrors’ edges), a final concentration ratio of approximately 42 was achieved.

A CuO film was grown on the surface of the copper pipe to promote solar selectivity of the receiver’s copper surface. An aqueous oxidant method was used to grow the CuO in-situ (Huang, et al., 2007):

The copper receiver was washed and polished with moderate grit sandpaper. Sodium hypochlorite and sodium hydroxide was slowly added to distilled water (250 mL batches $C_{NaClO} = 0.7 \left[\frac{mol}{L} \right]$, $C_{NaOH} = 2.1 \left[\frac{mol}{L} \right]$). The solution was mixed for 10 minutes and heated to 60 °C. It was then applied to the copper surface with a chemically resistant paint brush and left for 10 minutes before being rinsed with distilled water. Externally heated water was circulated through the receiver tube at approximately 60 °C during the curing process.

The resulting CuO layer appeared consistent visually. However, it did tarnish quickly. The black coating turned to a dull grey over a period of about 12 hours.

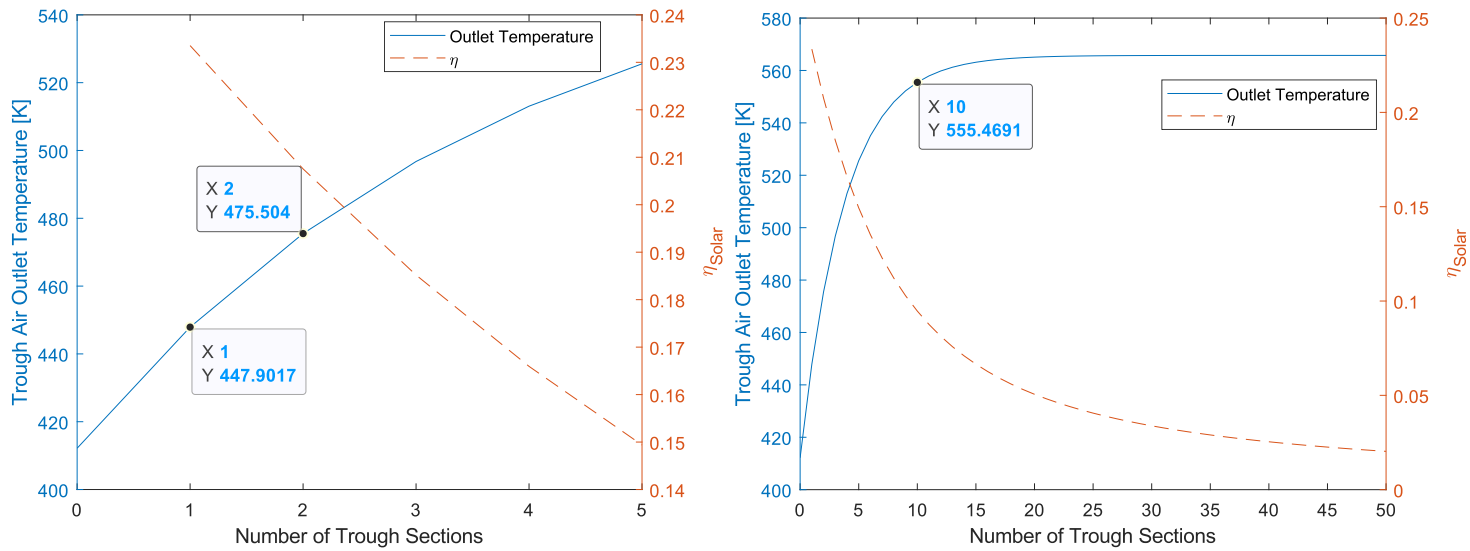


Figure 5.1.1 2: Number of Apparatus Trough Sections and their Expected Performance at 90% turbocharger Choke Point with 5m/s Wind

According to Huang et al. (2007), the CuO layer that formed should have an absorptivity and emissivity of approximately 95% and 45% respectively. This was not verified.

The old CuO layer was replaced for each run of the experiment. In the morning the mirrors were detached and the old layer of CuO was removed with sandpaper. A fresh application was applied using the method described above.

The numerically intensive linear receiver model was in the process of being derived and programmed concurrently with the design and fabrication of the linear collector sections. At the time when fabrication of the receiver sections had begun, it was unknown to what extent the penalty of not using a cover on the receiver would be with respect to convective heat losses. At the time it was hoped to at least provide somewhat substantial heating to the air stream, even if solar efficiency was relatively poor.

Approximately two months into fabrication of the receiver sections, one full receiver section was completed. A second section was 50% constructed. At this time the first version of the intensive linear receiver model was successfully implemented in code. The model showed that the apparatus's receivers coupled to a GT0632SZ were expected to perform relatively poorly.

With the expectation of targeting operating conditions for the GT0632SZ corresponding to 90% mass flow rate of the choke point (see Section 4.2 for this derivation), the outlet temperature of the compressor is approximately 413 K at a corrected mass flow rate of 5.8 lb/min. Figure 5.1.1 2 shows the expected performance of the apparatus's SEC field sections connected in series at these inlet conditions.

From Figure 5.1.1 2, using one trough section yields an expected solar efficiency of 23.5% while raising the air temperature from 413 K to 448 K. Additional trough sections produce diminishing returns in temperature gain and hamper cumulative efficiency. Two trough sections are expected to produce an SEC outlet temperature

of 475 K for a cumulative solar efficiency of 21%. The maximum temperature the troughs are capable of achieving with wind at 5 m/s is expected to be about 565 K.

Figure 5.1.1 2, for the operating conditions of the GT0632SZ this point of maximum air outlet temperature occurs at about 10 trough sections with little benefit derived for more trough sections. With 10 trough sections an outlet temperature of 555 K is achieved at a cumulative solar efficiency of 9.5%.

It was apparent that the trough sections designed for the apparatus would only heat the air by less than 30 K per section and far below the target combustion chamber inlet temperature of 923 K (650 °C – as detailed in the introduction of Section 4). Although this was disappointing, it was not unexpected for receivers not having covers.

Therefore, since by far the majority of the heat required to run the apparatus must come from combustion, only the trough section that was under construction at this point in time was completed. In total, two trough sections were made available to the apparatus to act as the air preheating stage.

5.1.2 Design of the Combustion Chamber

The combustion chamber used for the apparatus was constructed using offcuts of pipe and sheet metal from the workshop. The targeted dimensions of the flame tube followed the heuristics outlined in Section 2.8.

The GT0632SZ turbocharger has an inducer diameter of 22.63 mm for a cross-sectional area of 402.22 mm². From the heuristics, the target dimensions of the combustion chamber were to be: a length of 137 mm, a flame tube diameter of 68 mm and a combustion chamber diameter of 79 mm. Figures 2.8 1 and 2 depict the orientation and layout of the combustion chamber.

The combustion chamber housing was made from a piece of steel pipe with an OD of 89 mm and ID of 81 mm. The flame tube was made from a piece of steel pipe with an OD of 76 mm and an ID of 69 mm. The bulkhead and end ring were made from an offcut piece of 3 mm sheet metal.

The combustion chamber was mounted underneath the primary SEC. The flame tube pipe protruded from the combustion chamber about 300 mm and joined directly to the turbine inlet of the turbocharger. This was achieved by a coupling and a laser cut flange welded to the end of the flame tube with the appropriate holes to screw into the turbine inlet. Furnace caulking sealant was used to form a gasket between the flange and turbine inlet.

The holes for the flame tube were made using 2.5 mm, 5 mm and 8 mm drill bits for the primary, secondary and dilution bands respectively. The primary band was made from 24 holes drilled in two 12-hole bands with the centre of the two bands 28 mm from the bulkhead end. The secondary and dilution bands were made up from 4 holes each. The secondary band centre was located 49 mm from the

bulkhead end and the dilution band centre was located 69 mm from the bulkhead end. Figure 2.8 2 illustrates these bands.

During fabrication, the flame tube and end ring were mistakenly welded directly to the flange instead of to the bulkhead. Without a spacer pipe between the end of the flame tube and the flange, the flame tube length was insufficient to mate with the back of the combustion chamber section against the bulkhead. The remaining original length of pipe from which the flame tube was made was insufficient to make another flame tube.

Before making a new flame tube, the current configuration was tested to see if the flame tube may be made to work in this fashion before purchasing a new pipe for this purpose. To do so, a cap was welded onto the bulkhead end of the flame tube to perform the function of the bulkhead wall. The flame tube assembly was then inserted into the combustion chamber pipe and welded to the flange.

The holes for the propane inlet and spark plug were drilled through the combustion chamber wall and into the flame tube. A standard brass propane fitting was used as the injector. It was plugged at the end with fire sealant and a 2 mm hole drilled into its side (Figure 5.1.2 1). The injector was positioned approximately 15 mm from the flame tube bulkhead cap end (Figure 5.1.2 2). The spark plug was located between the secondary and dilution bands with the intent for it to perform its purpose without being located too close to the very high temperature primary zone.

The flame tube operated without issue in this configuration once the flame was lit. The process of lighting the flame using the spark plug required very fine adjustment of the fuel flow while the gas turbine was powered externally with a leaf blower (later the leaf blower was replaced by setting the ESC throttle to 100%) until combustion would begin. Since the combustion chamber worked, another one was not built to rectify this issue. It is unknown to what extent the lack of encouraged vortex flow within the annular region of the combustion chamber affected fuel combustion.

In spite of the difficulty of lighting the flame, it was assumed that most of the fuel was indeed burnt within the flame tube. For calculation purposes, it was assumed that all of the fuel was burnt prior to the turbine.



Figure 5.1.2 1: Propane Injector



Figure 5.1.2 2: Flame Tube Fuel Inlet and Spark Plug

5.1.3 Turbocharger modification and Power Extraction

Having constructed the SECs and working combustion chamber, the next step was to try and extract power from the very small turbocharger.

The initial intent was to operate the turbocharger as a turboshaft – that is to extend the shaft of the CHRA and link it mechanically directly to a generator. It became apparent that the very high rotational speed of the CHRA during normal operation would complicate this. Commercial gear train transmissions were first investigated. Unfortunately, the highest input shaft speed transmissions available were in the order of 25 000 - 30 000 RPM.

From Figure 4.1 1 it can be seen that the GT0632SZ has an operational range between 120 000 and 300 000 RPM. The BLDC motor available to be used as a generator was an Xnova 4025-1120KV 1.5Y. The motor had an operational voltage of 25 V, for a rated operating speed of 28 000 RPM at an efficiency of 94%. It was clear that a high gear ratio transmission system would be required to couple the turboshaft to a generator.

The first attempt at slowing the input speed to the generator was by means of a belt type transmission system.

Conveniently, the end of the compressor shaft was measured to have a diameter of 6 mm with an M4 0.5 mm fine pitch left-hand thread on the end of the shaft from manufacture. A silver steel 6 mm rod was cut and used to extend the shaft of the CHRA out from the compressor. An offcut of a solid 50 mm PVC cylinder was balanced and machined to fit on the shaft of the motor. V-shaped grooves measuring approximately 2.5 mm deep were machined into the silver steel shaft and the PVC cylinder. A 2mm nylon wire was then used to connect the two pieces. Figure 5.1.3 1 shows the fabricated belt drive gearing system that was first attempted.



Figure 5.1.3 1: Initial Belt Transmission for the Turboshaft

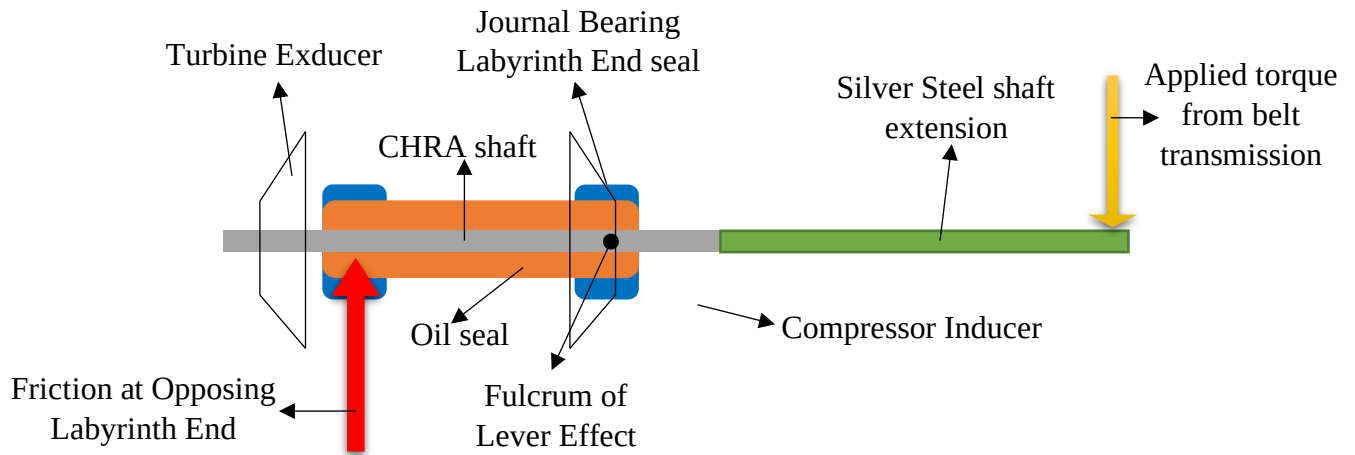


Figure 5.1.3 2: Friction in Journal Bearing from the Applied Torque of the Belt Transmission

In this configuration, the belt transmission had an effective gear ratio of approximately 8.3:1. Therefore, the motor would be able to extract electricity from the turbocharger within the motor's specification for a turbine speed of up to 233 000 RPM. By overdriving the motor 15%, the target turbine operating speed of 270 000 RPM – that is at 90% of the GT0632SZ's choke air flow rate – would be achieved.

In practice, however, the belt drive system proved unfeasible. To allow power transmission between the gas turbine and the generator, a relatively high tension was required in the nylon rope to prevent slippage of the belt. The shaft extension acted as a lever upon the CHRA. The torque applied to the CHRA from the tension in the shaft belt generated a substantial amount of friction within the CHRA's journal bearing structure. Figure 5.1.3 2 illustrates how this phenomenon was made manifest.

The gas turbine was unable to overcome the friction applied by the belt transmission upon the CHRA's journal bearing. Gradually increasing the tension in the belt while the turbine was running would eventually slow and stop the turbine before the generator would begin spinning.

The length of the shaft extension to the CHRA was already only enough to protrude 50 mm from the inducer's collar. It could not be shortened further to lessen the lever effect while still being practical to extract power from the turbine. At this point the use of a belt type transmission for this turbocharger was abandoned.

The purpose of this transmission so far was to mechanically couple the gas turbine to a generator. An alternate approach was taken at this point:

It was observed that the weight of the shaft extension would still permit the gas turbine to operate with no load. An idea was formulated to convert the shaft itself into the rotor for a generator built around the compressor's inducer collar. This would essentially permit the extraction of power from the turbine without a transmission. This generator would therefore in effect operate as an in-runner BLDC motor.

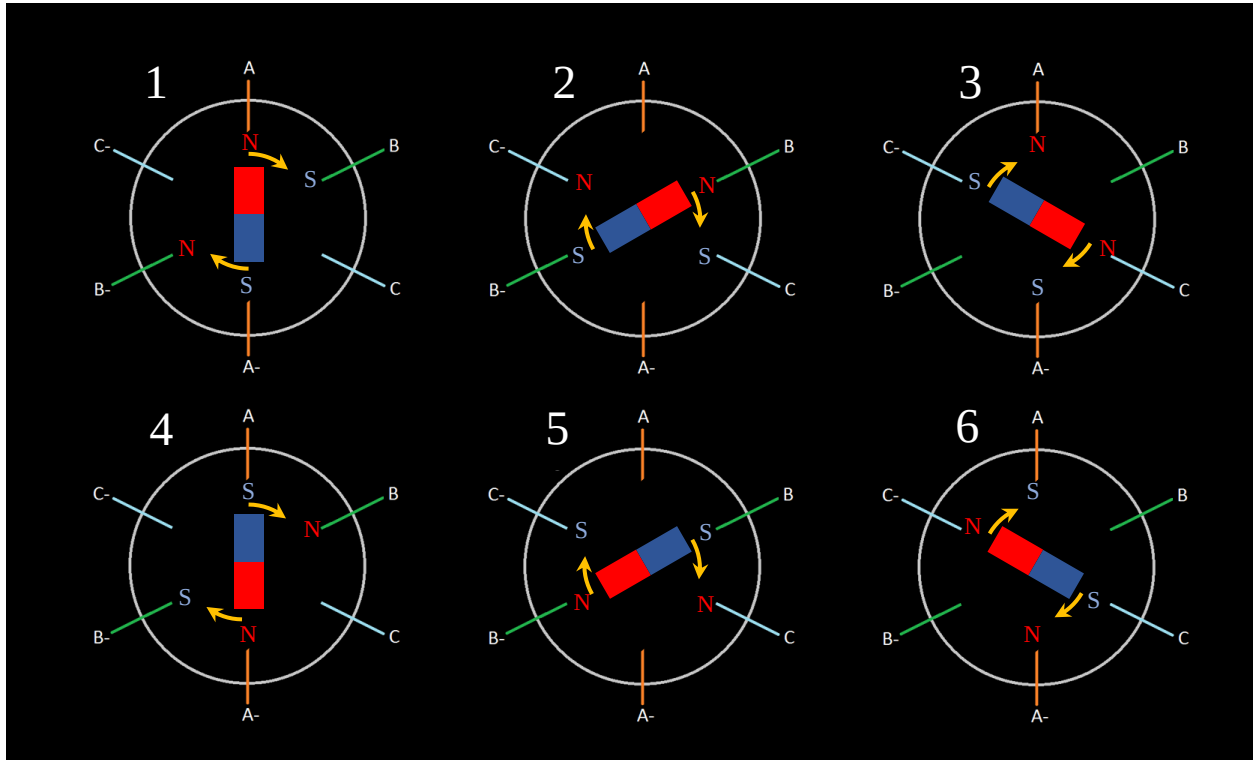


Figure 5.1.3 3: Motor/Generator Stator Coil Layout and Phase Order with Applied Torque Shown During Motor Operation

Extracting power in this fashion with a crude hand made in-runner BLDC motor would invariably be very inefficient: The large spacing between the inducer collar, stator coils and the magnets on the rotor would introduce significant losses. Hand wound coils without an appropriate winding form would further waste space. Coil holders without lamination sheets would also permit wasteful eddy currents and generate heat within the generator itself.

Purposely fabricating an efficient generator for this turbocharger was beyond the scope of this research. Nevertheless, a relatively straightforward modification to the gas turbine was made by converting the shaft into the rotor of an electric generator. This exercise was more of a proof-of-concept than a complete solution to the problem of extracting power from the very small turbocharger.

A crude motor/generator was fabricated to test this method of power extraction. The design was similar to that of a conventional in-runner BLDC motor. A commercial BLDC ESC was used for turbine start-up. Power output from the turbine was pure Three Phase AC. A set of relays was used to switch the motor/generator between power input and power extraction.

Power extraction was done by means of a Three Phase Rectifier connected to a solar MPPT controller. The MPPT controller output to four 7 Ah 12 V Lead Acid Batteries in series. The batteries were carefully individually charged/discharged to 12.5 V before each test.

It was necessary to keep the weight of the shaft extension as low as possible in order to limit the friction within the journal bearings of the turbocharger.

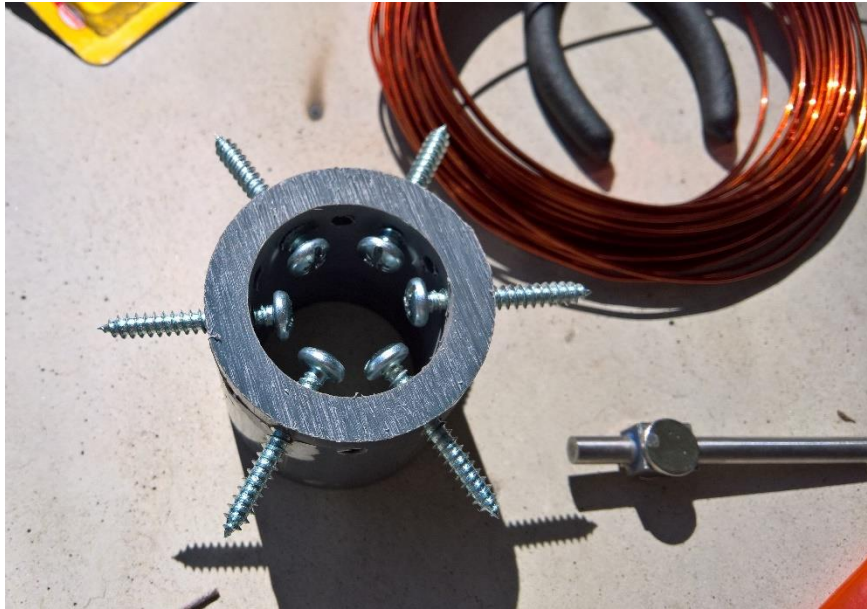


Figure 5.1.3 4: Fabrication of a crude BLDC Motor / 3 Phase Generator

It would have been ideal for a single diametrically magnetized ring magnet to be attached to the end of the shaft extension. Over 10 local magnet suppliers were contacted to enquire about a magnet of appropriate size. Unfortunately, the shortest waiting period to obtain supply was quoted as 4 months.

Two small neodymium N50 10x1 mm disc magnets were therefore attached on opposite sides to an 8 mm cube made from a piece of machined and balanced square bar. The balanced cube provided a surface upon which to adhere the magnets while keeping them parallel to one another. The magnets were oriented such that the visible face on one side of the cube was the opposite polarity of the face visible on the other side of the cube (i.e., to match the rotor polarity in Figure 5.1.3 3).

The stator was made from six coils, implying a high KV rating of the motor/generator for a relatively low output voltage from the generator. A high KV rating for the motor/generator is necessary because of the very high operating angular velocity of the turbine. Figure 5.1.3 4 depicts the coil arrangement of the motor/generator.

The choices of commercial solar and wind turbine MPPT equipment that was available had efficient input voltages between approximately 20 – 140 V.

With six coils, it would also be relatively simple to bypass three of the coils to double the KV rating (halving output voltage) of the generator if necessary.

The coils were arbitrarily labelled in a clockwise order as A, B, C, A-, B-, C-. A, B and C were wound clockwise with A-, B- and C- wound counter-clockwise with respect to the initial lead of the winding. Each coil was connected in series with the coil on the opposite side of the stator (i.e., the end lead from A was connected in series with the beginning lead of A-). Each coil was wound with 28 turns of 1.25 mm magnet wire. Figure 5.1.3 5 shows this stator layout during operation of the gas turbine.

Figure 5.1.3 3 shows the applied torque and field arrangement with the setup operating as a motor. An induced electric pole at a coil from the rotor's perspective will also induce the opposite pole at the opposite end of the stator. During generator operation, induced current would be in the same direction for both coils.

The final decision for the experimental motor/generator was to connect the windings in either a Wye or Delta configuration. A Delta connection would maximise the KV value of the motor/generator but it would also provide a secondary circuit for circulating current (Drury & Hughes, 2013). Drury and Hughes (2013) recommend the use of a Wye connection in order to limit this potential for harmonic responses from the motor/generator.

The leads from A, B- and C were connected together to form the Wye connection of the motor/generator. The leads from A-, B and C- were the final input/output leads of the experimental in-runner BLDC motor/generator.

With six coils in three phases and two poles for the rotor, each rotation of the rotor would produce a single positive and negative AC pulse to each phase. The angular velocity of the motor/generator (and the CHRA of the gas turbine) could be easily measured by measuring the frequency of the generated AC on any phase:

$$freq [Hz] = 60 \cdot RPM_{turbojet}$$

The motor/generator performed well as a motor in testing. The commercial ESC was able to spin-up the gas turbine CHRA from rest. Figure 5.1.3 5 shows the rotor spinning at an applied DC voltage of about 16 V. The efficiency during motor operation was not measured, but it was assumed to be very low. The screws supporting the coils were subject to significant eddy currents which generated substantial heat. The insulating tape between the screws and coils began to smoke and melt after about 30 seconds of motor operation during the first motor test run.

The KV rating of the motor/generator was measured by means of a drill press. The CHRA shaft extension was placed inside of a drill press chuck. The drill was set to



Figure 5.1.3 5: Gas Turbine Start-up using the Experimental In-runner BLDC Motor/Generator

Table 5.1.3 1: Measuring Motor/Generator KV Rating

Drill Press RPM	V _{RMS} Measured	KV	KV with k=0.95
460	0.111	2930	3080
1500	0.359	2950	3110
2580	0.63	2900	3000

a known RPM and turned on. The spinning rotor was then lowered into the stator housing on the drill press platform. The generated AC voltage between two leads was monitored while the height of the rotor was adjusted to find the position of maximum generated voltage.

The KV value for the motor/generator was calculated as follows (Drury & Hughes, 2013):

$$KV \left[\frac{RPM}{V} \right] = \frac{Drill\ Speed\ [RPM]}{\sqrt{2} \cdot V_{AC,RMS} \cdot k}$$

The purpose of the adjusting factor k is to account for voltage drop within the coils of the motor/generator during operation (Drury & Hughes, 2013). When operating as a motor, this voltage drop across the coils is additive to Back EMF. As a generator, this voltage is dropped across the coils by induced current.

During KV measurement however, no current is flowing through the coils. Therefore, the output voltage during generator operation will be lower than the no-load KV measurement case. Commercial BLDC motors drop approximately 5% applied voltage across the coils themselves (Drury & Hughes, 2013).

Table 5.1.3 1 shows the measured KV of the motor/generator that was attached to the turbocharger's compressor collar.

Virtually all of the wind turbine MPPT charge controllers that were available had a turbine over-speed protection function. When contacting suppliers, it was found that no wind turbine charge controllers were available without an overspeed function of about 500 Hz. Since the output of AC from the motor/generator was expected to be in excess of 4300 Hz (260 000 RPM), the three phase AC needed to be rectified to DC and smoothed with a capacitor before being fed into a DC solar MPPT charge controller.

A 30A 48V solar MPPT controller manufactured by Y-Solar was used to extract power from the gas turbine. This MPPT controller was chosen for its wide input voltage range of 50-190 VDC and claimed 98% efficiency with an input voltage 1.5 to 2 times that of the battery bank. For the motor/generator with a KV value of about 3050 operating at 270 000 RPM, the peak output DC voltage after rectification would be approximately 88 V. This was within the MPPT's efficient operating range of 72 to 96 V when charging a 48 V battery bank.

In theory this setup would make it possible to extract some power from the gas turbine. However, the efficiency of this method of power extraction was expected to be very low due to the nature of the hand-made motor/generator.



Figure 5.1.4 1: Solar Panel Shading Mechanism used by the Heliostat

5.1.4 Electronics, Control and Automation

Two tasks were identified as necessary to be automated. This first task was the heliostat function of the parabolic troughs. The second task was to perform a safe start-up of the gas turbine.

Automation of these tasks was done by means of a programmable Single Board Computer. A Raspberry Pi 3 B was used for its abundance of GPIO pins as well as its compatibility with SPI and I²C protocols. The single Pi could therefore communicate with several microcontrollers while operating a relatively large number of relays simultaneously.

The heliostat function was performed by means of a pair of solar panels to monitor the position of the sun relative to the collector's aperture. The output voltages of

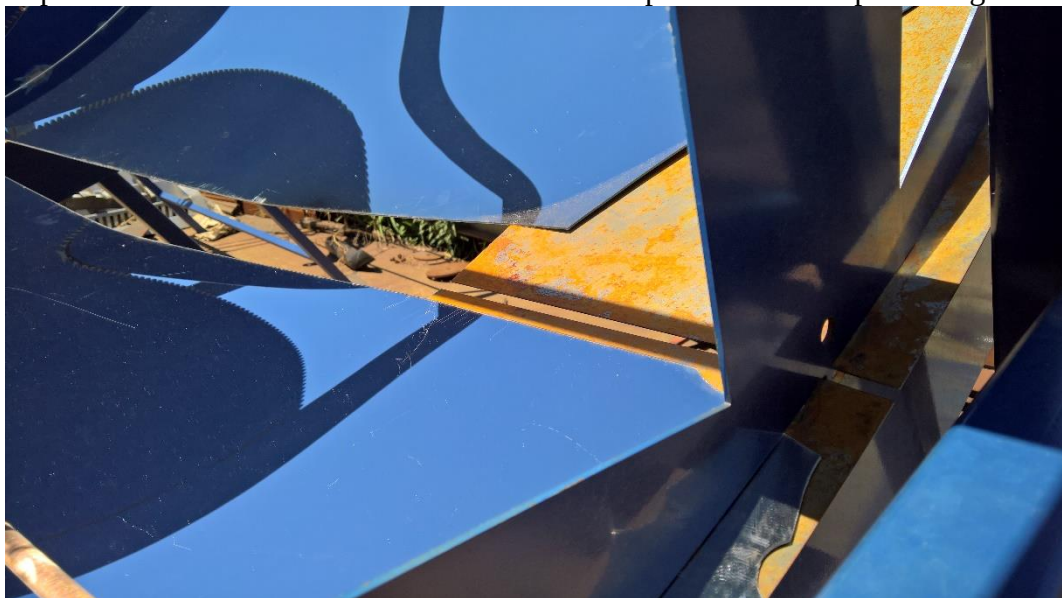


Figure 5.1.4 2: Receiver Shadow Cast upon Concentrator Mirror Supports Indicating Accurate Solar Tracking

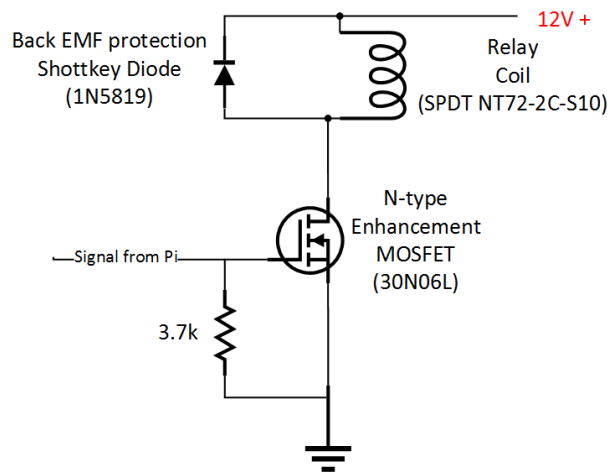


Figure 5.1.4 5: MOSFET Based Relay Actuation Circuit

The altitude of the collector's aperture was controlled by a small latching DC winch as detailed Appendix D. The direction of the winch was controlled by the direction of current through the winch using the circuit described in Figure 5.1.4 4.

The relays connected to the Pi were each actuated by the MOSFET based circuit described in Figure 5.1.4 5. The circuit includes a pull-down resistor on the gate of the MOSFET to prevent sending a false signal to the winch when leaving the connection to one of the Pi's GPIO pins floating. To further protect the Pi and MOSFET, a diode was placed across the relay's coil to provide a path for Back EMF to circulate and dissipate as the coil is unpowered.

All of the code used for the heliostat function is available Appendix G. Greater detail of the design of the control circuit is described in Appendix D.

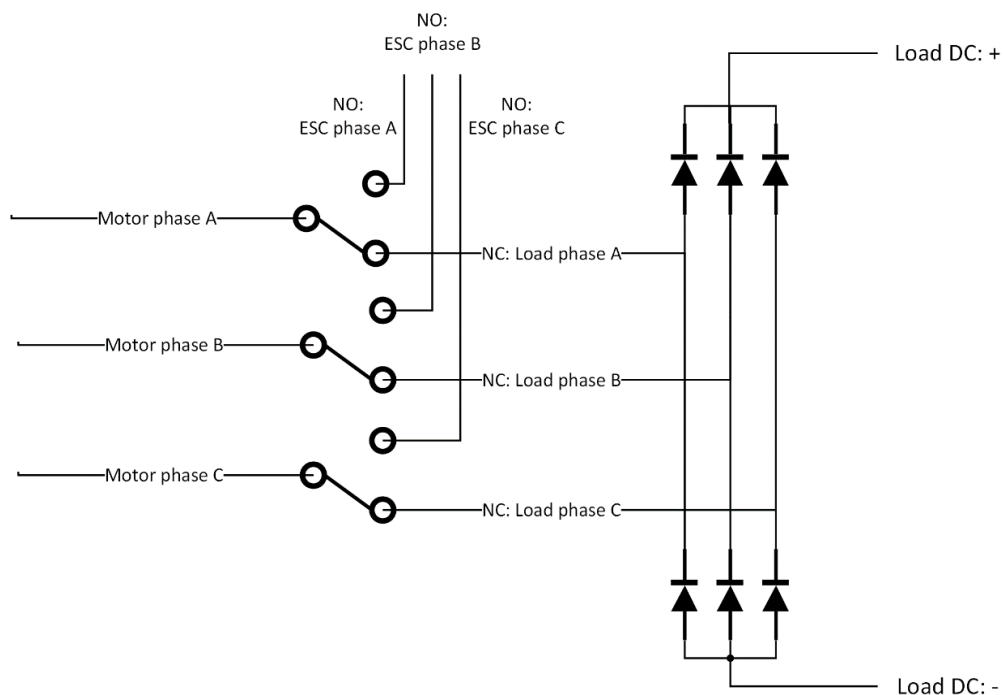


Figure 5.1.4 6: Motor/Generator ESC Input and Load Output Selection Circuit

The start-up function of the gas turbine was managed by the Pi. In the normal state of the apparatus, the leads from the motor/generator are connected to the solar MPPT controller. In the start-up state of the apparatus, the leads of the motor/generator were connected to a BLDC ESC (Maytech 40A-BEC-E). This was achieved by the circuit in Figure 5.1.4 6.

5.2 Experimental Setup and Performance

The apparatus was now capable of operating the gas turbine and extracting some power with the the handmade motor/generator. At this stage in testing the apparatus, a number of issues were acknowledged:

Firstly, the receiver of the apparatus lacked a vacuum cover. The receiver would therefore be subject to substantial convective losses. It was expected that each trough section would only raise the temperature of the air passing through it by approximately 35 K (Figure 5.1.1 2). This is certainly unfeasible for heating air to generate useful work. However, the bare copper receiver tube would be useful in practise to test the operation of the rest of the apparatus.

The second problem with the apparatus is the very low expected efficiency of the hand-made motor/generator. The purpose of the in-runner style motor/generator was to operate as a proof of concept at extracting some power from the gas turbine. The very high operating angular velocity of the CHRA and physical size of the turbocharger limited the ability to modify the turbocharger to operate as a turboshaft.

The two trough sections of the apparatus were stored overnight within the warehouse. A forklift was used to move the trough sections outside. The trough sections were plumbed together using a short length of flexible temperature resistant hose. The hose (Primaflex M9 51 mm) was rated for 3 bar_g and 300 °C. Teflon tape was wrapped around the outer diameter of the copper tubing of the receivers to match the ID of the hose and ensure a tight fit.

The operating pressure of the apparatus was measured shortly after the compressor outlet. Compressor outlet temperature was measured with a handheld infrared thermometer on a section of the tube spray-painted black. Two K-Type thermocouples were placed before and after the flame tube. Turbine angular velocity was measured with a multimeter (a Fluke 115) set to frequency mode across one of the output phases.

The intended experimental procedure was as follows:

- Prior to start-up of the gas turbine:
 - Charge/discharge all batteries to their intended voltages.
 - Initiate the solar tracking python script on the Pi and visually confirm the shadow of the absorber lines up with the centre gap of the mirrors. Stop the script and return the troughs to resting position.

- (This was necessary to avoid feeding air at too high a temperature to the coupling tube between the two trough sections)
- Preheat the oil that will be gravity fed to the turbocharger (Sasol 5W-40 was used, therefore oil was preheated to 115 °C).
 - Initiate the start-up python script to excite the relays and complete the circuit between the motor/generator and the BLDC ESC.
 - To start the gas turbine:
 - Initiate the solar tracking python script on the Pi and visually confirm the shadow of the absorber lines up with the centre gap of the mirrors.
 - Engage the sparker connected to the spark plug.
 - Engage the ESC at 100 % throttle.
 - Slowly open the valve controlling propane flow until audio confirmation that flame tube is lit (it produced a distinctive howling sound).
 - Disconnect the sparker.
 - End the start-up python script to return the ESC first to idle, and then disconnect the ESC from the motor/generator and reconnect the motor/generator to the MPPT controller.
 - Adjust the propane flow such that turbine operation is stable (i.e., the audible tone produced by the turbocharger does not fluctuate and AC output frequency is steady).

The first trial run without a load was done on the 12th September 2016 at approximately 12 PM with both trough sections.

The gas turbine was lit successfully and the fuel flow rate of the propane was adjusted to an AC output frequency of approximately 3 000 Hz. This corresponds to a CHRA angular velocity of 180 000 RPM. At this point the compressor outlet pressure was measured as 1.38 bar. According to the Compressor Map for the GT0632SZ (Appendix B), the corrected mass flow rate of the turbocharger was approximately 4.2 lb/min.

The apparatus was run at this constant fuel flow rate for approximately five minutes to allow the system to reach steady-state. The outlet temperature from the compressor was measured as 74.4 °C. Flame tube inlet temperature was measured at 161.3 °C. Flame tube outlet temperature fluctuated about 750 °C.

The next measurement to be taken was the RMS AC voltage from one of the phases from the motor/generator. The expected reading was to be approximately 42 VAC. The measured reading started at about 5 VAC and was slowly dropping while it was being measured.

The experiment was stopped in order to determine the cause of the very low output voltage. After some investigation it was found that the two magnets on the rotor had lost the majority of their magnetism.

The magnets were replaced for a second test to be run. The second experiment was performed on the following day at midday following the same procedure. Only one trough was used for this test to reduce the setup time necessary to perform the experiment. The new KV rating of the motor/generator was not remeasured prior to this experiment.

The primary purpose of this test was to determine if the magnets would fail again, and if so to determine the cause of this failure. A secondary objective was to try and extract some power from the gas turbine engine.

The engine was started and set to an AC output frequency of approximately 3000 Hz. At this point the MPPT controller responded and began applying a varying load to the gas turbine engine. After about 10 seconds an MPPT point was found. The engine was left to run at these conditions for 2 minutes to reach steady state. The results were then recorded.

Fuel flow rate was increased in two stages to obtain output frequencies of 4000 and 4500 Hz. Each stage was left for 2 minutes to reach steady state before results were recorded. 4500 Hz was arbitrarily chosen as the maximum target for the gas turbine to prevent accidentally over speeding the CHRA during the initial test session. The intention was for future tests to target frequencies corresponding to 90% choke flow rate of the compressor to maximise engine efficiency and output, as detailed in Section 4.2.

When the test had completed, the gas turbine was shut down. When the rotor of the motor/generator had stopped turning, the author touched the rotor while checking the magnets and received a superficial burn to the fingers. It was initially a surprise that the rotor had reached a temperature high enough to burn skin on the cold compressor side.

Upon investigation of the manufacturer's specification sheet of the N10 magnets used in the apparatus, it was found that the Curie point for the magnets was rated at

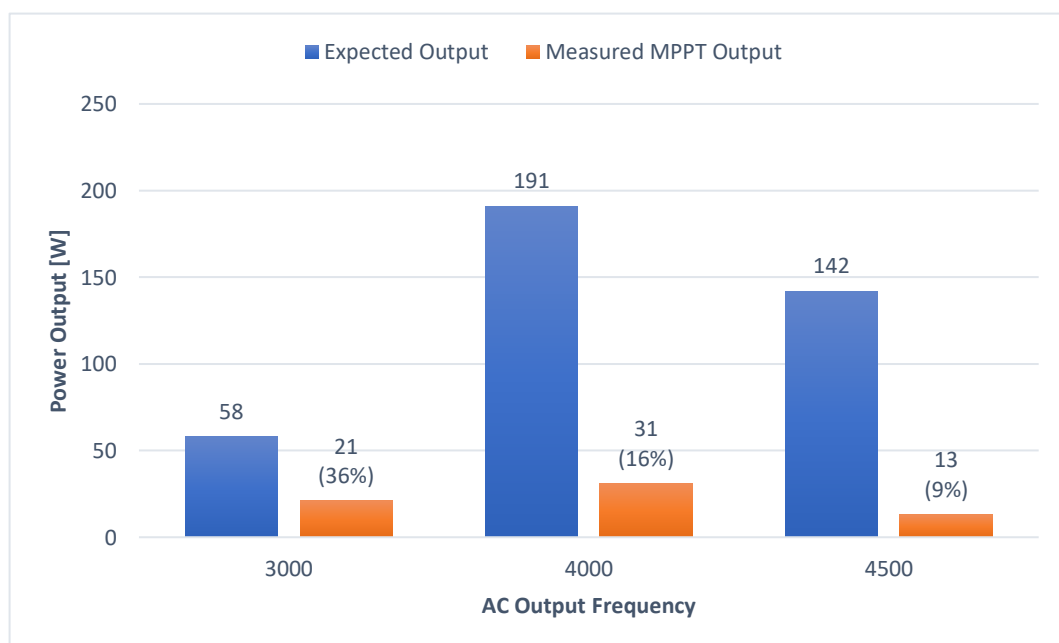


Figure 5.2 1: Experiment Power Output Results

80 °C. Heat was being conducted through the CHRA from the turbine side of the turbocharger to the compressor inducer. The silver steel shaft extension from the compressor impeller readily conducted heat toward the magnets, heating them past their Curie point. This explains the loss in magnetism during the experiments.

Figure 5.2 1 shows the measured power output of the gas turbine operating at increasing turbine speeds. The expected power output for the gas turbine was calculated according to the measured operating pressure ratio, estimated mass flow rate from the compressor map, and measured turbine inlet temperature.

The power output realized from the engine was extremely poor. This was somewhat expected with the handmade motor/generator. The fraction of extracted power decreased between each point of measurement. At the beginning of the test, 36% of the estimated available power was successfully extracted. By the end of the test, only 9% of the estimated available power was successfully extracted.

This gradual decrease in electrical output efficiency may be as a result of gradual heating of the magnets. The gradual loss in magnetism of the rotor would result in a weaker magnetic field strength of the rotor, thereby decreasing the efficiency of the handmade motor/generator.

It must be noted that only the output wattage from the MPPT controller which was visible on its screen was recorded. The input voltage to the MPPT controller was not recorded. The KV rating of the new magnets on the rotor was also not remeasured before the test. The KV rating of the motor/generator and how it shifted during the experiment is unknown.

Figure 5.2 2 outlines the measured performance of the copper tube receiver. As the inlet temperature to the receiver tube increased, the heat collection efficiency of the receiver decreased. The copper pipe receiver performed poorly, with estimated solar thermal collection efficiencies between 18 and 8%. However, Figure 5.1.1 2 predicts the solar thermal collection efficiency of the receiver section of a single trough collector to be in the order of 24%.

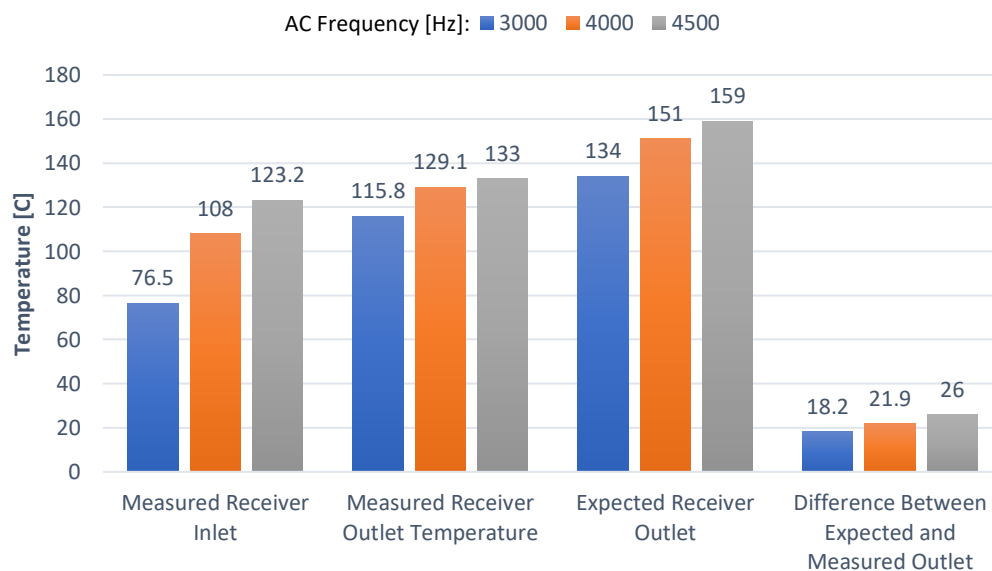


Figure 5.2 2: Experimental Receiver Performance at Various AC Output Frequencies

Table 5.2 1: Summary of Experimental Results of the 2nd Run on September 13th 2016

AC Frequency Target [Hz]	3000	4000	4500
Receiver Inlet Temperature [°C]	76.5	108	123.2
Flame Tube Inlet / Receiver Outlet Temperature [°C]	115.8	129.1	133
Flame Tube Outlet Temperature [°C]	765	742	792
Measured MPPT Output Power [W]	21	31	13
Estimated Flowrate from Compressor Map [corrected lb/min]	4.2	5.8	7
Expected Receiver Outlet Temperature [°C]	134	151	159
Δ Expected to Measured Receiver Outlet Temperature [°C]	18.2	21.9	26
Estimated $\eta_{collector}$	0.183	0.136	0.0825
Estimated net $Q_{HTF,absorbed}$ [W]	1026	762	462
Expected Output Power [W]	58	191	142
Estimated Flame Tube Heating Duty [W]	18 018	23 483	30 591
Estimated Thermal Efficiency	0.12%	0.13%	0.04%

The increasing difference in expected to measured receiver outlet temperature may indicate that substantial heat losses were experienced along the piping both to and from the receiver and turbocharger, where this issue was exacerbated with increasing flowrate. The flame tube was covered with a fibreglass turbocharger insulation wrap. The rest of the apparatus did not have any additional insulation.

In retrospect, a significant surface area was left uninsulated; the entire length of pipe from the compressor outlet to the flame tube inlet – including the receiver itself – was uninsulated. The receiver outlet temperature was measured at the flame tube inlet, approximately 2 m of pipe away from the actual receiver outlet. This may account for the increasing difference between expected and measured receiver outlet temperature. As the AC frequency increased, so too did the flow rate of air through the piping, and the more turbulent mass flow would achieve better heat conduction through the uninsulated piping.

The actual points of temperature measurement in the apparatus were: immediately following the compressor outlet; the flame tube inlet; and the flame tube outlet. Heat transfer calculations assumed that the temperature of the receiver outlet was the same as the compressor outlet, and that receiver outlet temperature was the same as flame tube inlet temperature.

The substantial surface area of uninsulated piping available for heat losses in the apparatus means the calculated values for $\eta_{collector}$ in the results Table 5.2 1 reflect



Figure 5.2 3: QR Code Link to YouTube Playlist of Apparatus during Operation

the efficiency of the collector as a whole and underestimates the true values of $\eta_{receiver}$.

Two short videos were recorded during the test and were uploaded to YouTube. The first video shows the exhaust from the turbine exducer with a visible flame front. This suggests that the flame tube may not have a long enough depletion zone. The second video shows the rotor spinning under self-power. A link to the videos is provided by the QR code in Figure 5.2 3.

5.3 Discussion

This section details the procedure in the development of an initial proof of concept solar hybrid gas turbine engine for electricity production, targeting chiefly domestic scale applications.

The experiment was inconclusive in as far deciding on the viability of using a linear focus collector as a high temperature heat source to be used in conjunction with a modified turbocharger gas turbine. Further research using a more efficient turbocharger and collector would help to gather data to come to a conclusion on the viability of this approach.

When operating in a hybrid fashion where CSP was used to preheat an air stream prior to combustion, the apparatus was indeed shown to produce electricity. However, thermal efficiencies were calculated to be below 0.1%.

The gas turbine was made from a modified motorcycle turbocharger while the accompanying parabolic trough linear collectors were fabricated for this specifically for this apparatus.

The three primary objectives of the experiment were: to demonstrate electricity extraction from a modified commercial vehicle turbocharger; to demonstrate a linear receiver based air preheating stage for a Brayton Cycle Heat Engine; and to measure real world performance and efficiency metrics for the receiver and engine.

All three objectives were successfully met. However, the poor performance of the apparatus was disappointing relative to the effort expended for its fabrication. This was primarily the result of the combination of a very small and inherently inefficient turbocharger together operating the receiver without a cover to mitigate convection losses.

The apparatus consisted of two parabolic trough sections, each with an aperture of 5.6 m² and an effective concentration ratio of 42. Only one receiver section was used during the experiment which produced measurable results.

The turbocharger used in the apparatus was a Garrett GT0632SZ. A propane fuelled flame tube was built to function as the gas turbine's combustion chamber. The flame tube was sized according to the dimensions compressor's inducer.

The very high operating rotational velocity and diminutive size of the turbocharger's CHRA complicated the modification procedure necessary to extract

power from the gas turbine. The solution to this was to mount magnets onto an extension of the shaft on the compressor side of the turbocharger. Hand wound coils were positioned around the magnets. The shaft extension was essentially converted into a BLDC in-runner motor/generator.

It was possible to start the gas turbine by connecting the handmade motor/generator to an ESC to rotate the CHRA and subsequently light the combustion chamber. Power was extracted by connecting the three-phase output of the motor/generator to a rectifier with a solar MPPT charge controller as the load. In theory, the MPPT function of the charge controller would maximise the power output of the gas turbine for the given operating conditions of the entire apparatus.

It became evident that custom built receiver sections would need to be fabricated to handle the relatively high expected inlet, operating, and outlet temperatures of the receiver. Commercially available shielded receivers were rated for temperatures up to about 130 °C and designed primarily for heating water. Fabricating a receiver section with a solar selective high temperature stable cermet coating and including a vacuum sealed cover was beyond the scope of this project.

As an exercise in a proof of concept, a domestic copper pipe was used as the receiver. The surface of the pipe was converted to CuO to increase the absorptivity of the copper. This was done by surface sanding and application of a sodium hydroxide / sodium hypochlorite solution.

The unshielded copper pipe receiver was simulated using the numerically intensive linear receiver model from Section 3 and was expected to obtain a solar thermal heat transfer efficiency of about 24% at standard operating conditions for the gas turbine. During testing, the entire collector section was estimated to have obtained a solar thermal heat transfer efficiency of between 18 and 8%. This may be explained by the lack of insulation of connecting pipes before and after the receiver itself within the collector.

During the experiment the single collector was able to preheat the air stream at a duty of between 0.462 and 1.02 kW. The combustor operated at a duty of between 18.0 and 30.6 kW. Though relatively low duty and at a poor efficiency, the linear receiver preheating section did function as a preheating stage for the gas turbine.

The unshielded bare copper receiver achieved a temperature rise of 10 to 39 °C when using one of the fabricated collectors with the turbocharger. This collector produced a maximum outlet temperature of 133 °C.

Power output from the engine was extremely poor. Reported electrical output power from the MPPT controller ranged between 13 and 31 W over the course of the experiment. The handmade motor/generator is estimated to have operated at an efficiency of between 36 and 9%. Overall total thermal efficiency conversion of net heat to electrical work was calculated to be between 0.12 and 0.042%.

The use of the handmade motor/generator coupled to the turbocharger-based gas turbine was a successful method of power extraction, but not an effective one. The large air gap between the rotor and stator coils together with the non-laminated steel forms for the coil centres implies a low expected efficiency of the motor/generator.

The magnets on the rotor were susceptible to conducting heat through the shaft extension of the turbocharger itself.

During multiple experiments, the magnets would conduct heat in this manner and eventually reach a temperature beyond their Curie point. They would gradually lose their magnetism during the test and would be completely defunct within about 10 minutes of gas turbine operation. This may explain the drop in the measured motor/generator efficiency as time progressed during the test.

In summary, the small Garrett GT0632SZ motorcycle turbocharger is not suited to this application. The very high angular velocity of its CHRA during operation make it impractical to couple mechanically to an external generator. Its small size means that tolerances required for modifications are very small. The compressor and turbine operate at relatively low efficiencies compared to larger turbochargers. The turbine requires a relatively high inlet temperature before it is capable of powering the compressor itself.

The use of a larger turbocharger in future experiments would likely solve these issues. However, a larger turbocharger would require a much greater heating duty in order to be operational.

Fabrication and heliostatic operation of the collector unit was relatively successful. The receiver section of the collector represents an area for substantial improvement in future experiments. Ideally, future tests of the collector would be done with a receiver consisting of a high temperature solar selective surface together with a vacuum cover to mitigate heat losses. From Section 4 it has been shown that with the use of contemporary materials in receiver construction, HTF outlet temperatures of 650 – 700 °C should be possible with a large enough collector aperture.

An important modification that should be made to the apparatus for future experiments is the insulation of all piping to and from the collector, and not just insulation of the combustion chamber in its current state.

The measured efficiency of the turbocharger based solar hybrid gas turbine Brayton Cycle Heat Engine was very low – well below 1% during operation. While the apparatus used materials that were far from ideal, even a 10-fold increase in operating efficiency is not impressive compared to conventional CSP or PV technologies. There is also an inherent limitation to the maximum practical combustion chamber inlet temperature of 650 °C to meet emissions standards.

From Section 4.2, it was shown that turbocharger turbine outlet temperatures can readily exceed 650 °C. This essentially nullifies the use of a CSP field as a preheating system for a turbocharger gas turbine as the target combustion chamber inlet temperature may be obtained simply through heat transfer with the turbine outlet stream.

This is true for any type of CSP based preheating technology coupled together with modified turbocharger-based gas turbine engines, not just linear receivers.

With all of these factors considered, it may be prudent to conclude that the use of modified vehicle turbochargers as gas turbine Brayton Cycle Heat Engines in

conjunction with linear receivers in a hybrid configuration for preheating is feasible, but not viable.

Turbocharger-based gas turbines are better suited to be powered either exclusively through combustion or solely through CSP. Operation with a heat recycling unit is vitally important in either case.

Each trough cost approximately R9'280 to fabricate for an aperture of 5.6 m² for a total price of R1'657 per kW of solar energy. If electric power could be extracted at an efficiency 9% as per Table 4.3 1, then electricity could be produced at a cost of R18'409 per kW_e.

ARTsolar is a wholesale solar panel manufacturer in Durban. A 300W 60 cell monocrystalline pergium solar module sells for R1638.75 incl. (ARTsolar, 2019). Producing solar power from these modules therefore costs R5'462.5 per kW_e.

Operating a modified turbocharger BCHE in a pure solar fashion is analogous to using PV panels to generate electricity.

Producing electricity with modified vehicle turbochargers and troughs such as those used in the apparatus is therefore 3 to 4 times more expensive than using PV panels without taking into account the capital cost of the turbocharger or running costs such as oil consumption and maintenance.

This does not take into account the advantages of CSP heat engine operation such as: low-cost heat storage, dispatchability, hybrid operation during inclement weather, expedient ramp up, or the utility of turned down operation.

The cost per kW_e of electricity may be high compared to PVs, however, collection of heat energy is separate to electricity production. This utility may be an important factor when deciding between the two technologies, and would be highly depended on the particular use case.

High temperature linear focus receivers may be better matched with high temperature Rankine Cycles, such as sCO₂. These heat engine cycles may also use hydrocarbon combustion as a backup heat source for hybrid operation during night or inclement weather.

5.4 Conclusion

The small Garrett GT0632SZ motorcycle turbocharger used in this experiment was not suited to this application. The very high angular velocity of its CHRA during operation make it impractical to couple mechanically to an external generator. Its diminutive size means that tolerances required for modifications are very tight. The compressor and turbine operate at relatively low efficiencies compared to larger turbochargers. The turbine requires a relatively high inlet temperature before it is capable of powering the compressor itself.

Fabrication and heliostatic operation of the collector unit itself was relatively successful.

Ultimately, the results of this experiment were inconclusive in providing insight into the viability of operating a modified commercial vehicle turbocharger as a pure solar or solar-hybrid Brayton Cycle Heat Engine.

A recommendation for future experiment using the apparatus would be to use a larger turbocharger. Additionally, it is also important to insulate all of piping to and from the collector, and not just insulation of the combustion chamber in its current state.

6. Conclusion

The first aim of this research was the development of a robust numerical model to simulate heat transfer dynamics within a linear focus receiver; particularly at operating temperatures much greater than that of conventional implementations of Parabolic Trough Collectors. The derived numerically intensive linear receiver model includes the ability to handle virtually any liquid or gas phase HTF, conventional and contemporary materials of construction for a wide range of receiver dimensions at a variety of atmospheric conditions.

The second aim of this research was to use this intensive linear receiver model to simulate the operation of a heat engine attached to a high temperature linear focus receiver of arbitrary dimensions. The model was used to parametrically analyse and optimise a setup with a Carnot Engine to determine the upper bound of performance of such an engine powered by this arbitrary receiver. Other more realistic heat engines such as Brayton Cycle Engines were studied too.

This acts as a motivation for intentionally using linear focus receivers at moderate to high temperatures through the aid of contemporary materials of construction. This challenges the heuristic that linear receivers are only suitable for relatively low temperature (and implied low thermal efficiency) operation.

The final aim of this research was to determine the feasibility and viability of using modified vehicle turbochargers as Brayton Cycle Engines in conjunction with high temperature linear receivers.

In Section 4.4.3 it was shown to be feasible to operate a turbocharger based engine purely from solar energy at approximately the same solar efficiency as a PV setup, however, the multitude of moving parts, unitary scalability in approximately 65 kW_e sections, required regular maintenance and oil consumption harm the viability of the technology compared to PV alternatives - especially at communal domestic and small commercial scales.

Turbocharger based engines operating as gas turbines in a hybrid fashion with CSP preheating were also investigated. There exists the limitation of a combustion chamber inlet temperature of 650 °C to control emissions from the engine. Preheating air beyond this point somewhat nullifies a renewables approach of power production. By burning hydrocarbons, even the most efficient turbocharger turbines produce outlet temperatures well above 700 °C. In this case, the SEC field may be replaced by a heat exchanger between the turbine outlet and the combustion chamber inlet.

As a result of this, it is unviable to use CSP of any technology to act as a preheating stage for turbocharger based gas turbines that burn hydrocarbons. Turbine outlet temperatures vastly exceed the combustion chamber inlet temperature limit of 650 °C during standard gas turbine operation. A heat exchanger would perform the same task as the solar field to preheat air while being more reliable and far less expensive to fabricate and operate.

A proof of concept in operating a turbocharger as a solar hybrid gas turbine engine was performed by the construction of an experimental apparatus. While the results of the experiment were ultimately disappointing, the apparatus itself provides a platform for future research.

The experiment itself was inconclusive in determining the viability of using a linear focus receiver as a high temperature heat source in conjunction with a modified vehicle turbocharger gas turbine.

6.1 Significant Findings

The points which follow highlight the more significant findings obtained during the development of this research.

- When designing the dimensions of a linear focus receiver, it was found that it is not always the case to simply target as high a concentration ratio as possible. This is a non-intuitive result since a lower CR implies a greater relative surface area is available for losses. At too great a CR, the efficiency of heat transfer to the HTF in-fact decreases.
- The primary heat transport mechanism of losses away from the receiver's absorber is radiative transfer between the absorber surface and the inside of the receiver's cover. Therefore, the emissivity of the absorber surface is the driving factor of collector losses at practically all receiver temperatures. A slight decrease in absorber surface emissivity results in a large increase to collector efficiency, especially at moderate to high temperatures
- There exists an ideal CR to AR ratio for a given HTF flowrate, receiver materials and receiver dimensions which maximise the efficiency of heat transfer from the collector to the HTF. For a fixed CR (as is the case in purchasing commercial linear focus receiver sections) there exists an ideal ratio of HTF flow rate to AR (that is the number of receiver sections).
- When using modified turbochargers as BCHEs, it was found that it was beneficial to operate the turbocharger at as high a pressure ratio as possible. The thermodynamic gain in operating at a higher pressure ratio was greater than the decrease in compressor efficiencies when under these conditions.
- It was shown through the intensive linear receiver model that achieving a collector outlet temperature of 650 °C should be readily possible with the use of modern materials of construction for the receiver. An arbitrary linear focus receiver was calculated to operate in the region of 65% cumulative efficiency at this temperature. This implies that there is a potential economic argument to be made on using linear focus receivers at these temperatures instead of more expensive point focus technologies such as HFCs and PDRs.
- It was found that it is unviable to use modified vehicle turbochargers together with any form of SEC where the STE would be used to preheat air prior to a fuel combustion stage. Since the combustion chamber inlet temperature is limited to a maximum of 650 °C to limit harmful emissions, turbine outlet temperatures far exceed 700 °C when maximising thermal

efficiency of the engine itself. Therefore, the entire solar preheating stage may be replaced by a single heat exchanger, thereby significantly simplifying construction and decreasing operating costs.

- When modelling a Garrett GTX5533 as a modified vehicle turbocharger BCHE at an operating temperature of 1000 °C, the net heat absorbed thermal efficiency with a heat recycling unit was calculated as approximately 33%. This is very close to the net heat absorbed thermal efficiency of approximately 30% found by Le Roux et al. and their HFC based modified turbocharger CSP BCHE with a heat recycling unit.
- A proof-of-concept apparatus was fabricated to operate a motorcycle turbocharger as a solar hybrid BCHE. While the motorcycle turbocharger did technically successfully operate as an engine and produce useful and measurable work, its performance was disappointing with measured thermal efficiencies below 0.1 %.

6.2 Recommendations for Future Research

The combination of bare copper pipe and a very small motorcycle turbocharger hindered the effectiveness of the experiment. The parabolic trough collectors of the apparatus themselves performed well. The experiment would benefit if it were to be performed with more ideal choices for the receiver and turbocharger in conjunction with the current apparatus.

Emissions from the apparatus were not tested. It is unknown how effective the flame tube was at fuel mixing and combusting the fuel at a predominantly lean air-fuel ratio. This may be an important factor especially when using larger turbochargers.

The intensive linear receiver model was derived from well-studied functions of real-world phenomena. Attempts were made to source real world performance data of moderate temperature CSP installations from around South Africa, but no requests for information were successful. The intensive linear receiver model would benefit if it were to be compared with measured real-world data. This would help to validate the model itself.

If it could be shown that the intensive linear receiver model is a reasonably effective representation of the performance of a linear focus receiver at moderate to high temperatures, then the model may be used as a tool for the hypothetical design of CSP power plants.

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Appendices

A. Physical Phenomena Modelling Functions

The following functions were used as the core functions in the numerically intensive linear receiver model.

McAdams (1954) performed a large series of experiments on the effects of the Reynolds of fluid flow. The Reynolds number may be used to relate forced and free convecting air along a cylinder carrying a heated fluid to the Nusselt number of the air flow (McAdams, 1954). Duffie & Beckman (2013) recommends using the values obtained by McAdams for parabolic troughs while also increasing the values of the coefficients in the equation by 25% in order to account for outdoors conditions. This yields the following (Duffie & Beckman, 2013, p. 165):

$$Nu = \begin{cases} 0.4 + 0.54 Re^{0.52} & 0.1 < Re < 1000 \\ 0.3 Re^{0.6} & 1000 < Re < 50000 \end{cases} \quad (A 1)$$

$$(A 2)$$

Where:

$$Re = \frac{\rho_{air} v_{wind} D_{receiver,outer}}{\mu_{air}} \quad (A 3)$$

$$Nu = \frac{h_{wind} D_{receiver,outer}}{k_{air}} \quad (A 4)$$

h_{wind} is the convective heat-transfer coefficient due to wind, v_{wind} is the wind velocity and $D_{receiver,outer}$ is the total diameter of receiver including all insulating layers. ρ, μ, k are the density, viscosity and heat conductivity of ambient air.

For the case of air at standard conditions (Duffie & Beckman, 2013, p. 330):

$$\frac{\rho_{air}}{\mu_{air}} = \frac{1}{v_{air}} = \frac{1.232}{1.754 \cdot 10^{-5} \left[\frac{s}{m^2} \right]} = 68\,673.36 \left[\frac{s}{m^2} \right] \quad (A 5)$$

To determine the value of the heat conductivity of air, the following equation may be used for dry air (Kadoya, et al., 1985):

$$k_{air}(T_r, \rho_r) = \Lambda \cdot (\lambda_T(T_r) + \lambda_\rho(\rho_r)) \quad (A 6)$$

Where:

$$\lambda_T(T) = 0.2395T_r + 0.0064T_r^{0.5} + 1 - 1.96261T_r^{-1} + 2.0038T_r^{-2} - 1.0755T_r^{-3} + 0.2294T_r^{-4}$$

$$\lambda_\rho(\rho) = 0.4022\rho_r + 0.3566\rho_r^2 - 0.1631\rho_r^3 + 0.1380\rho_r^4 - 0.0201\rho_r^5$$

$$T_r = \frac{T}{T^*}; \rho_r = \frac{\rho}{\rho^*}$$

$$T^* = 132.5 [K]; \rho^* = 314.3 \left[\frac{kg}{m^3} \right]$$

$$\Lambda = 2.59778 \cdot 10^{-2} \left[\frac{W}{m \cdot K} \right] \quad (A 7)$$

The heat capacity of air as a function of temperature is given by (Kyle, 1984):

$$C_{p,air} = 28.11 + 0.1967 \cdot 10^{-2}T + 0.4802 \cdot 10^{-5}T^2 - 1.966 \cdot 10^{-9}T^3 \left[\frac{kJ}{kmol \cdot K} \right] \quad (A 8)$$

For conductive heat transfer between a smooth-walled cylindrical receiver and the HTF, the following functions have shown to be a remarkably accurate reflection of real-world performance for Reynolds Numbers between 2300 and 5×10^6 and Bulk HTF Prandtl numbers between 0.5 and 2000 (Forristall, 2003, pp. 8-9):

$$Q_{receiver-HTF} = h_{T_{HTF,bulk}} D_{absorber,inner} \pi L (T_{absorber} - T_{HTF,bulk}) \quad (A 9)$$

$$h_{T_{HTF,bulk}} = \frac{Nu_{D_{receiver,inner}} k_{T_{HTF,bulk}}}{D_{receiver,inner}} \quad (A 10)$$

$$Nu_{D_{receiver,inner}} = \frac{\left(\frac{f_{fric}}{8}\right) (Re_{D_{receiver,inner}} - 1000) Pr_{T_{HTF,bulk}} \left(\frac{Pr_{T_{HTF,bulk}}}{Pr_{T_{receiver}}}\right)^{0.11}}{1 + 12.7 \left(Pr_{T_{HTF,bulk}}^{\frac{2}{3}} - 1\right) \sqrt{\frac{f_{fric}}{8}}} \quad (A 11)$$

$$f_{fric} = (1.82 * \log_{10}(Re_{D_{receiver,inner}}) - 1.64)^{-2} \quad (A 12)$$

$$Re_{D_{receiver,inner}} = \frac{\left(\frac{4\dot{m}_{HTF}}{\pi D_{receiver,inner}}\right)}{\mu_{HTF,T_{HTF,bulk}}} \quad (A 13)$$

$$Pr(T) = \frac{\mu_{HTF,T_{bulk \text{ or } receiver}} C_{p,HTF,T_{bulk \text{ or } receiver}}}{k_{HTF,T_{bulk \text{ or } receiver}}} \quad (A 14)$$

A transparent tube may be used on a linear focus receiver in an effort to inhibit convection losses by forming an envelope around the receiver. The effectiveness of the envelope may be increased by pulling a vacuum on the selected gas which occupies the annular volume around the receiver. At pressures of or below approximately 1 Torr (133.322 Pa) the primary heat convection mechanism is molecular conduction. At pressures above 1 Torr, free convection becomes driving. The following provides a smooth transition between each driving mechanism (Forristall, 2003, pp. 11-13):

For the case of a vacuum (less than or equal to 1 Torr):

$$Q_{vacuum,convection} = \pi L D_{absorber,outer} h_{vacuum} (T_{absorber,surface} - T_{cover,inner}) \quad (A 15)$$

$$h_{vacuum} = \frac{k_{std}}{\left(\frac{D_{absorber,outer}}{2} \ln\left(\frac{D_{cover,inner}}{D_{absorber,outer}}\right) + b\lambda \left(\frac{D_{absorber,outer}}{D_{cover,inner}} + 1\right)\right)} \quad (A 16)$$

$$\lambda = 2.331 * \frac{10^{-20} (T_{absorber,surface} - T_{cover,inner})}{2 \left(\frac{760}{101325}\right) p_{vacuum} \delta_{gas}^2} \quad (A 17)$$

Table A 1: Constants for Convection in A High Vacuum Annulus (Forristall, 2003, p. 13)

	Vacuum Gas	k_{std} [W/m.K]	b	δ_{gas}
Vacu Air Hydr	Air	0.02551	1.571	3.53E-8
	Hydrogen	0.1769	1.581	2.40E-8
	Argon	0.01777	1.886	3.80E-8

Where P_{vacuum} is the pressure of the gas in Pa, P_{atm} is the atmospheric pressure within the envelope's annulus volume. The constants for these formulae are listed in Table A 1.

For the case of an envelope pressure above 1 Torr assuming an ideal gas:

$$Q_{vacuum,convection} = \frac{2.425Lk_{T_{avg}}(T_{aborber,surface} - T_{cover,inner}) \left(\frac{Pr_{T_{avg}} Ra_{vac}}{0.861 + Pr_{T_{avg}}} \right)^{\frac{1}{4}}}{\left(1 + \left(\frac{D_{absorber,outer}}{D_{cover,inner}} \right)^{\frac{3}{5}} \right)^{\frac{5}{4}}} \quad (A 18)$$

With:

$$Ra_{vac} = \frac{g(T_{aborber,surface} - T_{cover,inner})D_{absorber,outer}^3}{T_{avg}\alpha_{gas,T_{avg}}\nu_{gas,T_{avg}}} \quad (A 19)$$

$$T_{avg} = \frac{(T_{aborber,surface} + T_{cover,inner})}{2} \quad (A 20)$$

Where $\alpha_{gas,T_{avg}}$ is the vacuum gas' thermal diffusivity at the average envelope temperature, $k_{T_{avg}}$ is the vacuum gas's thermal conductance at the average envelope temperature, and $Pr_{T_{avg}}$ is the Prandtl number of the vacuum gas at the average envelope temperature.

Duffie and Beckman (2013) propose a similar function to model heat transfer within an annulus from pure heat conduction at higher pressures to free molecular heat transfer at lower pressures. These functions are based on the findings by Raithby and Hollands (1975) and Raithby et al. (1977) (Duffie & Beckman, 2013, pp. 153, 154, 329):

$$\frac{k_{effective}}{k} = 0.386 \left(\frac{Pr \cdot Ra^*}{0.861 + Pr} \right) \quad (A 21)$$

$$Ra^* = \frac{\left(\ln \left(\frac{D_{cover,inner}}{D_{absorber,outer}} \right) \right)^4 Ra_L}{\left(\frac{D_{cover,inner} - D_{absorber,outer}}{2} \right)^3 \left(D_{cover,inner}^{-\frac{3}{5}} + D_{absorber,outer}^{-\frac{3}{5}} \right)^5} \quad (A 22)$$

$$Ra_L = \frac{g(T_{aborber,surface} - T_{cover,inner}) \left(\frac{D_{cover,inner} - D_{absorber,outer}}{2} \right)^3}{T_{avg}\alpha_{gas,T_{avg}}\nu_{gas,T_{avg}}} \quad (A 23)$$

Or:

$$\frac{k_{effective}}{k} = \left(1 + \frac{(2(9\gamma-5)\lambda)}{(\gamma+1) \ln\left(\frac{D_{cover,inner}}{D_{absorber,outer}}\right)} \cdot \left(\frac{1}{D_{cover,inner}} + \frac{1}{D_{absorber,outer}} \right) \right)^{-1} \quad (A 24)$$

$$\lambda = 1.381 \cdot 10^{-23} \frac{T_{avg}}{\sqrt{2}\pi p \delta^2} \quad (A 25)$$

With γ being the ratio of specific heats, λ being the mean free path of gaseous molecules, and δ being the molecular diameter of the gas.

For the case of air as the annulus gas, $\gamma \approx 1.4$, $\delta \approx 3.5 \cdot 10^{-10}$ (Duffie & Beckman, 2013, p. 154).

In practice, the higher value of $\frac{k_{effective}}{k}$ is chosen between Equations A 21 and 24, and used in conjunction with:

$$k_{effective} = \left(\frac{k_{effective}}{k} \right) * k \quad (A 26)$$

$$Q_{vacuum,convection} = \frac{2\pi k_{effective} L (T_{absorber,surface} - T_{cover,inner})}{\ln\left(\frac{D_{cover,inner}}{D_{absorber,outer}}\right)} \quad (A 27)$$

In addition to radiative losses, the absorber surface emits long wave radiation. Without a cover surrounding the absorber, the sky and physical surroundings are the objects to which the heat energy is radiated. With a solar transparent cover surrounding the absorber, the cover itself is the surface to which the absorber radiates.

If it is assumed that the solar transparent cover is opaque to all of the longwave radiation emitted from the absorber, and it is assumed the gas in the envelope is completely transparent to the longwave radiation, and it is assumed the absorber surface and cover surface behave as greybodies (that is their values for absorptivity, emissivity, transmissivity and reflectance are constant for all wavelengths and/or wavelength distributions), then it may be shown that (Forristall, 2003, p. 14):

$$Q_{radiation,absorber-cover} = \frac{(\sigma \pi L D_{absorber,outer} (T_{absorber,surface}^4 - T_{cover,inner}^4))}{\left(\frac{1}{\epsilon_{absorber}} + \frac{(1 - \epsilon_{cover}) D_{absorber,outer}}{\epsilon_{cover} D_{cover,inner}} \right)} \quad (A 28)$$

While the assumptions above are not a true reflection on reality, the errors produced by the assumptions are relatively small (Forristall, 2003, p. 14; Touloukian & DeWitt, 1972).

And finally, for radiation emitted from the receiver to the surroundings (Forristall, 2003, pp. 16-17):

$$Q_{\text{radiation,receiver-surroundings}} = \sigma \pi L D_{\text{receiver},\epsilon} \epsilon_{\text{receiver,outer}} (T_{\text{receiver},\epsilon}^4 - T_{\text{sky,blackbody}}^4) \quad (\text{A } 29)$$

Where $D_{\text{receiver},\epsilon}$ is the diameter of the receiver from which the radiative emission to the surroundings originates (i.e. the outer diameter of the absorber if there is no cover, or the outer diameter of the cover when present), $\epsilon_{\text{receiver,outer}}$ and $T_{\text{receiver},\epsilon}$ are the emissivity and temperature of that same surface.

When modelling linear receivers in this fashion, simplification for the cover is suggested by assuming the cover acts as a greybody. In the case of glass a reasonable value for the envelope gap is 6mm with glass's thermal conductivity = $1.05 \frac{\text{W}}{\text{m.K}}$ and $\epsilon = 0.86$ where this emissivity is not a function of temperature (Forristall, 2003, pp. 18-19; Touloukian & DeWitt, 1972). The vacuum pressure of commercial linear absorber covers lie in the region of about 0.013 Pa (Price, et al., 2004).

The effective temperature of the sky needs to be found in which it acts as the longwave radiation sink for the receiver. The Swinbank Formula may be used to estimate the temperature of the sky with respect to the thermal downwelling (longwave radiation flux density) emitted from the sky onto earth (Goforth, et al., 2002). This value may then be used to calculate the equivalent blackbody temperature of the sky. It has been calculated that (Goforth, et al., 2002; Swinbank, 1963):

$$T_{\text{sky,blackbody}} = \left(\frac{8.778 \cdot 10^{-13} \cdot T_{\text{air}}^{5.852} \cdot RH^{0.07195}}{\epsilon_{\text{sky}} \sigma} \right)^{\frac{1}{4}} \quad (\text{A } 30)$$

The sky's emissivity is a function of relative humidity (RH), and may be estimated as (Tang, et al., 2004):

$$\epsilon_{\text{sky}} = 0.711 + 0.56 \left(\frac{T_{\text{dew-point}} - 273.15}{100} \right) + 0.73 \left(\frac{T_{\text{dew-point}} - 273.15}{100} \right)^2 \quad (\text{A } 31)$$

B. Turbocharger Maps

The turbocharger compressor and turbine performance curves listed below are those that were used in Section 4. These high-resolution maps were necessary to develop the simulation functions described in Section 4.1.

The maps were sourced from Garret's website and most recent brochure (Garrett, 2018; Garrett, 2016).

GT0632SZ, 32 mm, 50 Trim, 0.32 A/R

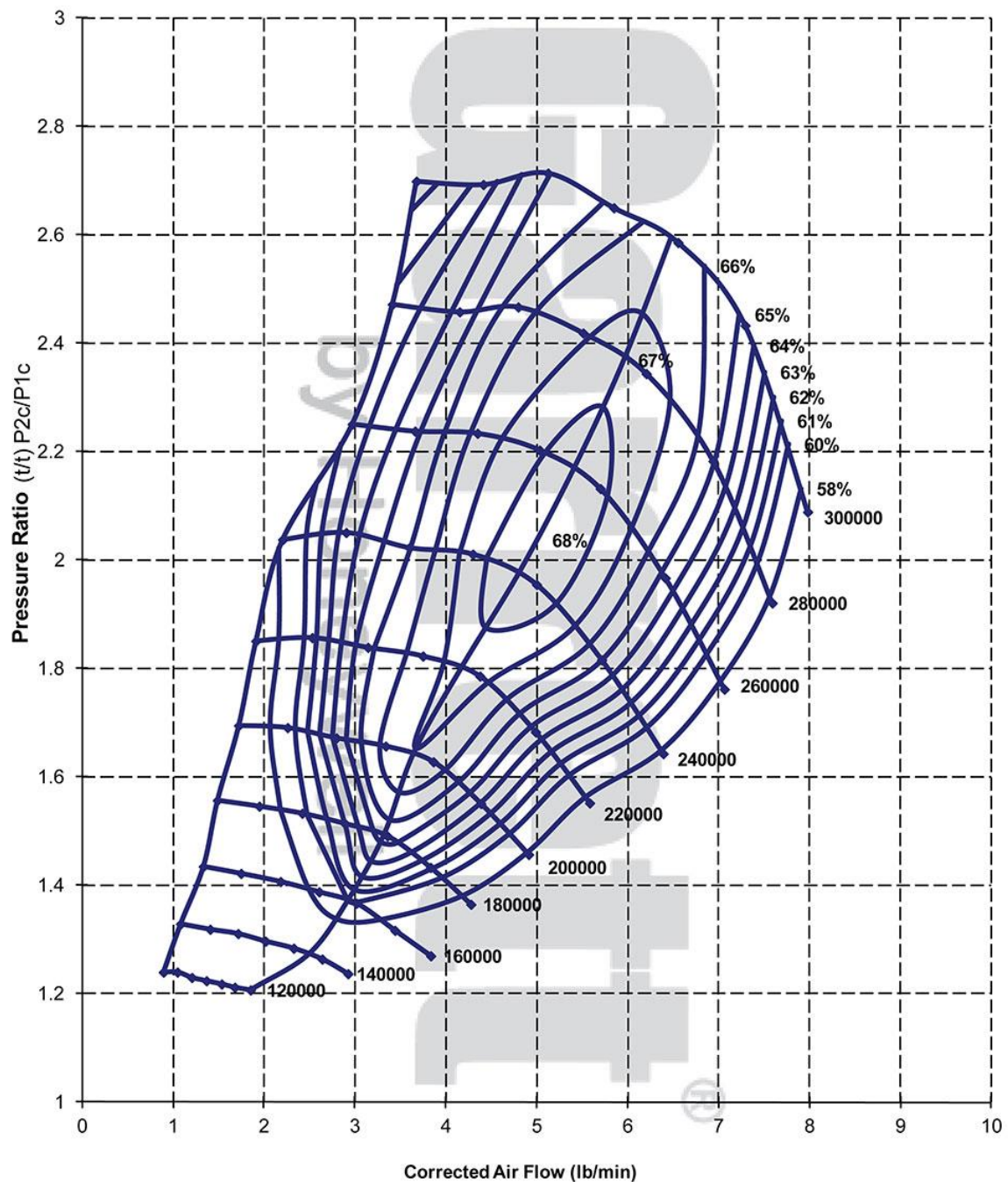


Figure B 1: GT0632SZ Compressor Map

The GT0632SZ is a micro sized turbocharger designed for use with motorcycles and snow bikes (Figures B 1 and 4). The GTX3584 had the highest each turbine and compressor peak efficiencies within Garret's catalogue (Figures B 2 and 5). The GTX5533R had the highest operating pressure ratio of all of the turbochargers in Garret's catalogue (Figures B 3 and 6).

GTX3584RS

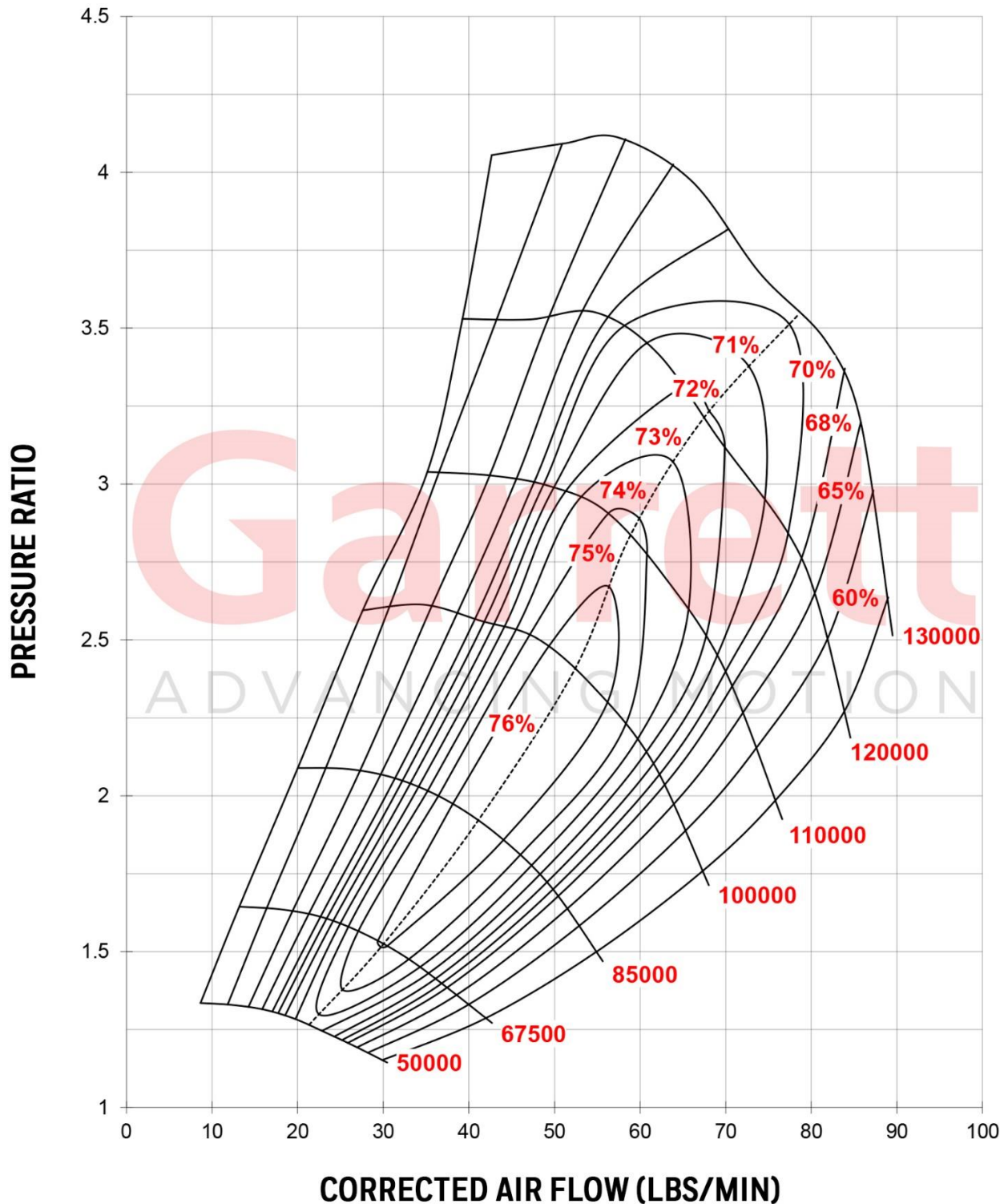


Figure B 2: GTX3584 Compressor Map

The reasons for the choice of each turbocharger used for modelling are detailed in Section 4.1.

GTX5533-98mm

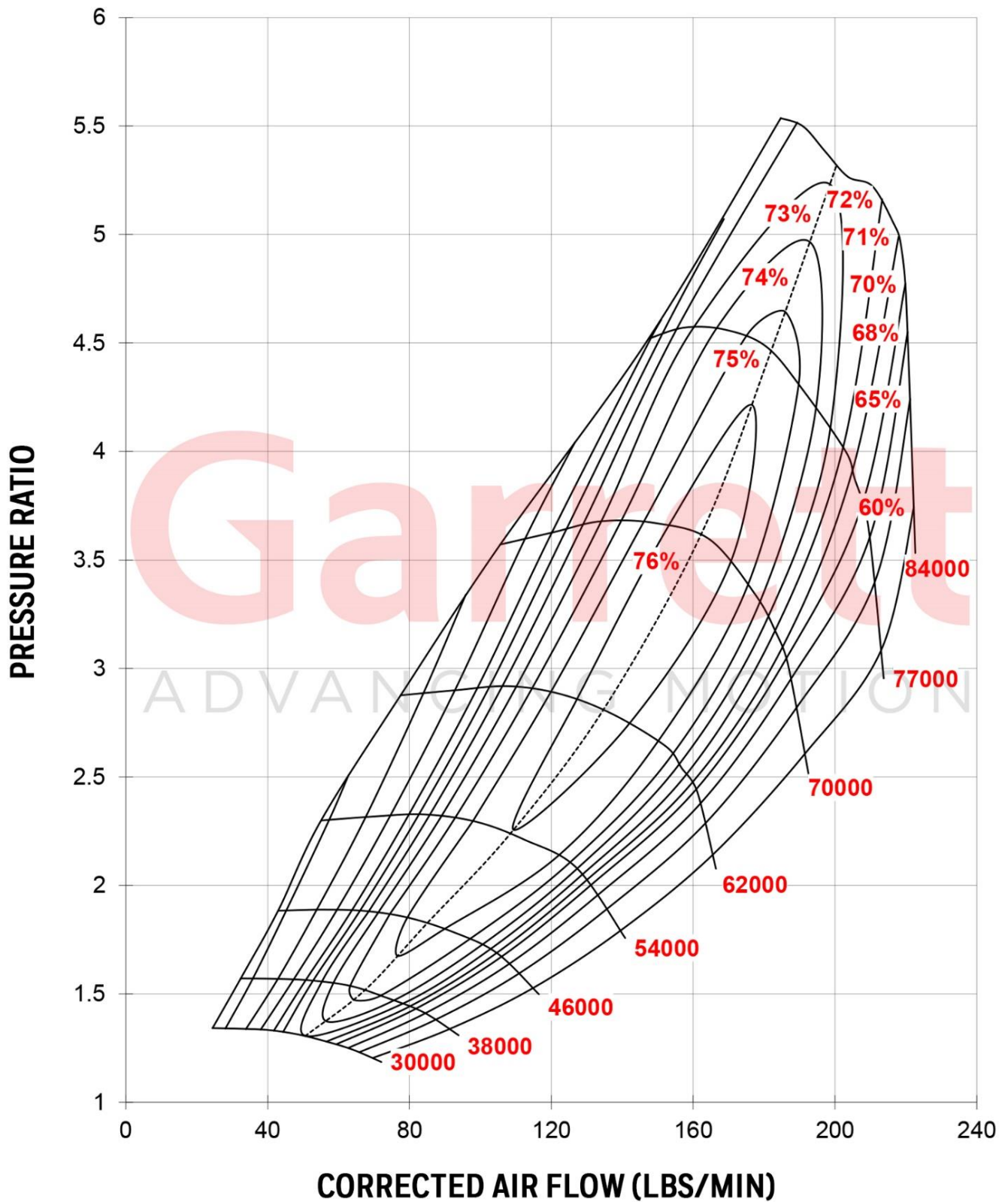
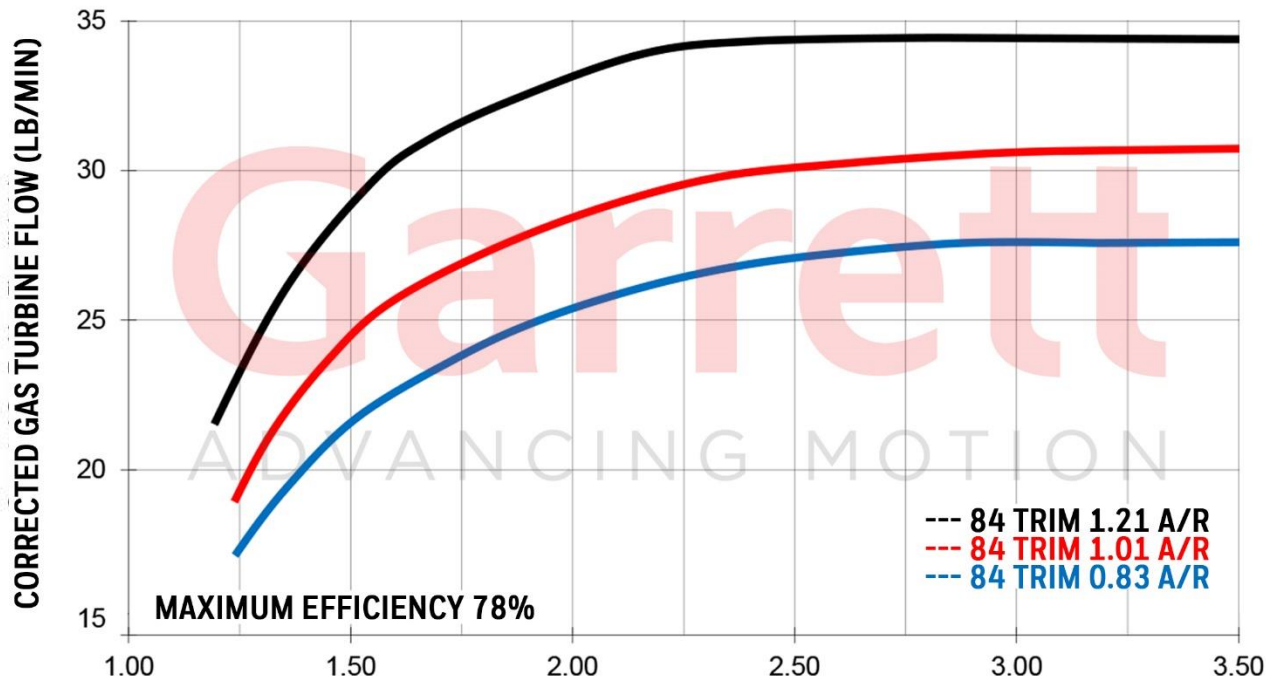
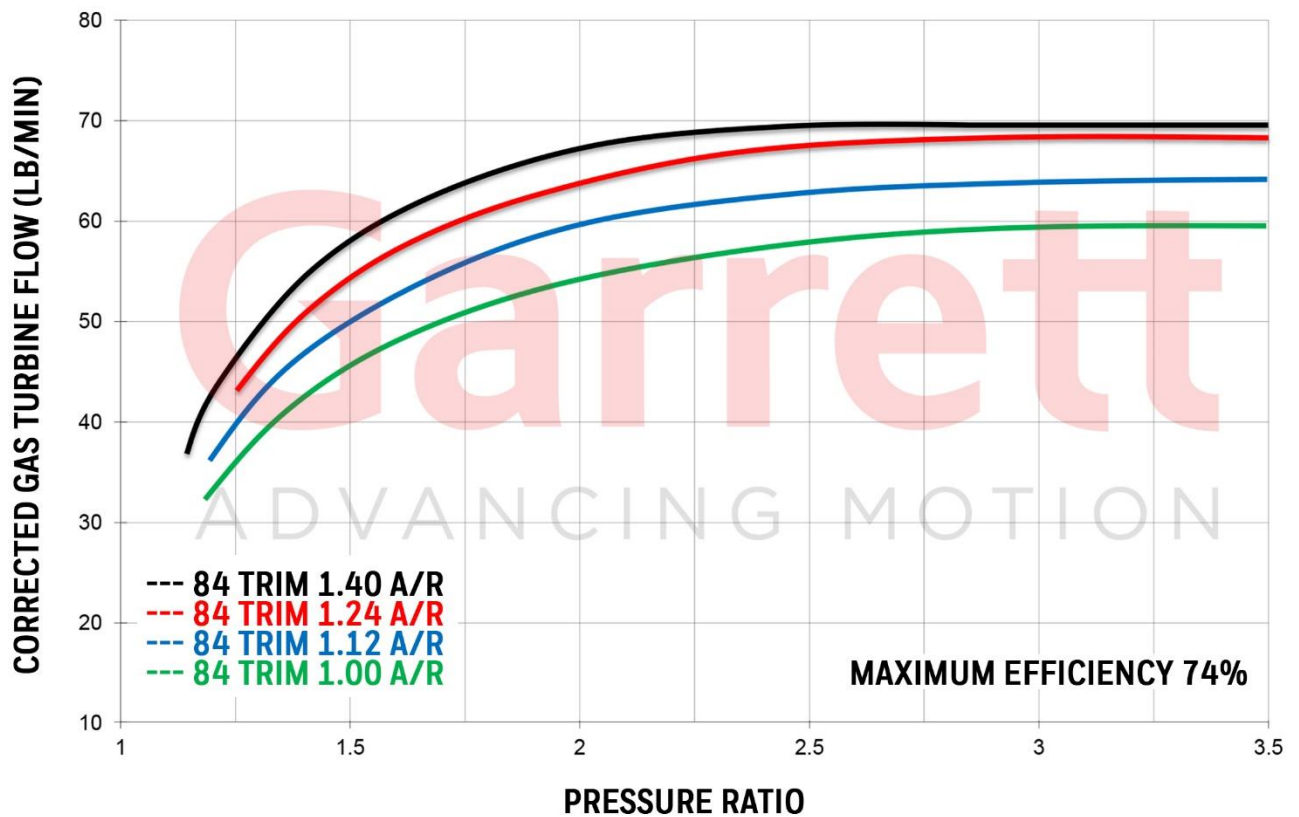


Figure B 3: GTX5533R Compressor Map

GTX3584RS



GTX5533-98mm



C. List of Purchased Materials

Table C 1 lists the prices of the items and materials purchased used in the fabrication of the apparatus used in Section 5.

Table C 2 lists the estimated production cost of each linear collector unit in USD for comparison to PV technologies as used in Section 6.

Table C 1: List of Purchased Materials for the Apparatus

Item	Quantity	Cost Per Unit	Total
45mm copper tubes	3x5.5m	943.36	2830.08
15mm copper tubes	2x5.5m	281.58	563.16
2000 lb _f winch	1	875.00	875.00
BLDC motor controller (Maytech 40A BEC)	1	356.00	356.00
Propane fittings	1	337.44	337.44
Raspberry Pi 3	1	987.00	987.00
4s LiPo 4AH	1	989.95	989.95
5V LVR	1	4.54	4.54
Diodes	20	0.63	12.60
Relays	20	11.82	236.40
Breadboard	1	55.01	55.01
Arduino breakout cables	3	23.00	69.00
3P DC Rectifier DPC35A 1K6V	1	42.00	42.00
10m 5mm wire	1	49.90	49.90
Weather resistant paint	1 tin	399.00	399.00
Silver solder with flux	1 pack	213.00	213.00
3200x2050mm acrylic mirrors	3	2283.04	6849.12
Magnet wire 16m 1.25mm	1	98.40	98.40
K-Type Thermocouple (RS-Components Stainless Steel 1100°C)	1	272.20	272.20
TM902C K-Type Thermocouple reader	1	175.00	175.00
10 000V sparker	1	11.99USD=R163.06	163.06
Exhaust wrap 15m	2	595.00	595.00
MCP3008 SPI ADC	2	46.07	46.07
Xnova 4025-1120KV 1.5 Y	1	2995.00	2995.00
BMS 6mm rod 2m	1	28.00	28.00
Silver Steel 6mm rod 2m	1	52.00	52.00
M4x0.5 Left hand tap + die HSS	1	14.85USD=R240.57	240.57
Y-Solar 30A 48V MPPT controller	1	63.55 USD = R1025.30	1025.30
			R 19534.50

Table C 2: Estimated Cost of Production for Each Linear Collector Unit

Item	USD
80.1kg Carbon Steel Plate	208.26
Collector Tube 2m x 42mm	32.47
Cleaning Tube 2m x 15mm	7.16
Plate 15.52m 50mm x 2mm = 23.87kg	31.03
2x2mx3m	168.06
Battery and PSU for Pi	26.36
Bearings and fittings	34.93
Pi 2 B+	44.90
Winch	59.02
Wires and detectors	13.29
Solder/Welding	23.36
Total per Trough	648.83

D. Trough Design Development

The following subsection details the design and fabrication of the linear collector units used in the experiments detailed in Section 4.

Each collector unit consisted of two A-frames made from square tube connected together by a length of square tube near each foot (Figure D 1). A sleeve was welded on each side of the frame at the points where the square tube crossed at the apex of the A. Each sleeve extended toward the inside of the frame a short distance. Two large weatherproof bearings were mounted by their inner diameters to the sleeves on the inside of the frame. The bearings were then welded to the sleeves, such that their outer diameters rotated freely.

Two large rectangular plates were then welded to the outer diameter of each bearing on the inside of the frame. By welding square tube to each corner of these plates, the plates were made to be rotated synchronously. Finally, a pipe was inserted through the inner diameter of sleeves such that the volume enclosed by the plates within the frame rotates about this stationary receiver pipe. There is no mechanical loading on the receiver pipe other than its own weight since it passes through the inside of the stationary sleeves.

By setting the centre of the bearing mountings for the plates as the focus of a parabola, the receiver pipe is made to be the linear focal point of the parabolic trough. Appropriate channels were cut on each plate along a parabolic arc such that a flexible acrylic mirror may be inserted and suspended between the two plates. For this reason, the plates were given the name of the mirror brace.

Thin offcuts of sheet metal were inserted within the channels to act as a lip for the flexible acrylic mirrors. Finally, thin square bars were welded across the underside

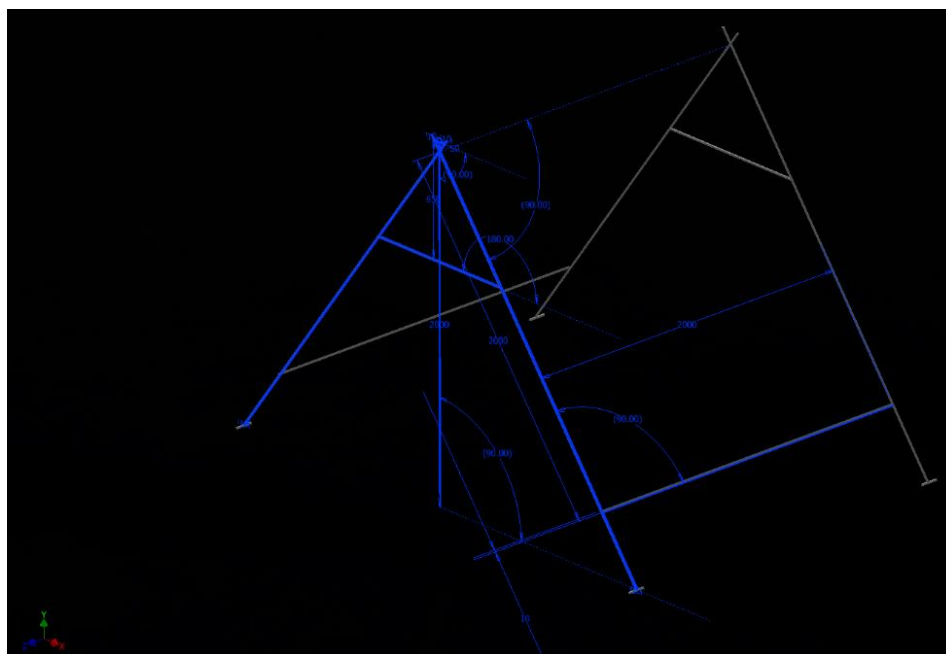


Figure D 1: A-Frame design for the Parabolic Trough



Figure D 2: A Completed Frame and Mirror Brace Assembly

of each lip to the other side of the frame to act as support for the flexible mirrors. Figure D 2 shows one of the completed mirror brace assemblies.

The troughs were oriented in an E-W fashion such that it was only necessary to rotate in a Northward direction. A counterweight of about 1 kg was added to the lower bar of the mirror brace on the South side, as seen in Figure D 3; where south is to the left, and north is to the right.

A small inexpensive latching winch (2000 lb_f) was placed in the centre of the frame bar on the South side of the collector and coupled to the lower mirror brace bar on the North side. Reeling in the winch pulls on the mirror brace and lowers the altitude of the aperture in a Northward direction (i.e., rotating the brace clockwise in Figure D 3). Reeling out the winch increases the altitude of the aperture Southwards.

This meant the trough could effectively rotate from about 30° above the horizon in a Northward direction to about 20° South of normal to the ground.



Figure D 3: Manually Testing Heliostat Winch Operation

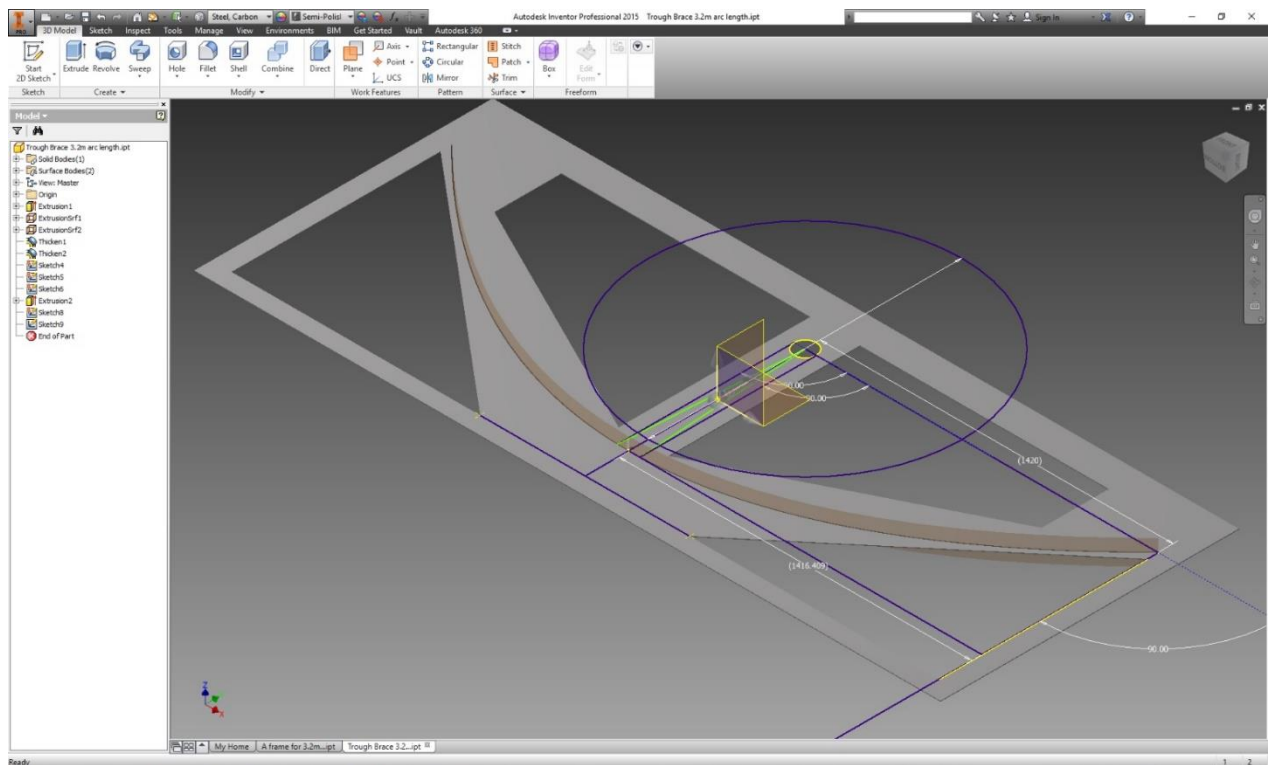


Figure D 4: Final CAD Sketch of the Mirror Brace in Autodesk

The gearing on the winch produced a relatively slow rotation of the trough. It required about 5 minutes to rotate the trough from one endpoint to the other. This afforded a substantial granularity of control over the altitude of the parabolic trough concentrator.

To maximise the aperture of the collector, the largest contiguous piece of sheet metal stocked at the fabrication warehouse was chosen for the mirror brace. This sheet metal had the dimensions of 3000 x 1200 x 3 mm. Conveniently, the largest piece of acrylic mirror which could be sourced measured 3200 x 2050 x 3mm. The parabolic curve of the mirror could be made shallow enough such that most of the width of the mirror brace would be effective aperture.

The acrylic mirror was cut in half along the width yielding two 1600 x 2050 x 3 mm sections. The intention behind this was to add a gap between the mirror sections to allow wind and water from rain and cleaning to pass through to minimise mechanical loading on the fragile mirrors.

A parabolic curve was parameterised to fit the arc length of the two 1.6 m mirror sections. The centre of the receiver was set as the focus and the aperture of the collector was kept as wide as reasonably possible.

Unused sections of the mirror brace were cut out to reduce unnecessary weight. These offcut sections were used to make other parts in day-to-day operations at the facility. The final design of the mirror brace may be seen in Figure D 4.

The function defining the parabolic curve in which the mirror is to be seated in the brace must next be found. The general form for a parabolic curve in Cartesian

coordinates with a vertex of (h, k) having a distance of p from vertex to focus is (Stroud & Booth, 2013):

$$(x - h)^2 = 4p(y - k)$$

Which may be rewritten in the form:

$$y = ax^2 + bx + c$$

Where:

$$a = \frac{1}{4p}; b = -\frac{h}{2p}; c = \frac{h^2}{4p} + k$$

$$h = -\frac{b}{2a}; k = \frac{4ac - b^2}{4a}$$

And may be parameterized in the form:

$$x(t) = 2pt + h$$

$$y(t) = pt^2 + k$$

The focus therefore exists at the point:

$$F = \left(-\frac{b}{2a}, \frac{1 - (b^2 - 4ac)}{4a} \right)$$

Setting the vertex at the origin and $h = k = 0$

$$F = \left(0, \frac{1}{4a} \right) = (0, p)$$

$$x(t) = 2pt$$

$$y(t) = pt^2$$

By arbitrarily assigning $t \in [-1, 1]$ so that the maximum height of the parabolic curve is level with the focal point, the value of p must satisfy the conditions that the maximum x and y points of the parabola must fit within the brace and the arc length of the parabola must be at least the width of the mirror to be used (3200mm).

For the brace dimensions 3000 x 1200 mm, or rather from origin 1500mm either direction in the x axis, $x(t) = 2pt \therefore \max p_x = 750$. Similarly, $\max p_y = 1000 \therefore \max p = 750$:

$$p \in [0, 750)$$

As for the arc length:

$$\int_{\alpha}^{\beta} dS \geq 3200$$

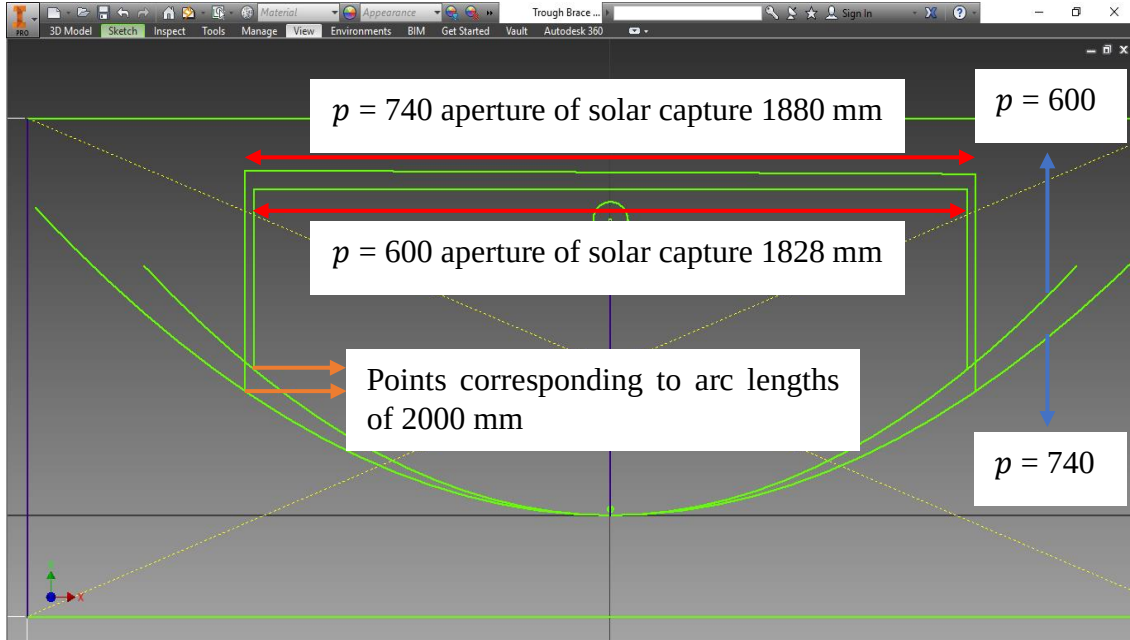


Figure D 5: Parameterising the Parabolic Function to generate target Arc Lengths

$$\begin{aligned}
 \therefore \int_{-1}^1 \left(\sqrt{\left(\frac{dx}{dt}\right)^2 + \left(\frac{dy}{dt}\right)^2} \right) dt &\geq 3200 \\
 &= 2p \int_{-1}^1 \left(\sqrt{1+t^2} \right) dt \geq 3200 \\
 &= 2p(\sqrt{2} + \operatorname{arcsinh}(1)) \approx 4.5912p \geq 3200
 \end{aligned}$$

Therefore, values of p satisfying the constraints of Equations above are:

$$p \in [696.986, 750)$$

The largest piece of plate available for use was 3000 x 1200 mm sections made from 3mm thick material. The largest piece of acrylic mirror that could be easily sourced nearby was from Maizey in Edenvale, measuring 3200 x 2050 x 3 mm. This was a convenient size as it was approximately the width of the brace to be used.

Figure D 5 shows the effect changing the p value has on the arc length of the parabola for the piece of sheet metal available to act as the frame to hold the mirrors.

A p value close to the maximum should be chosen so as to maximise the available surface area for solar capture in the x direction but with enough space between the brace and parabolic mirror seat ends to provide rigidity. The difference in aperture between the minimum and maximum p value in this case, however, is less than 2%.

A p value of 710 was chosen so that the arc length of the seat made in the brace for the mirror was slightly lengthier than the width of the mirror itself to ease the installation of the mirror into the brace. The arc length of the seat for the mirror was therefore 59.7mm longer than the width of the mirror.

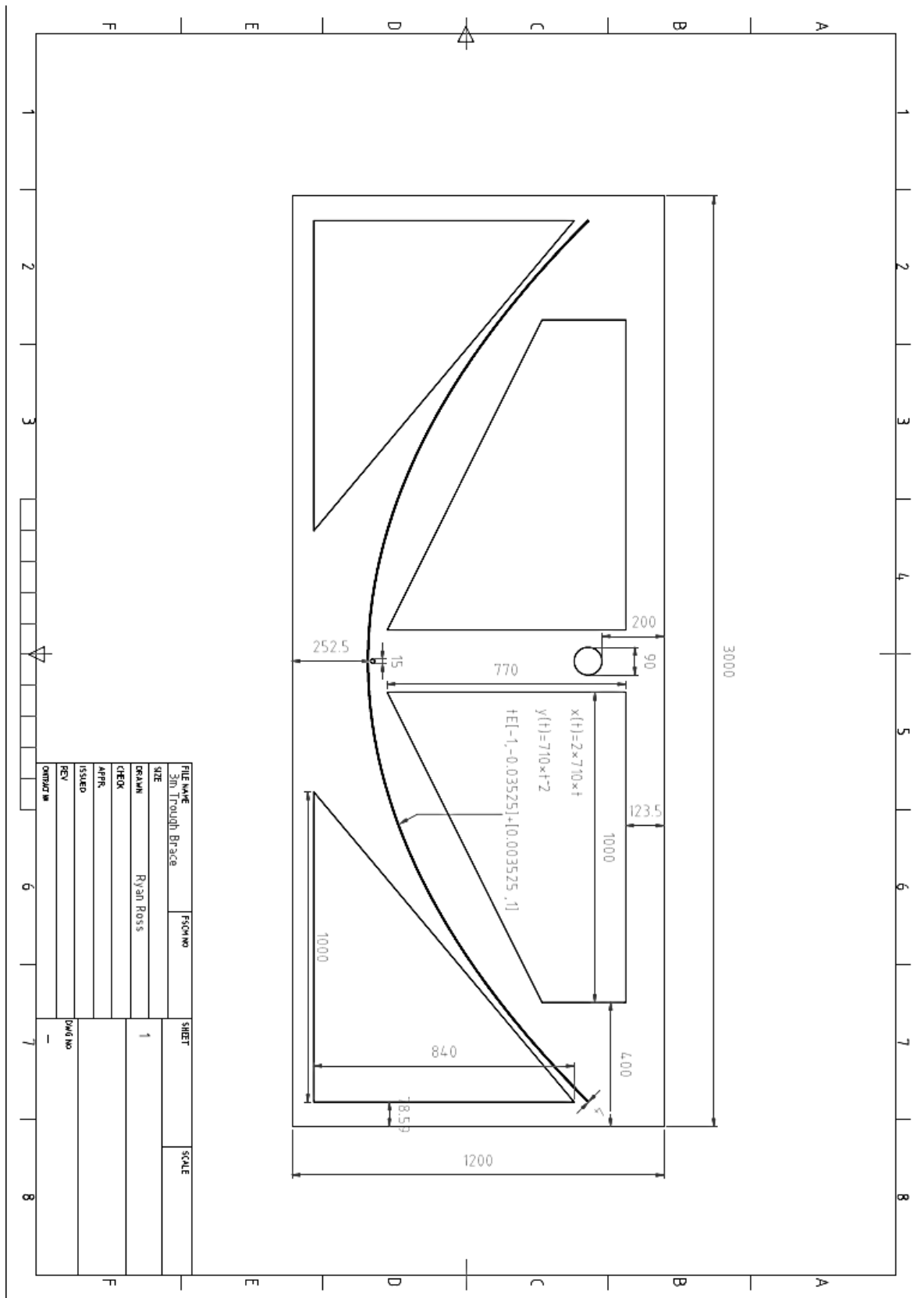


Figure D 6: CAD Drawing of the Mirror Brace sent to the Laser Cutter

A small gap was left in the middle of the parabola to provide a space for water to escape during rain or washing. The values of t used to generate the gap were $t \in$

$[-1 - 0.003525]$ and $t \in [0.003525, 1]$. This left a central gap of 10mm for an overall arc length for the seat of the mirror of 3250mm.

A 15mm hole was included in the brace to attach a thin pipe under the receiver. This pipe may be used for cleaning by drilling appropriate holes to spray soapy water onto the mirrors.

Figure D 6 was the final mirror brace design which was sent to the laser cutting machine. Figure D 7 is the CAD drawing that was used as a reference to make the A frames.

2.5m long offcut lengths of the 50mm square tubing were regularly available. The A-frame was designed to use these lengths to raise the braces high enough off the ground to permit 90 degrees of rotation in either direction. The legs each used a 2.5m long piece which were to be placed 2.9m apart from each other measured on the ground, as well as meet at 2.4m along each leg for the weld. This would place the bottom of the section for the brace bearing to be welded to at 1.95m above the ground giving a clearance of 0.45m for the brace when the trough was rotated at 90 degrees. The A-frame was made to be 2.05m in length to match the acrylic mirrors.

The assembly procedure for the frame of the trough is relatively straightforward. With an A-frame made, two 50mm round tube offcuts approximately 40mm in length are welded to the upmost meeting point of the legs each side of the frame positioned inwards. Bearings were slid over the round tube and spot welded in place. The braces were then be slid over the bearings each side and spot welded in place. Some 10mm square tube was then spot welded to four corners of each brace to add some rigidity to the collector. Finally, the receiver tube could be slid into the trough through the middle of the bearings.

As for the electronics control circuit used:

The BLDC ESC was powered by a 4S 4 Ah Li-Po battery. The Pi was powered from a portable 2 A 10 Ah USB power bank. A 12 V 7 Ah lead-acid battery was used as the power source for the relays and winch. Four 12 V 7 Ah lead-acid batteries were connected in series as the load for the MPPT controller. The 12 V batteries were all carefully charged/discharged to 12.5 V before each test.

A PHD DPC35A 1K6V was used for the three-phase rectifier. It measured a forward voltage drop of 0.42 V across each phase pin. A 160 V 100 μ F aluminium electrolytic capacitor was connected across the rectifier's DC output to smooth the input to the MPPT controller.

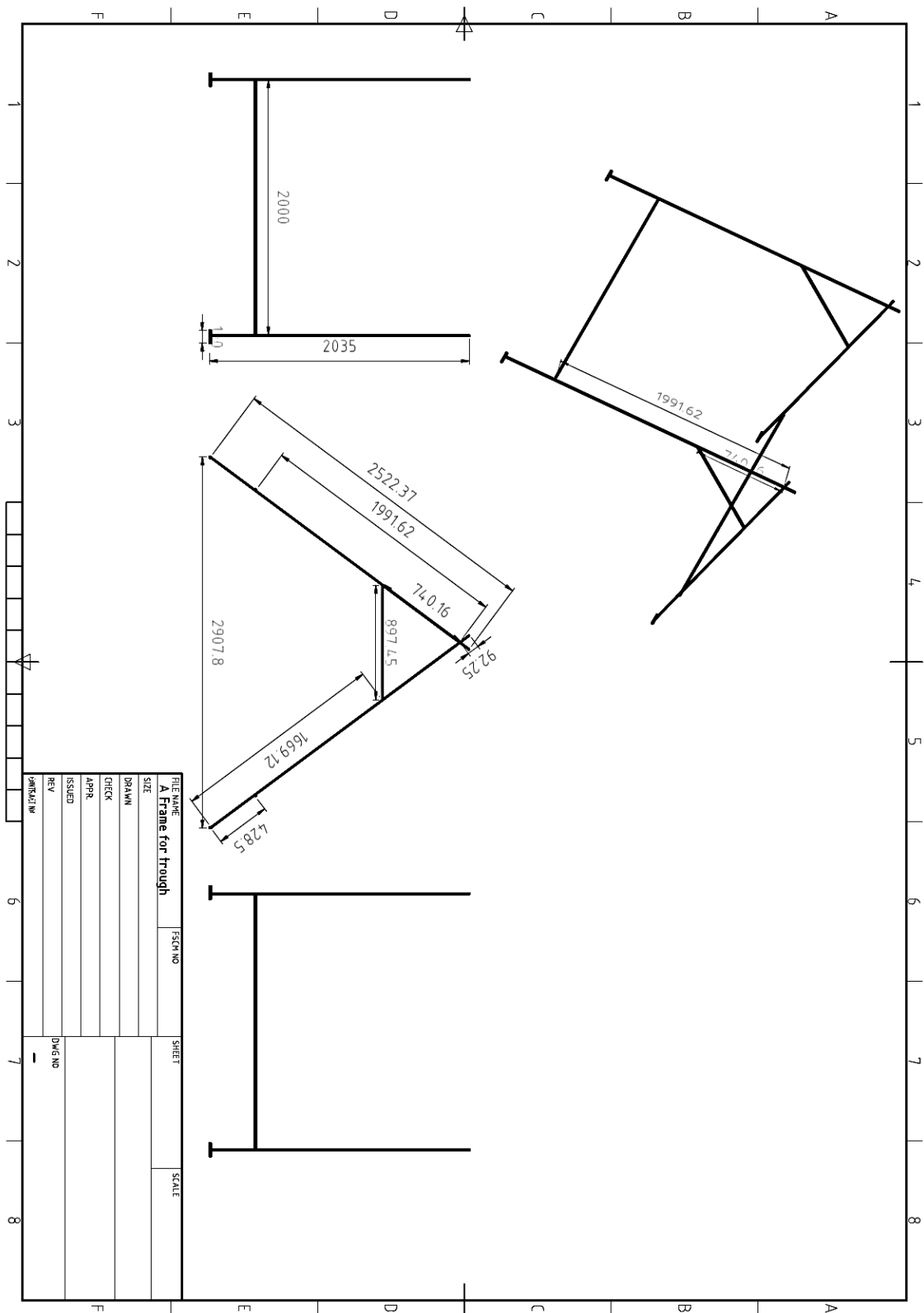


Figure D 7: A Frame Construction Guide

E. Experimental Data

Table E 1 lists the recorded data and calculation results from the 2nd experimental run of the apparatus. This is detailed in Section 5.2.

Table E 1: Results from the 2nd Experimental Run Performed on 13th September 2016

Freq Target [Hz]	3000	4000	4500
Comp Outlet / Measured Receiver Inlet Temp [°C]	76.5	108	123.2
$P_{operating}$ [bar_g]	1.35	1.81	1.92
Estimated Flowrate from Compressor Map [lb/min]	4.2	5.8	7
Receiver Outlet / Flame Tube Inlet Temperature [°C]	115.8	129.1	133
Expected Receiver Outlet Temperature [°C]	134	151	159
Flame Tube Outlet / Turbine Inlet Temperature [°C]	765	742	792
Expected Output Power [W]	58	191	142
Measured MPPT Output [W]	21	31	13
Difference Between Expected and Measured Receiver Outlet [°C]	18.2	21.9	26
Estimated Flame Tube Heating Duty	18018	23483	30591
Estimated Output Power [W]	1026	762	462
Estimated $\eta_{receiver}$	0.183214	0.136071	0.0825
Estimated Thermal Efficiency %	0.11655	0.13201	0.042496

F. MATLAB

This section contains all of the code used in the Dissertation. To recreate all of the simulation results as used in Sections 2,3,4 and 5, only the `_main_ MATLAB` script `DOALL.m` needs to be called.

All of the functions referenced in `DOALL.m` are listed in Section F.2.

F.1 DOALL.m

```
%Run DOALL.m to perform the Dissertation
%Does all of the processing for the dissertation. It takes a few
hours to complete. Some operations have been parallelized to
significantly increase processing speed. Final run for
dissertation performed on MATLAB R2019a
%Figure 2.4.1
Tair_ground_and_sky=25+273.15;
T_H = linspace(Tair_ground_and_sky,2500,100);
sigma = 5.670373.*10.^-8;%Stephan-Boltzman Constant%
eta_Carnot=1-((Tair_ground_and_sky)./(T_H));
I=1000;
eta_C_10 = (1-((sigma.*(T_H.^4))./(I.*10))).*(1-
((Tair_ground_and_sky)./(T_H)));
length_C_10 = length(T_H(eta_C_10>0))+1;
eta_C_50 = (1-((sigma.*(T_H.^4))./(I.*50))).*(1-
((Tair_ground_and_sky)./(T_H)));
length_C_50 = length(T_H(eta_C_50>0))+1;
eta_C_100 = (1-((sigma.*(T_H.^4))./(I.*100))).*(1-
((Tair_ground_and_sky)./(T_H)));
length_C_100 = length(T_H(eta_C_100>0))+1;
eta_C_250 = (1-((sigma.*(T_H.^4))./(I.*250))).*(1-
((Tair_ground_and_sky)./(T_H)));
length_C_250 = length(T_H(eta_C_250>0))+1;
eta_C_1000 = (1-((sigma.*(T_H.^4))./(I.*1000))).*(1-
((Tair_ground_and_sky)./(T_H)));
length_C_1000 = length(T_H(eta_C_1000>0))+1;
eta_C_2000 = (1-((sigma.*(T_H.^4))./(I.*2000))).*(1-
((Tair_ground_and_sky)./(T_H)));
length_C_2000 = length(T_H(eta_C_2000>0))+1;
eta_C_5000 = (1-((sigma.*(T_H.^4))./(I.*5000))).*(1-
((Tair_ground_and_sky)./(T_H)));
length_C_5000 = length(T_H(eta_C_5000>0))+1;
figure
plot(T_H,eta_Carnot,T_H(1:length_C_10),eta_C_10(1:length_C_10),'--
',T_H(1:length_C_50),eta_C_50(1:length_C_50),'-.',...

T_H(1:length_C_100),eta_C_100(1:length_C_100),'o',T_H(1:length_C_2
50),eta_C_250(1:length_C_250),'+',...

T_H(1:length_C_1000),eta_C_1000(1:length_C_1000),'*',T_H(1:length_
C_2000),eta_C_2000(1:length_C_2000),'.',...
T_H(1:length_C_5000),eta_C_5000(1:length_C_5000),'s');
legend('\eta_{Carnot}','C=10','C=50','C=100','C=250','C=1000','C=2
000','C=5000');
xlabel('T_H');
ylabel('\eta_{thermal,Black Body}');
%Figure 2.4.2
I=1000;
Tair=25+273.15;
```

```

C=linspace(10,500,50);
[ Tmax, Topt, eta_opt ] = getCSPdata( I,C,Tair );
figure
plot(C,Tmax,'r',C,Topt,'b--');
xlabel('Solar Concentration Ratio C');
ylabel('Temperature [K]');
legend('Maximum obtainable temperature','Optimal operating
temperature');
%Exergy optimization vs black body figure EDIT: Not used since
Tmax calced
%here is from stagnation temp calculated for only radiative losses
C=linspace(10,5000,500);
[ Tmax, Topt, ~ ] = getCSPdata( I,C,Tair );
Texergyopt=(Tmax.*Tair).^0.5;
figure
plot(C,Topt,'b--',C,Texergyopt,'r');
xlabel('Solar Concentration Ratio C')
ylabel('Temperature [K]');
legend('Optimal power output (Black Body Radiative Heat
Losses)','Optimal exergy recovery (Linear Heat Losses)');
%----- Intensive Linear Receiver Model Section 3.2
L=5;Q_solar=5000;T_HTF_bulk=80+273;p_HTF=2*101325;T_air=25+273;
D_absorber=0.045;D_cover_inner=D_absorber+(12/1000);D_cover_outer=
D_cover_inner+(20/1000);
p_air=101325;p_vac=0.013;epsilon_absorber=0.1;k_cover=1.005;epsilo
n_cover=0.86;
windspeed=2;T_dewpoint_air=14+273;RelativeHumidity=50;mdot=0.01;
alpha_absorber=0.9;rho_mirror=0.93;tau_cover=0.96;gamma_tracking=0
.94;
F_soiling=0.97;
%Do examples of arbitrary air receiver First number of numerical
slices for
%integral over receiver
Q_in=Q_solar;
graphSteps=200;
numSectionsForIntegral=round(logspace(log10(1),log10(5000),graphSt
eps));%the number of sections must be an integer
Q_HTF_numsections(graphSteps)=0;%Reserve memory for vector
parfor i=1:graphSteps

Q_HTF_numsections(i)=sum(sim_Receiver_air_HTF_air_vac(numSectionsF
orIntegral(i),L,mdot,Q_in,T_HTF_bulk,p_HTF,T_air,...

D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorbe
r,k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity)
);
    disp(i);%visual indication program not crashed
end
TrueAns = Q_HTF_numsections(end);
Q_HTF_error_percent=((Q_HTF_numsections-TrueAns)./TrueAns).*100;
figure
loglog(numSectionsForIntegral,Q_HTF_error_percent);
xlabel('Number of slices along receiver length used for numerical
integration');
ylabel('Percentage Error in calculating Q absorbed by HTF compared
to 5000 slices');
%Measure time and accuracy difference between 517 slices and 5000
slices
tic
Q_HTF_5000Slices=sum(sim_Receiver_air_HTF_air_vac(5000,L,mdot,Q_in
,T_HTF_bulk,p_HTF,T_air,D_absorber,D_cover_inner,...

```

```

D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,w
indspeed,T_dewpoint_air,RelativeHumidity));
toc
tic
Q_HTF_517Slices=sum(sim_Receiver_air_HTF_air_vac(517,L,mdot,Q_in,T
_HTF_bulk,p_HTF,T_air,D_absorber,D_cover_inner,...

D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,w
indspeed,T_dewpoint_air,RelativeHumidity));
toc
CalculationErrorPercent = (Q_HTF_517Slices-
Q_HTF_5000Slices)/Q_HTF_517Slices*100;
%3.2 graphs
numSections=500;
L=40;
Q_solar=L*1000;
Q_in=Q_solar*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*F_
soiling;
Q_optical_losses=Q_solar-Q_in;
[Q_HTF,Q_loss_rad,Q_loss_conv,T_out,T_absorber,T_cover_inner,T_cov
er_outer,count,Lsection] = sim_Receiver_air_HTF_air_vac...

(numSections,L,mdot,Q_in,T_HTF_bulk,p_HTF,T_air,D_absorber,D_cover
_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,...

k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
Q_optical_losses_per_Lsize=Q_optical_losses/numSections;
Q_solar_per_Lsize=Q_solar/numSections;
Q_in_per_Lsize=Q_in/numSections;
eta_Q_HTF_instant=Q_HTF./Q_solar_per_Lsize;
Q_HTF_cumulative(numSections)=0;
Q_HTF_cumulative(1)=Q_HTF(1);
Q_losses_total=Q_optical_losses_per_Lsize+Q_loss_rad+Q_loss_conv;
for i=2:numSections
    Q_HTF_cumulative(i)=Q_HTF_cumulative(i-1)+Q_HTF(i);
end
eta_receiver_cumulative =
Q_HTF_cumulative./((Q_solar/L*Lsection));
frac_convect_losses=(Q_loss_conv./(Q_losses_total));
frac_rad_losses=(Q_loss_rad./(Q_losses_total));
frac_optical_losses=Q_optical_losses_per_Lsize./Q_losses_total;
figure
[hAx,hLine1,hLine2] =
plotyy(Lsection,T_out,Lsection,eta_receiver_cumulative);
xlabel('Length along Receiver [m]')
ylabel(hAx(1),'T_{HTF} [K]') % left y-axis
ylabel(hAx(2),'Cumulative Heat Collection Efficiency
\eta_{receiver}') % right y-axis
figure
plot(Lsection,eta_Q_HTF_instant,Lsection,frac_convect_losses,'r--
',Lsection,frac_rad_losses,'g-',...
Lsection,frac_optical_losses,'k:');
xlabel('Length along Receiver [m]')
legend('Instantaneous Receiver Efficiency
\eta_{receiver}','Instantaneous Convective Fraction of losses
\Re_{convection}',...
'Instantaneous Radiative Fraction of losses
\Re_{radiation}','Instantaneous Optical Fraction of losses
\Re_{optical}');
%Comparison of cover and no cover

```

```

Q_in=Q_solar*alpha_absorber*rho_mirror*gamma_tracking*F_soiling;%i
.e. tau_cover=1 for no cover
Q_optical_losses=Q_solar-Q_in;
Q_in_per_Lsize_nocover=Q_in/numSections;
Q_optical_losses_per_Lsize=Q_optical_losses/numSections;
Q_solar_per_Lsize=Q_solar/numSections;
[Q_HTF_nocover,Q_loss_rad_nocover,Q_loss_conv_nocover,T_out_nocove
r,T_absorber_nocover,count_nocover,Lsection_nocover] = ...

sim_Receiver_air_HTF_coverless(numSections,L,mdot,Q_in,T_HTF_bulk,
p_HTF,T_air,D_absorber,p_air,epsilon_absorber,windspeed,...
    T_dewpoint_air,RelativeHumidity);
eta_Q_HTF_instant_nocover=Q_HTF_nocover./Q_solar_per_Lsize;
Q_losses_total=Q_optical_losses_per_Lsize+Q_loss_rad_nocover+Q_los
s_conv_nocover;
frac_convect_losses_nocover=(Q_loss_conv_nocover./(Q_losses_total))
;
frac_rad_losses_nocover=(Q_loss_rad_nocover./(Q_losses_total));
frac_optical_losses_nocover=Q_optical_losses_per_Lsize./Q_losses_t
otal;
figure
plot(Lsection,eta_Q_HTF_instant_nocover,Lsection,frac_convect_losse
s_nocover,'r--',Lsection,frac_rad_losses_nocover,...
    'g-.',Lsection,frac_optical_losses_nocover,'k:');
xlabel('Length along Receiver [m]')
legend('Instantaneous Receiver Efficiency
\eta_{receiver}','Instantaneous Convective Fraction of losses
\Re_{convection}','...
    'Instantaneous Radiative Fraction of losses
\Re_{radiation}','Instantaneous Optical Fraction of losses
\Re_{optical}');
%Now do comparision, black body, W:Al2O3,AlyTi1-y(OxN1-
x),coverless Black
%body
alpha_absorber=1;epsilon_absorber=1;
Q_in=Q_solar*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*F_
soiling;
Q_optical_losses=Q_solar-Q_in;
Q_in_per_Lsize_blackbody=Q_in/numSections;
Q_optical_losses_per_Lsize=Q_optical_losses/numSections;
Q_solar_per_Lsize=Q_solar/numSections;
[Q_HTF_blackbody,Q_loss_rad_blackbody,Q_loss_conv_blackbody,T_out_
blackbody,T_absorber_blackbody,T_cover_inner_blackbody,...
    T_cover_outer_blackbody,count_blackbody,Lsection_blackbody] =
sim_Receiver_air_HTF_air_vac(numSections,L,mdot,Q_in,...

T_HTF_bulk,p_HTF,T_air,D_absorber,D_cover_inner,D_cover_outer,p_ai
r,p_vac,epsilon_absorber,k_cover,epsilon_cover,...
    windspeed,T_dewpoint_air,RelativeHumidity);
eta_Q_HTF_instant_blackbody=Q_HTF_blackbody./Q_solar_per_Lsize;
Q_losses_total=Q_optical_losses_per_Lsize+Q_loss_rad_blackbody+Q_l
oss_conv_blackbody;
frac_convect_losses_blackbody=(Q_loss_conv_blackbody./(Q_losses_tot
al));
frac_rad_losses_blackbody=(Q_loss_rad_blackbody./(Q_losses_total))
;
frac_optical_losses_blackbody=Q_optical_losses_per_Lsize./Q_losses
_total;
%AlyTi1-y(OxN1-x)
alpha_absorber=0.91;epsilon_absorber=0.14;
Q_in=Q_solar*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*F_
soiling;

```

```

Q_optical_losses=Q_solar-Q_in;
Q_in_per_Lsize_AlyTil=Q_in/numSections;
Q_optical_losses_per_Lsize=Q_optical_losses/numSections;
Q_solar_per_Lsize=Q_solar/numSections;
[Q_HTF_AlyTil,Q_loss_rad_AlyTil,Q_loss_conv_AlyTil,T_out_AlyTil,T_
absorber_AlyTil,T_cover_inner_AlyTil,...
    T_cover_outer_AlyTil,count_AlyTil,Lsection_AlyTil] =
sim_Receiver_air_HTF_air_vac(numSections,L,mdot,...

Q_in,T_HTF_bulk,p_HTF,T_air,D_absorber,D_cover_inner,D_cover_outer
,p_air,p_vac,epsilon_absorber,k_cover,...
    epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
eta_Q_HTF_instant_AlyTil=Q_HTF_AlyTil./Q_solar_per_Lsize;
Q_losses_total=Q_optical_losses_per_Lsize+Q_loss_rad_AlyTil+Q_loss
_conv_AlyTil;
frac_convex_losses_AlyTil=(Q_loss_conv_AlyTil./(Q_losses_total));
frac_rad_losses_AlyTil=(Q_loss_rad_AlyTil./(Q_losses_total));
frac_optical_losses_AlyTil=Q_optical_losses_per_Lsize./Q_losses_to
tal;
%Now compare 4 cases
figure
plot(Lsection,eta_Q_HTF_instant,Lsection,eta_Q_HTF_instant_nocover
,'r--',Lsection,eta_Q_HTF_instant_blackbody,...
    'g-.',Lsection,eta_Q_HTF_instant_AlyTil,'k:');
xlabel('Length along Receiver [m]')
ylabel('Instantaneous Receiver Efficiency \eta_{receiver}')
legend('W:Al_2O_3 with Pyrex cover','W:Al_2O_3 without
cover','Blackbody with Pyrex cover',...
    'Al_yTi_{1-y}(O_xN_{1-x}) with Pyrex cover');
%Now compare vac gap size for W:Al_2O_3 20m,
L=5;Q_solar=L*1000;
epsilon_absorber=0.1;alpha_absorber=0.9;
Q_in=Q_solar*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*F_
soiling;
D_gap=linspace(8/1000,100/1000,100);%gap spacing from 4mm to 50mm,
conventional is 12mm
D_cover_inner_compare=D_absorber+D_gap;
D_cover_outer_compare=D_cover_inner_compare+0.02;%10mm thick glass
Q_solar_per_Lsize=Q_solar/numSections;
Q_HTF_total(100)=0;Q_loss_rad_total(100)=0;Q_loss_conv_total(100)=
0;
Q_HTF_gap(100,numSections)=0;
Q_loss_rad_gap(100,numSections)=0;
Q_loss_conv_gap(100,numSections)=0;
T_absorber_gap(100,numSections)=0;
T_cover_outer_gap(100,numSections)=0;
T_cover_inner_gap(100,numSections)=0;
T_out_gap(100,numSections)=0;
for i=1:100

[Q_HTF,Q_loss_rad,Q_loss_conv,T_out,T_absorber,T_cover_inner,T_cov
er_outer,count,Lsection] = ...

sim_Receiver_air_HTF_air_vac(numSections,L,mdot,Q_in,T_HTF_bulk,p_
HTF,T_air,D_absorber,...

D_cover_inner_compare(i),D_cover_outer_compare(i),p_air,p_vac,epsil
on_absorber,k_cover,...
    epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
Q_HTF_gap(i,1:numSections)=Q_HTF;
Q_loss_rad_gap(i,1:numSections)=Q_loss_rad;
Q_loss_conv_gap(i,1:numSections)=Q_loss_conv;

```

```

T_out_gap(i,1:numSections)=T_out;
T_absorber_gap(i,1:numSections)=T_absorber;
T_cover_outer_gap(i,1:numSections)=T_cover_outer;
T_cover_inner_gap(i,1:numSections)=T_cover_inner;
Q_HTF_total(i)=sum(Q_HTF);
Q_loss_rad_total(i)=sum(Q_loss_rad);
Q_loss_conv_total(i)=sum(Q_loss_conv);
disp(i);
end
figure
plot(D_gap*1000/2,Q_HTF_total)
xlabel('Vacuum Gap Width [mm]')
ylabel('Total Q_{HTF}')
Lsize=Lsection(2)-Lsection(1);
Q_loss_rad_gap=Q_loss_rad_gap./Lsize;
Q_loss_conv_gap=Q_loss_conv_gap./Lsize;
Q_loss_total=Q_loss_conv_gap+Q_loss_rad_gap;
figure
plot(Lsection,Q_loss_rad_gap(1,1:end),Lsection,Q_loss_rad_gap(100,
1:end),'--',Lsection,...

Q_loss_conv_gap(1,1:end),Lsection,Q_loss_conv_gap(end,1:end),'--
',Lsection,Q_loss_total(1,1:end),...
':',Lsection,Q_loss_total(100,1:end),'-.')
xlabel('Length along Receiver [m]')
ylabel('Heat Loss [W/m]')
legend('Radiation Loss, Gap=4mm','Radiation Loss,
Gap=50mm','Convective Loss, Gap=4mm',...
'Convective Loss, Gap=50mm','Radiation+Convective Loss,
Gap=4mm','Radiation+Convective Loss, Gap=50mm')
figure
plot(Lsection,T_cover_outer_gap(1,1:end),Lsection,T_cover_outer_ga
p(end,1:end),'--',...

Lsection,T_cover_inner_gap(1,1:end),'-.',Lsection,T_cover_inner_ga
p(end,1:end),':')
xlabel('Length along Receiver [m]')
ylabel('Cover Temperatures [K]')
legend('Cover Outer, Gap=4mm','Cover Outer, Gap=50mm','Cover
Inner, Gap=4mm','Cover Inner, Gap=50mm')
%Incidence examples Wits on 15th Feb 2019
dayInt=43511;Latitude=-26.187829;Longitude=28.028137;TimeZone=2;
L=5;Q_solar=5000;T_HTF_bulk=80+273;p_HTF=2*101325;T_air=25+273;
D_absorber=0.045;D_cover_inner=D_absorber+(12/1000);D_cover_outer=
D_cover_inner+(20/1000);
p_air=101325;p_vac=0.013;epsilon_absorber=0.1;k_cover=1.005;epsilo
n_cover=0.86;
windspeed=2;T_dewpoint_air=14+273;RelativeHumidity=50;mdot=0.01;
alpha_absorber=0.9;rho_mirror=0.93;tau_cover=0.96;gamma_tracking=0
.94;
F_soiling=0.97;numSections=500;
Q_in=Q_solar*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*F_
soiling;
troughSteps=500;secondsPerTimeStep=(24/troughSteps)*60*60;
timeOfDayFraction=linspace(0,1,troughSteps);
Q_HTF_incidenceEW(troughSteps)=0;
theta_degEW(troughSteps)=0;K_EW(troughSteps)=0;
for i=1:troughSteps
    [corectedSolarElevationDeg, solarAzimuthEofNDeg, ~] =
solarEleAzi(dayInt, timeOfDayFraction(i),...
    Latitude, Longitude, TimeZone);
    troughElivationRad=degtorad(corectedSolarElevationDeg);

```

```

        troughAzimuthRad=0; %North Facing
        troughvect=[sin(troughAzimuthRad)*cos(troughElivationRad)
cos(troughAzimuthRad)*cos(troughElivationRad)
sin(troughElivationRad)];
        sunElivationRad=deg2rad(corectedSolarElevationDeg);
        sunAzimuthRad=deg2rad(solarAzimuthEofNDeg);
        azimuthDifferenceRad=abs(sunAzimuthRad-troughAzimuthRad);
        elevationDifferenceRad=abs(sunElivationRad-
troughElivationRad);
        if(azimuthDifferenceRad>pi)
            azimuthDifferenceRad=2*pi-azimuthDifferenceRad;
        end

        if(elevationDifferenceRad>pi)
            elevationDifferenceRad=2*pi-elevationDifferenceRad;
        end
        sunvect=[sin(sunAzimuthRad)*cos(sunElivationRad)
cos(sunAzimuthRad)*cos(sunElivationRad) sin(sunElivationRad)];
        %tan(theta)=norm((u x v) / (u dot v))
        theta_degEW(i) =
atan2d(norm(cross(troughvect,sunvect)),dot(troughvect,sunvect));
        K_EW(i)=calcKincidenceIST(theta_degEW(i));
        if(corectedSolarElevationDeg>0 && theta_degEW(i)<90 &&
K_EW(i)>0 && azimuthDifferenceRad<=pi/2 &&
elevationDifferenceRad<=pi/2)%Sun must be up and infront of PTC,
glazing not total reflect DNI

Q_in=Q_solar*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*F_
soiling*K_EW(i);

Q_HTF_incidenceEW(i)=sum(sim_Receiver_air_HTF_air_vac(numSections,
L,mdot,Q_in,T_HTF_bulk,...

p_HTF,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,eps
ilon_absorber,k_cover,...

epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity));
    end
    disp(['Incidence Calc E-W ', num2str(i), ' of ',
num2str(troughSteps)])
end
%Solar Transit Logics
Q_HTF_incidenceNS(troughSteps)=0;
theta_degNS(troughSteps)=0;K_NS(troughSteps)=0;
for i=1:troughSteps
    [corectedSolarElevationDeg, solarAzimuthEofNDeg, ~] =
solarEleAzi(dayInt, timeOfDayFraction(i),...
Latitude, Longitude, TimeZone);
    troughElivationRad=deg2rad(corectedSolarElevationDeg);
    if(solarAzimuthEofNDeg<180)
        troughAzimuthRad=deg2rad(90);%i.e. East facing
    else
        troughAzimuthRad=deg2rad(270);%i.e. West facing
    end
    troughvect=[sin(troughAzimuthRad)*cos(troughElivationRad)
cos(troughAzimuthRad)*cos(troughElivationRad)
sin(troughElivationRad)];
    sunElivationRad=deg2rad(corectedSolarElevationDeg);
    sunAzimuthRad=deg2rad(solarAzimuthEofNDeg);
    azimuthDifferenceRad=abs(sunAzimuthRad-troughAzimuthRad);
    elevationDifferenceRad=abs(sunElivationRad-
troughElivationRad);

```

```

        if(azimuthDifferenceRad>pi)
            azimuthDifferenceRad=2*pi-azimuthDifferenceRad;
        end

        if(elevationDifferenceRad>pi)
            elevationDifferenceRad=2*pi-elevationDifferenceRad;
        end
        sunvect=[sin(sunAzimuthRad)*cos(sunElivationRad)
cos(sunAzimuthRad)*cos(sunElivationRad) sin(sunElivationRad)];
        %tan(theta)=norm((u x v) / (u dot v))
        theta_degNS(i) =
atan2d(norm(cross(troughvect,sunvect)),dot(troughvect,sunvect));
        K_NS(i)=calcKincidenceIST(theta_degNS(i));
        if(corectedSolarElevationDeg>0 && theta_degNS(i)<90 &&
K_NS(i)>0 && azimuthDifferenceRad<=pi/2 &&
elevationDifferenceRad<=pi/2)%Sun must be up and infront of PTC,
glazing not total reflect DNI

Q_in=Q_solar*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*F_
soiling*K_NS(i);

Q_HTF_incidenceNS(i)=sum(sim_Receiver_air_HTF_air_vac(numSections,
L,mdot,Q_in,T_HTF_bulk,...

p_HTF,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,eps
ilon_absorber,...

k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity));
    end
    disp(['Incidence Calc N-S ', num2str(i), ' of ',
num2str(troughSteps)])
end
E_HTF_kWh_EW=sum((Q_HTF_incidenceEW(Q_HTF_incidenceEW>0)).*seconds
PerTimeStep)/3600000;
E_HTF_kWh_NS=sum((Q_HTF_incidenceNS(Q_HTF_incidenceNS>0)).*seconds
PerTimeStep)/3600000;
figure
plot(timeOfDayFraction*24,Q_HTF_incidenceNS,timeOfDayFraction*24,Q
_HTF_incidenceEW,'--')
xlabel('Time of Day')
ylabel('Q_{HTF} [W]')
legend('N-S Oriented: Total collection = 38.18 kWh', 'E-W
Oriented: Total collection = 21.62 kWh')
dayInt=43637; %Winter Solstice at Wits, June 21st 2019
Q_HTF_incidenceEW=0;theta_degEW=0;K_EW=0;
Q_HTF_incidenceEW(troughSteps)=0;
theta_degEW(troughSteps)=0;K_EW(troughSteps)=0;
for i=1:troughSteps
    [corectedSolarElevationDeg, solarAzimuthEofNDeg, ~] =
solarEleAzi(dayInt, timeOfDayFraction(i), Latitude, Longitude,
TimeZone);
    troughElivationRad=degtorad(corectedSolarElevationDeg);
    troughAzimuthRad=0; %North Facing
    troughvect=[sin(troughAzimuthRad)*cos(troughElivationRad)
cos(troughAzimuthRad)*cos(troughElivationRad)
sin(troughElivationRad)];
    sunElivationRad=degtorad(corectedSolarElevationDeg);
    sunAzimuthRad=degtorad(solarAzimuthEofNDeg);
    azimuthDifferenceRad=abs(sunAzimuthRad-troughAzimuthRad);
    elevationDifferenceRad=abs(sunElivationRad-
troughElivationRad);
    if(azimuthDifferenceRad>pi)

```

```

        azimuthDifferenceRad=2*pi-azimuthDifferenceRad;
    end

    if(elevationDifferenceRad>pi)
        elevationDifferenceRad=2*pi-elevationDifferenceRad;
    end
    sunvect=[sin(sunAzimuthRad)*cos(sunElivationRad)
cos(sunAzimuthRad)*cos(sunElivationRad) sin(sunElivationRad)];
    %tan(theta)=norm((u x v) / (u dot v))
    theta_degEW(i) =
atan2d(norm(cross(troughvect,sunvect)),dot(troughvect,sunvect));
    K_EW(i)=calcKincidenceIST(theta_degEW(i));
    if(corectedSolarElevationDeg>0 && theta_degEW(i)<90 &&
K_EW(i)>0 && azimuthDifferenceRad<=pi/2 &&
elevationDifferenceRad<=pi/2)%Sun must be up and infront of PTC,
glazing not total reflect DNI

Q_in=Q_solar*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*F_
soiling*K_EW(i);

Q_HTF_incidenceEW(i)=sum(sim_Receiver_air_HTF_air_vac(numSections,
L,mdot,Q_in,T_HTF_bulk,p_HTF,T_air,D_absorber,D_cover_inner,D_cove
r_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,windspe
ed,T_dewpoint_air,RelativeHumidity));
    end
    disp(['Winter Incidence Calc E-W ', num2str(i), ' of ',
num2str(troughSteps)])
    end
    %Solar Transit Logics
    Q_HTF_incidenceNS=0;
    theta_degNS=0;K_NS=0;
    Q_HTF_incidenceNS(troughSteps)=0;
    theta_degNS(troughSteps)=0;K_NS(troughSteps)=0;
    for i=1:troughSteps
        [corectedSolarElevationDeg, solarAzimuthEofNDeg, ~] =
solarEleAzi(dayInt, timeOfDayFraction(i), Latitude, Longitude,
TimeZone);
        troughElivationRad=degtorad(corectedSolarElevationDeg);
        if(solarAzimuthEofNDeg<180)
            troughAzimuthRad=degtorad(90);%i.e. East facing
        else
            troughAzimuthRad=degtorad(270);%i.e. West facing
        end
        troughvect=[sin(troughAzimuthRad)*cos(troughElivationRad)
cos(troughAzimuthRad)*cos(troughElivationRad)
sin(troughElivationRad)];
        sunElivationRad=degtorad(corectedSolarElevationDeg);
        sunAzimuthRad=degtorad(solarAzimuthEofNDeg);
        azimuthDifferenceRad=abs(sunAzimuthRad-troughAzimuthRad);
        elevationDifferenceRad=abs(sunElivationRad-
troughElivationRad);
        if(azimuthDifferenceRad>pi)
            azimuthDifferenceRad=2*pi-azimuthDifferenceRad;
        end

        if(elevationDifferenceRad>pi)
            elevationDifferenceRad=2*pi-elevationDifferenceRad;
        end
        sunvect=[sin(sunAzimuthRad)*cos(sunElivationRad)
cos(sunAzimuthRad)*cos(sunElivationRad) sin(sunElivationRad)];
        %tan(theta)=norm((u x v) / (u dot v))

```

```

        theta_degNS(i) =
atan2d(norm(cross(troughvect,sunvect)),dot(troughvect,sunvect));
        K_NS(i)=calcKincidenceIST(theta_degNS(i));
        if(corectedSolarElevationDeg>0 && theta_degNS(i)<90 &&
K_NS(i)>0 && azimuthDifferenceRad<=pi/2 &&
elevationDifferenceRad<=pi/2)%Sun must be up and infront of PTC,
glazing not total reflect DNI

Q_in=Q_solar*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*F_
soiling*K_NS(i);

Q_HTF_incidenceNS(i)=sum(sim_Receiver_air_HTF_air_vac(numSections,
L,mdot,Q_in,T_HTF_bulk,p_HTF,T_air,D_absorber,D_cover_inner,D_cove
r_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,windspe
ed,T_dewpoint_air,RelativeHumidity));
    end
    disp(['Winter Incidence Calc N-S ', num2str(i), ' of ',
num2str(troughSteps)])
end
E_HTF_kWh_EW_winter=sum((Q_HTF_incidenceEW(Q_HTF_incidenceEW>0)).*
secondsPerTimeStep)/3600000;
E_HTF_kWh_NS_winter=sum((Q_HTF_incidenceNS(Q_HTF_incidenceNS>0)).*
secondsPerTimeStep)/3600000;
figure
plot(timeOfDayFraction*24,Q_HTF_incidenceNS,timeOfDayFraction*24,Q
_HTF_incidenceEW,'--')
xlabel('Time of Day')
ylabel('Q_{HTF} [W]')
legend('N-S Oriented: Total collection = 27.92 kWh', 'E-W
Oriented: Total collection = 28.50 kWh')
%Now do carnot engine, total thermal efficiency
L_list=linspace(0,30,100);
Q_solar_list=L_list*1000;
Q_HTF_L_opti(100)=0;
T_out_L_opti(100)=0;
T_HTF_bulk=T_air;
eta_thermal_L_opti(100)=0;
for i=1:100

Q_in=Q_solar_list(i)*alpha_absorber*rho_mirror*tau_cover*gamma_tra
cking*F_soiling;

[Q_HTF,~,~,T_out,~,~,~,~,~]=sim_Receiver_air_HTF_air_vac(numSectio
ns,L_list(i),mdot,Q_in,T_HTF_bulk,p_HTF,T_air,D_absorber,D_cover_i
nner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_co
ver,windspeed,T_dewpoint_air,RelativeHumidity);
    Q_HTF_L_opti(i)=sum(Q_HTF);
    T_out_L_opti(i)=T_out(end);
    disp(['Arbitrary Receiver Length Opti ', num2str(i), ' of
100']);
    eta_thermal_L_opti(i)=Q_HTF_L_opti(i).*(1-
((T_air)./(T_out_L_opti(i))))./Q_solar_list(i);
end
figure
[hAx,hLine1,hLine2] =
plotyy(L_list,eta_thermal_L_opti,L_list,T_out_L_opti);
xlabel('Lenght of Receiver [m]')
ylabel(hAx(1), '\eta_{thermal}') % left y-axis
ylabel(hAx(2), 'T_{HTF,outlet} [K]') % right y-axis
%Parametric Analysis Start with parameterizing Length and Width
numParametricFineness=100;%this.... takes a while
CR_max=100;%Conveniently Selected beforehand for nice graphs :)

```

```

L_max=15;%Conveniently Selected beforehand for nice graphs :)
%CR=Aperture/Image | ... Aperture=Image.CR=(pi*D/2) CR
Width_CR_max=pi.*D_absorber.*0.5.*CR_max;
L_list= linspace(0,L_max,numParametricFineness+1);
Width_list= linspace(0,Width_CR_max,numParametricFineness+1);
L_list=L_list(2:end);%i.e. no 0 lengths
Width_list=Width_list(2:end);%i.e. no 0 widths
L_result_vector(numParametricFineness.^2)=0;
Width_result_vector(numParametricFineness.^2)=0;
Wcarnot_result_vector(numParametricFineness.^2)=0;
Eta_thermal_result_vector(numParametricFineness.^2)=0;
Tout_result_vector(numParametricFineness.^2)=0;
position=0;
I=1000;%W/m^2
for m=1:numParametricFineness
    for n=1:numParametricFineness
        position=position+1;
        L_now=L_list(m);
        W_now=Width_list(n);
        L_result_vector(position)=L_now;
        Width_result_vector(position)=W_now;
        Q_solar=I.*L_now.*W_now;

Q_in=Q_solar.*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*F_soiling;

[Q_HTF,~,~,T_out,~,~,~,~,~]=sim_Receiver_air_HTF_air_vac(numSections,L_now,mdot,Q_in,T_HTF_bulk,p_HTF,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
        Q_HTF=sum(Q_HTF);
        Tout_result_vector(position)=(T_out(end));
        Wcarnot_result_vector(position)=Q_HTF.*(1-((T_air)./(Tout_result_vector(position))));

Eta_thermal_result_vector(position)=Wcarnot_result_vector(position)./Q_solar;
        disp(['Parametric Analysis Length vs Width ',
num2str(position), ' of ', num2str(numParametricFineness.^2)]);
    end
end
x=L_result_vector(:);
y=Width_result_vector(:);
z=Eta_thermal_result_vector(:);
dx=L_list(2)-L_list(1);
dy=Width_list(2)-Width_list(1);
x_edge=min(x):dx:max(x);
y_edge=min(y):dy:max(y);
[X,Y]=meshgrid(x_edge,y_edge);
F = TriScatteredInterp(x,y,z);
Z= F(X,Y);
figure
mesh(X,Y,Z)
xlabel('Length of Receiver [m]');ylabel('Reciever Aperture Width [m]');zlabel('\eta_{thermal}');
figure
contour(X,Y,Z,[0.15 0.2 0.25 0.3 0.31 0.32 0.33 0.332], 'ShowText', 'on')
xlabel('Length of Receiver [m]');ylabel('Reciever Aperture Width [m]');
legend('\eta_{thermal}');
%Now parameterize mdot and L

```

```

mdot_result_vector=0;
numParametricFineness=100;
mdot_list=linspace(0,3,numParametricFineness+1);
mdot_list=mdot_list(2:end);
mdot_result_vector(numParametricFineness.^2)=0;
L_list=linspace(0,1600,numParametricFineness+1);
L_list=L_list(2:end);%i.e. no 0 lengths
L_result_vector=0;L_result_vector(numParametricFineness.^2)=0;
Wcarnot_result_vector=0;Wcarnot_result_vector(numParametricFineness.^2)=0;
Eta_thermal_result_vector=0;Eta_thermal_result_vector(numParametricFineness.^2)=0;
Tout_result_vector=0;Tout_result_vector(numParametricFineness.^2)=0;
position=0;
for m=1:numParametricFineness
    for n=1:numParametricFineness
        position=position+1;
        L_now=L_list(m);
        mdot_now=mdot_list(n);
        L_result_vector(position)=L_now;
        mdot_result_vector(position)=mdot_now;
        Q_solar=I.*L_now.*1.4137;

Q_in=Q_solar.*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*_soiling;

[Q_HTF,~,~,T_out,~,~,~,~,~]=sim_Receiver_air_HTF_air_vac(numSections,L_now,mdot_now,Q_in,T_HTF_bulk,p_HTF,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
        Q_HTF=sum(Q_HTF);
        Tout_result_vector(position)=(T_out(end));
        Wcarnot_result_vector(position)=Q_HTF.*(1-((T_air)./(Tout_result_vector(position))));

Eta_thermal_result_vector(position)=Wcarnot_result_vector(position)./Q_solar;
        disp(['Parametric Analysis Length vs mdot ',
num2str(position), ' of ', num2str(numParametricFineness.^2)]);
    end
end
x=L_result_vector(:);
y=mdot_result_vector(:);
z=Eta_thermal_result_vector(:);
dx=L_list(2)-L_list(1);
dy=mdot_list(2)-mdot_list(1);
x_edge=min(x):dx:max(x);
y_edge=min(y):dy:max(y);
[X,Y]=meshgrid(x_edge,y_edge);
F = TriScatteredInterp(x,y,z);
Z= F(X,Y);
figure
mesh(X,Y,Z)
xlabel('Length of Receiver [m]');ylabel('Air Mass Flow Rate [kg/s]');zlabel('\eta_{thermal}');
figure
contour(X,Y,Z,[0.15 0.2 0.25 0.3 0.35 0.4 0.412],'ShowText','on');
xlabel('Length of Receiver [m]');ylabel('Air Mass Flow Rate [kg/s]');
legend('\eta_{thermal}');

```

```

xlabel('Length of Receiver [m]');ylabel('Air Mass Flow Rate
[kg/s]');
legend('\eta_{thermal}');
%Wind Examples
L=6.75;W=1.414;Q_solar=L.*W.*1000;p_HTF=2*101325;T_air=25+273;T_HT
F_bulk=T_air;
D_absorber=0.045;D_cover_inner=D_absorber+(12/1000);D_cover_outer=
D_cover_inner+(20/1000);
p_air=101325;p_vac=0.013;epsilon_absorber=0.1;k_cover=1.005;epsilo
n_cover=0.86;
windspeed=2;T_dewpoint_air=14+273;RelativeHumidity=50;mdot=0.01;
alpha_absorber=0.9;rho_mirror=0.93;tau_cover=0.96;gamma_tracking=0
.94;
F_soiling=0.97;numSections=500;chartFineness=200;
windspeed_list=linspace(0.01,10,chartFineness);
Q_in=Q_solar*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*F_
soiling;
Q_HTF_covered(chartFineness)=0;Q_HTF_coverless(chartFineness)=0;
for i=1:chartFineness
    [Q_HTF,~,~,~,~,~,~,~,~] =
sim_Receiver_air_HTF_air_vac(numSections,L,mdot,Q_in,T_HTF_bulk,p_
HTF,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsil
on_absorber,k_cover,epsilon_cover,windspeed_list(i),T_dewpoint_air
,RelativeHumidity);
    Q_HTF_covered(i)=sum(Q_HTF);
    [Q_HTF,~,~,~,~,~,~,~] =
sim_Receiver_air_HTF_coverless(numSections,L,mdot,Q_in,T_HTF_bulk,
p_HTF,T_air,D_absorber,p_air,epsilon_absorber,windspeed_list(i),T_
dewpoint_air,RelativeHumidity);
    Q_HTF_coverless(i)=sum(Q_HTF);
    disp(['Wind effect cover and coverless ', num2str(i), ' of ',
num2str(chartFineness)]);
end
eta_receiver_covered=Q_HTF_covered./Q_solar;
eta_receiver_coverless=Q_HTF_coverless./Q_solar;
figure
plot(windspeed_list,eta_receiver_covered,windspeed_list,eta_receiv
er_coverless,'--')
xlabel('Windspeed [m/s]')
ylabel('Cumulative Collector Efficiency \eta_{collector}')
legend('With Cover', 'Without Cover')
%Liquid phase example Marlotherm SH
D_absorber=0.045;D_cover_inner=D_absorber+(12/1000);D_cover_outer=
D_cover_inner+(20/1000);
mdot=3.5;
numParametricFineness=50;%this.... takes a while
CR_max=100;%60
L_max=800;%440
%CR=Aperture/Image | ... Aperture=Image.CR=(pi*D/2)CR
Width_CR_max=pi.*D_absorber.*0.5.*CR_max;
L_list=linspace(0,L_max,numParametricFineness+1);
Width_list=linspace(0,Width_CR_max,numParametricFineness+1);
L_list=L_list(2:end);%i.e. no 0 lengths
Width_list=Width_list(2:end);%i.e. no 0 widths
L_result_vector=0;L_result_vector(numParametricFineness.^2)=0;
Width_result_vector=0;Width_result_vector(numParametricFineness.^2
)=0;
Wcarnot_result_vector=0;Wcarnot_result_vector(numParametricFinenes
s.^2)=0;
Eta_thermal_result_vector=0;Eta_thermal_result_vector(numParametri
cFineness.^2)=0;

```

```

Tout_result_vector=0;Tout_result_vector(numParametricFineness.^2)=
0;
position=0;
I=1000;%W/m^2
for m=1:numParametricFineness
    for n=1:numParametricFineness
        position=position+1;
        L_now=L_list(m);
        W_now=Width_list(n);
        L_result_vector(position)=L_now;
        Width_result_vector(position)=W_now;
        Q_solar=I.*L_now.*W_now;

Q_in=Q_solar.*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*F
_soiling;

[Q_HTF,~,~,T_out,~,~,~,~,~]=sim_Receiver_marlothermSH_HTF_air_vac(
numSections,L_now,mdot,Q_in,T_HTF_bulk,T_air,D_absorber,D_cover_in
ner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cov
er,windspeed,T_dewpoint_air,RelativeHumidity);
        Q_HTF=sum(Q_HTF);
        Tout_result_vector(position)=(T_out(end));
        Wcarnot_result_vector(position)=Q_HTF.*(1-
((T_air)./(Tout_result_vector(position))));

Eta_thermal_result_vector(position)=Wcarnot_result_vector(position
)./Q_solar;
        disp(['Parametric Analysis Length vs Width ',
num2str(position), ' of ', num2str(numParametricFineness.^2)]);
    end
end
x=L_result_vector(:);
y=Width_result_vector(:);
z=Eta_thermal_result_vector(:);
t=Tout_result_vector(:);
x=x(t<623.15); %reject temps above oil marlotherm max temp
y=y(t<623.15);
z=z(t<623.15);
t=t(t<623.15);
x_edge=min(x):dx:max(x);
y_edge=min(y):dy:max(y);
[X,Y]=meshgrid(x_edge,y_edge);
F = TriScatteredInterp(x,y,z);
Z= F(X,Y);
figure
mesh(X,Y,Z)
xlabel('Length of Receiver [m]');ylabel('Reciever Aperture Width
[m]');zlabel('\eta_{thermal}');
figure
contour(X,Y,Z,'ShowText','on')
xlabel('Length of Receiver [m]');ylabel('Reciever Aperture Width
[m]');
legend('\eta_{thermal}');
%----- Chapter 4 Simulate a Garrett GT0632SZ 32mm
p_HTF=2*101325;T_air=25+273;T_HTF_bulk=T_air;
p_air=101325;p_vac=0.013;epsilon_absorber=0.1;k_cover=1.005;epsilo
n_cover=0.86;
windspeed=2;T_dewpoint_air=14+273;RelativeHumidity=50;
alpha_absorber=0.9;rho_mirror=0.93;tau_cover=0.96;gamma_tracking=0
.94;F_soiling=0.97;
D_absorber=0.032;D_cover_inner=D_absorber+(12/1000);D_cover_outer=
D_cover_inner+(20/1000);

```

```

mdot=5*0.45359237/60;
I=1000;W=6;numSections=500;
eta_compressor=.68;eta_turbine=.56;p_in=101325;p_operating=p_in*2;
p_out=p_in;
L=linspace(1,50,100);Tin=T_HTF_bulk;
P_out=0;P_out(100)=0;Tout_turb=0;Tout_turb(100)=0;Q_solar=0;Q_solar(100)=0;
for i=1:100
    Q_solar(i)=I.*L(i).*W;

Q_in(i)=Q_solar(i).*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*F_soiling;
    [P_out(i),Tout_turb(i)] =
sim_SEC_Turbocharger_air_covered(eta_compressor,eta_turbine,p_in,p_operating,p_out,numSections,L(i),mdot,Q_in(i),Tin,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
    disp(i);
end
P_out(P_out<0)=0;%negative power output is no power output - much easier to understand graph
etaEngine = P_out./Q_solar;
figure
yyaxis left
plot(L,Tout_turb)
xlabel('Length of Receiver [m]');
ylabel('Turbine Outlet Temperature [K]');
yyaxis right
plot(L,etaEngine,'--')
xlabel('Length of Receiver [m]');
ylabel('\eta_{thermal}');
legend('Turbine Outlet Temperature','\eta_{thermal}')
fineness=100;
vectorLength=fineness.*fineness;%Optimise for grain depth vs. processing time in dissertation. 50x50 is pretty good balance
P_out=0;P_out(vectorLength)=0;Tout_turb=0;Tout_turb(vectorLength)=0;Q_solar=0;Q_in=0;
Q_solar(vectorLength)=0;Q_in(vectorLength)=0;L=0;L(vectorLength)=0;W=0;W(vectorLength)=0;
Wlist=linspace(1,28,fineness);
Llist=linspace(1,20,fineness);
position=1;
for i=1:fineness
    for n=1:fineness
        L(position)=Llist(i);
        W(position)=Wlist(n);
        position=position+1;
    end
end
parfor i=1:vectorLength
    Q_solar(i)=I.*L(i).*W(i);

Q_in(i)=Q_solar(i).*alpha_absorber*rho_mirror*tau_cover*gamma_tracking*F_soiling;
    [P_out(i),Tout_turb(i)] =
sim_SEC_Turbocharger_air_covered(eta_compressor,eta_turbine,p_in,p_operating,p_out,numSections,L(i),mdot,Q_in(i),Tin,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
end
P_out(P_out<0)=0;
etaEngine = P_out./Q_solar;

```

```

max(etaEngine)
W(etaEngine==max(etaEngine))
L(etaEngine==max(etaEngine))
x=W(:);
y=L(:);
z=etaEngine(:);
dx=Wlist(2)-Wlist(1);
dy=Llist(2)-Llist(1);
x_edge=min(x):dx:max(x);
y_edge=min(y):dy:max(y);
[X,Y]=meshgrid(x_edge,y_edge);
F = TriScatteredInterp(x,y,z);
Z= F(X,Y);
figure
contour(X,Y,Z,'ShowText','on')
xlabel('Reciever Aperture Width [m]');ylabel('Length of Receiver [m]')
legend('\eta_{thermal}');
%Big turbo GTX3584RS
%          comp      turb      PR      lb/min  turbine inducer
%GTX3584RS  0.76      0.78      2.25      50       68 mm GTX5008R  0.8
0.76
%2.25      75       99 mm
fineness=100; %how many steps along x axis for graphs.
D_absorber=0.068;D_cover_inner=D_absorber+(12/1000);D_cover_outer=
D_cover_inner+(20/1000);
p_air=101325;p_vac=0.013;epsilon_absorber=0.1;k_cover=1.005;epsilo
n_cover=0.86;
mdot=50*0.45359237/60;p_in=101325;p_operating=p_in*2.25;p_out=p_in
;I=1000;numSections=500;
eta_compressor=.76;eta_turbine=.78;W=8;windspeed=5;T_air=25+273;T_
HTF_bulk=T_air;
alpha_absorber=0.9;rho_mirror=0.93;tau_cover=0.96;gamma_tracking=0
.94;F_soiling=0.97;
L=linspace(1,200,fineness);Tin=T_HTF_bulk;Q_in=0;Q_in(fineness)=0;
T_dewpoint_air=14+273;RelativeHumidity=50;
P_out=0;P_out(fineness)=0;Tout_turb=0;Tout_turb(fineness)=0;Q_sola
r=0;Q_solar(fineness)=0;
parfor i=1:fineness
    Q_solar(i)=I.*L(i).*W;

Q_in(i)=Q_solar(i).*alpha_absorber*rho_mirror*tau_cover*gamma_trac
king*F_soiling;
    [P_out(i),Tout_turb(i)] =
sim_SEC_Turbocharger_air_covered(eta_compressor,eta_turbine,p_in,p
_operating,p_out,numSections,L(i),mdot,Q_in(i),Tin,T_air,D_absorbe
r,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover
,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
    %disp(i);
end
P_out(P_out<0)=0;%negative power ouput is no power output - much
easier to understand graph
etaEngine = P_out./Q_solar;
figure
yyaxis left
plot(L,Tout_turb)
xlabel('Length of Receiver [m]');
ylabel('Turbine Outlet Temperature [K]');
yyaxis right
plot(L,etaEngine,'--')
xlabel('Length of Receiver [m]');
ylabel('\eta_{thermal}');

```

```

legend('Turbine Outlet Temperature','\eta_{thermal}')
%Now need to find optimal L for a range of mdots for GTX3584RS
using PR
%function.
mdot=linspace(22.53,77.33,fineness);%in lb/min
eta_comp(fineness)=0;
eta_comp((mdot<25.17)|(mdot>=59.95))=0.74;
eta_comp((mdot>=25.17&mdot<29.77)|(mdot>=56.22&mdot<59.95))=0.75;
eta_comp(mdot>=63.87)=0.73;
eta_comp(mdot>=68.13)=0.72;
eta_comp(mdot>=72.67)=0.71;
eta_comp(eta_comp==0)=0.76;
p_operating=p_in.*(0.040347785.*mdot+0.492855643);
mdot=mdot*0.45359237/60; %convert to kg/s
%p_total and p_static are very very close (6th sig fig different)
for
%compressor outlet for temps between ambient and 700K, so assume
PR
%total/total is equal to static. This saves recursive iterative
functions
%to find "real" total PR.
mdot_corr_Factor =
calcMdot_corr_air(1,D_absorber,p_in,T_air);%factor for division to
mdot real
%mdot_corr = mdot .* f(D_abosorber,p_in,T_in) = mdot .*
mdot_corr_Factor
mdot=mdot./mdot_corr_Factor;
L_optimal=0;L_optimal(fineness)=0;etaEngine_optimal=0;etaEngine_op
timal(fineness)=0;
L=linspace(1,200,fineness);
for n=1:fineness
    mdot_now=mdot(n);
    eta_comp_now=eta_comp(n);
    p_operating_now=p_operating(n);
    parfor i=1:fineness
        Q_solar(i)=I.*L(i).*W;

Q_in(i)=Q_solar(i).*alpha_absorber*rho_mirror*tau_cover*gamma_trac
king*F_soiling;
        [P_out(i),Tout_turb(i)] =
sim_SEC_Turbocharger_air_covered(eta_comp_now,eta_turbine,p_in,p_o
perating_now,p_out,numSections,L(i),mdot_now,Q_in(i),Tin,T_air,D_a
bsorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k
_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
    end
    P_out(P_out<0)=0;%negative power ouput is no power output -
much easier to understand graph
    etaEngine = P_out./Q_solar;
    etaEngine_optimal(n)=max(etaEngine);
    L_optimal(n)=L(etaEngine==max(etaEngine));
    disp(['Length for optimal engine eta GTX3584RS for various
flow rates ',num2str(n),' of ', num2str(fineness),' completed']);
end
figure
yyaxis left
plot(mdot,L_optimal)
xlabel('True Mass flow');
ylabel('Optimal Receiver Length [m]');
yyaxis right
plot(mdot,etaEngine_optimal,'.')
xlabel('True Mass flow [kg/s]');
ylabel('Optimal \eta_{thermal}');

```

```

legend('Optimal Receiver Length','Optimal \eta_{thermal}')
%Now do GTX5533R GEN II 98mm
%           comp      turb      PR      lb/min  turbine inducer
%GTX5533RGII.98  0.76    0.74    3.5     160      112 mm
fineness=100;numSections=500;
D_absorber=0.112;D_cover_inner=D_absorber+(12/1000);D_cover_outer=
D_cover_inner+(20/1000);
p_air=101325;p_vac=0.013;epsilon_absorber=0.1;k_cover=1.005;epsilo
n_cover=0.86;p_in=101325;
p_out=p_in;I=1000;eta_compressor=.76;eta_turbine=.74;W=8;windspeed
=5;T_air=25+273;T_HTF_bulk=T_air;
alpha_absorber=0.9;rho_mirror=0.93;tau_cover=0.96;gamma_tracking=0
.94;F_soiling=0.97;
Tin=T_HTF_bulk;Q_in=0;Q_in(fineness)=0;T_dewpoint_air=14+273;Relat
iveHumidity=50;
P_out=0;P_out(fineness)=0;Tout_turb=0;Tout_turb(fineness)=0;Q_sola
r=0;Q_solar(fineness)=0;
mdot=linspace(56.13559322,198.5084746,fineness);%in lb/min
eta_comp(fineness)=0;%reading compressor map reading along x axis
left to right
eta_comp(mdot>=56)=0.73;
eta_comp(mdot>=64.10847458)=0.74;
eta_comp(mdot>=76.47457627)=0.75;
eta_comp(mdot>=109.3423729)=0.76;
eta_comp(mdot>=176.5423729)=0.75;
eta_comp(mdot>=185.979661)=0.74;
eta_comp(mdot>=192.8135593)=0.73;
p_operating=p_in.*(7.98261E-07.*(mdot.^3)-
0.000165779.*(mdot.^2)+0.026722678.*mdot+0.245310835);
mdot=mdot*0.45359237/60;%convert to kg/s
%p_total and p_static are very very close (6th sig fig different)
for
%compressor outlet for temps between ambient and 700K, so assume
PR
%total/total is equal to static. This saves recursive iterative
functions
%to find "real" total PR.
mdot_corr_Factor =
calcMdot_corr_air(1,D_absorber,p_in,T_air);%factor for division to
mdot real
%mdot_corr = mdot .* f(D_abosorber,p_in,T_in) = mdot .*
mdot_corr_Factor
mdot=mdot./mdot_corr_Factor;
L_optimal=0;L_optimal(fineness)=0;etaEngine_optimal=0;etaEngine_op
timal(fineness)=0;
L=linspace(1,300,fineness);
for n=1:fineness
    mdot_now=mdot(n);
    eta_comp_now=eta_comp(n);
    p_operating_now=p_operating(n);
    parfor i=1:fineness
        Q_solar(i)=I.*L(i).*W;

Q_in(i)=Q_solar(i).*alpha_absorber*rho_mirror*tau_cover*gamma_trac
king*F_soiling;
        [P_out(i),Tout_turb(i)] =
sim_SEC_Turbocharger_air_covered(eta_comp_now,eta_turbine,p_in,p_o
perating_now,p_out,numSections,L(i),mdot_now,Q_in(i),Tin,T_air,D_a
bsorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k
_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
    end

```

```

P_out(P_out<0)=0;%negative power ouput is no power output -
much easier to understand graph
etaEngine = P_out./Q_solar;
etaEngine_optimal(n)=max(etaEngine);
L_optimal(n)=L(etaEngine==max(etaEngine));
disp(['Length for optimal engine eta GTX55 for various flow
rates ',num2str(n), ' of ',num2str(fineness), ' completed']);
end
figure
yyaxis left
plot(mdot,L_optimal)
xlabel('True Mass flow');
ylabel('Optimal Receiver Length [m]');
yyaxis right
plot(mdot,etaEngine_optimal, '.')
xlabel('True Mass flow [kg/s]');
ylabel('Optimal \eta_{thermal}');
legend('Optimal Receiver Length', 'Optimal \eta_{thermal}')
etabrayton=(1-1./((p_operating./p_in).^(0.4/1.4)));
figure
yyaxis left
plot(p_operating./p_in,etabrayton,p_operating./p_in,etaEngine_opti
mal, '.');
ylabel('Efficiency');
xlabel('Operating Pressure Ratio');
yyaxis right
plot(p_operating./p_in,L_optimal, '--')
legend('\eta_{Brayton}', '\eta_{thermal}', 'Optimal Receiver Length
[m]')
%650 C receiver length finding Turbocharger corr.flow PR
%eta_c eta_tu Inducer mm GT0632SZ 5.832752613
2.322889616
%0.67 0.56 32 GTX3584RS 70.51420839
3.314803717 0.72
%0.78 68 GTX5533 Gen II 98mm 180.4149153 4.358176478 0.75
0.74 112
%Do GT0632SZ
clear;fineness=100;numSections=500;p_air=101325;p_in=p_air;mdot=5.
832752613;
D_absorber=0.032;eta_compressor=.67;eta_turbine=.56;p_operating=p_
in.*2.322889616;
D_cover_inner=D_absorber+(12/1000);D_cover_outer=D_cover_inner+(20
/1000);
p_vac=0.013;epsilon_absorber=0.1;k_cover=1.005;epsilon_cover=0.86;
p_out=p_in;I=1000;W=8;windspeed=5;T_air=25+273;T_HTF_bulk=T_air;
alpha_absorber=0.9;rho_mirror=0.93;tau_cover=0.96;gamma_tracking=0
.94;F_soiling=0.97;
Tin=T_HTF_bulk;Q_in=0;Q_in(fineness)=0;T_dewpoint_air=14+273;Relat
iveHumidity=50;
P_out=0;P_out(fineness)=0;Tout_turb=0;Tout_turb(fineness)=0;
mdot=mdot*0.45359237/60; %convert to kg/s
mdot_corr_Factor = calcMdot_corr_air(1,D_absorber,p_in,T_air);
%mdot_corr = mdot .* f(D_abosorber,p_in,T_in) = mdot .*
mdot_corr_Factor
mdot=mdot./mdot_corr_Factor;
[Tout_comp,DeltaH_Compressor_Polytropic_GT06] =
sim_Turbocharger_Section_Compressor_Air(Tin,eta_compressor,p_in,p_
operating,mdot)
L=linspace(1,5,500);Q_HTF(500)=0;T_out(500)=0;
Q_solar=I.*L.*W;
Q_in=Q_solar.*alpha_absorber.*rho_mirror.*tau_cover.*gamma_trackin
g.*F_soiling;

```

```

parfor i=1:500
    [Q_HTF_temp,~,~,T_out_temp,~,~,~,~,~] =
    sim_Receiver_air_HTF_air_vac(numSections,L(i),mdot,Q_in(i),Tout_co
    mp,p_operating,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,
    p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_
    air,RelativeHumidity);
    Q_HTF(i)=sum(Q_HTF_temp);
    T_out(i)=T_out_temp(end);
end
figure
plot(L,T_out);
%This is to display results of manual search - faster than doing a
newton
%iteration to find 650C outlet temp
format long
Tout_compGT06=Tout_comp;
LGT06=L((T_out<924 & T_out>923))
QhtfGT06=Q_HTF((T_out<924 & T_out>923))
QsolarGT06=Q_solar((T_out<924 & T_out>923))
QinGT06=Q_in((T_out<924 & T_out>923))
Tcombustor_inGT06=T_out((T_out<924 & T_out>923))
%Do GTX3584RS
fineness=100;numSections=500;p_air=101325;p_in=p_air;mdot=70.51420
839;
D_absorber=0.068;eta_compressor=0.72;eta_turbine=0.78;p_operating=
p_in.*3.314803717;
D_cover_inner=D_absorber+(12/1000);D_cover_outer=D_cover_inner+(20
/1000);
p_vac=0.013;epsilon_absorber=0.1;k_cover=1.005;epsilon_cover=0.86;
p_out=p_in;I=1000;W=8;windspeed=5;T_air=25+273;T_HTF_bulk=T_air;
alpha_absorber=0.9;rho_mirror=0.93;tau_cover=0.96;gamma_tracking=0
.94;F_soiling=0.97;
Tin=T_HTF_bulk;Q_in=0;Q_in(fineness)=0;T_dewpoint_air=14+273;Relat
iveHumidity=50;
P_out=0;P_out(fineness)=0;Tout_turb=0;Tout_turb(fineness)=0;
mdot=mdot*0.45359237/60; %convert to kg/s
mdot_corr_Factor = calcMdot_corr_air(1,D_absorber,p_in,T_air);
mdot=mdot./mdot_corr_Factor;
[Tout_comp,DeltaH_Compressor_Polytropic_GTX35] =
sim_Turbocharger_Section_Compressor_Air(Tin,eta_compressor,p_in,p_
operating,mdot)
L=linspace(1,50,500);Q_HTF(500)=0;T_out(500)=0;
Q_solar=I.*L.*W;
Q_in=Q_solar.*alpha_absorber.*rho_mirror.*tau_cover.*gamma_trackin
g.*F_soiling;
parfor i=1:500
    [Q_HTF_temp,~,~,T_out_temp,~,~,~,~,~] =
    sim_Receiver_air_HTF_air_vac(numSections,L(i),mdot,Q_in(i),Tout_co
    mp,p_operating,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,
    p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_
    air,RelativeHumidity);
    Q_HTF(i)=sum(Q_HTF_temp);
    T_out(i)=T_out_temp(end);
end
figure
plot(L,T_out);
Tout_compGTX35=Tout_comp;
LGTX35=L((T_out<924 & T_out>923))
QhtfGTX35=Q_HTF((T_out<924 & T_out>923))
QsolarGTX35=Q_solar((T_out<924 & T_out>923))
Tcombustor_inGTX35=T_out((T_out<924 & T_out>923))
QinGTX35=Q_in((T_out<924 & T_out>923))

```

```

%Do GTX5533
fineness=100;numSections=500;p_air=101325;p_in=p_air;mdot=180.4149
153;
D_absorber=0.112;eta_compressor=0.75;eta_turbine=0.74;p_operating=
p_in.*4.358176478;
D_cover_inner=D_absorber+(12/1000);D_cover_outer=D_cover_inner+(20
/1000);
p_vac=0.013;epsilon_absorber=0.1;k_cover=1.005;epsilon_cover=0.86;
p_out=p_in;I=1000;W=8;windspeed=5;T_air=25+273;T_HTF_bulk=T_air;
alpha_absorber=0.9;rho_mirror=0.93;tau_cover=0.96;gamma_tracking=0
.94;F_soiling=0.97;
Tin=T_HTF_bulk;Q_in=0;Q_in(fineness)=0;T_dewpoint_air=14+273;Relat
iveHumidity=50;
P_out=0;P_out(fineness)=0;Tout_turb=0;Tout_turb(fineness)=0;
mdot=mdot*0.45359237/60; %convert to kg/s
mdot_corr_Factor = calcMdot_corr_air(1,D_absorber,p_in,T_air);
mdot=mdot./mdot_corr_Factor;
[Tout_comp,DeltaH_Compressor_Polytropic_GTX55] =
sim_Turbocharger_Section_Compressor_Air(Tin,eta_compressor,p_in,p_
operating,mdot)
L=linspace(1,150,500);Q_HTF(500)=0;T_out(500)=0;
Q_solar=I.*L.*W;
Q_in=Q_solar.*alpha_absorber.*rho_mirror.*tau_cover.*gamma_trackin
g.*F_soiling;
parfor i=1:500
    [Q_HTF_temp,~,~,T_out_temp,~,~,~,~,~] =
sim_Receiver_air_HTF_air_vac(numSections,L(i),mdot,Q_in(i),Tout_co
mp,p_operating,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,
p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_
air,RelativeHumidity);
    Q_HTF(i)=sum(Q_HTF_temp);
    T_out(i)=T_out_temp(end);
end
figure
plot(L,T_out);
Tout_compGTX55=Tout_comp;
LGTX55=L((T_out<924 & T_out>923))
QhtfGTX55=Q_HTF((T_out<924 & T_out>923))
QsolarGTX55=Q_solar((T_out<924 & T_out>923))
Tcombustor_inGTX55=T_out((T_out<924 & T_out>923))
QinGTX55=Q_in((T_out<924 & T_out>923))
%Burn fuel for temps up to 1000C for turb inlet. Eta overall,
Tout, Pout Do
%GT0632SZ
mdot=5.832752613;L=LGTX55;mdot=mdot*0.45359237/60; %convert to kg/s
mdot_corr_Factor =
calcMdot_corr_air(1,D_absorber,p_in,T_air);mdot=mdot./mdot_corr_Fa
ctor;
D_absorber=0.032;eta_compressor=.67;eta_turbine=.56;p_operating=p_
in.*2.322889616;
D_cover_inner=D_absorber+(12/1000);D_cover_outer=D_cover_inner+(20
/1000);
T_turb_inlet =
linspace(924,1273,fineness);H_fuel_added(fineness)=0;
P_out=0;P_out(fineness)=0;Tout_turb=0;Tout_turb(fineness)=0;DeltaH
_Turbine_Polytropic=0;
DeltaH_Turbine_Polytropic(fineness)=0;eta_thermal_solar_HC=0;eta_t
hermal_solar_HC(fineness)=0;
eta_thermal_net=0;eta_thermal_net(fineness)=0;
for i=1:fineness

```

```

H_fuel_added(i)=mdot.*calcH_air_Delta(T_turb_inlet(i),Tcombustor_i
nGT06);
    [Tout_turb(i),DeltaH_Turbine_Polytropic(i)] =
sim_Turbocharger_Section_Turbine_Air(T_turb_inlet(i),eta_turbine,p
_operating,p_out,mdot);
    P_out(i)=-
1*(DeltaH_Turbine_Polytropic(i)+DeltaH_Compressor_Polytropic_GT06)
;

eta_thermal_solar_HC(i)=(P_out(i)./(H_fuel_added(i)+QsolarGT06));
    eta_thermal_net(i)=(P_out(i)./(H_fuel_added(i)+QhtfGT06));
end
figure
yyaxis left
plot(H_fuel_added./1000,eta_thermal_solar_HC,H_fuel_added./1000,et
a_thermal_net,'-.');
ylabel('\eta');xlabel('Fuel Combustion Contribution [kW]');
yyaxis right
plot(H_fuel_added./1000,P_out./1000,'--');
ylabel('Electrical Power Output [kW]')
legend('\eta_{thermal,Solar+HC}','\eta_{thermal,net to
HTF}','Electrical Power Output')
figure
plot(H_fuel_added./1000,Tout_turb,H_fuel_added./1000,T_turb_inlet,
'-.');
ylabel('Temperature [K]');xlabel('Fuel Combustion Contribution
[kW]');
legend('Turbine Outlet','Turbine Inlet')
%Do GTX3584RS
mdot=70.51420839;
D_absorber=0.068;eta_compressor=0.72;eta_turbine=0.78;p_operating=
p_in.*3.314803717;
L=LGTX35;mdot=mdot*0.45359237/60; %convert to kg/s
mdot_corr_Factor =
calcMdot_corr_air(1,D_absorber,p_in,T_air);mdot=mdot./mdot_corr_Fa
ctor;
D_cover_inner=D_absorber+(12/1000);D_cover_outer=D_cover_inner+(20
/1000);
T_turb_inlet =
linspace(924,1273,fineness);H_fuel_added(fineness)=0;
P_out=0;P_out(fineness)=0;Tout_turb=0;Tout_turb(fineness)=0;DeltaH
_Turbine_Polytropic=0;
DeltaH_Turbine_Polytropic(fineness)=0;eta_thermal_solar_HC=0;eta_t
hermal_solar_HC(fineness)=0;
eta_thermal_net=0;eta_thermal_net(fineness)=0;
for i=1:fineness

H_fuel_added(i)=mdot.*calcH_air_Delta(T_turb_inlet(i),Tcombustor_i
nGTX35);
    [Tout_turb(i),DeltaH_Turbine_Polytropic(i)] =
sim_Turbocharger_Section_Turbine_Air(T_turb_inlet(i),eta_turbine,p
_operating,p_out,mdot);
    P_out(i)=-
1*(DeltaH_Turbine_Polytropic(i)+DeltaH_Compressor_Polytropic_GTX35
);

eta_thermal_solar_HC(i)=(P_out(i)./(H_fuel_added(i)+QsolarGTX35));
    eta_thermal_net(i)=(P_out(i)./(H_fuel_added(i)+QhtfGTX35));
end
figure
yyaxis left

```

```

plot(H_fuel_added./1000,eta_thermal_solar_HC,H_fuel_added./1000,et
a_thermal_net,'-.');
ylabel('\eta');xlabel('Fuel Combustion Contribution [kW]');
yyaxis right
plot(H_fuel_added./1000,P_out./1000,'--');
ylabel('Electrical Power Output [kW]')
legend('\eta_{thermal,Solar+HC}','\eta_{thermal,net to
HTF}','Electrical Power Output')
figure
plot(H_fuel_added./1000,Tout_turb,H_fuel_added./1000,T_turb_inlet,
'-.');
ylabel('Temperature [K]');xlabel('Fuel Combustion Contribution
[kW]');
legend('Turbine Outlet','Turbine Inlet')
%Do GTX55
mdot=180.4149153;
D_absorber=0.112;eta_compressor=0.75;eta_turbine=0.74;p_operating=
p_in.*4.358176478;
L=L GTX55;mdot=mdot*0.45359237/60; %convert to kg/s
mdot_corr_Factor =
calcMdot_corr_air(1,D_absorber,p_in,T_air);mdot=mdot./mdot_corr_Fa
ctor;
D_cover_inner=D_absorber+(12/1000);D_cover_outer=D_cover_inner+(20
/1000);
T_turb_inlet =
linspace(924,1273,fineness);H_fuel_added(fineness)=0;
P_out=0;P_out(fineness)=0;Tout_turb=0;Tout_turb(fineness)=0;DeltaH
_Turbine_Polytropic=0;
DeltaH_Turbine_Polytropic(fineness)=0;eta_thermal_solar_HC=0;eta_t
hermal_solar_HC(fineness)=0;
eta_thermal_net=0;eta_thermal_net(fineness)=0;
for i=1:fineness

H_fuel_added(i)=mdot.*calcH_air_Delta(T_turb_inlet(i),Tcombustor_i
nGTX55);
    [Tout_turb(i),DeltaH_Turbine_Polytropic(i)] =
sim_Turbocharger_Section_Turbine_Air(T_turb_inlet(i),eta_turbine,p
_operating,p_out,mdot);
    P_out(i)=-
1*(DeltaH_Turbine_Polytropic(i)+DeltaH_Compressor_Polytropic_GTX55
);

eta_thermal_solar_HC(i)=(P_out(i)./(H_fuel_added(i)+QsolarGTX55));
eta_thermal_net(i)=(P_out(i)./(H_fuel_added(i)+QhtfGTX55));
end
figure
yyaxis left
plot(H_fuel_added./1000,eta_thermal_solar_HC,H_fuel_added./1000,et
a_thermal_net,'-.');
ylabel('\eta');xlabel('Fuel Combustion Contribution [kW]');
yyaxis right
plot(H_fuel_added./1000,P_out./1000,'--');
ylabel('Electrical Power Output [kW]')
legend('\eta_{thermal,Solar+HC}','\eta_{thermal,net to
HTF}','Electrical Power Output')
figure
plot(H_fuel_added./1000,Tout_turb,H_fuel_added./1000,T_turb_inlet,
'-.');
ylabel('Temperature [K]');xlabel('Fuel Combustion Contribution
[kW]');
legend('Turbine Outlet','Turbine Inlet')
%Graph etas and Hin for all turbos vs turb inlet temp

```

```

fineness=50;
mdot=5.832752613;L=LGT06;mdot=mdot*0.45359237/60; %convert to kg/s
mdot_corr_Factor =
calcMdot_corr_air(1,D_absorber,p_in,T_air);mdot=mdot./mdot_corr_Fa
ctor;
D_absorber=0.032;eta_compressor=.67;eta_turbine=.56;p_operating=p_
in.*2.322889616;
D_cover_inner=D_absorber+(12/1000);D_cover_outer=D_cover_inner+(20
/1000);
T_turb_inlet =
linspace(Tout_compGT06,3500,fineness);H_fuel_added=0;H_fuel_added(
fineness)=0;
P_out=0;P_out(fineness)=0;Tout_turb=0;Tout_turb(fineness)=0;DeltaH
_Turbine_Polytropic=0;
DeltaH_Turbine_Polytropic(fineness)=0;eta_thermal_solar_HC=0;eta_t
hermal_solar_HC(fineness)=0;
eta_thermal_net=0;eta_thermal_net(fineness)=0;
for i=1:fineness

H_fuel_added(i)=mdot.*calcH_air_Delta(T_turb_inlet(i),Tout_compGT0
6);
    [Tout_turb(i),DeltaH_Turbine_Polytropic(i)] =
sim_Turbocharger_Section_Turbine_Air(T_turb_inlet(i),eta_turbine,p
_operating,p_out,mdot);
    P_out(i)=-
1*(DeltaH_Turbine_Polytropic(i)+DeltaH_Compressor_Polytropic_GT06)
;
end
P_out(P_out<0)=0;%i.e. no power output
eta_thermal_solar_HC=(P_out./(H_fuel_added));
T_turb_inlet_GT06=T_turb_inlet;
eta_thermal_solar_HC_GT06=eta_thermal_solar_HC;
H_fuel_added_GT06=H_fuel_added;
%Do GTX3584RS
mdot=70.51420839;
D_absorber=0.068;eta_compressor=0.72;eta_turbine=0.78;p_operating=
p_in.*3.314803717;
L=LGTX35;mdot=mdot*0.45359237/60; %convert to kg/s
mdot_corr_Factor =
calcMdot_corr_air(1,D_absorber,p_in,T_air);mdot=mdot./mdot_corr_Fa
ctor;
D_cover_inner=D_absorber+(12/1000);D_cover_outer=D_cover_inner+(20
/1000);
T_turb_inlet =
linspace(Tout_compGTX35,3500,fineness);H_fuel_added(fineness)=0;
P_out=0;P_out(fineness)=0;Tout_turb=0;Tout_turb(fineness)=0;DeltaH
_Turbine_Polytropic=0;
DeltaH_Turbine_Polytropic(fineness)=0;eta_thermal_solar_HC=0;eta_t
hermal_solar_HC(fineness)=0;
eta_thermal_net=0;eta_thermal_net(fineness)=0;
for i=1:fineness

H_fuel_added(i)=mdot.*calcH_air_Delta(T_turb_inlet(i),Tout_compGTX
35);
    [Tout_turb(i),DeltaH_Turbine_Polytropic(i)] =
sim_Turbocharger_Section_Turbine_Air(T_turb_inlet(i),eta_turbine,p
_operating,p_out,mdot);
    P_out(i)=-
1*(DeltaH_Turbine_Polytropic(i)+DeltaH_Compressor_Polytropic_GTX35
);
end
P_out(P_out<0)=0;%i.e. no power output

```

```

eta_thermal_solar_HC=(P_out./(H_fuel_added));
T_turb_inlet_GTX35=T_turb_inlet;
eta_thermal_solar_HC_GTX35=eta_thermal_solar_HC;
H_fuel_added_GTX35=H_fuel_added;
%Do GTX55
mdot=180.4149153;
D_absorber=0.112;eta_compressor=0.75;eta_turbine=0.74;p_operating=
p_in.*4.358176478;
L=L GTX55;mdot=mdot*0.45359237/60; %convert to kg/s
mdot_corr_Factor =
calcMdot_corr_air(1,D_absorber,p_in,T_air);mdot=mdot./mdot_corr_Fa
ctor;
D_cover_inner=D_absorber+(12/1000);D_cover_outer=D_cover_inner+(20
/1000);
T_turb_inlet =
linspace(Tout_compGTX55,3500,fineness);H_fuel_added(fineness)=0;
P_out=0;P_out(fineness)=0;Tout_turb=0;Tout_turb(fineness)=0;DeltaH
_Turbine_Polytropic=0;
DeltaH_Turbine_Polytropic(fineness)=0;eta_thermal_solar_HC=0;eta_t
hermal_solar_HC(fineness)=0;
eta_thermal_net=0;eta_thermal_net(fineness)=0;
for i=1:fineness

H_fuel_added(i)=mdot.*calcH_air_Delta(T_turb_inlet(i),Tout_compGTX
55);
[Tout_turb(i),DeltaH_Turbine_Polytropic(i)] =
sim_Turbocharger_Section_Turbine_Air(T_turb_inlet(i),eta_turbine,p
_operating,p_out,mdot);
P_out(i)=-
1*(DeltaH_Turbine_Polytropic(i)+DeltaH_Compressor_Polytropic_GTX55
);
end
P_out(P_out<0)=0;%i.e. no power output
eta_thermal_solar_HC=(P_out./(H_fuel_added));
T_turb_inlet_GTX55=T_turb_inlet;
eta_thermal_solar_HC_GTX55=eta_thermal_solar_HC;
H_fuel_added_GTX55=H_fuel_added;
figure
yyaxis left
plot(T_turb_inlet_GT06,eta_thermal_solar_HC_GT06,'s',T_turb_inlet_
GTX35,eta_thermal_solar_HC_GTX35,'x',T_turb_inlet_GTX55,eta_therma
l_solar_HC_GTX55,'.');
ylabel('\eta_{thermal,net to HTF}');xlabel('Turbine Inlet
Temperature [K]');
yyaxis right
plot(T_turb_inlet_GT06,H_fuel_added_GT06./1000./5.832752613,T_turb
_inlet_GTX35,H_fuel_added_GTX35./1000./70.51420839,T_turb_inlet_GT
X55,H_fuel_added_GTX55./1000./180.4149153);
ylabel('Net Heat Added per kg Real Air Mass Flow [kW.s/kg]');
legend('\eta - GT0632SZ','\eta - GTX3584RS','\eta - GTX5533 Gen
II. 98mm','Heat Addition - GT0632SZ','Heat Addition -
GTX3584RS','Heat Addition - GTX5533 Gen II. 98mm');
fineness=100;
%Do GTX55 graphs with and without recycle recycle plot GTX55
fineness=400;
mdot=180.4149153;
D_absorber=0.112;eta_compressor=0.75;eta_turbine=0.74;p_operating=
p_in.*4.358176478;
mdot=mdot*0.45359237/60; %convert to kg/s
mdot_corr_Factor =
calcMdot_corr_air(1,D_absorber,p_in,T_air);mdot=mdot./mdot_corr_Fa
ctor;

```

```

D_cover_inner=D_absorber+(12/1000);D_cover_outer=D_cover_inner+(20
/1000);
Tin_turb=linspace(800,1500,fineness);%~800 K minimum inlet temp to
generate power
%no recycle
P_out_noRe=0;P_out_noRe(fineness)=0;
Tout_turb_noRe=0;Tout_turb_noRe(fineness)=0;DeltaH_Turbine_Polytro
pic_noRe=0;DeltaH_Turbine_Polytropic_noRe(fineness)=0;
Q_net_req_noRe=0;Q_net_req_noRe(fineness)=0;eta_thermal_noRe=0;eta
_thermal_noRe(fineness)=0;
for i=1:fineness
    [Tout_turb_noRe(i),DeltaH_Turbine_Polytropic_noRe(i)] =
sim_Turbocharger_Section_Turbine_Air(Tin_turb(i),eta_turbine,p_ope
rating,p_out,mdot);

Q_net_req_noRe(i)=mdot.*calcH_air_Delta(Tin_turb(i),Tout_compGTX55
);
    P_out_noRe(i)=-
1*(DeltaH_Turbine_Polytropic_noRe(i)+DeltaH_Compressor_Polytropic_
GTX55);
    P_out_noRe(P_out_noRe<0)=0;
    eta_thermal_noRe(i)=P_out_noRe(i)./Q_net_req_noRe(i);
end
%recycle
P_out=0;P_out(fineness)=0;
Tout_turb=0;Tout_turb(fineness)=0;DeltaH_Turbine_Polytropic=0;Delt
aH_Turbine_Polytropic(fineness)=0;
Q_net_req=0;Q_net_req(fineness)=0;T_HE_out=0;T_HE_out(fineness)=0;
eta_thermal=0;eta_thermal(fineness)=0;
for i=1:fineness
    [Tout_turb(i),DeltaH_Turbine_Polytropic(i)] =
sim_Turbocharger_Section_Turbine_Air(Tin_turb(i),eta_turbine,p_ope
rating,p_out,mdot);
    if(Tout_turb(i)+50>Tout_compGTX55)
        T_HE_out(i)=Tout_turb(i)-50;
        if(T_HE_out<Tin_turb(i))

Q_net_req(i)=mdot.*calcH_air_Delta(Tin_turb(i),T_HE_out(i));
        P_out(i)=-
1*(DeltaH_Turbine_Polytropic(i)+DeltaH_Compressor_Polytropic_GTX55
);
            if(P_out(i)<0)
                P_out(i)=0;%negative power doesnt generate
electricity)
            end
            eta_thermal(i)=P_out(i)./Q_net_req(i);
        end
    else
        P_out=0;
    end
    P_out(P_out<0)=0;
end
figure
plot(Tin_turb,eta_thermal,Tin_turb,eta_thermal_noRe,'--')
xlabel('Turbine Inlet Temperature [K]');ylabel('\eta_{thermal,net
to HTF}');
legend('With Heat Recycle Unit','Without Heat Recyle Unit');
%700C no recycle
Tin_turb=700+273;
[Tout_turb,DeltaH_Turbine_Polytropic] =
sim_Turbocharger_Section_Turbine_Air(Tin_turb,eta_turbine,p_operat
ing,p_out,mdot);

```

```

P_out=-
1*(DeltaH_Turbine_Polytropic+DeltaH_Compressor_Polytropic_GTX55);
Q_net_req=mdot.*calhH_air_Delta(Tin_turb,Tout_compGTX55);
eta_thermal_netIn=P_out./Q_net_req;
L=linspace(1,200,fineness);
Q_solar=I.*L.*W;
Q_in=Q_solar.*alpha_absorber.*rho_mirror.*tau_cover.*gamma_tracking.*F_soiling;
Q_HTF_solar=0;Q_HTF_solar(fineness)=0;T_out_receiver=0;T_out_receiver(fineness)=0;
parfor i=1:fineness
    [Q_HTF,~,~,T_out,~,~,~,~,~] =
sim_Receiver_air_HTF_air_vac(numSections,L(i),mdot,Q_in(i),Tout_compGTX55,p_operating,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
    Q_HTF_solar(i)=sum(Q_HTF);
    T_out_receiver(i)=T_out(end);
end
L_noRe=L(T_out_receiver>(699+273)&T_out_receiver<(701+273))
T_out_rec_noRe=T_out_receiver(T_out_receiver>(699+273)&T_out_receiver<(701+273))
Q_in_noRe=Q_in(T_out_receiver>(699+273)&T_out_receiver<(701+273))
Q_solar_noRe=Q_solar(T_out_receiver>(699+273)&T_out_receiver<(701+273))
Q_HTF_solar_noRe=Q_HTF_solar(T_out_receiver>(699+273)&T_out_receiver<(701+273))
eta_netin_noRe=P_out./Q_HTF_solar_noRe
eta_solar_noRe=P_out./Q_solar_noRe
eta_receiver_noRe=Q_HTF_solar_noRe./Q_solar_noRe
%recycle
Tin_turb=700+273;
[Tout_turb,DeltaH_Turbine_Polytropic] =
sim_Turbocharger_Section_Turbine_Air(Tin_turb,eta_turbine,p_operating,p_out,mdot);
P_out=-
1*(DeltaH_Turbine_Polytropic+DeltaH_Compressor_Polytropic_GTX55);
T_HE_out=Tout_turb-50;
Q_net_req=mdot.*calhH_air_Delta(Tin_turb,T_HE_out);
eta_thermal_netIn=P_out./Q_net_req;
L=linspace(1,100,fineness);
Q_solar=I.*L.*W;
Q_in=Q_solar.*alpha_absorber.*rho_mirror.*tau_cover.*gamma_tracking.*F_soiling;
Q_HTF_solar=0;Q_HTF_solar(fineness)=0;T_out_receiver=0;T_out_receiver(fineness)=0;
parfor i=1:fineness
    [Q_HTF,~,~,T_out,~,~,~,~,~] =
sim_Receiver_air_HTF_air_vac(numSections,L(i),mdot,Q_in(i),T_HE_out,p_operating,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
    Q_HTF_solar(i)=sum(Q_HTF);
    T_out_receiver(i)=T_out(end);
end
%add to table
L_Re=L(T_out_receiver>(700+273)&T_out_receiver<(700.5+273))
T_out_rec_Re=T_out_receiver(T_out_receiver>(700+273)&T_out_receiver<(700.5+273))
Q_in_Re=Q_in(T_out_receiver>(700+273)&T_out_receiver<(700.5+273))
Q_solar_Re=Q_solar(T_out_receiver>(700+273)&T_out_receiver<(700.5+273))

```

```

Q_HTF_solar_Re=Q_HTF_solar(T_out_receiver>(700+273)&T_out_receiver
<(700.5+273))
eta_netin_Re=P_out./Q_HTF_solar_Re
eta_solar_Re=P_out./Q_solar_Re
eta_receiver_Re=Q_HTF_solar_Re./Q_solar_Re
%Show trough effectiveness for GT06 apparatus setup with wind 5
m/s with
%copper pipe
clear
fineness=100;numSections=500;p_air=101325;p_in=p_air;mdot=5.832752
613;
D_absorber=0.042;eta_compressor=.67;eta_turbine=.56;p_operating=p
in.*2.322889616;
p_vac=0.013;epsilon_absorber=0.45;k_cover=1.005;epsilon_cover=0.86
;
p_out=p_in;I=1000;W=2.8;windspeed=5;T_air=25+273;T_HTF_bulk=412.18
;
alpha_absorber=0.95;rho_mirror=0.93;gamma_tracking=0.94;F_soiling=
0.97;
Tin=T_HTF_bulk;Q_in=0;Q_in(fineness)=0;T_dewpoint_air=14+273;Relat
iveHumidity=50;
P_out=0;P_out(fineness)=0;Tout_turb=0;Tout_turb(fineness)=0;
mdot=mdot*0.45359237/60; %convert to kg/s
mdot_corr_Factor = calcMdot_corr_air(1,D_absorber,p_in,T_air);
%mdot_corr = mdot .* f(D_abosorber,p_in,T_in) = mdot .*
mdot_corr_Factor
mdot=mdot./mdot_corr_Factor;
%Graph to show it is pretty much pointless doing more than 1
collector
NumberTroughs=1:5;
L=2.*NumberTroughs;%Each trough section is 2m long
numGraphPoints=length(L);
TroughT_out(numGraphPoints)=0;TroughEtaSolar(numGraphPoints)=0;
for i=1:numGraphPoints
    Q_solar=I.*L(i).*W;

Q_in=Q_solar.*alpha_absorber.*rho_mirror.*gamma_tracking.*F_soilin
g;
    [Q_HTF,~,~,T_out,~,~,~] =
sim_Receiver_air_HTF_coverless(numSections,L(i),mdot,Q_in,T_HTF_bu
lk,p_operating,T_air,D_absorber,p_air,epsilon_absorber,windspeed,T
_dewpoint_air,RelativeHumidity);
    TroughT_out(i)=T_out(end);
    TroughEtaSolar(i)=sum(Q_HTF)./Q_solar;
end
figure
yyaxis left
plot(cat(2,[0],NumberTroughs),cat(2,[412.18],TroughT_out))%i.e.
show if no trough used
ylabel('Trough Air Outlet Temperature [K]');xlabel('Number of
Trough Sections');
yyaxis right
plot(NumberTroughs,TroughEtaSolar,'--')
ylabel('\eta_{Solar}')
legend('Outlet Temperature','\eta')
windspeed=5;
NumberTroughs=1:50;
L=2.*NumberTroughs;%Each trough section is 2m long
numGraphPoints=length(L);
TroughT_out(numGraphPoints)=0;TroughEtaSolar(numGraphPoints)=0;
for i=1:numGraphPoints
    Q_solar=I.*L(i).*W;

```

```

Q_in=Q_solar.*alpha_absorber.*rho_mirror.*gamma_tracking.*F_soiling;
g;
    [Q_HTF,~,~,T_out,~,~,~] =
sim_Receiver_air_HTF_coverless(numSections,L(i),mdot,Q_in,T_HTF_bulk,
p_operating,T_air,D_absorber,p_air,epsilon_absorber,windspeed,T_dewpoint_
air,RelativeHumidity);
    TroughT_out(i)=T_out(end);
    TroughEtaSolar(i)=sum(Q_HTF)./Q_solar;
end
figure
yyaxis left
plot(cat(2,[0],NumberTroughs),cat(2,[412.18],TroughT_out))%i.e.
show if no trough used
ylabel('Trough Air Outlet Temperature [K]');xlabel('Number of
Trough Sections');
yyaxis right
plot(NumberTroughs,TroughEtaSolar,'--')
ylabel('\eta_{Solar}')
legend('Outlet Temperature','\eta')
%Apparatus with water flowing through it
clear
fineness=100;numSections=500;p_air=101325;p_in=p_air;mdot=.1;
D_absorber=0.042;eta_compressor=.67;eta_turbine=.56;p_operating=p_in;
D_cover_inner=D_absorber+(12/1000);D_cover_outer=D_cover_inner+(20
/1000);
epsilon_absorber=0.45;k_cover=1.005;L=2;p_vac=0.013;
I=1000;W=2.8;windspeed=5;T_air=25+273;T_HTF_bulk=T_air;
alpha_absorber=0.95;rho_mirror=0.93;gamma_tracking=0.94;F_soiling=
0.97;
Tin=T_HTF_bulk;T_dewpoint_air=14+273;RelativeHumidity=50;
Q_solar=I.*L.*W;epsilon_cover=0.86;
Q_in=Q_solar.*alpha_absorber.*rho_mirror.*gamma_tracking.*F_soiling;
g;
[Q_HTF,Q_loss_rad,Q_loss_conv,T_out,T_absorber,count,Lsection] =
sim_Receiver_water_HTF_coverless(numSections,L,mdot,Q_in,T_HTF_bulk,
T_air,D_absorber,p_air,epsilon_absorber,windspeed,T_dewpoint_
air,RelativeHumidity);
sum(Q_HTF)/Q_in
T_out(end)

```

F.2 Intensive Linear Receiver Model Functions

```

function [rho,cp,k,alpha_t,mu,nu] =
AIRPROPERTIES(p,T,want_rho,want_cp,want_k,want_alpha_t,want_mu,want_nu)
%p in Pa, T in K, rho in kg/m3, cp in J/kg.K, k in W/m.K, alpha_t
in m2/s,
%mu in kg/m.s, nu in m2/s

%input p,T, get desired values. Only use memory and perform
calculation for
%wanted air properties. This keeps number of functions down and
program
%speed high, especially for 1000s of iterations.
rho=0;cp=0;k=0;alpha_t=0;mu=0;nu=0; %return nul
T=T-273.15; % functions require T in Celcius
if(want_rho)
    rho=(1./(T.*0.002833191+0.774043)).*(p./101325);

```

```

end
if(want_cp)
    cp=( (4.73252.*10.^(-11)).*(T.^4)+(-2.07769.*10.^(-
7)).*(T.^3)+(0.000238738).*(T.^2)+(0.106809446).*T+999.9268821);
end
if(want_k)
    k=( (4.63986.*10.^(-12)).*(T.^3)+(-2.5217.*10.^(-
8)).*(T.^2)+(7.56219.*10.^(-5)).*T+0.023635444);
end
if(want_alpha_t)
    alpha_t=( (-1.50263.*10.^(-14)).*(T.^3)+(9.71026.*10.^(-
11)).*(T.^2)+(1.39146.*10.^(-7)).*T+1.84666.*10.^(-
5)).*(101325./p);
end
if(want_mu)
    mu=( (-3.28024.*10.^(-18)).*(T.^4)+(1.56331.*10.^(-
14)).*(T.^3)+(-3.02471.*10.^(-11)).*(T.^2)+(4.87271.*10.^(-
8)).*T+1.72057.*10.^(-5));
end
if(want_nu)
    nu=( (4.18424.*10.^(-18)).*(T.^4)+(-2.64477.*10.^(-
14)).*(T.^3)+(9.60235.*10.^(-11)).*(T.^2)+(8.70674.*10.^(-
8)).*T+1.34434.*10.^(-5)).*(101325./p);
end
end

```

```

function [mdot_corrected] =
calc_mdot_Total_air(mdot,D,p_static,T_static)
%Find corrected mass flow in a pipe, so that turbharger maps can
be
%made to account for the affects of temperature and pressure on
enthalpy
%flow
[p_total,T_total] = calc_p_Total_air(mdot,D,p_static,T_static);
mdot_corrected=mdot.*((T_total./288.15).^0.5)./(p_total./101325);
end

```

```

function [p_total,T_total] =
calc_p_Total_air(mdot,D_absorber,p_static,T_static)
%Find total pressure of flow in a pipe, so that turbharger maps
can be
%made to account for the affects of temperature and pressure on
enthalpy
%flow
[T_total,cp] = calcT_Total_air(mdot,D_absorber,p_static,T_static);
%C_(v,air)=C_(p,air)-287.058 [J/(kg?K)]
cv=cp-287.058;
gamma=cp./cv;
p_total=p_static+(T_total./T_static).^(gamma./(gamma-1));
end

```

```

function [DeltaH] = calcH_air_Delta(Tfinal,Tinitial)
%Enthalpy change of air over a temperature.
%Units J/kg
%Cp_air = (4.73252?10^(-11))T^4 + (-2.07769?10^(-7))T^3 +
(0.000238738)T^2
%+ (0.106809446)T + 999.9268821 J/kg.K
%DeltaH=int(Cp)dT

```

```

a=4.73252E-11;b=-2.07769E-
7;c=0.000238738;d=0.106809446;e=999.9268821;
H_final=((a.*(Tfinal.^5))./5)+((b.*(Tfinal.^4))./4)+((c.*(Tfinal.^
3))./3)+((d.*(Tfinal.^2))./2)+e.*Tfinal;
H_initial=((a.*(Tinitial.^5))./5)+((b.*(Tinitial.^4))./4)+((c.*(Ti
nitial.^3))./3)+((d.*(Tinitial.^2))./2)+e.*Tinitial;
DeltaH=H_final-H_initial;
end

```

```

function [K] = calcKincidenceIST(phi_deg)
%CALCKINCIDENSEIST Industrial Solar Technology Corporation product
PTC
%incidence factor, for use in approximating conventional receiver
tube with
%anti-glazing coating
K=1+0.0003718*(phi_deg/cosd(phi_deg))-
0.00003985*(phi_deg)^2/cosd(phi_deg));
end

```

```

function [mdot_corr] = calcMdot_corr_air(mdot,D,p_static,T_static)
%Input real mass flow rate, get out corrected mass flow rate to
sea level
T_STP=288.15;p_STP=101325;%Constants to compare to sea level inlet
conditions
[p_total,T_total] = calc_p_Total_air(mdot,D,p_static,T_static);
mdot_corr = mdot.*(((T_total./T_STP).^0.5)./(p_total./p_STP));
end

```

```

function [Q] =
calcQ_absorber_to_cover_convection(L,D_absorber_outer,D_cover_inne
r,T_absorber_surface,T_cover_inner,p_vacuum)
%calculate convection heat transfer within an anulus vacuum cover,
with air
%as the gas.
g=9.81; %gravity constant
T_avg = (T_absorber_surface+T_cover_inner)/2;
gamma = 1.4; %cp/cv air
delta = 3.5E-10; % molecular diameter air
Q(length(p_vacuum))=0;
for i=1:length(p_vacuum)
[~,cp,k,alpha_t,mu,nu] =
AIRPROPERTIES(p_vacuum(i),T_avg,0,1,1,1,1,1);
Pr = mu.*cp./k;
Ra_L = g.*(T_absorber_surface-
T_cover_inner).*((D_cover_inner-
D_absorber_outer)./2).^3./(T_avg.*alpha_t.*nu);
Ra_star_1 =
((log(D_cover_inner./D_absorber_outer)).^4).*Ra_L;
Ra_star_2a = (((D_cover_inner-D_absorber_outer)./2).^3);
Ra_star_2b = (((D_cover_inner).^(-
3/5))+((D_absorber_outer).^(-3/5))).^5;
Ra_star = Ra_star_1./(Ra_star_2a.*Ra_star_2b);
k_eff_1_on_k = 0.386.*((Pr.*Ra_star)./(0.861+Pr));
lambda = 1.381E-
23.*(T_avg)./(2^0.5).*pi.*p_vacuum(i).*((delta).^2));

```

```

        k_eff_2_on_k = (1+((2.*lambda.*(9.*gamma-
5))./(gamma+1).*(log(D_cover_inner./D_absorber_outer)))).*(((1./D
_cover_inner)+(1./D_absorber_outer))).^-1;
        k_eff_on_k = max([k_eff_1_on_k k_eff_2_on_k]);
        k_eff = k_eff_on_k.*k;
        Q(i)=2.*pi.*k_eff.*L.*(T_absorber_surface-
T_cover_inner)./(log(D_cover_inner./D_absorber_outer));
    end
end

```

```

function [Q] =
calcQ_absorber_to_cover_radiation(L,D_absorber,D_cover_inner,epsil
on_absorber,epsilon_cover,T_absorber,T_cover_inner)
% Calculate radiative heat transfer between absorber surface and
% surrounding cover.
sigma = 5.670373.*10.^-8;%Stephan-Boltzman Constant%
p1 = sigma.*pi.*L.*D_absorber.*((T_absorber.^4)-
(T_cover_inner.^4));
p2 = (1./epsilon_absorber)+(((1-
epsilon_cover).*D_absorber)./(epsilon_cover.*D_cover_inner));
Q = p1./p2;
end

```

```

function [Q] =
calcQ_absorber_to_cover_total(L,D_absorber,D_cover_inner,T_absorbe
r,T_cover_inner,p_vacuum,epsilon_absorber,epsilon_cover)
%Total Q to cover from absorber
rad =
calcQ_absorber_to_cover_radiation(L,D_absorber,D_cover_inner,epsil
on_absorber,epsilon_cover,T_absorber,T_cover_inner);
conv =
calcQ_absorber_to_cover_convection(L,D_absorber,D_cover_inner,T_ab
sorber,T_cover_inner,p_vacuum);
Q=rad+conv;
end

```

```

function [Q] =
calcQ_cover_conduction(L,k_cover,D_cover_inner,D_cover_outer,T_cov
er_inner,T_cover_outer)
%Calculate conduction through the cover material, used for
iteration
%purposes to find T_absorber when a cover exists
Q=2.*pi.*L.*k_cover.*(T_cover_inner-
T_cover_outer)./(log(D_cover_outer./D_cover_inner));
end

```

```

function [Q_radiation,Q_convection] =
calcQ_receiver_to_ambient_air(L,D_outer,T_outer,epsilon_outer,T_sk
y_blackbody,T_air,h_wind)
%Calculates ambient heat losses when outer surface temperature of
the
%receiver is known.
sigma = 5.670373.*10.^-8;%Stephan-Boltzman Constant%
Q_radiation = sigma.*pi.*L.*D_outer.*epsilon_outer.*((T_outer.^4)-
(T_sky_blackbody.^4));
Q_convection = h_wind.*pi.*L.*D_outer.*(T_outer-T_air);
end

```

```

function [Q] =
calcQ_receiver_to_ambient_air_total(L,D_outer,T_outer,epsilon_oute
r,T_sky_blackbody,T_air,h_wind)
%Finds total Q to atm
[Q_radiation,Q_convection] =
calcQ_receiver_to_ambient_air(L,D_outer,T_outer,epsilon_outer,T_sk
y_blackbody,T_air,h_wind);
Q=Q_radiation+Q_convection;
end

```

```

function [Q] =
calcQ_receiver_to_HTF_for_air(D_receiver_inner,mdot,T_absorber,T_H
TF_bulk,p_HTF,L)
%calculate heat transfer to HTF from absorber surface when air is
the HTF.
%note that only valid for 2300 < Re_D_receiver_inner < 5E6 and 0.5
<
%Pr_T_bulk < 2000
[~,cp_bulk,k_bulk,~,mu_bulk,~] =
AIRPROPERTIES(p_HTF,T_HTF_bulk,0,1,1,0,1,0);
[~,cp_absorber,k_absorber,~,mu_absorber,~] =
AIRPROPERTIES(p_HTF,T_absorber,0,1,1,0,1,0);
Re_D_receiver_inner =
((4.*mdot)./(pi.*D_receiver_inner))./(mu_bulk);
Pr_T_bulk = mu_bulk.*cp_bulk./k_bulk;
Pr_T_absorber = mu_absorber.*cp_absorber./k_absorber;
f_fric=(1.82.*log10(Re_D_receiver_inner)-1.64).^(-2);
p1 = (f_fric./8).*(Re_D_receiver_inner-1000).*(Pr_T_bulk);
p2 = 1+12.7.*(((Pr_T_bulk.^(2/3))-1).*((f_fric./8).^0.5));
p3 = (Pr_T_bulk./Pr_T_absorber)^0.11;
Nu_D_receiver_inner = (p1.*p3)./p2;
h_T_HTF_bulk = Nu_D_receiver_inner.*k_bulk./D_receiver_inner;
Q = h_T_HTF_bulk.*D_receiver_inner.*pi.*L.*(T_absorber-
T_HTF_bulk);
%Error Notification
if(Re_D_receiver_inner<2300||Re_D_receiver_inner>5E6)
disp(['ERROR: Re_D_receiver_inner= '
num2str(Re_D_receiver_inner)]);
end
if(Pr_T_bulk>2000)
disp(['ERROR: Pr_T_bulk= ' num2str(Pr_T_bulk)]);
end
end
end

```

```

function [Q] =
calcQ_receiver_to_HTF_for_marlothermSH(D_receiver_inner,mdot,T_abs
orber,T_HTF_bulk,L)
%calculate heat transfer to HTF from absorber surface when
marlotherm is the HTF.
%note that only valid for 2300 < Re_D_receiver_inner < 5E6 and 0.5
<
%Pr_T_bulk < 2000
[cp_bulk,k_bulk,mu_bulk] = MarlothermSH_properties(T_HTF_bulk);
[cp_absorber,k_absorber,mu_absorber] =
MarlothermSH_properties(T_absorber);
Re_D_receiver_inner =
((4.*mdot)./(pi.*D_receiver_inner))./(mu_bulk);

```

```

Pr_T_bulk = mu_bulk.*cp_bulk./k_bulk;
Pr_T_absorber = mu_absorber.*cp_absorber./k_absorber;
f_fric=(1.82.*log10(Re_D_receiver_inner)-1.64).^(-2);
p1 = (f_fric./8).*(Re_D_receiver_inner-1000).*(Pr_T_bulk);
p2 = 1+12.7.*(((Pr_T_bulk.^(2/3))-1).*((f_fric./8).^0.5));
p3 = (Pr_T_bulk./Pr_T_absorber)^0.11;
Nu_D_receiver_inner = (p1.*p3)./p2;
h_T_HTF_bulk = Nu_D_receiver_inner.*k_bulk./D_receiver_inner;
Q = h_T_HTF_bulk.*D_receiver_inner.*pi.*L.*(T_absorber-
T_HTF_bulk);
>Error Notification
if(Re_D_receiver_inner<2300||Re_D_receiver_inner>5E6)
    disp(['ERROR: Re_D_receiver_inner= '
num2str(Re_D_receiver_inner)]);
end
if(Pr_T_bulk>2000)
    disp(['ERROR: Pr_T_bulk= ' num2str(Pr_T_bulk)]);
end
end

function [Q] =
calcQ_receiver_to_HTF_for_waterLiq(D_receiver_inner,mdot,T_absorber,
T_HTF_bulk,L)
%calculate heat transfer to HTF from absorber surface when water
is the HTF.
%note that only valid for 2300 < Re_D_receiver_inner < 5E6 and 0.5
<
%Pr_T_bulk < 2000
[cp_bulk,k_bulk,mu_bulk] = Water_properties(T_HTF_bulk);
[cp_absorber,k_absorber,mu_absorber] =
Water_properties(T_absorber);
Re_D_receiver_inner =
((4.*mdot)./(pi.*D_receiver_inner))./(mu_bulk);
Pr_T_bulk = mu_bulk.*cp_bulk./k_bulk;
Pr_T_absorber = mu_absorber.*cp_absorber./k_absorber;
f_fric=(1.82.*log10(Re_D_receiver_inner)-1.64).^(-2);
p1 = (f_fric./8).*(Re_D_receiver_inner-1000).*(Pr_T_bulk);
p2 = 1+12.7.*(((Pr_T_bulk.^(2/3))-1).*((f_fric./8).^0.5));
p3 = (Pr_T_bulk./Pr_T_absorber)^0.11;
Nu_D_receiver_inner = (p1.*p3)./p2;
h_T_HTF_bulk = Nu_D_receiver_inner.*k_bulk./D_receiver_inner;
Q = h_T_HTF_bulk.*D_receiver_inner.*pi.*L.*(T_absorber-
T_HTF_bulk);
>Error Notification
if(Re_D_receiver_inner<2300||Re_D_receiver_inner>5E6)
    disp(['ERROR: Re_D_receiver_inner= '
num2str(Re_D_receiver_inner)]);
end
if(Pr_T_bulk>2000)
    disp(['ERROR: Pr_T_bulk= ' num2str(Pr_T_bulk)]);
end
end

function [DeltaS] = calcS_air_Delta(Tin,Tout,Pin,Pout)
%Entropy generated by air compression/decompression. Can Newton
Iteration
%to find isentropic conditions. Units J/kg.
%Cp_air = (4.73252?10^(-11))T^4 + (-2.07769?10^(-7))T^3 +
(0.000238738)T^2

```

```

%+ (0.106809446)T + 999.9268821 J/kg.K
%DeltaS = (int(Cp/T)dT)final-(int(Cp/T)dT)initial -Rln(Pout/Pin)
R=8.3144598; %Gas constant J/K?1?mol?1
Mair=28.9647/1000; %kg/mol
a=4.73252E-11;b=-2.07769E-
7;c=0.000238738;d=0.106809446;e=999.9268821;
%convert Cp to J/mol.K to use in thermo equation
a=a.*Mair;b=b.*Mair;c=c.*Mair;d=d.*Mair;e=e.*Mair;
DeltaS_int_final=((a.*(Tout).^4)./4)+((b.*(Tout).^3)./3)+((c.*(Tou
t).^2)./2)+((d.*(Tout)))+e.*log(Tout);
DeltaS_int_initial=((a.*(Tin).^4)./4)+((b.*(Tin).^3)./3)+((c.*(Tin
).^2)./2)+((d.*(Tin)))+e.*log(Tin);
DeltaS_per_mol=DeltaS_int_final-DeltaS_int_initial-
R.*log(Pout./Pin);
DeltaS=DeltaS_per_mol./Mair;
end

```

```

function [T_cover_inner] =
calcT_cover_inner_known_conduction_Touter(L,k_cover,D_cover_inner,
D_cover_outer,T_cover_outer,Q)
%Calculate inner cover temp when known outer cover temp and Q that
needs to
%be conducted, for finding range of permissible absorber and cover
temps
%during iterations
T_cover_inner=(Q.*(log(D_cover_outer./D_cover_inner))./(2.*pi.*L.*
k_cover))+T_cover_outer;
end

```

```

function [T_total,cp] = calcT_Total_air(mdot,D,p_static,T_static)
%Find total temperature of flow in a pipe, so that turbharger maps
can be
%made to account for the affects of temperature and pressure on
enthalpy
%flow
[rho,cp,~,~,~,~] = AIRPROPERTIES(p_static,T_static,1,1,0,0,0,0);
Mair=28.9647/1000; %kg/mol
T_total=T_static+(((4.*mdot)./(pi.*(D.^2).*rho)).^2)./(2.*cp);
end

```

```

function [T_cover_inner,Q_inner,count] =
findT_cover_inner_air_vac(L,D_absorber,D_cover_inner,epsilon_absor
ber,epsilon_cover,T_absorber,p_vac,Q_target_losses,T_air)
%Newton's Bisection Method to find temp of cover inner surface
which
%balances energy from collector, HTF and losses
%Newton Iterations Setup
count = 1;
countlimit = 32; % 4 decimal places usually found at 24
iterations. This provides some wiggle room before terminating loop
solution_tolerance = 0.0001; % how tolerant root finding must be.
4 decimal places here
T_cover_inner_high = T_absorber;
T_cover_inner_low = T_air;
%Begin iteration
T_cover_inner_middle = (T_cover_inner_high+T_cover_inner_low)/2;
T_cover_inner_difference = T_cover_inner_high-T_cover_inner_low;
Q_rad_to_cover_low =
calcQ_absorber_to_cover_radiation(L,D_absorber,D_cover_inner,epsil
on_absorber,epsilon_cover,T_absorber,T_cover_inner_low);

```

```

Q_conv_to_cover_low =
calcQ_absorber_to_cover_convection(L,D_absorber,D_cover_inner,T_absorber,T_cover_inner_low,p_vac);
Q_rad_to_cover_middle =
calcQ_absorber_to_cover_radiation(L,D_absorber,D_cover_inner,epsilon_absorber,epsilon_cover,T_absorber,T_cover_inner_middle);
Q_conv_to_cover_middle =
calcQ_absorber_to_cover_convection(L,D_absorber,D_cover_inner,T_absorber,T_cover_inner_middle,p_vac);
Q_losses_low = Q_rad_to_cover_low+Q_conv_to_cover_low;
Q_losses_middle = Q_rad_to_cover_middle+Q_conv_to_cover_middle;
Q_difference_middle = Q_target_losses-Q_losses_middle;
Q_difference_low = Q_target_losses-Q_losses_low;
%catch if impossible to produce losses targeted
if(Q_difference_low>=Q_target_losses)
    T_cover_inner=T_cover_inner_low;
    Q_inner=Q_losses_low;
    return
end
while(T_cover_inner_difference>solution_tolerance &&
sign(Q_difference_middle)~=0 && count<=countlimit)
    if(sign(Q_difference_low)*sign(Q_difference_middle)==-1)
        T_cover_inner_high = T_cover_inner_middle;
    else
        T_cover_inner_low=T_cover_inner_middle;
    end
    T_cover_inner_middle =
(T_cover_inner_high+T_cover_inner_low)/2;
    T_cover_inner_difference = T_cover_inner_high-
T_cover_inner_low;
    Q_rad_to_cover_low =
calcQ_absorber_to_cover_radiation(L,D_absorber,D_cover_inner,epsilon_absorber,epsilon_cover,T_absorber,T_cover_inner_low);
    Q_conv_to_cover_low =
calcQ_absorber_to_cover_convection(L,D_absorber,D_cover_inner,T_absorber,T_cover_inner_low,p_vac);
    Q_rad_to_cover_middle =
calcQ_absorber_to_cover_radiation(L,D_absorber,D_cover_inner,epsilon_absorber,epsilon_cover,T_absorber,T_cover_inner_middle);
    Q_conv_to_cover_middle =
calcQ_absorber_to_cover_convection(L,D_absorber,D_cover_inner,T_absorber,T_cover_inner_middle,p_vac);
    Q_losses_low = Q_rad_to_cover_low+Q_conv_to_cover_low;
    Q_losses_middle =
Q_rad_to_cover_middle+Q_conv_to_cover_middle;
    Q_difference_middle = Q_target_losses-Q_losses_middle;
    Q_difference_low = Q_target_losses-Q_losses_low;
    count=count+1;
end %end of iterations with escape mechanisms
if(count>countlimit)
    disp('Count limit reached searching for T_Cover_inner');
end
T_cover_inner = T_cover_inner_middle; %Return temp and count. Must
check count to count limit to search for error.
Q_inner=Q_losses_middle;
end

function [T_cover_inner] =
findT_cover_inner_for_req_loss_known_Touter(Q_target,L,k_cover,D_c
over_inner,D_cover_outer,T_cover_outer)
%Find outer temp of cover inner face for known target losses and
cover

```

```

%outer temp
count = 1;
countlimit = 32; % 4 decimal places usually found at 24
iterations. This provides some wiggle room before terminating loop
solution_tolerance = 0.0001; % how tolerant root finding must be.
4 decimal places here
T_cover_high = 1380; %arbitrary, about melting point of copper
T_cover_low = T_cover_outer;
T_cover_mid = (T_cover_high+T_cover_low)/2;
T_cover_difference = T_cover_high-T_cover_low;
Q_losses_low =
calcQ_cover_conduction(L,k_cover,D_cover_inner,D_cover_outer,T_cov
er_low,T_cover_outer);
Q_losses_mid =
calcQ_cover_conduction(L,k_cover,D_cover_inner,D_cover_outer,T_cov
er_mid,T_cover_outer);
Q_difference_middle=Q_target-Q_losses_mid;
Q_difference_low = Q_target-Q_losses_low;
while(T_cover_difference>solution_tolerance &&
sign(Q_difference_middle)~=0 && count<=countlimit)
    if(sign(Q_difference_low)*sign(Q_difference_middle)==-
1)
        T_cover_high = T_cover_mid;
    else
        T_cover_low=T_cover_mid;
    end
    T_cover_mid = (T_cover_high+T_cover_low)/2;
    T_cover_difference = T_cover_high-T_cover_low;
    Q_losses_low =
calcQ_cover_conduction(L,k_cover,D_cover_inner,D_cover_outer,T_cov
er_low,T_cover_outer);
    Q_losses_mid =
calcQ_cover_conduction(L,k_cover,D_cover_inner,D_cover_outer,T_cov
er_mid,T_cover_outer);
    Q_difference_middle=Q_target-Q_losses_mid;
    Q_difference_low = Q_target-Q_losses_low;
    count=count+1;
end
T_cover_inner=T_cover_mid;
if(count>countlimit)
    disp('Error in finding root, T_cover_inner search
countlimit exceeded');
end
end

function [T_cover_outer,count] =
findT_cover_outer(L,k_cover,D_cover_inner,D_cover_outer,T_cover_in
ner,Q_target_losses)
%Newton's Bisection Method to find temp of cover outer surface
which
%balances conduction through cover equal to losses from absorber
surface to
%cover inner surface
%Newton Iterations Setup
count = 1;
countlimit = 32; % 4 decimal places usually found at 24
iterations. This provides some wiggle room before terminating loop
solution_tolerance = 0.0001; % how tolerant root finding must be.
4 decimal places here
T_cover_outer_high = T_cover_inner;

```

```

T_cover_outer_low = 0;
%Begin iteration
T_cover_outer_middle = (T_cover_outer_high+T_cover_outer_low)/2;
T_cover_outer_difference = T_cover_outer_high-T_cover_outer_low;
Q_cover_low=calcQ_cover_conduction(L,k_cover,D_cover_inner,D_cover_
_outer,T_cover_inner,T_cover_outer_low);
Q_cover_middle=calcQ_cover_conduction(L,k_cover,D_cover_inner,D_co
ver_outer,T_cover_inner,T_cover_outer_middle);
Q_difference_middle=Q_target_losses-Q_cover_middle;
Q_difference_low = Q_target_losses-Q_cover_low;
    while(T_cover_outer_difference>solution_tolerance &&
sign(Q_difference_middle)~=0 && count<=countlimit)
        if(sign(Q_difference_low)*sign(Q_difference_middle)==-
1)
            T_cover_outer_high = T_cover_outer_middle;
        else
            T_cover_outer_low=T_cover_outer_middle;
        end
        T_cover_outer_middle =
(T_cover_outer_high+T_cover_outer_low)/2;
        T_cover_outer_difference = T_cover_outer_high-
T_cover_outer_low;

Q_cover_low=calcQ_cover_conduction(L,k_cover,D_cover_inner,D_cover_
_outer,T_cover_inner,T_cover_outer_low);

Q_cover_middle=calcQ_cover_conduction(L,k_cover,D_cover_inner,D_co
ver_outer,T_cover_inner,T_cover_outer_middle);
Q_difference_middle=Q_target_losses-Q_cover_middle;
Q_difference_low = Q_target_losses-Q_cover_low;
count=count+1;
    end
    T_cover_outer=T_cover_outer_middle;
    if(count>countlimit)
        disp('Error in finding root, T_cover_outer search
countlimit exceeded');
    end
end

function [T_outer] =
findT_cover_outer_for_req_losses(Q_losses_target,L,D_outer,epsilon_
_outer,T_sky_blackbody,T_air,h_wind)
%Find outer temp of cover surface for known target losses
count = 1;
countlimit = 32; % 4 decimal places usually found at 24
iterations. This provides some wiggle room before terminating loop
solution_tolerance = 0.0001; % how tolerant root finding must be.
4 decimal places here
T_cover_high = 1380; %arbitrary, about melting point of copper
T_cover_low = T_air;
T_cover_mid = (T_cover_high+T_cover_low)/2;
T_cover_difference = T_cover_high-T_cover_low;
Q_losses_low =
calcQ_receiver_to_ambient_air_total(L,D_outer,T_cover_low,epsilon_
_outer,T_sky_blackbody,T_air,h_wind);
Q_losses_mid =
calcQ_receiver_to_ambient_air_total(L,D_outer,T_cover_mid,epsilon_
_outer,T_sky_blackbody,T_air,h_wind);
Q_difference_middle=Q_losses_target-Q_losses_mid;
Q_difference_low = Q_losses_target-Q_losses_low;

```

```

        while(T_cover_difference>solution_tolerance &&
sign(Q_difference_middle)~=0 && count<=countlimit)
            if(sign(Q_difference_low)*sign(Q_difference_middle)==-
1)
                T_cover_high = T_cover_mid;
            else
                T_cover_low=T_cover_mid;
            end
            T_cover_mid = (T_cover_high+T_cover_low)/2;
            T_cover_difference = T_cover_high-T_cover_low;
            Q_losses_low =
calcQ_receiver_to_ambient_air_total(L,D_outer,T_cover_low,epsilon_
outer,T_sky_blackbody,T_air,h_wind);
            Q_losses_mid =
calcQ_receiver_to_ambient_air_total(L,D_outer,T_cover_mid,epsilon_
outer,T_sky_blackbody,T_air,h_wind);
            Q_difference_middle=Q_losses_target-Q_losses_mid;
            Q_difference_low = Q_losses_target-Q_losses_low;
            count=count+1;
        end
        T_outer=T_cover_mid;
        if(count>countlimit)
            disp('Error in finding root, T_cover_outer search
countlimit exceeded');
        end
    end
end

```

```

function [T_out_air,count] = findT_HTF_Air_out(mdot,Qin,T_in)
%Find outlet temperature of air as the HTF in an absorber section
once Qin
%is known by Newton's Bisection Method
count = 1;
countlimit = 32; % 4 decimal places usually found at 24
iterations. This provides some wiggle room before terminating loop
solution_tolerance = 0.0001; % how tolerant root finding must be.
4 decimal places here
T_out_air_high = 2000; %Arbitrary high temp, higher than most
metal melting points
T_out_air_low = 0; %Arbitrary for establishing iterations
%Begin iteration setup
T_out_air_middle = (T_out_air_high+T_out_air_low)./2;
T_out_air_difference = T_out_air_high-T_out_air_low;
%Q_ref=((4.73252E-11).*(T.^5)./5)+((-2.07769E-
7).*(T.^4)./(4))+((0.000238738).*(T.^3)./(3))+((0.106809446).*(T.^
2)./(2))+999.9268821.*T)
Q_out_air_low = ((4.73252E-11).*((T_out_air_low.^5)-
(T_in.^5))./5)+((-2.07769E-7).*((T_out_air_low.^4)-
(T_in.^4))./(4))+((0.000238738).*((T_out_air_low.^3)-
(T_in.^3))./(3))+((0.106809446).*((T_out_air_low.^2)-
(T_in.^2))./(2))+999.9268821.*(T_out_air_low-T_in));
Q_out_air_middle = ((4.73252E-11).*((T_out_air_middle.^5)-
(T_in.^5))./5)+((-2.07769E-7).*((T_out_air_middle.^4)-
(T_in.^4))./(4))+((0.000238738).*((T_out_air_middle.^3)-
(T_in.^3))./(3))+((0.106809446).*((T_out_air_middle.^2)-
(T_in.^2))./(2))+999.9268821.*(T_out_air_middle-T_in));
Q_difference_middle = Qin-(mdot.*Q_out_air_middle);
Q_difference_low = Qin-(mdot.*Q_out_air_low);
        while(T_out_air_difference>solution_tolerance &&
sign(Q_difference_middle)~=0 && count<=countlimit)
            if(sign(Q_difference_low)*sign(Q_difference_middle)==-1)

```

```

        T_out_air_high = T_out_air_middle;
    else
        T_out_air_low=T_out_air_middle;
    end
    T_out_air_middle = (T_out_air_high+T_out_air_low)./2;
    T_out_air_difference = T_out_air_high-T_out_air_low;
    Q_out_air_low = ((4.73252E-11).*((T_out_air_low.^5)-
(T_in.^5))./5)+((-2.07769E-7).*((T_out_air_low.^4)-
(T_in.^4))./4))+((0.000238738).*((T_out_air_low.^3)-
(T_in.^3))./3))+((0.106809446).*((T_out_air_low.^2)-
(T_in.^2))./2))+999.9268821.*(T_out_air_low-T_in));
    Q_out_air_middle = ((4.73252E-11).*((T_out_air_middle.^5)-
(T_in.^5))./5)+((-2.07769E-7).*((T_out_air_middle.^4)-
(T_in.^4))./4))+((0.000238738).*((T_out_air_middle.^3)-
(T_in.^3))./3))+((0.106809446).*((T_out_air_middle.^2)-
(T_in.^2))./2))+999.9268821.*(T_out_air_middle-T_in));
    Q_difference_middle = Qin-(mdot.*Q_out_air_middle);
    Q_difference_low = Qin-(mdot.*Q_out_air_low);
    count=count+1;
end %end of iterations with escape mechanisms
T_out_air=T_out_air_middle;
end

```

```

function [T_out_marlothermSH,count] =
findT_HTF_MarlothermSH_out(mdot,Qin,T_in)
%Find outlet temperature of air as the HTF in an absorber section
once Qin
%is known by Newton's Bisection Method
count = 1;
countlimit = 32; % 4 decimal places usually found at 24
iterations. This provides some wiggle room before terminating loop
solution_tolerance = 0.0001; % how tolerant root finding must be.
4 decimal places here
T_out_marlothermSH_high = 2000; %Arbitrary high temp, higher than
most metal melting points
T_out_marlothermSH_low = 0; %Arbitrary for establishing iterations
%Begin iteration setup
T_out_marlothermSH_middle =
(T_out_marlothermSH_high+T_out_marlothermSH_low)./2;
T_out_marlothermSH_difference = T_out_marlothermSH_high-
T_out_marlothermSH_low;
%Q_ref=((1.14256228807313E-
07).*(T.^3)./(3))+((0.00368518354710305).*(T.^2)./(2))+1.47685714
285714.*T)
Q_out_marlothermSH_low = ((1.14256228807313E-
07).*((T_out_marlothermSH_low.^3)-
(T_in.^3))./3))+((0.00368518354710305).*((T_out_marlothermSH_low.^
2)-(T_in.^2))./2))+1.47685714285714.*(T_out_marlothermSH_low-
T_in));
Q_out_marlothermSH_middle = ((1.14256228807313E-
07).*((T_out_marlothermSH_middle.^3)-
(T_in.^3))./3))+((0.00368518354710305).*((T_out_marlothermSH_midd
le.^2)-
(T_in.^2))./2))+1.47685714285714.*(T_out_marlothermSH_middle-
T_in));
Q_out_marlothermSH_low=Q_out_marlothermSH_low.*1000;%kJ to J
Q_out_marlothermSH_middle=Q_out_marlothermSH_middle.*1000;%kJ to J
Q_difference_middle = Qin-(mdot.*Q_out_marlothermSH_middle);
Q_difference_low = Qin-(mdot.*Q_out_marlothermSH_low);
while(T_out_marlothermSH_difference>solution_tolerance &&
sign(Q_difference_middle)~=0 && count<=countlimit)
    if(sign(Q_difference_low)*sign(Q_difference_middle)==-1)

```

```

        T_out_marlothermSH_high =
T_out_marlothermSH_middle;
        else
T_out_marlothermSH_low=T_out_marlothermSH_middle;
        end
        T_out_marlothermSH_middle =
(T_out_marlothermSH_high+T_out_marlothermSH_low)./2;
        T_out_marlothermSH_difference = T_out_marlothermSH_high-
T_out_marlothermSH_low;
        Q_out_marlothermSH_low = ((1.14256228807313E-
07).*((T_out_marlothermSH_low.^3)-
(T_in.^3))./(3))+((0.00368518354710305).*((T_out_marlothermSH_low.^
2)-(T_in.^2))./(2)))+(1.47685714285714.*(T_out_marlothermSH_low-
T_in));
        Q_out_marlothermSH_middle = ((1.14256228807313E-
07).*((T_out_marlothermSH_middle.^3)-
(T_in.^3))./(3))+((0.00368518354710305).*((T_out_marlothermSH_midd
le.^2)-(T_in.^2))./(2)))+(1.47685714285714.*(T_out_marlothermSH_middle-
T_in));
        Q_out_marlothermSH_low=Q_out_marlothermSH_low.*1000;%kJ to
J

Q_out_marlothermSH_middle=Q_out_marlothermSH_middle.*1000;%kJ to J
        Q_difference_middle = Qin-
(mdot.*Q_out_marlothermSH_middle);
        Q_difference_low = Qin-(mdot.*Q_out_marlothermSH_low);
        count=count+1;
    end %end of iterations with escape mechanisms
T_out_marlothermSH=T_out_marlothermSH_middle;
end

function [T_out_water,count] = findT_HTF_Water_out(mdot,Qin,T_in)
%Find outlet temperature of air as the HTF in an absorber section
once Qin
%is known by Newton's Bisection Method
count = 1;
countlimit = 32; % 4 decimal places usually found at 24
iterations. This provides some wiggle room before terminating loop
solution_tolerance = 0.0001; % how tolerant root finding must be.
4 decimal places here
T_out_water_high = 500; %Arbitrary high temp, higher than most
metal melting points
T_out_water_low = 273; %Arbitrary for establishing iterations
%Begin iteration setup
T_out_water_middle = (T_out_water_high+T_out_water_low)./2;
T_out_water_difference = T_out_water_high-T_out_water_low;
A=3.81469814333049E-07;B=-
0.000567808525638521;C=0.32679547657916;D=-
84.65825105625;E=12410.839133103;
%integral_ref=(A.*T^5)./5+(B.*T^4)./4+(C.*T^3)./3+(D.*T^2)./2+E.*T
Q_out_water_low_intFinal=(A.*T_out_water_low^5)./5+(B.*T_out_water
_low^4)./4+(C.*T_out_water_low^3)./3+(D.*T_out_water_low^2)./2+E.*
T_out_water_low;
Q_out_water_low_intInitial=(A.*T_in^5)./5+(B.*T_in^4)./4+(C.*T_in^
3)./3+(D.*T_in^2)./2+E.*T_in;
Q_out_water_low=Q_out_water_low_intFinal-
Q_out_water_low_intInitial;
Q_out_water_middle_intFinal=(A.*T_out_water_middle^5)./5+(B.*T_out
_water_middle^4)./4+(C.*T_out_water_middle^3)./3+(D.*T_out_water_m
iddle^2)./2+E.*T_out_water_middle;

```

```

Q_out_water_middle_intInitial=Q_out_water_low_intInitial;
Q_out_water_middle=Q_out_water_middle_intFinal-
Q_out_water_middle_intInitial;
Q_difference_middle = Qin-(mdot.*Q_out_water_middle);
Q_difference_low = Qin-(mdot.*Q_out_water_low);
    while(T_out_water_difference>solution_tolerance &&
sign(Q_difference_middle)~=0 && count<=countlimit)
        if(sign(Q_difference_low)*sign(Q_difference_middle)==-1)
            T_out_water_high = T_out_water_middle;
        else
            T_out_water_low=T_out_water_middle;
        end
        T_out_water_middle =
(T_out_water_high+T_out_water_low)./2;
        T_out_water_difference = T_out_water_high-T_out_water_low;

Q_out_water_low_intFinal=(A.*T_out_water_low^5)./5+(B.*T_out_water
_low^4)./4+(C.*T_out_water_low^3)./3+(D.*T_out_water_low^2)./2+E.*
T_out_water_low;

Q_out_water_low_intInitial=(A.*T_in^5)./5+(B.*T_in^4)./4+(C.*T_in^
3)./3+(D.*T_in^2)./2+E.*T_in;
    Q_out_water_low=Q_out_water_low_intFinal-
Q_out_water_low_intInitial;

Q_out_water_middle_intFinal=(A.*T_out_water_middle^5)./5+(B.*T_out
_water_middle^4)./4+(C.*T_out_water_middle^3)./3+(D.*T_out_water_m
iddle^2)./2+E.*T_out_water_middle;
    Q_out_water_middle_intInitial=Q_out_water_low_intInitial;
    Q_out_water_middle=Q_out_water_middle_intFinal-
Q_out_water_middle_intInitial;
    Q_difference_middle = Qin-(mdot.*Q_out_water_middle);
    Q_difference_low = Qin-(mdot.*Q_out_water_low);
    count=count+1;
    end %end of iterations with escape mechanisms
T_out_water=T_out_water_middle;
end

function [T_sky_blackbody,h_wind] =
get_ambient_air_losses_constants(T_dewpoint_air,T_air,RelativeHumi
dity,p_air,windspeed,D_outer)
%Get constants for calcQ_receiver_to_ambient_air so that it doesnt
waste time
%redoing same calcs each iteration step. This and other
optimizations
%significantly speed up calculation speed (~3 hours to ~1 hour)
%T_sky_blackbody
sigma = 5.670373.*10.^-8;%Stephan-Boltzman Constant%
T_dewpoint_air_C = T_dewpoint_air-273.15;
epsilon_sky =
0.711+0.56.*(T_dewpoint_air_C./100)+0.73.*((T_dewpoint_air_C./100)
.^2);
T_sky_blackbody = ((8.778E-
13.*(T_air.^(5.852)).*(RelativeHumidity.^(0.07195)))./(epsilon_sky
.*sigma)).^(1/4);
%h_wind
[rho_air,~,k_air,~,mu_air,~] =
AIRPROPERTIES(p_air,T_air,1,0,1,0,1,0);
Re_air = rho_air.*windspeed.*D_outer./mu_air;
    if(Re_air<1000)
        Nu = 0.4+0.54.*(Re_air.^0.52);
    else

```

```

        Nu = 0.3.*(Re_air.^0.6);
    end
h_wind = Nu.*k_air./D_outer;
>Error notification for out of range of calculation
    if(Re_air<=0.1)
        disp(['WARNING: Re_air IS LOWER THAN THRESHOLD OF 0.1.
Re_air = ' num2str(Re_air)]);
    end
    if(Re_air>=50000)
        disp(['WARNING: Re_air IS HIGHER THAN THRESHOLD OF 50000.
Re_air = ' num2str(Re_air)]);
    end
end

```

```

function [ Tmax, Topt, eta_opt ] = getCSPdata( I,C,Tair )
%Calculate the Maximum receiver temperature, optimal heat engine
operating
%temperature and overall optimum efficiency for a solar
concentrated power
%heat engine
%Note that this function will accept a matrix of C, but not of I
or Tc
sigma = 5.670373.*10.^-8; %Stefan-Boltzmann constant in SI units
ToptCalc = 0;
eta_optCalc = 0; %Variable size matrices for Topt and eta_opt in
the case
        %of a matrix of C values parsed
Tmax=(I.*C./sigma).^0.25; %Calculate matrix of Tmax values
    for n =1:length(C)
        v=[1, -0.75.*Tair, 0, 0 , 0, -Tair.*I.*C(n)./(4.*sigma)];
        s = roots(v);
        s = s(imag(s)==0);%Use only the real root calculated
        ToptCalc(n) = s;
        eta_optCalc(n) = (1-
(sigma.*(ToptCalc(n).^4))./(I.*C(n))).*(1-(Tair)./(ToptCalc(n)));
    end
eta_opt = eta_optCalc; %returns the variable size matrix as an
absolute
        %sized matrix matching C
Topt = ToptCalc;
end

```

```

function [cp_marlo,k_marlo,mu_marlo] = MarlothermSH_properties(T)
%p in Pa, T in K, rho in kg/m3, cp in J/kg.K, k in W/m.K,
%mu in kg/m.s, nu in m2/s
%input T get desired values for heat transfer to Marlotherm as HTF
t=T-273.15; % functions require T in Celcius
if(t>350 || t<-5)
    disp(['Error, marlothermSH temperature out of bounds Tin K= ',
num2str(T)])
end
cp_marlo=1000.*(1.14256E-
07.*(t.^2)+0.003685184.*t+1.476857143);%kJ to J
k_marlo=(-2.76426E-09).*(t.^2)+(-0.000130847).*t+(0.133201504);
rho_marlo=(-4.60711E-06).*(t.^2)+(-
0.713166003).*(t)+(1058.311278);
nu_marlo_inv=((2.03452E-05).*(t.^2)+(0.001417023).*(t)+(-
0.02232397));%
nu_marlo=1./nu_marlo_inv;%mm^2/s
nu_marlo=nu_marlo./1000000;%mm^2/s to m^2/s

```

```

mu_marlo=rho_marlo.*nu_marlo;
end

function
[Q_HTF,Q_loss_rad,Q_loss_conv,T_out,T_absorber,T_cover_inner,T_cov
er_outer,count,Lsection] =
sim_Receiver_air_HTF_air_vac(numSections,L,mdot,Q_in,T_HTF_bulk,p_
HTF,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsil
on_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,Relativ
eHumidity)
%Perform sim_Receiver_Section_air_HTF_air_vac as an intigral over
a number
%of specified sections. The higher the number of sections, the
more
%accurate. Linear processing cost.
if(numSections==1)%catch only doing a single section

[Q_HTF(1),Q_loss_rad(1),Q_loss_conv(1),T_out(1),T_absorber(1),T_co
ver_inner(1),T_cover_outer(1),count(1)]=sim_Receiver_Section_air_H
TF_air_vac(L,mdot,Q_in,T_HTF_bulk,p_HTF,T_air,D_absorber,D_cover_i
nner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_co
ver,windspeed,T_dewpoint_air,RelativeHumidity);
Lsection=L;
else

Q_HTF(numSections)=0;Q_loss_rad(numSections)=0;Q_loss_conv(numSect
ions)=0;T_out(numSections)=0;

T_absorber(numSections)=0;T_cover_inner(numSections)=0;T_cover_out
er(numSections)=0;count(numSections)=0;
Lsection=linspace(0,L,numSections+1);
Lsection=Lsection(2:end);%get rid of 0 distance point - only
want spacings
Lsize=Lsection(2)-Lsection(1);
Qsection=Q_in./numSections;%assume uniform distribution of Q
across receiver
%calc first point data from inlet conditions

[Q_HTF(1),Q_loss_rad(1),Q_loss_conv(1),T_out(1),T_absorber(1),T_co
ver_inner(1),T_cover_outer(1),count(1)]=sim_Receiver_Section_air_H
TF_air_vac(Lsize,mdot,Qsection,T_HTF_bulk,p_HTF,T_air,D_absorber,D
_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,ep
silon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
%calc point data with info from previous point
for i=2:numSections

[Q_HTF(i),Q_loss_rad(i),Q_loss_conv(i),T_out(i),T_absorber(i),T_co
ver_inner(i),T_cover_outer(i),count(i)]=sim_Receiver_Section_air_H
TF_air_vac(Lsize,mdot,Qsection,T_out(i-
1),p_HTF,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,
epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,Re
lativeHumidity);
end
end
end

function
[Q_HTF,Q_loss_rad,Q_loss_conv,T_out,T_absorber,count,Lsection] =
sim_Receiver_air_HTF_coverless(numSections,L,mdot,Q_in,T_HTF_bulk,

```

```

p_HTF,T_air,D_absorber,p_air,epsilon_absorber,windspeed,T_dewpoint
_air,RelativeHumidity)
%Similar to sim_Receiver_air_HTF_air_vac but without a cover over
the
%receiver
if(numSections==1)%catch only doing a single section

[Q_HTF(1),Q_loss_rad(1),Q_loss_conv(1),T_absorber(1),T_out(1),count
(1)]=sim_Receiver_Section_air_HTF_coverless(L,mdot,Q_in,T_HTF_bulk,
p_HTF,T_air,D_absorber,p_air,epsilon_absorber,windspeed,T_dewpoint
_air,RelativeHumidity);
    Lsection=L;
else

Q_HTF(numSections)=0;Q_loss_rad(numSections)=0;Q_loss_conv(numSections)=0;T_out(numSections)=0;
    T_absorber(numSections)=0;count(numSections)=0;
    Lsection=linspace(0,L,numSections+1);
    Lsection=Lsection(2:end);%get rid of 0 distance point - only
want spacings
    Lsize=Lsection(2)-Lsection(1);
    Qsection=Q_in./numSections;%assume uniform distribution of Q
across receiver
    %calc first point data from inlet conditions

[Q_HTF(1),Q_loss_rad(1),Q_loss_conv(1),T_absorber(1),T_out(1),count
(1)]=sim_Receiver_Section_air_HTF_coverless(Lsize,mdot,Qsection,T
_HTF_bulk,p_HTF,T_air,D_absorber,p_air,epsilon_absorber,windspeed,
T_dewpoint_air,RelativeHumidity);
    %calc point data with info from previous point
    for i=2:numSections

[Q_HTF(i),Q_loss_rad(i),Q_loss_conv(i),T_absorber(i),T_out(i),count
(i)]=sim_Receiver_Section_air_HTF_coverless(Lsize,mdot,Qsection,T
_out(i-
1),p_HTF,T_air,D_absorber,p_air,epsilon_absorber,windspeed,T_dewpoint
_air,RelativeHumidity);
        end
    end
end

function
[Q_HTF,Q_loss_rad,Q_loss_conv,T_out,T_absorber,T_cover_inner,T_cover
_outer,count,Lsection] =
sim_Receiver_marlothermSH_HTF_air_vac(numSections,L,mdot,Q_in,T_HTF
_bulk,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon
_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity)
%Perform sim_Receiver_Section_air_HTF_air_vac as an integral over
a number
%of specified sections. The higher the number of sections, the
more
%accurate. Linear processing cost.
if(numSections==1)%catch only doing a single section

[Q_HTF(1),Q_loss_rad(1),Q_loss_conv(1),T_out(1),T_absorber(1),T_cover
_inner(1),T_cover_outer(1),count(1)]=sim_Receiver_Section_marlothermSH
_HTF_air_vac(L,mdot,Q_in,T_HTF_bulk,T_air,D_absorber,D_cover_inner,D_cover
_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint
_air,RelativeHumidity);

```

```

    Lsection=L;
else

Q_HTF(numSections)=0;Q_loss_rad(numSections)=0;Q_loss_conv(numSections)=0;T_out(numSections)=0;

T_absorber(numSections)=0;T_cover_inner(numSections)=0;T_cover_outer(numSections)=0;count(numSections)=0;
    Lsection=linspace(0,L,numSections+1);
    Lsection=Lsection(2:end);%get rid of 0 distance point - only want spacings
    Lsize=Lsection(2)-Lsection(1);
    Qsection=Q_in./numSections;%assume uniform distribution of Q across receiver
    %calc first point data from inlet conditions

[Q_HTF(1),Q_loss_rad(1),Q_loss_conv(1),T_out(1),T_absorber(1),T_cover_inner(1),T_cover_outer(1),count(1)]=sim_Receiver_Section_marlothermSH_HTF_air_vac(Lsize,mdot,Qsection,T_HTF_bulk,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
    %calc point data with info from previous point
    for i=2:numSections

[Q_HTF(i),Q_loss_rad(i),Q_loss_conv(i),T_out(i),T_absorber(i),T_cover_inner(i),T_cover_outer(i),count(i)]=sim_Receiver_Section_marlothermSH_HTF_air_vac(Lsize,mdot,Qsection,T_out(i-1),T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);
    end
end
end

function
[Q_HTF,Q_loss_rad,Q_loss_conv,T_out,T_absorber,T_cover_inner,T_cover_outer,count] =
sim_Receiver_Section_air_HTF_air_vac(L,mdot,Q_in,T_HTF_bulk,p_HTF,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity)
%Iterative calculation to find variables of an operating section of the
%receiver.
T_max = 5000;%Arbitrary maximum material temperature. 1360K is approximate copper melting point. 5000K for testing purposes
%Bisection Iterations Setup
iterations_count = 1;
countlimit = 32; % 4 decimal places usually found at 24 iterations. This provides some wiggle room before terminating loop
solution_tolerance = 0.0001; % how tolerant root finding must be. 4 decimal places here
[T_sky_blackbody,h_wind] =
get_ambient_air_losses_constants(T_dewpoint_air,T_air,RelativeHumidity,p_air,windspeed,D_cover_outer);
T_absorber_high = T_max;
T_absorber_low = max(T_air,T_HTF_bulk);
%Find Max Temp of absorber such that all Q_in goes to HTF. If this is not
%done, error persists in search for T_cover_outer in next bisection search
%for T_absorber i.e. real root for that search does not exist

```

```

T_max_highT_high=T_absorber_high;
T_max_highT_low=T_absorber_low;
T_max_highT_mid=(T_max_highT_high+T_max_highT_low)/2;
T_max_highT_difference=T_max_highT_high-T_max_highT_low;
Q_HTF_highT_mid =
calcQ_receiver_to_HTF_for_air(D_absorber,mdot,T_max_highT_mid,T_HTF
F_bulk,p_HTF,L);
Q_HTF_highT_low =
calcQ_receiver_to_HTF_for_air(D_absorber,mdot,T_max_highT_low,T_HTF
F_bulk,p_HTF,L);
Q_difference_middle = Q_in-Q_HTF_highT_mid;
Q_difference_low = Q_in-Q_HTF_highT_low;
    while(T_max_highT_difference>solution_tolerance &&
sign(Q_difference_middle)~=0 && iterations_count<=countlimit)
        if(sign(Q_difference_low)*sign(Q_difference_middle)==-1)
            T_max_highT_high = T_max_highT_mid;
        else
            T_max_highT_low=T_max_highT_mid;
        end
        T_max_highT_mid=(T_max_highT_high+T_max_highT_low)/2;
        T_max_highT_difference=T_max_highT_high-T_max_highT_low;
        Q_HTF_highT_mid =
calcQ_receiver_to_HTF_for_air(D_absorber,mdot,T_max_highT_mid,T_HTF
F_bulk,p_HTF,L);
        Q_HTF_highT_low =
calcQ_receiver_to_HTF_for_air(D_absorber,mdot,T_max_highT_low,T_HTF
F_bulk,p_HTF,L);
        Q_difference_middle = Q_in-Q_HTF_highT_mid;
        Q_difference_low = Q_in-Q_HTF_highT_low;
        iterations_count=iterations_count+1;
    end %end of iterations with escape mechanisms
T_max_all_Q_to_HTF=T_max_highT_mid;
if(iterations_count>countlimit)
    disp('Error: HTF cannot absorb all Qin for given length of
pipe even with no losses');
end
%Begin bisection search for T_absorber
count=1;
T_absorber_high = T_max_all_Q_to_HTF;
T_absorber_mid = (T_absorber_high+T_absorber_low)/2;
T_absorber_difference = T_absorber_high-T_absorber_low;
Q_HTF_mid =
calcQ_receiver_to_HTF_for_air(D_absorber,mdot,T_absorber_mid,T_HTF
_bulk,p_HTF,L);
Q_required_loss_mid = Q_in-Q_HTF_mid;
T_cover_outer_mid =
findT_cover_outer_for_req_losses(Q_required_loss_mid,L,D_cover_out
er,epsilon_cover,T_sky_blackbody,T_air,h_wind);
T_cover_inner_mid =
findT_cover_inner_for_req_loss_known_Touter(Q_required_loss_mid,L,
k_cover,D_cover_inner,D_cover_outer,T_cover_outer_mid);
Q_to_cover_mid =
calcQ_absorber_to_cover_total(L,D_absorber,D_cover_inner,T_absorbe
r_mid,T_cover_inner_mid,p_vac,epsilon_absorber,epsilon_cover);
Q_error_mid=Q_required_loss_mid-Q_to_cover_mid;
Q_HTF_low =
calcQ_receiver_to_HTF_for_air(D_absorber,mdot,T_absorber_low,T_HTF
_bulk,p_HTF,L);
Q_required_loss_low = Q_in-Q_HTF_low;
T_cover_outer_low =
findT_cover_outer_for_req_losses(Q_required_loss_low,L,D_cover_out
er,epsilon_cover,T_sky_blackbody,T_air,h_wind);

```

```

T_cover_inner_low =
findT_cover_inner_for_req_loss_known_Touter(Q_required_loss_low,L,
k_cover,D_cover_inner,D_cover_outer,T_cover_outer_low);
Q_to_cover_low =
calcQ_absorber_to_cover_total(L,D_absorber,D_cover_inner,T_absorber_low,T_cover_inner_low,p_vac,epsilon_absorber,epsilon_cover);
Q_error_low=Q_required_loss_low-Q_to_cover_low;
while(T_absorber_difference>solution_tolerance &&
sign(Q_error_mid)~=0 && count<=countlimit)
    if(sign(Q_error_low)*sign(Q_error_mid)==-1)
        T_absorber_high = T_absorber_mid;
    else
        T_absorber_low=T_absorber_mid;
    end
    T_absorber_mid = (T_absorber_high+T_absorber_low)/2;
    T_absorber_difference = T_absorber_high-
T_absorber_low;
    Q_HTF_mid =
calcQ_receiver_to_HTF_for_air(D_absorber,mdot,T_absorber_mid,T_HTF_bulk,p_HTF,L);
    Q_required_loss_mid = Q_in-Q_HTF_mid;
    T_cover_outer_mid =
findT_cover_outer_for_req_losses(Q_required_loss_mid,L,D_cover_outer,epsilon_cover,T_sky_blackbody,T_air,h_wind);
    T_cover_inner_mid =
findT_cover_inner_for_req_loss_known_Touter(Q_required_loss_mid,L,
k_cover,D_cover_inner,D_cover_outer,T_cover_outer_mid);
    Q_to_cover_mid =
calcQ_absorber_to_cover_total(L,D_absorber,D_cover_inner,T_absorber_mid,T_cover_inner_mid,p_vac,epsilon_absorber,epsilon_cover);
    Q_error_mid=Q_required_loss_mid-Q_to_cover_mid;
    Q_HTF_low =
calcQ_receiver_to_HTF_for_air(D_absorber,mdot,T_absorber_low,T_HTF_bulk,p_HTF,L);
    Q_required_loss_low = Q_in-Q_HTF_low;
    T_cover_outer_low =
findT_cover_outer_for_req_losses(Q_required_loss_low,L,D_cover_outer,epsilon_cover,T_sky_blackbody,T_air,h_wind);
    T_cover_inner_low =
findT_cover_inner_for_req_loss_known_Touter(Q_required_loss_low,L,
k_cover,D_cover_inner,D_cover_outer,T_cover_outer_low);
    Q_to_cover_low =
calcQ_absorber_to_cover_total(L,D_absorber,D_cover_inner,T_absorber_low,T_cover_inner_low,p_vac,epsilon_absorber,epsilon_cover);
    Q_error_low=Q_required_loss_low-Q_to_cover_low;
    count=count+1;
end
T_absorber=T_absorber_mid;
if(count>countlimit)
    disp('Error in T_absorber, search countlimit exceeded');
end
%assign outputs correctly
Q_HTF=Q_HTF_mid;
T_cover_inner=T_cover_inner_mid;
T_cover_outer=T_cover_outer_mid;
[Q_loss_rad,Q_loss_conv]=calcQ_receiver_to_ambient_air(L,D_cover_outer,T_cover_outer,epsilon_cover,T_sky_blackbody,T_air,h_wind);
T_out=findT_HTF_Air_out(mdot,Q_HTF,T_HTF_bulk);
end

```

```

function
[Q_HTF,Q_loss_rad,Q_loss_conv,T_absorber,T_out,iterations_count] =
sim_Receiver_Section_air_HTF_coverless(L,mdot,Q_in,T_HTF_bulk,p_HTF,T_air,D_absorber,p_air,epsilon_absorber,windspeed,T_dewpoint_air,RelativeHumidity)
%Iterative calculation to find variables of an operating section
of the
%receiver.
T_max = 1360;%Arbitrary maximum material temperature. 1360K is
approximate copper melting point.
%Newton Iterations Setup
iterations_count = 1;
countlimit = 32; % 4 decimal places usually found at 24
iterations. This provides some wiggle room before terminating loop
solution_tolerance = 0.0001; % how tolerant root finding must be.
4 decimal places here
[T_sky_blackbody,h_wind] =
get_ambient_air_losses_constants(T_dewpoint_air,T_air,RelativeHumidity,p_air,windspeed,D_absorber);
T_absorber_high = T_max;
T_absorber_low = max(T_air,T_HTF_bulk);
T_absorber_middle = (T_absorber_high+T_absorber_low)/2;
T_absorber_difference = T_absorber_high-T_absorber_low;
Q_HTF_low =
calcQ_receiver_to_HTF_for_air(D_absorber,mdot,T_absorber_low,T_HTF_bulk,p_HTF,L);
[Q_radiation_atm_low,Q_convection_atm_low] =
calcQ_receiver_to_ambient_air(L,D_absorber,T_absorber_low,epsilon_absorber,T_sky_blackbody,T_air,h_wind);
Q_losses_total_low = Q_radiation_atm_low+Q_convection_atm_low;
Q_difference_low=Q_in-Q_HTF_low-Q_losses_total_low;
Q_HTF_middle =
calcQ_receiver_to_HTF_for_air(D_absorber,mdot,T_absorber_middle,T_HTF_bulk,p_HTF,L);
[Q_radiation_atm_middle,Q_convection_atm_middle] =
calcQ_receiver_to_ambient_air(L,D_absorber,T_absorber_middle,epsilon_absorber,T_sky_blackbody,T_air,h_wind);
Q_losses_total_middle =
Q_radiation_atm_middle+Q_convection_atm_middle;
Q_difference_middle=Q_in-Q_HTF_middle-Q_losses_total_middle;
%Iterations begin
while(T_absorber_difference>solution_tolerance &&
sign(Q_difference_middle)~=0 && iterations_count<=countlimit)
    if(sign(Q_difference_low)*sign(Q_difference_middle)==-1)
        T_absorber_high = T_absorber_middle;
    else
        T_absorber_low=T_absorber_middle;
    end
    T_absorber_middle = (T_absorber_high+T_absorber_low)/2;
    T_absorber_difference = T_absorber_high-T_absorber_low;
    Q_HTF_low =
calcQ_receiver_to_HTF_for_air(D_absorber,mdot,T_absorber_low,T_HTF_bulk,p_HTF,L);
    [Q_radiation_atm_low,Q_convection_atm_low] =
calcQ_receiver_to_ambient_air(L,D_absorber,T_absorber_low,epsilon_absorber,T_sky_blackbody,T_air,h_wind);
    Q_losses_total_low =
Q_radiation_atm_low+Q_convection_atm_low;
    Q_difference_low=Q_in-Q_HTF_low-Q_losses_total_low;
    Q_HTF_middle =
calcQ_receiver_to_HTF_for_air(D_absorber,mdot,T_absorber_middle,T_HTF_bulk,p_HTF,L);

```

```

        [Q_radiation_atm_middle,Q_convection_atm_middle] =
        calcQ_receiver_to_ambient_air(L,D_absorber,T_absorber_middle,epsilon_
        on_absorber,T_sky_blackbody,T_air,h_wind);
        Q_losses_total_middle =
        Q_radiation_atm_middle+Q_convection_atm_middle;
        Q_difference_middle=Q_in-Q_HTF_middle-
        Q_losses_total_middle;
        iterations_count=iterations_count+1;
    end %end of iterations with escape mechanisms
    Q_HTF=Q_HTF_middle;
    Q_loss_rad=Q_radiation_atm_middle;
    Q_loss_conv=Q_convection_atm_middle;
    T_absorber=T_absorber_middle;
    T_out=findT_HTF_Air_out(mdot,Q_HTF,T_HTF_bulk);
end

```

```

function
[Q_HTF,Q_loss_rad,Q_loss_conv,T_out,T_absorber,T_cover_inner,T_cov
er_outer,count] =
sim_Receiver_Section_marlothermSH_HTF_air_vac(L,mdot,Q_in,T_HTF_bu
lk,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilo
n_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,Relative
Humidity)
%Iterative calculation to find variables of an operating section
of the
%receiver.
T_max = 623;
iterations_count = 1;
countlimit = 32; % 4 decimal places usually found at 24
iterations. This provides some wiggle room before terminating loop
solution_tolerance = 0.0001; % how tolerant root finding must be.
4 decimal places here
[T_sky_blackbody,h_wind] =
get_ambient_air_losses_constants(T_dewpoint_air,T_air,RelativeHumi
dity,p_air,windspeed,D_cover_outer);
T_absorber_high = T_max;
T_absorber_low = max(T_air,T_HTF_bulk);
%Find Max Temp of absorber such that all Q_in goes to HTF. If this
is not
%done, error persists in search for T_cover_outer in next
bisection search
%for T_absorber i.e. real root for that search does not exist
T_max_highT_high=T_absorber_high;
T_max_highT_low=T_absorber_low;
T_max_highT_mid=(T_max_highT_high+T_max_highT_low)/2;
T_max_highT_difference=T_max_highT_high-T_max_highT_low;
Q_HTF_highT_mid =
calcQ_receiver_to_HTF_for_marlothermSH(D_absorber,mdot,T_max_highT
_mid,T_HTF_bulk,L);
Q_HTF_highT_low =
calcQ_receiver_to_HTF_for_marlothermSH(D_absorber,mdot,T_max_highT
_low,T_HTF_bulk,L);
Q_difference_middle = Q_in-Q_HTF_highT_mid;
Q_difference_low = Q_in-Q_HTF_highT_low;
while(T_max_highT_difference>solution_tolerance &&
sign(Q_difference_middle)~=0 && iterations_count<=countlimit)
    if(sign(Q_difference_low)*sign(Q_difference_middle)==-1)
        T_max_highT_high = T_max_highT_mid;
    else
        T_max_highT_low=T_max_highT_mid;
    end
end

```

```

end
T_max_highT_mid=(T_max_highT_high+T_max_highT_low)/2;
T_max_highT_difference=T_max_highT_high-T_max_highT_low;
Q_HTF_highT_mid =
calcQ_receiver_to_HTF_for_marlothermSH(D_absorber,mdot,T_max_highT
_mid,T_HTF_bulk,L);
Q_HTF_highT_low =
calcQ_receiver_to_HTF_for_marlothermSH(D_absorber,mdot,T_max_highT
_low,T_HTF_bulk,L);
Q_difference_middle = Q_in-Q_HTF_highT_mid;
Q_difference_low = Q_in-Q_HTF_highT_low;
iterations_count=iterations_count+1;
end %end of iterations with escape mechanisms
T_max_all_Q_to_HTF=T_max_highT_mid;
if(iterations_count>countlimit)
disp('Error: HTF cannot absorb all Qin for given length of
pipe even with no losses');
end
%Begin bisection search for T_absorber
count=1;
T_absorber_high = T_max_all_Q_to_HTF;
T_absorber_mid = (T_absorber_high+T_absorber_low)/2;
T_absorber_difference = T_absorber_high-T_absorber_low;
Q_HTF_mid =
calcQ_receiver_to_HTF_for_marlothermSH(D_absorber,mdot,T_absorber_
mid,T_HTF_bulk,L);
Q_required_loss_mid = Q_in-Q_HTF_mid;
T_cover_outer_mid =
findT_cover_outer_for_req_losses(Q_required_loss_mid,L,D_cover_out
er,epsilon_cover,T_sky_blackbody,T_air,h_wind);
T_cover_inner_mid =
findT_cover_inner_for_req_loss_known_Touter(Q_required_loss_mid,L,
k_cover,D_cover_inner,D_cover_outer,T_cover_outer_mid);
Q_to_cover_mid =
calcQ_absorber_to_cover_total(L,D_absorber,D_cover_inner,T_absorbe
r_mid,T_cover_inner_mid,p_vac,epsilon_absorber,epsilon_cover);
Q_error_mid=Q_required_loss_mid-Q_to_cover_mid;
Q_HTF_low =
calcQ_receiver_to_HTF_for_marlothermSH(D_absorber,mdot,T_absorber_
low,T_HTF_bulk,L);
Q_required_loss_low = Q_in-Q_HTF_low;
T_cover_outer_low =
findT_cover_outer_for_req_losses(Q_required_loss_low,L,D_cover_out
er,epsilon_cover,T_sky_blackbody,T_air,h_wind);
T_cover_inner_low =
findT_cover_inner_for_req_loss_known_Touter(Q_required_loss_low,L,
k_cover,D_cover_inner,D_cover_outer,T_cover_outer_low);
Q_to_cover_low =
calcQ_absorber_to_cover_total(L,D_absorber,D_cover_inner,T_absorbe
r_low,T_cover_inner_low,p_vac,epsilon_absorber,epsilon_cover);
Q_error_low=Q_required_loss_low-Q_to_cover_low;
while(T_absorber_difference>solution_tolerance &&
sign(Q_error_mid)~=0 && count<=countlimit)
if(sign(Q_error_low)*sign(Q_error_mid)==-1)
T_absorber_high = T_absorber_mid;
else
T_absorber_low=T_absorber_mid;
end
T_absorber_mid = (T_absorber_high+T_absorber_low)/2;
T_absorber_difference = T_absorber_high-T_absorber_low;

```

```

        Q_HTF_mid =
        calcQ_receiver_to_HTF_for_marlothermSH(D_absorber,mdot,T_absorber_
        mid,T_HTF_bulk,L);
        Q_required_loss_mid = Q_in-Q_HTF_mid;
        T_cover_outer_mid =
        findT_cover_outer_for_req_losses(Q_required_loss_mid,L,D_cover_out
        er,epsilon_cover,T_sky_blackbody,T_air,h_wind);
        T_cover_inner_mid =
        findT_cover_inner_for_req_loss_known_Touter(Q_required_loss_mid,L,
        k_cover,D_cover_inner,D_cover_outer,T_cover_outer_mid);
        Q_to_cover_mid =
        calcQ_absorber_to_cover_total(L,D_absorber,D_cover_inner,T_absorbe
        r_mid,T_cover_inner_mid,p_vac,epsilon_absorber,epsilon_cover);
        Q_error_mid=Q_required_loss_mid-Q_to_cover_mid;
        Q_HTF_low =
        calcQ_receiver_to_HTF_for_marlothermSH(D_absorber,mdot,T_absorber_
        low,T_HTF_bulk,L);
        Q_required_loss_low = Q_in-Q_HTF_low;
        T_cover_outer_low =
        findT_cover_outer_for_req_losses(Q_required_loss_low,L,D_cover_out
        er,epsilon_cover,T_sky_blackbody,T_air,h_wind);
        T_cover_inner_low =
        findT_cover_inner_for_req_loss_known_Touter(Q_required_loss_low,L,
        k_cover,D_cover_inner,D_cover_outer,T_cover_outer_low);
        Q_to_cover_low =
        calcQ_absorber_to_cover_total(L,D_absorber,D_cover_inner,T_absorbe
        r_low,T_cover_inner_low,p_vac,epsilon_absorber,epsilon_cover);
        Q_error_low=Q_required_loss_low-Q_to_cover_low;
        count=count+1;
    end
    T_absorber=T_absorber_mid;
    if(count>countlimit)
        disp('Error in T_absorber, search countlimit exceeded');
    end
    %assign outputs correctly
    Q_HTF=Q_HTF_mid;
    T_cover_inner=T_cover_inner_mid;
    T_cover_outer=T_cover_outer_mid;
    [Q_loss_rad,Q_loss_conv]=calcQ_receiver_to_ambient_air(L,D_cover_o
    uter,T_cover_outer,epsilon_cover,T_sky_blackbody,T_air,h_wind);
    T_out=findT_HTF_MarlothermSH_out(mdot,Q_HTF,T_HTF_bulk);
end

function
[Q_HTF,Q_loss_rad,Q_loss_conv,T_absorber,T_out,iterations_count] =
sim_Receiver_Section_waterLiq_HTF_coverless(L,mdot,Q_in,T_HTF_bulk
,T_air,D_absorber,p_air,epsilon_absorber,windspeed,T_dewpoint_air,
RelativeHumidity)
%Iterative calculation to find variables of an operating section
of the
%receiver.
T_max = 623.14;%Arbitrary maximum material temperature.
%Newton Iterations Setup
iterations_count = 1;
countlimit = 32; % 4 decimal places usually found at 24
iterations. This provides some wiggle room before terminating loop
solution_tolerance = 0.0001; % how tolerant root finding must be.
4 decimal places here

```

```

[T_sky_blackbody,h_wind] =
get_ambient_air_losses_constants(T_dewpoint_air,T_air,RelativeHumidity,p_air,windspeed,D_absorber);
T_absorber_high = T_max;
T_absorber_low = max(T_air,T_HTF_bulk);
T_absorber_middle = (T_absorber_high+T_absorber_low)/2;
T_absorber_difference = T_absorber_high-T_absorber_low;
Q_HTF_low =
calcQ_receiver_to_HTF_for_waterLiq(D_absorber,mdot,T_absorber_low,
T_HTF_bulk,L);
[Q_radiation_atm_low,Q_convection_atm_low] =
calcQ_receiver_to_ambient_air(L,D_absorber,T_absorber_low,epsilon_absorber,T_sky_blackbody,T_air,h_wind);
Q_losses_total_low = Q_radiation_atm_low+Q_convection_atm_low;
Q_difference_low=Q_in-Q_HTF_low-Q_losses_total_low;
Q_HTF_middle =
calcQ_receiver_to_HTF_for_waterLiq(D_absorber,mdot,T_absorber_middle,
T_HTF_bulk,L);
[Q_radiation_atm_middle,Q_convection_atm_middle] =
calcQ_receiver_to_ambient_air(L,D_absorber,T_absorber_middle,epsilon_absorber,T_sky_blackbody,T_air,h_wind);
Q_losses_total_middle =
Q_radiation_atm_middle+Q_convection_atm_middle;
Q_difference_middle=Q_in-Q_HTF_middle-Q_losses_total_middle;
%Iterations begin
while(T_absorber_difference>solution_tolerance &&
sign(Q_difference_middle)~=0 && iterations_count<=countlimit)
    if(sign(Q_difference_low)*sign(Q_difference_middle)==-1)
        T_absorber_high = T_absorber_middle;
    else
        T_absorber_low=T_absorber_middle;
    end
    T_absorber_middle = (T_absorber_high+T_absorber_low)/2;
    T_absorber_difference = T_absorber_high-T_absorber_low;
    Q_HTF_low =
calcQ_receiver_to_HTF_for_waterLiq(D_absorber,mdot,T_absorber_low,
T_HTF_bulk,L);
    [Q_radiation_atm_low,Q_convection_atm_low] =
calcQ_receiver_to_ambient_air(L,D_absorber,T_absorber_low,epsilon_absorber,T_sky_blackbody,T_air,h_wind);
    Q_losses_total_low =
Q_radiation_atm_low+Q_convection_atm_low;
    Q_difference_low=Q_in-Q_HTF_low-Q_losses_total_low;
    Q_HTF_middle =
calcQ_receiver_to_HTF_for_waterLiq(D_absorber,mdot,T_absorber_middle,
T_HTF_bulk,L);
    [Q_radiation_atm_middle,Q_convection_atm_middle] =
calcQ_receiver_to_ambient_air(L,D_absorber,T_absorber_middle,epsilon_absorber,T_sky_blackbody,T_air,h_wind);
    Q_losses_total_middle =
Q_radiation_atm_middle+Q_convection_atm_middle;
    Q_difference_middle=Q_in-Q_HTF_middle-
Q_losses_total_middle;
    iterations_count=iterations_count+1;
end %end of iterations with escape mechanisms
Q_HTF=Q_HTF_middle;
Q_loss_rad=Q_radiation_atm_middle;
Q_loss_conv=Q_convection_atm_middle;
T_absorber=T_absorber_middle;
T_out=findT_HTF_Water_out(mdot,Q_HTF,T_HTF_bulk);
end

```

```

function
[Q_HTF,Q_loss_rad,Q_loss_conv,T_out,T_absorber,count,Lsection] =
sim_Receiver_water_HTF_coverless(numSections,L,mdot,Q_in,T_HTF_bulk,T_air,D_absorber,p_air,epsilon_absorber,windspeed,T_dewpoint_air,RelativeHumidity)
%Similar to sim_Receiver_air_HTF_air_vac but without a cover over the
%receiver
if(numSections==1)%catch only doing a single section

[Q_HTF(1),Q_loss_rad(1),Q_loss_conv(1),T_absorber(1),T_out(1),count(1)]=sim_Receiver_Section_waterLiq_HTF_coverless(L,mdot,Q_in,T_HTF_bulk,p_HTF,T_air,D_absorber,p_air,epsilon_absorber,windspeed,T_dewpoint_air,RelativeHumidity);
Lsection=L;
else

Q_HTF(numSections)=0;Q_loss_rad(numSections)=0;Q_loss_conv(numSections)=0;T_out(numSections)=0;
T_absorber(numSections)=0;count(numSections)=0;
Lsection=linspace(0,L,numSections+1);
Lsection=Lsection(2:end);%get rid of 0 distance point - only want spacings
Lsize=Lsection(2)-Lsection(1);
Qsection=Q_in./numSections;%assume uniform distribution of Q across receiver
%calc first point data from inlet conditions

[Q_HTF(1),Q_loss_rad(1),Q_loss_conv(1),T_absorber(1),T_out(1),count(1)]=sim_Receiver_Section_waterLiq_HTF_coverless(Lsize,mdot,Qsection,T_HTF_bulk,T_air,D_absorber,p_air,epsilon_absorber,windspeed,T_dewpoint_air,RelativeHumidity);
%calc point data with info from previous point
for i=2:numSections

[Q_HTF(i),Q_loss_rad(i),Q_loss_conv(i),T_absorber(i),T_out(i),count(i)]=sim_Receiver_Section_waterLiq_HTF_coverless(Lsize,mdot,Qsection,T_out(i-1),T_air,D_absorber,p_air,epsilon_absorber,windspeed,T_dewpoint_air,RelativeHumidity);
end
end
end

function [P_out,Tout_turb] =
sim_SEC_Turbocharger_air_covered(eta_compressor,eta_turbine,p_in,p_operating,p_out,numSections,L,mdot,Q_in,Tin,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity)
%Single pass simulation of a linear SEC with attached turbocharger as
%Brayton Cycle Heat Engine
[Tout_comp,DeltaH_Compressor_Polytropic] =
sim_Turbocharger_Section_Compressor_Air(Tin,eta_compressor,p_in,p_operating,mdot);
[~,~,~,T_out,~,~,~,~,~] =
sim_Receiver_air_HTF_air_vac(numSections,L,mdot,Q_in,Tout_comp,p_operating,T_air,D_absorber,D_cover_inner,D_cover_outer,p_air,p_vac,epsilon_absorber,k_cover,epsilon_cover,windspeed,T_dewpoint_air,RelativeHumidity);

```

```

[Tout_turb,DeltaH_Turbine_Polytropic] =
sim_Turbocharger_Section_Turbine_Air(T_out(end),eta_turbine,p_oper
ating,p_out,mdot);
P_out=-
1.*(DeltaH_Compressor_Polytropic+DeltaH_Turbine_Polytropic);
end

```

```

function [Tout_comp,DeltaH_Compressor_Polytropic] =
sim_Turbocharger_Section_Compressor_Air(Tin,eta_compressor,p_in,p_
out,mdot)
%Function to find outlet conditions of turbocharger compressor.
This is
%specifically for air.
%T in K, DeltaH_Compressor_Polytropic return val in Watts, working
val in
%J/kg
%DeltaH_Compressor_Polytropic used to work out power requirement
for
%compressor to run at these conditions.
%First, find isentropic outlet temp for given operating conditions
solution_tolerance=0.0001;
iterations_count = 1;
countlimit = 32;
Tlow = 0; %0K arbitrary starting point for Newton's Method
iterations
Thigh = 8400; %Max temp for 2axes concentrator - arbitrary
starting point
Tmiddle = (Thigh-Tlow).*0.5;
Tdiff = Thigh-Tlow;
DeltaSLow = calcS_air_Delta(Tin,Tlow,p_in,p_out);
DeltaSMid = calcS_air_Delta(Tin,Tmiddle,p_in,p_out);
while(Tdiff>solution_tolerance && sign(DeltaSMid)~=0 &&
iterations_count<=countlimit)
    if(sign(DeltaSLow)*sign(DeltaSMid)==-1)
        Thigh = Tmiddle;
    else
        Tlow=Tmiddle;
    end
    Tmiddle=(Thigh+Tlow).*0.5;
    Tdiff = Thigh-Tlow;
    DeltaSLow = calcS_air_Delta(Tin,Tlow,p_in,p_out);
    DeltaSMid = calcS_air_Delta(Tin,Tmiddle,p_in,p_out);
    iterations_count = iterations_count+1;
end
if(iterations_count>countlimit)
    disp('ERROR in turbo compressor outlet isentropic temp search.
Iterations exceed max allowed');
end
ToutIsentropic = Tmiddle;
DeltaHIsentropic = calcH_air_Delta(ToutIsentropic,Tin);
DeltaH_Compressor_Polytropic=DeltaHIsentropic./eta_compressor;
%Now want to find temp oulet for polytropic compression
iterations_count=1;
Tlow = 0; %0K arbitrary starting point for Newton's Method
iterations
Thigh = 8400; %Max temp for 2axes concentrator - arbitrary
starting point
Tmiddle = (Thigh-Tlow).*0.5;
Tdiff = Thigh-Tlow;

```

```

DeltaHFunctionLow=calcH_air_Delta(Tlow,Tin)-
DeltaH_Compressor_Polytropic;
DeltaHFunctionMiddle=calcH_air_Delta(Tmiddle,Tin)-
DeltaH_Compressor_Polytropic;
while(Tdiff>solution_tolerance&&sign(DeltaHFunctionMiddle)~=0 &&
iterations_count<=countlimit)
    if(sign(DeltaHFunctionLow)*sign(DeltaHFunctionMiddle)==-1)
        Thigh = Tmiddle;
    else
        Tlow=Tmiddle;
    end
    Tmiddle=(Thigh+Tlow).*0.5;
    Tdiff = Thigh-Tlow;
    DeltaHFunctionLow=calcH_air_Delta(Tlow,Tin)-
DeltaH_Compressor_Polytropic;
    DeltaHFunctionMiddle=calcH_air_Delta(Tmiddle,Tin)-
DeltaH_Compressor_Polytropic;
end
if(iterations_count>countlimit)
    disp('ERROR in turbo compressor outlet polytropic temp search.
Iterations exceed max allowed');
end
Tout_comp=Tmiddle;
DeltaH_Compressor_Polytropic=DeltaH_Compressor_Polytropic.*mdot;
end

function [Tout_turb,DeltaH_Turbine_Polytropic] =
sim_Turbocharger_Section_Turbine_Air(Tin,eta_turbine,p_operating,p
_out,mdot)
%Function to find outlet conditions of turbocharger turbine. This
is
%specifically for air.
%T in K, DeltaH_Turbine_Polytropic return val in Watts, working
val in
%J/kg
%DeltaH_Turbine_Polytropic used to work out power requirement for
%turbine to run at these conditions.
%First, find isentropic outlet temp for given operating conditions
solution_tolerance=0.0001;
iterations_count = 1;
countlimit = 32;
Tlow = 0; %0K arbitrary starting point for Newton's Method
iterations
Thigh = 8400; %Max temp for 2axes concentrator - arbitrary
starting point
Tmiddle = (Thigh-Tlow).*0.5;
Tdiff = Thigh-Tlow;
DeltaSLow = calcS_air_Delta(Tin,Tlow,p_operating,p_out);
DeltaSMid = calcS_air_Delta(Tin,Tmiddle,p_operating,p_out);
while(Tdiff>solution_tolerance && sign(DeltaSMid)~=0 &&
iterations_count<=countlimit)
    if(sign(DeltaSLow)*sign(DeltaSMid)==-1)
        Thigh = Tmiddle;
    else
        Tlow=Tmiddle;
    end
    Tmiddle=(Thigh+Tlow).*0.5;
    Tdiff = Thigh-Tlow;
    DeltaSLow = calcS_air_Delta(Tin,Tlow,p_operating,p_out);
    DeltaSMid = calcS_air_Delta(Tin,Tmiddle,p_operating,p_out);
    iterations_count = iterations_count+1;
end

```

```

end
if(iterations_count>countlimit)
    disp('ERROR in turbo turbine outlet isentropic temp search.
Iterations exceed max allowed');
end
ToutIsentropic = Tmiddle;
DeltaHIsentropic = calcH_air_Delta(ToutIsentropic,Tin);
DeltaH_Turbine_Polytropic=DeltaHIsentropic.*eta_turbine;
%Now want to find temp outlet for polytropic expansion
iterations_count=1;
Tlow = 0; %0K arbitrary starting point for Newton's Method
iterations
Thigh = 8400; %Max temp for 2axes concentrator - arbitrary
starting point
Tmiddle = (Thigh-Tlow).*0.5;
Tdiff = Thigh-Tlow;
DeltaHFunctionLow=calcH_air_Delta(Tlow,Tin)-
DeltaH_Turbine_Polytropic;
DeltaHFunctionMiddle=calcH_air_Delta(Tmiddle,Tin)-
DeltaH_Turbine_Polytropic;
while(Tdiff>solution_tolerance&&sign(DeltaHFunctionMiddle)~=0 &&
iterations_count<=countlimit)
    if(sign(DeltaHFunctionLow)*sign(DeltaHFunctionMiddle)==-1)
        Thigh = Tmiddle;
    else
        Tlow=Tmiddle;
    end
    Tmiddle=(Thigh+Tlow).*0.5;
    Tdiff = Thigh-Tlow;
    DeltaHFunctionLow=calcH_air_Delta(Tlow,Tin)-
DeltaH_Turbine_Polytropic;
    DeltaHFunctionMiddle=calcH_air_Delta(Tmiddle,Tin)-
DeltaH_Turbine_Polytropic;
end
if(iterations_count>countlimit)
    disp('ERROR in turbo turbine outlet polytropic temp search.
Iterations exceed max allowed');
end
Tout_turb=Tmiddle;
DeltaH_Turbine_Polytropic=DeltaH_Turbine_Polytropic.*mdot;
end

function [corectedSolarElevationDeg, solarAzimuthEofNDeg,
solarAzimuthDepartureFromNDeg ] = solarEleAzi( dayInt, absTime,
Latitude, Longitude, TimeZone )

JulianDay = dayInt+2415018.5+absTime-TimeZone/24;
JulianCentuary = (JulianDay-2451545)/36525;
GeomMeanLongSunDeg = mod((280.46646+JulianCentuary*(36000.76983 +
JulianCentuary*0.0003032)),360);
GeomMeanAnomSunDeg = 357.52911+JulianCentuary*(35999.05029 -
0.0001537*JulianCentuary);
EccentEarthOrbit = 0.016708634-
JulianCentuary*(0.000042037+0.0000001267*JulianCentuary);
SunEqofCtr = sin(degtorad(GeomMeanAnomSunDeg))*(1.914602-
JulianCentuary*(0.004817+0.000014*JulianCentuary))+sin(degtorad(2*
GeomMeanAnomSunDeg))*(0.019993-
0.000101*JulianCentuary)+sin(degtorad(3*GeomMeanAnomSunDeg))*0.000
289;
SunTrueLongDeg = GeomMeanLongSunDeg+SunEqofCtr;
%SunTrueAnomDeg = GeomMeanAnomSunDeg+SunEqofCtr;

```

```

%SunRadVectorAUs = (1.000001018*(1-
EccentEarthOrbit*EccentEarthOrbit))/(1+EccentEarthOrbit*cos(degtorad(SunTrueAnomDeg)));
SunAppLongDeg = SunTrueLongDeg-0.00569-
0.00478*sin(degtorad(125.04-1934.136*JulianCentuary));
MeanObliqEclipticDeg = 23+(26+((21.448-
JulianCentuary*(46.815+JulianCentuary*(0.00059-
JulianCentuary*0.001813))))/60)/60;
ObliqCorrDeg = MeanObliqEclipticDeg+0.00256*cos(degtorad(125.04-
1934.136*JulianCentuary));
%SunRtAscenDeg =
atan2d(cos(degtorad(ObliqCorrDeg))*sin(degtorad(SunAppLongDeg)),cos(degtorad(SunAppLongDeg)));
SunDeclinDeg =
rattodeg(asin(sin(degtorad(ObliqCorrDeg))*sin(degtorad(SunAppLongDeg))));
VarY =
tan(degtorad(ObliqCorrDeg/2))*tan(degtorad(ObliqCorrDeg/2));
EqofTimeMinutes =
4*rattodeg(VarY*sin(2*degtorad(GeomMeanLongSunDeg))-
2*EccentEarthOrbit*sin(degtorad(GeomMeanAnomSunDeg))+4*EccentEarthOrbit*VarY*sin(degtorad(GeomMeanAnomSunDeg))*cos(2*degtorad(GeomMeanLongSunDeg))-0.5*VarY*VarY*sin(4*degtorad(GeomMeanLongSunDeg))-1.25*EccentEarthOrbit*EccentEarthOrbit*sin(2*degtorad(GeomMeanAnomSunDeg)));
HASunriseDeg =
rattodeg(acos(cos(degtorad(90.833))/(cos(degtorad(Latitude))*cos(degtorad(SunDeclinDeg)))-
tan(degtorad(Latitude))*tan(degtorad(SunDeclinDeg))));
%SolarNoonLST = (720-4*Longitude-
EqofTimeMinutes+TimeZone*60)/1440;
%SunriseTimeLST = SolarNoonLST-HASunriseDeg*4/1440;
%SunsetTimeLST = SolarNoonLST+HASunriseDeg*4/1440;
SunlightDurationMinutes = 8*HASunriseDeg;
TrueSolarTimeMin = mod(absTime*1440+EqofTimeMinutes+4*Longitude-60*TimeZone,1440);
%HourAngle=0;
    if(TrueSolarTimeMin/4<0)
        HourAngle = TrueSolarTimeMin/4+180;
    else HourAngle = TrueSolarTimeMin/4-180;
    end
SolarZenithAngleDeg =
rattodeg(acos(sin(degtorad(Latitude))*sin(degtorad(SunDeclinDeg))+cos(degtorad(Latitude))*cos(degtorad(SunDeclinDeg))*cos(degtorad(HourAngle))));
SolarElevationAngleDeg = 90-SolarZenithAngleDeg;
%ApproxAtmosphericRefractionDeg = 0;
    if(SolarElevationAngleDeg>85) ApproxAtmosphericRefractionDeg = 0;
    else if(SolarElevationAngleDeg>5)
ApproxAtmosphericRefractionDeg =
58.1/tan(degtorad(SolarElevationAngleDeg))-
0.07/((tan(degtorad(SolarElevationAngleDeg)))^3)+0.000086/((tan(degtorad(SolarElevationAngleDeg)))^5);
        else if(SolarElevationAngleDeg>-0.575)
ApproxAtmosphericRefractionDeg = 1735+SolarElevationAngleDeg*(-518.2+SolarElevationAngleDeg*(103.4+SolarElevationAngleDeg*(-12.79+SolarElevationAngleDeg*0.711)));
        else ApproxAtmosphericRefractionDeg = -
20.772/tan(degtorad(SolarElevationAngleDeg));
        end
    end
end

```

```

        end
        ApproxAtmosphericRefractionDeg =
        ApproxAtmosphericRefractionDeg/3600;
        corectedSolarElevationDeg =
        SolarElevationAngleDeg+ApproxAtmosphericRefractionDeg;
        %solarAzimuthCWfromNorth = 0;
        if(HourAngle>0)
            solarAzimuthEofNDeg =
            mod(radtodeg(acos(((sin(degtorad(Latitude))*cos(degtorad(SolarZenithAngleDeg)))-
            sin(degtorad(SunDeclinDeg)))/(cos(degtorad(Latitude))*sin(degtorad
            (SolarZenithAngleDeg)))))+180,360);
            else solarAzimuthEofNDeg = mod(540-
            radtodeg(acos(((sin(degtorad(Latitude))*cos(degtorad(SolarZenithAngleDeg)))-
            sin(degtorad(SunDeclinDeg)))/(cos(degtorad(Latitude))*sin(degtorad
            (SolarZenithAngleDeg))))),360);
        end
        solarAzimuthEofNDeg=mod(solarAzimuthEofNDeg,360);
        if(solarAzimuthEofNDeg>180)
            solarAzimuthDepartureFromNDeg = -1*(360-
            solarAzimuthEofNDeg);
        else solarAzimuthDepartureFromNDeg = solarAzimuthEofNDeg;
        end
    end
end

```

```

function [cp_water,k_water,mu_water] = Water_properties(T)
%p in Pa, T in K, rho in kg/m3, cp in J/kg.K, k in W/m.K,
%mu in kg/m.s, nu in m2/s
%input T get desired values for heat trasfer to water as HTF
%if(T>373 || T<273)
%    disp(['Error, water temperature out of bounds Tin C= ',
num2str(T)])
%end
t=T/1000;A=-203.606;B=1523.29;C=-3196.413;D=2474.455;E=3.855326;
cp_water=A+B*t+C*(t^2)+D*(t^3)+E./(t^2);%this is in J/mol.K
cp_water=cp_water.*1000./18.0153;%Water=18.0153 g/mol -- J/mol.K
to J/kg.K
k_water=-0.0000111820460241513.*(T^2)+0.0083876978810663.*T-
0.900374816586921;
A=-3.7188;B=578.919;C=-137.546;
mu_water=exp(A+((B)./(C+T)));%this is mPa.s
mu_water=mu_water./1000;
end

```

G Python Operational Code

This section lists the code python code that was used to operate the linear receiver. The Raspberry Pi was connected to a WiFi network, and a host laptop was SSHed into the Pi to call the scripts. A keyboard interrupt signal was used as the method to break out of the loops and perform safe shutdown of the apparatus.

G.1 Main heliostat runtime

```
#solarController.py - main heliostat runtime. Attached potentiometer
changes mode from startup to solar tracking
import spidev
from time import sleep
import os
import RPi.GPIO as GPIO
#open SPI bus
spi = spidev.SpiDev()
spi.open(0,0)
#ADC pin layout
leftPanelChannel = 0
rightPanelChannel = 1
rheoChannel=2
#Pi's GPIO pin layout
clockwiseMosfet = 12
counterClockwiseMosfet = 16
motorToGeneratorRelay = 18
escPWM = 22
#globals
sleepTime = 0.2#frequency of main program
leftPanelReference=0.618#these values normalise the voltages from each
panel
rightPanelReference=0.637
deltaPanelRatioLimit=1.05 #this is the ratio of difference between the
panels to start the winch
minTotalPanelVoltage=0.4 #ensures running at daytime
winchMosfetTime=0.2#how long each step of the winch to be active
winchWaitTime=0.1#how long to wait for the mosfet to switch off and relay
to disconnect
#init GPIO pins
GPIO.setmode(GPIO.BOARD)
GPIO.setup(clockwiseMosfet,GPIO.OUT)
GPIO.setup(counterClockwiseMosfet,GPIO.OUT)
GPIO.setup(motorToGeneratorRelay,GPIO.OUT)
GPIO.setup(escPWM,GPIO.OUT)
pwm=GPIO.PWM(escPWM,50) #50Hz carrier for the signal, emulating PPM
(legacy mode for ESC)
pwm.start(1) #1 percent duty cycle
#Recieve binary over SPI from ADC and return processed float of 3.3V
def getReading(channel):
    rawData = spi.xfer([1, (8+channel) << 4, 0])
    processedData = ((rawData[1]&3) << 8) + rawData[2]
    chanelVoltage = processedData*3.3/float(1023)
    return chanelVoltage
#START OF MAIN
try:
    while True:
        leftPanelVoltage =
getReading(leftPanelChannel)/leftPanelReference
        rightPanelVoltage =
getReading(rightPanelChannel)/rightPanelReference
        rheoChannelVoltage = getReading(rheoChannel)
        #The rheostat sets the mode of the Pi to either sun track or
startup mode
        if (rheoChannelVoltage>1.6): #tracking mode
```

```

        print('Tracking mode') #debugging

    if((leftPanelVoltage+rightPanelVoltage)>minTotalPanelVoltage):
        print('Daytime confirmed')
        if(leftPanelVoltage>=rightPanelVoltage):
            panelRatio =
leftPanelVoltage/rightPanelVoltage
            if (panelRatio>deltaPanelRatioLimit):
                print('Rotating clockwise')
                GPIO.output(clockwiseMosfet,
True)
                sleep(winchMosfetTime)
                GPIO.output(clockwiseMosfet,
False)
                sleep(winchWaitTime)
            else:
                panelRatio =
rightPanelVoltage/leftPanelVoltage
                if (panelRatio>deltaPanelRatioLimit):
                    print('Rotating counter-
clockwise')
                    GPIO.output(counterClockwiseMosfet, True)
                    sleep(winchMosfetTime)
                    GPIO.output(counterClockwiseMosfet, False)
                    sleep(winchWaitTime)
            else:
                print('Night time detected')
                sleep(10)
                #pwm.stop() #doesnt make measurable CPU load
difference
            else: #startup mode
                print('Startup mode initiating in 2
seconds') #debugging
                GPIO.output(motorToGeneratorRelay, True)
                pwm.ChangeDutyCycle(2) #Ready acceleration ramp
                sleep(2)
                print('Full power for 15 seconds')
                pwm.ChangeDutyCycle(12) #PPM 100% throttle
                sleep(15)
                pwm.ChangeDutyCycle(1) #Back to min throttle
                #print(rheoChannelVoltage)
                print('Startup complete')
                GPIO.output(motorToGeneratorRelay, False)
                sleep(sleepTime)
#END OF MAIN
finally:
    GPIO.output(clockwiseMosfet, False)
    GPIO.output(counterClockwiseMosfet, False)
    GPIO.output(motorToGeneratorRelay, False)
    pwm.stop()
    GPIO.output(escPWM, False)
    GPIO.cleanup()

```

G.2 Component test functions

```

#rheoADCTest.py - tests the value read on a rheostat connected to ADC
channel 2
import spidev
from time import sleep
import os
import RPi.GPIO as GPIO
#open SPI bus
spi = spidev.SpiDev()
spi.open(0,0)
#initiate sensors

```

```

rheoChannel = 2
def getReading(channel):
    rawData = spi.xfer([1, (8+channel) << 4, 0])
    processedData = ((rawData[1]&3) << 8) + rawData[2]
    return processedData
try:
    while True:
        rheoVoltage = getReading(rheoChannel)*3.3/float(1023)
        # rightPanelVoltage = getReading(rightPanelChannel)*3.3/float(1023)
        # print(leftPanelVoltage)
        # print(rightPanelVoltage)
        print(rheoVoltage)
        #print(getReading(rheoChannel))
        sleep(0.2)
finally:
    GPIO.cleanup()

#rheoToSwitch.py - tests the command send from a rheostat to actuate
either the clockwise or counterclockwise relay and prints result
import spidev
from time import sleep
import os
import RPi.GPIO as GPIO
GPIO.setmode(GPIO.BOARD)
mosfetPin = 22
GPIO.setup(mosfetPin,GPIO.OUT)
#open SPI bus
spi = spidev.SpiDev()
spi.open(0,0)
#initiate sensors
leftPanelChannel = 0
rightPanelChannel = 1
sleepTime = 1
relyMode = 0
def getReading(channel):
    rawData = spi.xfer([1, (8+channel) << 4, 0])
    processedData = ((rawData[1]&3) << 8) + rawData[2]
    return processedData
try:
    while True:
        leftPanelVoltage =
getReading(leftPanelChannel)*3.3/float(1023)
        # rightPanelVoltage = getReading(rightPanelChannel)*3.3/float(1023)
        # print(leftPanelVoltage)
        # print(rightPanelVoltage)
        if(relyMode==0):
            if(leftPanelVoltage>=3.3/2):
                GPIO.output(mosfetPin, True)
                print("Turn on")
                relyMode = 1
            elif(relyMode==1):
                if(leftPanelVoltage<=3.3/2):
                    GPIO.output(mosfetPin, False)
                    print("Turn off")
                    relyMode = 0
        sleep(1)
finally:
    GPIO.output(mosfetPin, False)
    GPIO.cleanup()

#testWinch.py - Tests each direction of the winch
from time import sleep
import RPi.GPIO as GPIO
GPIO.setmode(GPIO.BOARD)
clockwisePin = 12
counterclockwisePin = 16
GPIO.setup(clockwisePin,GPIO.OUT)

```

```

GPIO.setup(counterclockwisePin,GPIO.OUT)
try:
    GPIO.output(clockwisePin, True)
    sleep(1)
    GPIO.output(clockwisePin, False)
    sleep(0.2)
    GPIO.output(counterclockwisePin, True)
    sleep(1)
    GPIO.output(counterclockwisePin, False)
    sleep(0.2)
finally:
    GPIO.cleanup()

#manualWinch.py - allows for manual control over winch direction using
the potentiometer
import spidev
from time import sleep
import os
import RPi.GPIO as GPIO
#open SPI bus
spi = spidev.SpiDev()
spi.open(0,0)
#ADC pin layout
rheoChannel=2
#Pi's GPIO pin layout
clockwiseMosfet = 12
counterClockwiseMosfet = 16
winchWaitTime=0.1#how long to wait for the mosfet to switch off and relay
to disconnect
sleepTime = 0.1#frequency of main program
#init
GPIO.setmode(GPIO.BOARD)
GPIO.setup(clockwiseMosfet,GPIO.OUT)
GPIO.setup(counterClockwiseMosfet,GPIO.OUT)
GPIO.output(clockwiseMosfet, False)
GPIO.output(counterClockwiseMosfet, False)
#Recieve binary over SPI from ADC and return processed float of 3.3V
def getReading(channel):
    rawData = spi.xfer([1, (8+channel) << 4, 0])
    processedData = ((rawData[1]&3) << 8) + rawData[2]
    chanelVoltage = processedData*3.3/float(1023)
    return chanelVoltage
try:
    while True:
        while (getReading(rheoChannel)<1.1):
            print('Rotating clockwise')
            GPIO.output(clockwiseMosfet, True)
            sleep(sleepTime)
            GPIO.output(clockwiseMosfet, False)
            GPIO.output(counterClockwiseMosfet, False)
        while (getReading(rheoChannel)>2.2):
            print('Rotating counter-clockwise')
            GPIO.output(counterClockwiseMosfet, True)
            sleep(sleepTime)
            GPIO.output(clockwiseMosfet, False)
            GPIO.output(counterClockwiseMosfet, False)
        print('Holding Position')
        sleep(sleepTime*2)
finally:
    GPIO.output(clockwiseMosfet, False)
    GPIO.output(counterClockwiseMosfet, False)
    GPIO.cleanup()

#servorotate.py - Tests pwm signal to control ESC by connecting to a
servo
import RPi.GPIO as GPIO

```

```

from time import sleep
GPIO.setmode(GPIO.BOARD)
servoPin=26
GPIO.setup(servoPin,GPIO.OUT)
pwm=GPIO.PWM(servoPin,50)
pwm.start(7)
sleeptime = 0.01
try:
    while(True):
        for i in range(0,180):
            DC=1./18.*(i)+2
            pwm.ChangeDutyCycle(DC)
            sleep(sleeptime)
        for i in range(180,0,-2):
            DC=1./18.*(i)+2
            pwm.ChangeDutyCycle(DC)
            sleep(sleeptime)
finally:
    pwm.stop()
    GPIO.cleanup()

```

H. Summary of Research Questions Answered

The following subsection summarizes explicit answers to the research questions that were outlined in the research dissertation proposal.

How do the dynamics of heat transfer and losses change across the surface of the receiver?

At an air HTF inlet temperature of 80 °C fed to a covered arbitrary receiver of reasonable dimensions and contemporary absorber materials, receiver efficiency is high and optical losses account for the vast majority of total losses at shorter lengths (Figures 3.2 2 and 3).

At low temperatures (0 to 4 m at 80-305 °C), the cumulative receiver efficiency is high (61-67%) and optical losses account for the vast majority of total losses (81% to 58% fraction of total losses).

At moderate temperatures (7.5 m at about 435 °C), optical losses are approximately equal to convective losses to the surrounding air (44% fraction each) with the remainder being radiative losses (12% fraction). At this point the receiver's instantaneous efficiency is about 40% with a cumulative efficiency of 54%.

At high temperatures (13.5 m at about 600 °C) convective losses are driving (52% fraction) with optical losses being secondary (33% fraction) and radiative losses the least (remaining 15% fraction). At this point the receiver's instantaneous efficiency is about 19% with a cumulative efficiency of 43%.

At what temperature does radiative heat losses become driving over convective losses?

For the arbitrary linear focus receiver studied, the cover does not reach a temperature high enough to have radiative losses to the atmosphere become driving.

However, radiative heat transfer between the absorber surface and the cover's inner surface is the primary mechanism of heat loss transfer away from the absorber's surface. The absorber's emittance is therefore the primary driver of heat losses from the receiver after optical losses are taken into account.

How important are shielding and surface materials to this?

A shield is absolutely essential to mitigate losses from the receiver. At a windspeed of 5 m/s, a cover reduces losses by a factor of 50 at moderate temperatures for the arbitrary receiver modelled.

May any conclusions be drawn on best practice for linear collector design?

A cover is absolutely essential for the receiver, irrespective of its operating temperature. The best practise to reduce heat losses from the receiver beyond optical losses is to have as low an absorber surface emittance as possible.

It is not always the case to target as high a concentration ratio as possible. As the temperature gradient between the absorber surface and HTF increases, radiative heat transfer or losses between the absorber surface and the cover grows exponentially. There exists an ideal combination of CR, mass flow rate and AR to maximise the efficiency of the linear focus SEC.

Linear collectors are usually operated with liquid phase HTFs. To what extent does using a gas phase HTF such as air affect heat transfer specifically in a linear receiver?

The effective heat conductivity of gas phase HTFs tends to be lower than that of liquid phase HTFs. If the HTF flow within the absorber is turbulent enough then effective heat transfer may still be effectively achieved. To obtain this turbulent flow, the receiver diameter may be kept small and the flow rate kept high. The diameter of the receiver needs to balance the effective CR, surface area for losses to the atmosphere, pressure drop along the receiver and temperature gradient between the absorber surface and the HTF to maximise heat transfer and engine thermal efficiency. This depends on the viscosity, heat capacity, heat conductivity and density of the HTF.

Is there any validity to the heuristic of relegating linear receivers to low temperature operation?

New materials of construction for the absorber surface – especially cermets – should allow for linear focus receivers to operate effectively at temperatures in the order of 700 °C. While there are many factors involved in receiver and collector design, it was shown that an appropriately designed linear focus receiver made from contemporary materials should operate with a heat transfer efficiency only about 12-21% lower than that of a point focus receiver. If the LCOE for conventional linear focus setups are 20-30% less than that of point focus installations, then it is worth investigating the intentional operation of linear focus receivers at relatively high temperatures.

Is there a significant enough thermodynamic or other benefits to justify operating linear collectors at elevated temperatures?

Operating at higher temperatures does imply a thermodynamic benefit. A higher operating temperature also allows for different more efficient heat engine cycles to be used. Low temperature heat engine operation is more or less limited to Rankine or low temperature Sterling cycles. Higher temperature operation permits the use of Brayton, Ericsson and Sterling cycles to be used.

From Table 4.3.1 it was shown that it should be possible to operate a modified turbocharger as a Brayton Cycle Heat Engine at a net thermal efficiency of 9.0%. This is within the same region of about 10% net thermal efficiency for conventional PTC, LFR, and HFC implementations (Table 2.3.1 1).

In reality therefore, there is little effective electricity output benefit in opting to use such a linear focus receiver based Brayton Cycle Engine over a conventional Rankine based cycle. There is however a major operational difference in the fact that Brayton Cycle engines (such as Gas Turbines) do not typically require cooling water for operation. This may be an especially important factor for operation in arid climates.

PV electricity production competes directly with such CSP based Brayton Cycle Engines. Commercial PV panels operate with efficiencies in the region of about 10-15% and like BCHEs do not require cooling water. The choice between the use of either technology depends on capital and operating costs and the cost effectiveness of energy storage in the form of heat or batteries. The fact that PVs require substantially less maintenance and control during operation than CSP BCHEs and the decentralized nature of their power production make PVs particularly attractive.

CSP-BCHEs therefore are likely to be preferred over PVs only in particularly niche circumstances, particularly if heat storage may be done in an economical manner.

To what extent do factors such as wind velocity change the effectiveness of linear focus receivers?

With an appropriate cover for the receiver, the surrounding wind velocity makes little difference to the effectiveness of the collector. Heat losses to atmosphere are primarily driven by the emissivity of the absorber surface. The cover outer surface is only capable of transferring the heat that it actually receives from the absorber to the atmosphere. Practically all available heat supplied to the cover may be effectively transferred to the atmosphere from as little as 2 m/s windspeed.

Is the operation of a modified vehicle turbocharger in a hybrid CSP-BCHE configuration feasible? Is the use of a linear collector feasible over the use of a normal point focus collector?

It has been shown that the operation of a turbocharger as a hybrid CSP-BCHE is feasible, but not viable.

In fact, it was shown that turbocharger turbine outlet temperatures readily exceed 700 °C. Given the limit of a combustion chamber inlet temperature of 650 °C, hybrid operation with a solar preheating stage is essentially unviable. The SEC preheating stage may be replaced by a single heat exchanger between the turbine outlet and the combustion chamber inlet, greatly simplifying engine operation.

Though technically feasible, such hybrid CSP-BCHE operation is not viable, irrespective of the SEC technology used.

Does a heat recycling stage provide substantial benefit to the turbocharger based BCHE?

Yes, a heat recycling stage provides a considerable benefit to the operation of the turbocharger based engine. A heat recuperator increases the net thermal efficiency of the engine in the region of about 63% (Table 4.3 1).

What is the environmental impact for the fabrication, installation, operation, maintenance and disposal of a modified vehicle turbocharger linear collector hybrid CSP-BCHE?

The apparatus was constructed almost exclusively with steel. Virtually every part is recyclable, including the acrylic mirrors. To the knowledge of the author, no toxic or especially harmful materials are present anywhere in the apparatus at any stage of its fabrication or operation.

The operation of the engine involves the combustion of propane gas as well as lubricating oil. Emissions from the apparatus would need to be measured and assessed for their impact on the environment. Operation of the engine produces a significant amount of noise which may be an especially important factor for smaller installations.

What opportunities exist for heat recovery of such a system?

The outlet temperature of the turbine was in excess of 500 °C. This heat may be readily used in a heat recovery system. At these temperatures, an ORC or even steam turbine may be used to generate additional electricity.

If the turbocharger based engine were to be used in a domestic or small commercial setting, it may be more advantageous instead to use the turbine exhaust to heat water. Practically all of the available thermal energy could be transferred to heating the water. This would reduce the need to use the generated electricity to run geysers to achieve the same effect, but would be subject to other electrical transmission losses.

How difficult is it to fabricate and control such a system?

The apparatus was designed and built with relatively simple tools and equipment. By far the most complicated piece of equipment used was the laser cutter. Laser cutters themselves are fairly ubiquitous in fabrication facilities.

While production of the apparatus was time consuming – primarily 100 hours spent welding per trough – it was not particularly difficult with respect to building tolerances and availability of raw materials.

The automated heliostat function of the trough produced excellent results, with the solar tracking capable of about 1 degree of accuracy. Engine start-up and operation was partially automated which left the operator free to concentrate on fuel flow control to the engine.

When optimised for materials and working components, what sort of efficiencies, sizes and duties should be expected for a modified vehicle turbocharger linear collector hybrid CSP-BCHE?

Operating turbochargers as hybrid CSP-BCHEs permits the preheating of air to 650 °C prior to the combustion chamber. Three turbochargers were simulated in this fashion with each connected to an 8 m wide PTC of appropriate length to achieve this combustion chamber inlet temperature. The turbochargers were run at an air flow rate and pressure ratio corresponding to 90% of their choke flow rate to maximise output work production and efficiency.

The three different size turbochargers were chosen to provide an insight across the range of commercially available models. A small motorcycle turbocharger, a medium sized and high efficiency turbocharger, and an excessively large turbocharger were simulated.

The small turbocharger required a collector field about 3 m in length, for an aperture of 24 m². This receiver was found to have a thermal transfer efficiency of solar to HTF of 62%. The efficient medium sized turbocharger required a collector field 49 m in length for an aperture of 395 m² at a receiver efficiency of 67%. The larger turbocharger required a field of 128 m, an aperture of 1026 m² for an efficiency of 64%.

A linear focus receiver made from contemporary materials should therefore be expected to operate in the region of 65% cumulative efficiency for an outlet temperature of about 650 C.

By burning enough fuel to achieve a turbine inlet temperature of 1000 °C, the small turbocharger was preheated with 24 kW of solar power and was capable of producing 3.6 kW_e by burning 18 kW of fuel for a strict fuel efficiency of about 20%. The efficient medium sized turbocharger was preheated with 395 kW of solar power and produced 65 kW_e by burning 210 kW of fuel, for a fuel efficiency of 31%. The large turbocharger was preheated with 1026 kW of solar power and produced 172 kW_e by burning 558 kW of fuel, for a fuel efficiency of 31%.

For the small turbocharger, the solar preheating duty was 1.32 times that of fuel combustion duty. For the efficient medium sized turbocharger, the solar preheating duty was 1.88 times that of fuel combustion duty. The large turbocharger was found to have a solar duty of 1.84 times that of fuel combustion duty.

Does it work at a domestic or larger scale? Is it applicable to rural application – especially somewhere with excess insolation and combustible gas reserves such as the Karoo?

Using solar power as a preheating mechanism for modified turbocharger gas turbines is certainly feasible, but it is unlikely to be viable. It has been shown that the whole preheating stage may be replaced by a straightforward heat recovery system especially as the inlet temperature to the turbine increases.

For modified turbocharger solar hybrid BCHEs, a fuel efficiency of about 30% is achieved with electrical power outputs of about 65 kW_e and above. As the size of the turbocharger decreases, its efficiency at producing electricity also decreases. For the smallest turbochargers, about 3.6 kW_e may be produced at a fuel efficiency of 20%.

Commercial generators are capable of operating at thermal efficiencies in the region of 20-35% and do not require a solar preheating stage (Baglione, 2007). It is unlikely that modified turbochargers would outperform already available generator technologies.

It is perhaps more prudent to burn the fuel directly without a solar preheating stage, or to forgo the combustion in the first place and operate in a strictly solar powered manner.

How does a rough estimate of total system cost compare with a similarly sized commercially bought PV system? Are the costs for each system within the same order of magnitude?

Appendix C contains a list of all purchased materials for the apparatus as well as a breakdown of assumed costs to produce a single trough section. To estimate the minimum cost per each trough section, only the essential components will be summed together. While the steel used for the collectors was sourced exclusively from recycled materials, it is necessary to approximate a production cost.

Each trough section was built from approximately 15.5 m of 50 mm x 2 mm square pipe. Each mirror brace weighed about 80 kg. Prior to painting the primer layer, one of the troughs was weighed and measured to be 184 kg.

Each trough cost approximately R9'280 to fabricate for an aperture of 5.6 m² for a total price of R1'657 per kW of solar energy. If electric power could be extracted at an efficiency 9% as per Table 4.3 1, then electricity could be produced at a cost of R18'409 per kW_e.

The value above excludes maintenance, capital costs of the turbocharger, transport, fabrication labour, and running costs.

The majority of materials used for the apparatus were purchased as part of larger day-to-day orders at the metalworks facility. The orders themselves were already fairly substantial commercial orders. It is therefore unlikely that associated costs of

larger scale production of the troughs would benefit significantly from an increased scale.

ARTsolar is a wholesale solar panel manufacturer in Durban. A 300W 60 cell monocrystalline pergium solar module sells for R1638.75 incl. (ARTsolar, 2019). Producing solar power from these modules therefore costs R5'462.5 per kW_e.

Producing electricity with modified vehicle turbochargers and troughs such as those used in the apparatus is therefore 3 to 4 times more expensive than using PV panels without taking into account the capital cost of the turbocharger or running costs such as oil consumption and maintenance.

While an argument could be made that the exhaust from the turbine may be easily used to heat water at a duty 3.33 times that of the electrical power output of the engine (this assumes a 30% net thermal energy to HTF to work efficiency of a solar hybrid turbocharger BCHE for 70% thermal energy available to heat water), the solar panels may also be simply adapted to circulate and heat water on the underside of the cells.

I. Table of Targeted Objectives and Outcomes

The following table summarises the success of targeted objectives and outcomes as outlined in Section 1 is available below.

	Achieved	Not Achieved
<i>Develop a robust model describing the heat transfer dynamics present for a linear focus receiver</i>	x	
<i>Use a modular approach such that new materials, substances and phenomena may be easily added in the future</i>	x	
<i>Implement the model in MATLAB</i>	x	
<i>Use MATLAB to perform parametric studies and hypothetical optimizations for theoretical and realistic heat engines</i>	x	
<i>Quantify the necessity of receiver covers especially at high temperatures</i>	x	
<i>Quantify the relevance of heat recycling units used in conjunction with modified turbocharger BCHEs</i>	x	
<i>Compare the design and performance of real-world linear focus CSP plants to that predicted by the model</i>		x
<i>Provide justification to challenge the heuristic that linear focus receivers are suitable only for low temperature heat engines and heat engine cycles</i>	x	
<i>Determine the feasibility and viability of using modified vehicle turbochargers in conjunction with CSP for electricity generation, especially at a domestic scale</i>	x	
<i>Provide a rough comparison between PVs and solar hybrid modified vehicle turbocharger BCHEs for electricity generation</i>	x	
<i>Build and test an apparatus running a modified vehicle turbocharger as a solar hybrid BCHE</i>	x	
<i>Compare measured performance of the apparatus to predicted performance</i>	x	
<i>Record the intricacies of building, operating, controlling and metering the apparatus such that lessons learnt may be applied in future studies</i>	x	