A solar concentrating photovoltaic / thermal collector

Joseph Sydney Coventry
June 2004

A thesis submitted for the degree of Doctor of Philosophy at the Australian National University
Declaration

This PhD thesis contains no material that has been accepted for the award of any other degree or diploma in any university. To the best of the author’s knowledge and belief, no material previously published or written by another person has been included in this thesis, except where due reference is made in the text.

Joe Coventry
June 2004
Acknowledgements

This thesis would not have been possible without the generous academic support of my two supervisors, Andrew Blakers and Keith Lovegrove. They have given me solid guidance throughout the PhD, and their optimism and commitment towards the cause of renewable energy is certainly inspirational. Many thanks also to Chris Bales in Sweden for his help as an advisor, and for bringing me up to speed with TRNSYS during his year at the ANU.

I would like to acknowledge the generosity and patience shown by the administration at the Centre for Sustainable Energy Systems and the Faculty of Engineering and Information Technology during the last four years. I have had the opportunity to conduct an international study tour, attend local and international conferences, and to take some time out while working on the Bruce Hall solar project. Both CSES and FEIT have been generous with financial support and encouraged extra-curricular developmental activities. Thanks also to the now defunct CRC for Renewable Energy for their funding support and good fun post-graduate conferences.

To my fellow CHAPS team, it has been a pleasure to work with you, up on the roof, in the labs and sitting around the table discussing ideas. A few people deserve special mention: James Cotsell for his enthusiasm and willingness to get things done, and especially for accelerating the construction of the long trough; Bruce Condon, for advice on all things measurement and electrical; John Smelting, for regularly getting his hands dirty to ‘get the job done’; Greg Burgess, for his help on cold winter nights with the photogrammetry; Will Keogh, for advice with LabView and the flash tester; and the team of people who have helped out in the workshop, particularly Luke Clayton, Jeff Brown, Tony Ashmore, Ben Nash and Jaap den Hartog.

My office mates have been absolutely fantastic. Holger Kreetz was a real motivator, not just on the soccer field, but in discussions on thermodynamics and the thesis in general during the early days. Thanks to Mike Dennis for teaching me about wires and sharing the frustrations of TRNSYS, and Evan Franklin for a shot and a half of coffee, Monday footy analysis and help with the flux measurements (in no particular order). Thank you also Dave, Liz, Paul and Tom for helping make the office such a great work environment, and top notch place to hang out!

Thank you friends and family for your support. Most of all, thank you Leonie for moving up to Canberra, marrying me and making life outside the thesis good.
Abstract

This thesis discusses aspects of a novel solar concentrating photovoltaic / thermal (PV/T) collector that has been designed to produce both electricity and hot water. The motivation for the development of the Combined Heat and Power Solar (CHAPS) collector is twofold: in the short term, to produce photovoltaic power and solar hot water at a cost which is competitive with other renewable energy technologies, and in the longer term, at a cost which is lower than possible with current technologies. To the author’s knowledge, the CHAPS collector is the first PV/T system using a reflective linear concentrator with a concentration ratio in the range 20-40x. The work contained in this thesis is a thorough study of all facets of the CHAPS collector, through a combination of theoretical and experimental investigation.

A theoretical discussion of the concept of ‘energy value’ is presented, with the aim of developing methodologies that could be used in optimisation studies to compare the value of electrical and thermal energy. Three approaches are discussed; thermodynamic methods, using second law concepts of energy usefulness; economic valuation of the hot water and electricity through levelised energy costs; and environmental valuation, based on the greenhouse gas emissions associated with the generation of hot water and electricity. It is proposed that the value of electrical energy and thermal energy is best compared using a simple ratio.

Experimental measurement of the thermal and electrical efficiency of a CHAPS receiver was carried out for a range of operating temperatures and fluid flow rates. The effectiveness of internal fins incorporated to augment heat transfer was examined. The glass surface temperature was measured using an infrared camera, to assist in the calculation of thermal losses, and to help determine the extent of radiation absorbed in the cover materials. FEA analysis, using the software package Strand7, examines the conductive heat transfer within the receiver body to obtain a temperature profile under operating conditions.

Electrical efficiency is not only affected by temperature, but by non-uniformities in the radiation flux profile. Highly non-uniform illumination across the cells was found to reduce the efficiency by about 10% relative. The radiation flux profile longitudinal to the receivers was measured by a custom-built flux scanning device. The results show significant fluctuations in the flux profile and, at worst, the minimum flux intensity is as much as 27% lower than the median. A single cell with low flux intensity limits the current and performance of all cells in series, causing a significant drop in overall output. Therefore, a detailed understanding of the causes of flux non-uniformities is essential for the design of a single-axis tracking PV trough concentrator. Simulation of the flux profile was carried out
using the ray tracing software Opticad, and good agreement was achieved between the simulated and measured results. The ray tracing allows the effect of the receiver supports, the gap between mirrors and the mirror shape imperfections to be examined individually.

A detailed analytical model simulating the CHAPS collector was developed in the TRNSYS simulation environment. The accuracy of the new component was tested against measured data, with acceptable results. A system model was created to demonstrate how sub-components of the collector, such as the insulation thickness and the conductivity of the tape bonding the cells to the receiver, can be examined as part of a long term simulation.
Foreword

The author would like to acknowledge colleagues at CSES for their contributions to the design and production of the CHAPS system. The receiver design was modified from the air cooled system used for the Rockingham PV trough project, which is a two-axis tracking PV concentrator system designed by the ANU (Smeltink et al., 2000). The author was responsible for many of the key changes to this design, in particular, the shift to a full aluminium extrusion (with the use of anti-corrosive additives in the cooling fluid) and the inclusion of internal fins to improve the heat transfer. The design team, led by James Cotsell, assisted with realising the design and fabricating the receivers. The author was also responsible for the system design change from two-axis tracking CHAPS systems to single-axis tracking long troughs. The detailed mechanical drawings for the single axis tracking system were coordinated by John Smeltink, and the manufacturing was outsourced. The mirrors were designed by Glen Johnston and Greg Burgess, and manufactured in the solar thermal workshop at the ANU. The monocrystalline silicon solar cells were manufactured in the photovoltaic laboratory at the ANU, by a dedicated and persistent team lead by Chris Holly. The solar tracking controller was developed and built by Mike Dennis. The author carried out the experimental work to examine the impact of non-uniform light across solar cells, but would like to acknowledge the work of Evan Franklin in developing a theoretical model to further explain the results. The author would like to acknowledge the contribution by Keith Lovegrove to chapter 3, which is largely taken from a co-authored journal paper (Coventry and Lovegrove, 2003). The author carried out all data gathering and analysis in this chapter, but Keith was very helpful in discussing the intricacies of the concept of energy value.

The following publications were produced during the course of the research project:

**Journal papers**


Conference papers


# Nomenclature and Abbreviations

## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>( A )</td>
<td>Area</td>
</tr>
<tr>
<td>( A_m )</td>
<td>Mirror aperture area</td>
</tr>
<tr>
<td>( A_s )</td>
<td>Nominal cross-sectional area for the fluid conduit (excluding fins)</td>
</tr>
<tr>
<td>( A_{ss} )</td>
<td>Cross-sectional area of the fluid conduit</td>
</tr>
<tr>
<td>( \dot{A} )</td>
<td>Exergy (or Availability)</td>
</tr>
<tr>
<td>( c_p )</td>
<td>Specific heat</td>
</tr>
<tr>
<td>( C_{p,col} )</td>
<td>Thermal capacitance of the solar collector</td>
</tr>
<tr>
<td>( C_0 )</td>
<td>Capital cost</td>
</tr>
<tr>
<td>( C_i )</td>
<td>Net cash flow generated at time ( t )</td>
</tr>
<tr>
<td>( D )</td>
<td>Diameter</td>
</tr>
<tr>
<td>( D_h )</td>
<td>Hydraulic diameter</td>
</tr>
<tr>
<td>( FF )</td>
<td>Fill factor</td>
</tr>
<tr>
<td>( F_H )</td>
<td>Carnavos correction factor</td>
</tr>
<tr>
<td>( F_{dirt} )</td>
<td>Scaling factor for dirt on a mirror</td>
</tr>
<tr>
<td>( F_{shade} )</td>
<td>Scaling factor for shading of a mirror</td>
</tr>
<tr>
<td>( F_{shape} )</td>
<td>Scaling factor for mirror shape error</td>
</tr>
<tr>
<td>( F_{uniformity} )</td>
<td>Scaling factor to account for the effect of non-uniform radiation on electrical output</td>
</tr>
<tr>
<td>( g )</td>
<td>Acceleration due to gravity</td>
</tr>
<tr>
<td>( \dot{G} )</td>
<td>Radiation flux intensity</td>
</tr>
<tr>
<td>( \dot{G}_T )</td>
<td>Total (direct and diffuse) radiation intensity</td>
</tr>
<tr>
<td>( \dot{G}_{d} )</td>
<td>Direct beam radiation flux intensity</td>
</tr>
<tr>
<td>( Gr )</td>
<td>Grashof number</td>
</tr>
<tr>
<td>( h )</td>
<td>Specific enthalpy</td>
</tr>
<tr>
<td>( h_c )</td>
<td>Heat transfer coefficient for convection</td>
</tr>
<tr>
<td>( J )</td>
<td>Local radiation flux intensity</td>
</tr>
<tr>
<td>( J )</td>
<td>Current</td>
</tr>
<tr>
<td>( J_0 )</td>
<td>Dark current, or reverse saturation current</td>
</tr>
<tr>
<td>( J_L )</td>
<td>Light generated current</td>
</tr>
<tr>
<td>( J_{mp} )</td>
<td>Current at the maximum power point</td>
</tr>
<tr>
<td>( J_{sc} )</td>
<td>Short circuit current</td>
</tr>
<tr>
<td>( k )</td>
<td>Thermal conductivity</td>
</tr>
<tr>
<td>( k_b )</td>
<td>Boltzmann's constant</td>
</tr>
<tr>
<td>( k_d )</td>
<td>Discount rate</td>
</tr>
<tr>
<td>( K )</td>
<td>Extinction coefficient</td>
</tr>
<tr>
<td>( kT/q )</td>
<td>Thermal voltage</td>
</tr>
<tr>
<td>( L )</td>
<td>Characteristic length</td>
</tr>
<tr>
<td>( m )</td>
<td>Mass</td>
</tr>
<tr>
<td>( \dot{m} )</td>
<td>Mass flow of fluid</td>
</tr>
<tr>
<td>( n )</td>
<td>Refractive index</td>
</tr>
<tr>
<td>( n_p )</td>
<td>Lifetime of a project</td>
</tr>
<tr>
<td>( Nu )</td>
<td>Nusselt number</td>
</tr>
<tr>
<td>( p )</td>
<td>Pressure</td>
</tr>
<tr>
<td>( P )</td>
<td>Perimeter of a fluid conduit</td>
</tr>
<tr>
<td>( P_n )</td>
<td>Nominal wetted perimeter for the fluid conduit (excluding fins)</td>
</tr>
<tr>
<td>( Pr )</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>( q )</td>
<td>Electronic charge</td>
</tr>
<tr>
<td>( Q )</td>
<td>Energy</td>
</tr>
<tr>
<td>( Q_{eq.elec} )</td>
<td>Equivalent electrical energy</td>
</tr>
<tr>
<td>( \dot{Q}_{th} )</td>
<td>Thermal output power</td>
</tr>
<tr>
<td>( \dot{Q}_{elec} )</td>
<td>Electrical output power</td>
</tr>
<tr>
<td>( \dot{Q} )</td>
<td>Rate of (heat) energy transfer</td>
</tr>
<tr>
<td>( \dot{Q}_{rad} )</td>
<td>Thermal heat loss due to radiation</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
</tr>
<tr>
<td>$\dot{q}_{sun}$</td>
<td>Solar radiation incident upon the receiver</td>
</tr>
<tr>
<td>$\dot{q}_{abs-cells}$</td>
<td>Radiation absorbed by the solar cells</td>
</tr>
<tr>
<td>$\dot{q}_{abs-glass}$</td>
<td>Radiation absorbed in the glass-silicone cover</td>
</tr>
<tr>
<td>$\dot{Q}_{rad}$</td>
<td>Thermal heat loss due to radiation from the glass surface</td>
</tr>
<tr>
<td>$\dot{Q}_{conv}$</td>
<td>Thermal heat loss due to convection loss from the glass surface</td>
</tr>
<tr>
<td>$\dot{Q}_{ins}$</td>
<td>Thermal heat transfer through the insulation</td>
</tr>
<tr>
<td>$\dot{Q}_{conv}^{*}$</td>
<td>Thermal heat loss due to convection loss from the insulation cover</td>
</tr>
<tr>
<td>$\dot{Q}_{rad}^{*}$</td>
<td>Thermal heat loss due to radiation from the insulation cover</td>
</tr>
<tr>
<td>$R_{cond}$</td>
<td>Thermal resistance for conduction</td>
</tr>
<tr>
<td>$R_{conv}$</td>
<td>Thermal resistance for convection</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>$R_s$</td>
<td>Series resistance</td>
</tr>
<tr>
<td>$R_{sh}$</td>
<td>Shunt resistance</td>
</tr>
<tr>
<td>$s$</td>
<td>Specific entropy</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature</td>
</tr>
<tr>
<td>$t$</td>
<td>Time</td>
</tr>
<tr>
<td>$T_0$</td>
<td>Environmental temperature</td>
</tr>
<tr>
<td>$T_m$</td>
<td>Fluid temperature</td>
</tr>
<tr>
<td>$T_f$</td>
<td>Film temperature (the average of the fluid and surface temperatures)</td>
</tr>
<tr>
<td>$T_s$</td>
<td>Surface temperature</td>
</tr>
<tr>
<td>$U$</td>
<td>Overall heat transfer coefficient = k/t</td>
</tr>
<tr>
<td>$u_m$</td>
<td>Mean fluid velocity</td>
</tr>
<tr>
<td>$u_{wind}$</td>
<td>Wind speed</td>
</tr>
<tr>
<td>$V$</td>
<td>Velocity of fluid</td>
</tr>
<tr>
<td>$V_{cc}$</td>
<td>Open circuit voltage</td>
</tr>
<tr>
<td>$V_{mp}$</td>
<td>Voltage at the maximum power point</td>
</tr>
<tr>
<td>$z$</td>
<td>Height</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>Absorption</td>
</tr>
<tr>
<td>$\alpha_H$</td>
<td>Helix angle of the fins = 0 for the CHAPS receiver</td>
</tr>
<tr>
<td>$\beta$</td>
<td>Temperature coefficient for the relationship between solar cell efficiency and temperature</td>
</tr>
<tr>
<td>$\delta$</td>
<td>Thickness</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>Azimuth angle</td>
</tr>
<tr>
<td>$\Delta t$</td>
<td>Small time interval</td>
</tr>
<tr>
<td>$e_g$</td>
<td>Emissivity of glass</td>
</tr>
<tr>
<td>$\eta_{pes}$</td>
<td>Primary-energy saving efficiency</td>
</tr>
<tr>
<td>$\eta_{power}$</td>
<td>Conversion efficiency of a conventional thermal power station</td>
</tr>
<tr>
<td>$\eta_{th}$</td>
<td>Thermal efficiency</td>
</tr>
<tr>
<td>$\eta_{elec}$</td>
<td>Electrical efficiency</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Angle of incidence of radiation</td>
</tr>
<tr>
<td>$\theta_{TR}$</td>
<td>Escape angle for Total Internal Reflection</td>
</tr>
<tr>
<td>$\theta_z$</td>
<td>Zenith angle</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Dynamic viscosity</td>
</tr>
<tr>
<td>$\mu_w$</td>
<td>Dynamic viscosity evaluated at the wall temperature</td>
</tr>
<tr>
<td>$v$</td>
<td>Kinematic viscosity = $\mu/\rho$</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Reflectivity</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>Stefan-Boltzmann constant</td>
</tr>
<tr>
<td>$= 5.67 \times 10^{-8} \text{ W.m}^{-2}.\text{K}^{-4}$</td>
<td></td>
</tr>
<tr>
<td>$\tau$</td>
<td>Transmission-absorption product</td>
</tr>
<tr>
<td>$= 1 - \rho$</td>
<td></td>
</tr>
<tr>
<td>$\tau$</td>
<td>Transmissivity</td>
</tr>
</tbody>
</table>
## Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>AM0</td>
<td>Air Mass 0, referring to the spectral distribution of sunlight outside the atmosphere</td>
</tr>
<tr>
<td>AMx</td>
<td>Air Mass 1.5, referring to the spectral distribution of sunlight when the sun is at angle $\cos^{-1}(1/x)$ from vertical</td>
</tr>
<tr>
<td>ANU</td>
<td>Australian National University</td>
</tr>
<tr>
<td>BOS</td>
<td>Balance of system</td>
</tr>
<tr>
<td>CHAPS</td>
<td>The Combined Heat and Power Solar collector</td>
</tr>
<tr>
<td>CPC</td>
<td>Compound Parabolic Concentrator</td>
</tr>
<tr>
<td>CSES</td>
<td>Centre for Sustainable Energy Systems, at the Australian National University</td>
</tr>
<tr>
<td>CSR</td>
<td>Circumsolar Ratio</td>
</tr>
<tr>
<td>DOE</td>
<td>U.S. Department of Energy</td>
</tr>
<tr>
<td>EQE</td>
<td>External Quantum Efficiency</td>
</tr>
<tr>
<td>FES</td>
<td>Fractional Energy Saving</td>
</tr>
<tr>
<td>GHG</td>
<td>Greenhouse gas</td>
</tr>
<tr>
<td>GOML</td>
<td>Glass On Metal Laminate - the material used to fabricate CHAPS mirrors</td>
</tr>
<tr>
<td>HWS</td>
<td>Hot water system</td>
</tr>
<tr>
<td>LEC</td>
<td>Levelised energy cost</td>
</tr>
<tr>
<td>LGBG</td>
<td>Laser Grooved Buried Grid</td>
</tr>
<tr>
<td>MPPT</td>
<td>Maximum power point tracker</td>
</tr>
<tr>
<td>NPV</td>
<td>Net present value</td>
</tr>
<tr>
<td>PT100</td>
<td>Temperature sensor using a platinum resistive device</td>
</tr>
<tr>
<td>PV</td>
<td>Photovoltaic</td>
</tr>
<tr>
<td>PV/T</td>
<td>Combined Photovoltaic / Thermal</td>
</tr>
<tr>
<td>SEF</td>
<td>Solar Energy Fraction</td>
</tr>
<tr>
<td>SHWS</td>
<td>Solar hot water system</td>
</tr>
<tr>
<td>SRCC</td>
<td>Solar Rating and Certification Corporation</td>
</tr>
<tr>
<td>TK</td>
<td>Thermocouple Type K</td>
</tr>
<tr>
<td>TRNSYS</td>
<td>A TRaNsient SYStem simulation program, used for solar system simulations</td>
</tr>
</tbody>
</table>
# Table of contents

Declaration .................................................................................................................................................. iii
Acknowledgements ................................................................................................................................. v
Abstract ................................................................................................................................................. vii
Foreword ................................................................................................................................................... ix
Nomenclature and Abbreviations ............................................................................................................ xi
  Nomenclature ........................................................................................................................................ xi
  Abbreviations ........................................................................................................................................ xiii
Table of contents ..................................................................................................................................... xv

## Introduction

1.1 Energy today ....................................................................................................................................... 1
1.2 Solar energy ....................................................................................................................................... 2
1.3 Objectives of this work ..................................................................................................................... 2
1.4 Thesis structure ............................................................................................................................... 3

## Background

2.1 The sun ............................................................................................................................................... 7
2.2 Photovoltaics ..................................................................................................................................... 8
  2.2.1 Concentrator solar cells ............................................................................................................... 11
2.3 Concentrator photovoltaic systems ............................................................................................... 13
2.4 Solar thermal .................................................................................................................................... 16
2.5 Combined photovoltaic – thermal ................................................................................................ 19
  2.5.1 Water cooled PV/T .................................................................................................................... 20
  2.5.2 Air cooled PV/T collectors .......................................................................................................... 21
  2.5.3 Concentrating PV/T collectors ................................................................................................. 21
2.6 Introduction to TRNSYS ................................................................................................................ 23
2.7 Heat transfer theory ....................................................................................................................... 24
  2.7.1 Convective heat transfer for internal flow .................................................................................. 24
  2.7.2 Convective heat transfer for external flow ............................................................................... 25

## Energy Value Comparison

3.1 Introduction ....................................................................................................................................... 27
3.2 Thermodynamic valuation ............................................................................................................... 28
  3.2.1 Energy ....................................................................................................................................... 28
  3.2.2 Primary-energy saving ............................................................................................................... 28
  3.2.3 Exergy ...................................................................................................................................... 29
3.3 Economic valuation ......................................................................................................................... 30
  3.3.1 Open Market Approach ............................................................................................................. 30
  3.3.2 Renewable Energy Market Approach ...................................................................................... 32
5.3 Heat transfer between the receiver and the fluid .......................................................... 73
  5.3.1 Determination of the rate of heat transfer in a receiver ........................................... 74
  5.3.2 Results and correlations for internal fins ................................................................. 75
5.4 Heat transfer from the surface of the receiver ............................................................ 77
  5.4.1 Radiative heat transfer ............................................................................................. 77
    5.4.1.1 Measurement of glass temperature ................................................................. 78
    5.4.1.2 Calculation of radiation losses ................................................................. 78
  5.4.2 Convective heat transfer ....................................................................................... 80
    5.4.2.1 Free convection ........................................................................................... 80
    5.4.2.2 Forced convection ....................................................................................... 80
    5.4.2.3 Mixed convection ....................................................................................... 81
    5.4.2.4 Convection calculations for a CHAPS receiver ............................................. 81
5.5 Heat transfer within the receiver materials ................................................................. 84
  5.5.1 Measurement of thermal resistance tests for various heat sinking tapes .................. 84
  5.5.2 Other materials .................................................................................................... 87
  5.5.3 Measured losses through the insulation ................................................................. 87
6.6 Simulation of the conduction using Strand7 ................................................................. 88
  6.6.1 Energy input ......................................................................................................... 89
  6.6.2 Energy loss ........................................................................................................... 90
  6.6.3 Base case ............................................................................................................. 91
  6.6.4 Validation ............................................................................................................ 92
  6.6.5 Sensitivity analysis ............................................................................................... 94
  6.6.6 Results of the Strand7 modelling ........................................................................... 98
    6.6.6.1 Wind speed and direction .......................................................................... 98
    6.6.6.2 Fluid temperature and flow rate .............................................................. 99
    6.6.6.3 Conductivity of the thermal tape ................................................................ 99

**Electrical Performance** Chapter 6.................................................................................. 101

6.1 Temperature dependency ............................................................................................ 101
  6.1.1 Measurement of I-V curves using the flash tester .................................................. 101
  6.1.2 Temperature dependency results from flash tester measurements ....................... 102
  6.1.3 Temperature dependency results from a full receiver ............................................ 103
6.2 Illumination profile ..................................................................................................... 104
  6.2.1 The sun shape ..................................................................................................... 105
6.3 Non-uniform illumination in the transverse direction ................................................. 106
  6.3.1 Modelling the effect of a non-uniform illumination profile .................................... 106
  6.3.2 Experimental comparison with the model .............................................................. 109
6.4 Non-uniform illumination in the longitudinal direction ............................................. 111
  6.4.1 The ‘Skywalker’ module - measurement of the longitudinal flux profile .............. 112
  6.4.2 Results from the skywalker module ...................................................................... 113
    6.4.2.1 Comparison of mirrors ............................................................................. 113
    6.4.2.2 Results from a single mirror for a range of incidence angles ....................... 115
    6.4.2.3 Attenuation of peaks and troughs .............................................................. 117
  6.4.3 Shape error of the mirror ..................................................................................... 118
  6.4.4 The effect of slope error on the reflected flux profile ............................................ 121
  6.4.5 Ray tracing – simulation of the longitudinal flux profile .................................... 124
A5 TRNSYS deck file for the system base case ................................................................. 240
Appendix B .................................................................................................................. 257
Appendix C .................................................................................................................. 261
Introduction

1.1 Energy today

A great challenge for the world energy industry today is to take action to slow global climate warming. While it is recognised that the earth’s climate varies naturally, the great majority of scientists now believe that greenhouse gas emissions attributable to human activities have increased the rate of climate change. The Third Assessment report from the Intergovernmental Panel on Climate Change (2001) states that “the global average surface temperature has increased over the 20th century by about 0.6°C” and this is projected to increase in the range 1.4 to 5.8°C over the period 1990 to 2100. The sea level is projected to rise in the range 0.11 m to 0.77 m, which will result in some flooding to low lying areas, predominately in poorer countries. Scientific evidence linking an increase in extreme weather events to increased greenhouse gas emissions has worried financial institutions, who are concerned that “worldwide economic losses due to natural disasters appear to be doubling every ten years” (United Nations Environment Programme, 2002). It is becoming well recognised by both governments and industry that climate change due to greenhouse gas emissions is fact.

According to the IPCC, the present atmospheric CO$_2$ concentration has not been exceeded in the past 420,000 years, and probably not in the last 20 million years. The rapid increase over the past century is because of human influence, with three-quarters of those emissions due to the burning of fossil fuels. In the longer term, fossil fuels are a finite resource and can be replaced by renewable energy sources. Recent International Energy Agency (2001) estimates are that oil will be depleted by 2100. Estimates of the remaining resources for gas and coal, including undiscovered resources, range from 170 to 200 years of supply for gas, and 200 years of supply for coal, at current production rates.

The challenge today is to mitigate climate change, and clearly the reduction of fossil fuel use is important. The challenge for the future is to replace fossil fuel to achieve sustainable energy generation.
1.2 Solar energy

Solar energy has the potential to play an important role in reducing greenhouse gas emissions. In many places, domestic solar hot water is already competitive with conventional hot water systems. It is estimated that the levelised energy cost of solar hot water in a sunny climate is around US 7.2 c/kWh (Coventry and Lovegrove, 2003). However a solar hot water system must compete with direct heat generation from gas and oil.

Solar electricity from photovoltaics is uncompetitive with fossil fuel based electricity in all but a few niche markets. A monthly survey of over 2000 photovoltaic modules by SolarBuzz (2004) gives the April 2004 module sales price as $5.85/Wp, and an electricity cost of US 39 c/kWh for a 2 kW domestic grid-connected system with battery backup.

Learning curve analysis suggests that without a ‘breakthrough’ technology, the strong growth in module production and the reduction in module cost seen over the past 20 years, must continue for another 15-20 years for solar electricity to compete with fossil fuel based electricity (Green, 2003a). Green suggests that module costs below US $2/Wp seem feasible in the medium term with conventional bulk silicon based photovoltaic technologies, but that costs below US $1/Wp are unlikely, yet necessary for photovoltaics to penetrate the energy market with large-scale applications. He advocates thin film silicon cells and novel “third generation” devices, but suggests that photovoltaic concentrators also have the potential for costs below US $1/Wp (Green, 2003b). A silicon roadmap developed by the National Renewable Energy Laboratory (Sopori et al., 2003) suggests that growth and cost reductions using conventional bulk silicon are possible for another 10 years, and that costs of US $1/Wp are achievable by 2012 with high efficiency silicon cells. However, Swanson suggests that alternative photovoltaic technologies such as concentrators and thin film solar cells will be required to continue the cost reductions beyond 2012 (Swanson, 2003).

1.3 Objectives of this work

This thesis discusses aspects of a novel solar concentrating photovoltaic / thermal collector that has been designed to produce both electricity and hot water. The hot water is produced at useful temperatures for applications such as building heating and domestic hot water, as well as many commercial and agricultural applications that require low-grade heat. The motivation for the development of the Combined Heat and Power Solar (CHAPS) collector is twofold: in the short term, to produce photovoltaic power and solar hot water at a cost which is competitive with other renewable energy technologies, and in the longer term, at a cost which is lower than possible with current technologies. To be capable of achieving this aim,
the CHAPS collectors must have an inherent advantage over other photovoltaic and thermal technologies. Most photovoltaic and thermal collectors used for similar applications are flat plate collectors. The primary advantage of the CHAPS system is that concentrating light allows a significant reduction in the area of solar cell coverage, the main cost driver in a flat plate system. The thermal energy generated could be considered as a byproduct due to the necessity to cool the cells, but in appropriate applications the thermal energy is equally valuable. A secondary advantage of the CHAPS system is the efficient use of space inherent in combining electrical and thermal energy generation, which may be advantageous on rooftops or in other applications where space is limited. The challenge in the development of the CHAPS system is to design a robust collector with a clear pathway to more rapid cost reduction than the incumbent flat plate technology, and to optimise the performance within the cost constraints. The latter is the subject of this thesis. The main objectives of this thesis are to address the key technical challenges in the design and operation of the CHAPS system, including:

- developing a detailed understanding of the physics of all components of the collector;
- understanding the sources of non-uniformities in the solar radiation flux profile that significantly affect electrical output;
- achieving good heat transfer from the solar cells to the cooling fluid to minimise the cell operating temperature and maximise thermal efficiency;
- achieving a robust receiver design that can withstand regular thermal cycling and radiation flux in excess of 100 times normal sunlight;
- developing a modelling tool that allows long term simulation of the collector.

### 1.4 Thesis structure

Chapter 2 gives an overview of the potential of the solar resource, as well as some background to the photovoltaics and solar thermal fields. A detailed review of research in the field of combined photovoltaic-thermal collectors is presented, which shows that most effort to date has focused on flat plate collectors using water or air as the working fluid. In addition, a review of concentrator PV applications has been undertaken, which shows that most applications use Fresnel lenses rather than reflective mirrors, and that point focus concentrators are more prevalent than linear concentrators.

Optimisation of a combined photovoltaic-thermal system involves drawing a comparison between the value of energy in the form of hot water and electricity. Chapter 3 discusses different techniques that may be used to draw this comparison, based on thermodynamic, economic and environmental methods of assessment. The concept of Equivalent Electrical Energy is introduced to relate thermal energy to electrical energy via a ratio.
Chapter 4 contains a detailed discussion of individual components of the CHAPS collector, including the solar cells, mirrors, receivers and sun tracking system. The optical performance of the mirror and receiver sub-components is examined over the entire solar spectrum to determine optical efficiency, and to identify the extent of radiation absorption in the cover materials.

Chapter 5 is concerned with the thermal performance of the CHAPS receiver, and includes experimental measurement of efficiency for a range of operating temperatures. Modes of heat transfer within the receiver are discussed, in particular the augmentation of heat transfer between the receiver and fluid by the inclusion of internal longitudinal fins. The validity of an empirical correlation that describes this heat transfer is tested experimentally. Thermal loss mechanisms are examined; radiation losses are measured by using a thermal imaging camera to determine the emissivity and temperature of the glass surface, conduction losses through the insulation are determined by experiment, and convection losses are estimated by calculation. The finite element analysis software Strand7 is used to examine the sensitivity of the thermal performance to changes in wind conditions and operating temperature. In addition, the Strand7 modelling is used to examine the solar cell temperature, which directly influences electrical efficiency.

Chapter 6 examines the electrical performance of the CHAPS receiver, and focuses in particular on the impact of non-uniform radiation flux profiles both transverse and longitudinal to the receiver. The dependency of the efficiency of the cells on temperature is quantified by experiment. The flux profile across the receiver is highly non-uniform, and the impact on cell performance is examined. It was found that the reduction in fill factor and open circuit voltage causes a drop in performance by around 5-15% depending on the temperature of the cells. Achieving good flux uniformity along the length of a receiver is perhaps the largest technical hurdle for PV concentrators. Low illumination on a single cell proportionally reduces its current, and hence affects the performance of all other cells in series. For linear concentrators, the effect of shading due to gaps between mirrors and receiver supports constrains the design and can reduce overall electrical output significantly. A key aspect of this thesis is the experimental examination of the flux profile, and the identification of methods to mitigate the problems and improve the electrical performance. The measured flux profiles are compared to simulated profiles generated using ray tracing software. The precise shape of the mirrors was measured using photogrammetry techniques, and the data incorporated into the ray tracing. It was found that a small degree of shape error significantly affects the flux profile. However, it was also found that shape error near the ends of the mirrors can have a positive impact, in part compensating for shading due to gaps between the mirrors.
In chapter 7, a new component is developed using the simulation software package TRNSYS, to simulate the CHAPS collector for the purpose of annual energy predictions. The theoretical formulation of the component is given in detail, and the component is calibrated using experimental data. A case study is developed to carry out annual simulations. The system model simulates a domestic hot water system, including thermal storage and a hot water load. The case study examines the effect of changes to the insulation thickness on the receiver and to changes in the conductivity of the tape bonding the solar cells to the receiver. The value of combining thermal and electrical generation for a domestic hot water system is discussed. The annual performance of the CHAPS system is compared to a typical flat plate solar hot water collector.
Background

Chapter 2

2.1 The sun

The surface temperature of the sun is around 6000 K. The centre of the sun is very much hotter, at around 20,000,000 K. For solar thermal considerations, it is adequate to consider the sun as a blackbody radiator at the surface temperature. However, the radiation emitted from the sun comes from a number of layers of different temperatures, which emit and absorb radiation of varying wavelengths.

The radiated energy that reaches the earth from the sun has different intensities at different wavelengths. Figure 2-1 compares the energy distribution one would expect from a perfect 6000 K black body at the mean earth-sun distance with that actually reaching the earth outside the atmosphere and on the ground.

![Figure 2-1. Solar spectral irradiance at AM0 and AM1.5.](image)

Air mass (AM) refers to the path length of sunlight through the atmosphere, and is approximated by $1/\cos(\theta)$, where the sun is at an angle $\theta$ to vertical. Air Mass Zero (AM0) is
the spectral distribution of sunlight outside the atmosphere, and has an integrated value of 1366.1 W.m\(^{-2}\) (ASTM E490).

For some wavelengths, atmospheric gases such as ozone, O\(_2\), H\(_2\)O, and CO\(_2\) strongly absorb light. This causes the absorption bands apparent in figure 2-1. Other sources of energy reduction include Rayleigh scattering by molecules in the atmosphere, and scattering by aerosols and dust particles. By the time light reaches the earth, the total energy density is about 970 W.m\(^{-2}\) for Air Mass 1.5. 1000 W.m\(^{-2}\) has become the standard for photovoltaic work.

### 2.2 Photovoltaics

The photovoltaics industry today generates annual revenues around US $3-4 billion. Demand has grown 20-25% annually over the past 20 years and PV module production was 742 MW in 2003 (Maycock, 2003, SolarBuzz, 2004) The installed capacity of grid-connected applications has outpaced that of off-grid during the past few years, with domestic applications the main growth area. The market for commercial grid-connected applications is expected to grow strongly during the next 10 years.

A photovoltaic solar cell is a semiconductor device, essentially a large diode, which converts sunlight into direct-current electricity. Over 90% of the world’s photovoltaic market is for crystalline silicon solar cells. Light with a wavelength of less than 1100 nm has enough energy to break a bond in the silicon, creating a free electron and a free hole per incident photon. These electrons and holes move through the crystal lattice. If an electron created in the p-region reaches the pn junction, it will cross to the n-region and accumulate with other electrons. The same process happens in reverse for the holes created in the n-region, and the effect is a voltage build up across the pn junction. Power can be extracted from the solar cell by connecting the p-region to the n-region through a load.

The External Quantum Efficiency (EQE) of a solar cell is a measure of the probability that photons incident on a solar cell contribute to the cell current. Comparison of the EQE for a good one-sun solar cell (SunPower Corporation, 2001) and the solar spectrum for AM1.5 (expressed as photons per unit area) shows close to 100% efficiency between 400 nm and 1000 nm (figure 2-2a). In other words, nearly all the available photons are creating electrons that contribute to the cell current. At wavelengths below 400 nm, many of the photons are lost due to increased reflection from the top surface of the cell and absorption in the anti-reflection coating itself. The electrons that are created are very close to the top surface and are mostly lost due to surface recombination. At wavelengths greater than 1100 nm, light
is only very weakly absorbed by silicon and does not contribute to electrical current generation. This is because silicon requires photons with energy greater than 1.1 eV, corresponding to wavelengths less than 1100 nm, to lift an electron from the valance band of the silicon atom to the conduction band where it can move about freely. The spectral responsivity of a solar cell is a measure of how many Amps are generated per Watt of incident light, and is plotted for the same solar cell in figure 2-2b. The spectral responsivity drops off for shorter wavelengths because the incoming photons have more energy than required to create a free electron. The excess energy is dissipated as heat.

Figure 2-2b also shows the cell output power curve across the solar spectrum based on the cell open circuit voltage and fill factor given by SunPower Corporation (2001) for the same cell. The area under this curve represents the electrical power generated by the cell. The remaining area under the sun’s spectral irradiance curve is power lost, a little due to optical losses, but mainly as heat. Therefore it is demonstrated that even a high quality silicon solar cell with close to ideal quantum efficiency will always have the bulk of the sun’s energy converted to heat. This is the premise for one of the motivations to study PV/T collectors – to significantly increase the fraction of solar energy utilised per unit area of collector.

![Figure 2.2](image)

*Figure 2-2. (a) External quantum efficiency and (b) spectral responsivity of a good quality solar cell from SunPower.*

Current-voltage (IV) curves are used to characterise solar cells. The shape of an IV curve is the same as a diode or solar cell in the dark. However the position of the curve with respect to the vertical axis, is proportional to the illumination, as shown in figure 2-3. Note that current is negative. References to an ‘increase’ in current throughout this thesis will refer to an increase in the negative direction.
The ideal relationship between current $J$ and voltage $V$ is given by the solar cell equation (Sze, 1985) as follows:

$$J = J_0 \left( \frac{qV}{e^{k_bT} - 1} \right) - J_L$$

(2-1)

When there is zero bias there is a reverse current, called short circuit current $J_{SC}$, equal to the magnitude of the light generated current $J_L$. The quantity $J_0$ is the dark current or reverse saturation current, $T$ is cell temperature, $q$ is electronic charge and $k_b$ is Boltzmann constant. For a silicon solar cell, when the forward bias across the solar cell rises to a value of around 0.6 to 0.7 V, a large forward current begins to flow, opposing the reverse current due to the illumination. The voltage where the net current is zero is called the open circuit voltage $V_{oc}$ and can be calculated as follows:

$$V_{oc} = \frac{k_bT}{q} \ln \left( \frac{J_L}{J_0} \right)$$

(2-2)

There is a point on the IV curve where the product of the current $J_{mp}$ and voltage $V_{mp}$ is maximum, called the maximum power point. The fill factor of a solar cell is defined as follows:

$$FF = \frac{V_{mp} \times J_{mp}}{V_{oc} \times J_{sc}}$$

(2-3)
Typical fill factors are around 70% to 80%.

The temperature of a photovoltaic module depends very much on the configuration of the installation. For example, a passively vented PV façade may run at around 60°C under ambient conditions of around 20°C. A thermally insulated PV façade may run as hot as 80°C, whereas an actively cooled façade can be cooled below 40°C (Krauter et al., 1999). The efficiency of a solar cell falls as the temperature increases, mainly due to a reduction in open circuit voltage, typically around 2.0 to 2.4 mV/°C. The reason for the sensitivity of voltage to temperature is that the reverse saturation current $J_0$ increases rapidly with temperature. Overall solar cell efficiency declines with temperature at a rate around 0.3 to 0.4 %/°C (refer to chapter 6 for experimental investigation of temperature dependency).

### 2.2.1 Concentrator solar cells

The ANU is currently manufacturing silicon solar cells for the CHAPS system that are around 20-22% efficient at 30x concentration. A prominent commercial producer of silicon concentrator cells is Sunpower Corporation, which produces back-contact cells with peak efficiency of 26% at 250x concentration (Swanson, 2000). Silicon solar cells for concentrator systems are designed to have low internal series resistance, since the high current density due to concentrated light means series resistance significantly affects the fill factor of the cell. Series resistance is made up of a contact resistance between the metal and semiconductor, ohmic resistance in the metal contacts and ohmic resistance in the semiconductor material. The other form of resistance is known as shunt resistance. This is caused by defects in the p-n junction as well as leaking currents along the edges of the solar cell. Shunt resistance losses are usually negligible at high concentration ratios. These resistances can be modelled in an equivalent circuit modelling a solar cell, such as figure 2-4. The circuit is comprised of a diode, a constant current generator, a shunt resistor and an internal series resistor.

![Equivalent Circuit Diagram](image)

**Figure 2-4. A solar cell equivalent circuit during normal operation.**

Based on the equivalent circuit, a relationship between current $I$ and voltage $V$ can be derived:
\[ I + I_L = I_0 \left( e^{\frac{q(V - IR_s)}{kT}} - 1 \right) + \frac{(V - IR_s)}{R_{sh}} \] (2.4)

Using typical values for a good solar cell (table 2-1), the IV curves for a range of series resistances have been plotted in figure 2-5. The graph shows that a one-sun solar cell, designed with a series resistance around 0.5 \(\Omega\text{cm}^2\), has a poor fill factor when illuminated to 30 suns. This can be compared to a typical ANU concentrator cell, with series resistance of 0.043 \(\Omega\text{cm}^2\).

**Table 2-1. Values used in the equivalent circuit equation**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>(R_{sh})</td>
<td>1000 (\Omega\text{cm}^2)</td>
</tr>
<tr>
<td>(J_0)</td>
<td>(10^{-13}) A/cm(^2)</td>
</tr>
<tr>
<td>(KT/q)</td>
<td>25.7 mV</td>
</tr>
<tr>
<td>(J_L)</td>
<td>38 mA/cm(^2)</td>
</tr>
</tbody>
</table>

**Figure 2-5a. I-V curves for a solar cell under 1 sun illumination for a range of values of series resistance.**

**Figure 2-5b. I-V curves for a solar cell under 30 suns illumination for a range of values of series resistance.**

While silicon is the most commonly used material for making concentrator solar cells, other compound semiconductor materials have been used. Gallium arsenide is the most common, and has been used extensively in the space industry where output power is more important than cost. Concentrator applications are beginning to see the use of multijunction solar cells with efficiencies in excess of 35%. Depending on the material from which they are made, solar cells vary in the efficiency by which they convert different parts of the solar spectrum to electricity. Multijunction cells achieve high overall efficiencies by having two or more cells stacked together, each optimally designed for a particular wavelength range within the solar spectrum. SHARP Corporation has recently produced triple-junction cells for concentrators...
using the edge margins of wafers processed for one sun space applications. The peak efficiency of 37.4% at 200x concentration, achieved in 2003, is a current world record (Takamoto, 2003). Spectrolab produces multijunction cells for concentrator applications, and achieved 36.9% peak efficiency in 2003 (Sherif et al., 2003). Research institutions currently working on multijunction cells for concentrators include the National Renewable Energy Laboratories in the United States (Duda et al., 2003) and Fraunhofer ISE in Germany (Bett et al., 2003).

2.3 Concentrator photovoltaic systems

Concentrator photovoltaic systems can be divided into two main categories: Fresnel lens refractors and parabolic reflectors. Both can be either point focus or linear focus concentrators. All concentrator systems with concentration ratios more than about 4x use direct radiation and not diffuse radiation, and therefore need to track the sun. Fresnel lenses and point focus mirrors must face directly towards the sun, and hence require two-axis tracking. Linear parabolic mirrors may track the sun on a single-axis only, and therefore can be mounted horizontally. Early experience with a variety of designs of photovoltaic concentrators found that refractive concentrators equipped with Fresnel lenses generally performed better than reflective concentrators at both the prototype module and operating system levels (Boes and Luque, 1993). This finding, along with a decline in government funding, led to a period of almost no activity in reflective concentrators. One reason is that the photovoltaic receiver assembly causes no shading, and large heat sinks may be used. Another reason is that the Fresnel lenses can be designed such that there is a near uniform radiation flux profile on the cells. However, mirrors are usually cheaper and allow for single-axis tracking and therefore simpler and cheaper mounting.

Figure 2-6. Cross-section of a Fresnel lens refractor (left) and a parabolic reflector (right).

The history of the development of photovoltaic concentrators is well covered by excellent summaries from Boes and Luque (1993) and Swanson (2000). Research on concentrating PV began in earnest in the mid-1970s and several large-scale demonstration projects were built, mostly using either point-focus Fresnel modules (such as the 3800 m² Solares system in Saudi
Arabia) or linear Fresnel modules (such as the 2000 m² ENTECH-3M system in Texas). The prevalence of Fresnel lenses has continued to the present, particularly point focus Fresnel systems. Amonix has recently installed over 600 kWp of concentrator modules in Arizona using point-focus Fresnel lenses and silicon cells. Amonix are conducting trials with multijunction cells for future systems. A collaborative project between Fraunhofer ISE and the Ioffe Institute has produced another point-focus Fresnel lens system called FLATCON™. The module uses dual junction cells with efficiency >30% from Fraunhofer ISE (Bett et al., 2003), and a 500x Fresnel concentrator consisting of a hermetically sealed glass box, with the Fresnel profile formed with silicone rubber on the inside of the glass (Rumyantsev et al., 2002). The module efficiency has been measured at 22.7%. The highest efficiency point focus Fresnel system to date is the 200 Wp 28% efficient module recently developed in Japan by Daido Metal and Daido Steel using injection-moulded dome shaped Fresnel lenses and multi-junction solar cells from SHARP Corporation (Araki et al., 2003). ENTECH uses linear concentrating Fresnel lenses for its fourth-generation concentrator module, but is currently looking at point focus lenses for use with multijunction cells (O'Neill et al., 2001). Sunpower, in conjunction with Ciudad University, is considering very low profile (3 cm) point focus Fresnel concentrators with very small solar cells (Terao et al., 2002).

In recent years there has been a small resurgence in the use of reflective optics for photovoltaic concentrators. Melbourne based company Solar Systems has been working on dish systems for some years, and recently commissioned a 220 kWp system serving a mini-grid on the Anangu Pitjantjatjara lands in central Australia (Lasich, 2003). The system comprises ten 130 m² dishes, each with 112 spherical mirror panels focusing light onto a 0.23 m² receiver. The receiver is made up of 64 sub-modules that use silicon concentrator cells from Sunpower (Solar Systems, 2004). Another recent dish project is the 115x concentration dish at the University of Ferrara, part of the European Union IDEOCONTE project. The system consists of 130 flat mirror facets focusing light onto a receiver with silicon concentrator cells. A 400 m³ dish installed at the Ben-Gurion University in Israel is proposed for testing with a photovoltaic receiver, but is still under development (Faiman et al., 2002). Another interesting project is the recent construction of a photovoltaic cavity converter (PVCC) by Ortabasi et al. (2002) that has four different cells arranged on the inside of a spherical cavity. Each cell has optimal efficiency within a limited wavelength range in the solar spectrum, and hence the cells are covered with Rugate filters that allow light of an appropriate wavelength to pass, and reflect the remainder. The cost effectiveness of the concept relies on very high concentrations of light. A prototype receiver has been built, but is yet to be fully tested.
In the medium concentration range (10x-40x) the 480 kWp EUCLIDES™ system, deployed in 1998, is the largest photovoltaic trough system (Sala et al., 2000). The collector has a split mirror arrangement, with each side of the trough focusing on a separate array of modified laser grooved buried grid (LGBG) silicon cells from BP Solar. The cells are cooled via a passive heat sink. The EUCLIDES™ system is the nearest equivalent to the CHAPS system, and as such shares many similar technical challenges, in particular with the effect of non-uniform illumination on the cells, as discussed by Anton et al. (2000).

The field of low concentration photovoltaic systems (less than 5x) has received some recent attention, driven by the advantage of using production line one sun cells with little or no modification, such as the LGBG cell in production by BP Solar (Bruton et al., 2002). Additionally, the use of non-imaging optics, such as compound parabolic concentrators (CPCs), is attractive for building integration. CPCs are low concentration non-tracking concentrators with a solar acceptance angle $\theta_a$, as shown in figure 2-7. If a CPC is designed not to track the sun, a wide acceptance angle of $\theta_a$ $>$ 30° is desirable, giving a concentration ratio of 2x or less. However if the CPC is tilted two or more times a year, and oriented in the east-west direction, the acceptance angle can be smaller giving concentration ratios up to 4x (Brogren, 2001). Some recent low concentration photovoltaic projects include the MaReCo collector (Brogren and Karlsson, 2002), the ARCHIMEDES V-trough system (Mohring and Gabler, 2002), and the PV Venetian Store (Alonso et al., 2002). However, there are many more systems based on similar optical systems.

![Compound parabolic concentrating (CPC) collector.](image)

All concentrating PV systems must have some form of cooling system for the solar cells. Most PV concentrators rely on passive cooling through fins, such as those shown in figure 2-8.
High concentration PV systems such as the Solar System dishes require active cooling to effectively remove heat, due to the high flux of solar energy incident on the cells. They also require effective heat transport from the cells to the cooling fluid. Because the efficiency of the cells is diminished with increasing temperature, the thermal energy collected by the cooling fluid is suited to low to medium temperature applications such as domestic hot water, building heating, pool heating and desalination.

### 2.4 Solar thermal

Solar thermal energy refers to direct conversion of the energy from the sun into useful thermal energy for applications such as domestic hot water, solar thermochemical processes, solar detoxification, distillation, drying of crops, building heating and of course, power generation. By far the largest part of the solar thermal industry is solar water heaters. The estimated deployment of solar thermal heating in the European Union countries was 10.8 million m\(^2\) in 2002, with 11.7% growth over the previous decade (European Solar Thermal Industry Federation, 2003a). The United States has seen a small decline in recent years in the total area of solar collectors, to 11.1 million m\(^2\) in 2001, three quarters of which is unglazed low temperature collectors primarily for pool heating. The largest and fastest growing market is China, with an estimated 40 million m\(^2\) of solar collectors installed in 2002, predominately evacuated glass tubes (European Solar Thermal Industry Federation, 2003b). Other major solar thermal markets are Japan, India, South Africa, and the Middle East, in particular Israel. Most solar hot water collectors are flat plate collectors with an absorber plate made of copper, aluminium or steel, and coated with a selective black paint, black chrome or black crystallographic metal alloy surface. The absorber plate is bonded or pressed to a copper or steel tube through which a fluid flows, usually water or a water-glycol mixture. A low iron tempered glass is used to protect the absorber surface and create an air gap for top-surface insulation. The back of the solar collector is insulated by material such as polyester or polyisocyanurate.
Figure 2-9a. Amonix 35 kWp Fresnel lens, point focus concentrator PV system at Prescott, Arizona.

Figure 2-9b. Test site for four 200 Wp Fresnel lens, point focus concentrator PV modules from Daido Steel & Daido Metal.

Figure 2-9c. ENTECH 100 kWp Fresnel lens, linear focus concentrator PV system at Fort Davis, Texas.

Figure 2-9d. University of Ferrara 100x concentration faceted dish concentrator.

Figure 2-9e. The 220 kWp PV dish concentrator system on the Anangu Pitjantjatjara lands, central Australia.

Figure 2-9f. The 33x concentration prototype EUCLIDES PV trough concentrator.
Flat plate collectors are used mainly for domestic hot water heating, pool heating, and to a lesser extent, space heating through radiators or a heating floor. These applications rarely need hot water delivered at more than about 65°C. Flat plate collectors rapidly lose efficiency at higher temperatures due to heat losses from the large collector area. Evacuated tube collectors are often used for applications where slightly higher temperatures are needed; for example, water at 80°C to 90°C is required to drive an absorption chiller for air conditioning. Evacuated tube collectors consist of either a coated fin-tube absorber strip surrounded by an evacuated glass tube, or a double layer glass tube, with the inner tube coated on its outside surface. Convection losses are minimal and so the collector can operate at higher temperatures. However, evacuated tube collectors are expensive relative to flat plate collectors and, with the exception of China, occupy a niche market.

There are many solar applications where high temperatures are required, and this can only be achieved by concentrating the sunlight. There is more flexibility in the method for light concentration for solar thermal than photovoltaic systems, as the flux distribution at the focus is not as critical. For this reason, concentration of light using reflective optics is suitable for most applications. Single-axis tracking, horizontally mounted systems allow much simpler pipe connections. Large collectors based on reflective optics dominate solar thermal concentrator applications today.

Concentrating solar thermal systems can produce high temperature heat that can be converted into electricity in conventional heat engines. The biggest and most well known solar thermal power plants, the SEGS I to IX plants, are located at Kramer Junction in the Californian desert. They consist of around 2.3 km² of parabolic mirrors, focusing light at concentration ratios of 60-80x onto evacuated tubes containing a mineral heat transfer fluid. Steam is produced by heat transfer to water at the power blocks, driving a standard Rankine-cycle power plant. The SEGS plants combined have a peak output of around 350 MW.
Power tower systems use an array of large individually tracking mirrors, known as heliostats, to focus light onto a central receiver mounted on top of a tower. Very high temperatures up to 1500°C can be achieved at concentration ratios around 1500x. Power tower systems have been successfully demonstrated in America with Solar One, which used water/steam as the heat transfer fluid, and Solar Two, which was an upgrade of Solar One and used molten salt as both the heat transfer fluid and the heat storage media. A third project (‘Solar Tres’) is planned in Spain, also using molten salt.

![Schematic diagram of a power tower.](image)

At the ANU, the use of dish concentrators has been advocated since the 1970s. A dish pilot facility (fourteen 20 m² dishes) using a 25 kWₑ water/steam engine-generator was built and tested in the 1980’s to power the remote Australian township of White Cliffs (Kanef, 1987). In 1994 the ANU built a prototype 400 m² ‘Big Dish’, connected to a 50 kWₑ steam engine (Lovegrove et al., 2003). The ANU has also worked on solar dish concentrator systems incorporating thermochemical storage of energy using ammonia (Lovegrove et al., 2004).

### 2.5 Combined photovoltaic – thermal

Combined photovoltaic – thermal (or PV/T) collectors describe any solar energy collector that produces both dc electricity from solar cells and useful heat. In the loosest sense, this definition includes semi-transparent PV panels such as dye-sensitized solar panels used for daylighting, where heat and light is transferred directly into a building. However, in general the term ‘PV/T collector’ refers to a system with active heat collection through a heat transfer fluid, usually air or water, and sometimes both. While not a new concept, PV/T collectors have not progressed far from the laboratories of research institutions around the world.
The main motivations for combined PV/T systems over separate PV and thermal systems are expected cost reductions due to the combined production, installation and mounting, and more efficient utilisation of the available sunlight, which results in a reduction in the installed collector area and alleviates potential problems with availability of roof space.

### 2.5.1 Water cooled PV/T

![Diagram of Water cooled PV/T](image)

*Figure 2-12. Side view of a single PV/T collector strip with water as the working fluid*

Figure 2-12 shows a common configuration for a PV/T collector strip, using water as the working fluid. The first detailed simulation of a PV/T system using collectors of this kind was developed at the University of Pennsylvania (Wolf, 1976). The results indicated that significant overall energy conversion gains could be achieved by the PV/T system compared to separate PV and hot water systems alone. The earliest published mathematical model of a PV/T collector was by Florschuetz (1979) of Arizona State University. He modified the well-known Hottel-Whillier model for flat plate thermal collectors to include a PV component, and his equations were later used for a TRNSYS simulation model of a PV/T collector (Solar Energy Laboratory, 2000). Since then, more detailed physical modelling of PV/T collectors has been carried out, investigating dynamic effects (Zondag et al., 2002) and various physical parameters such as mass flow (Prakash, 1994), the dimensions of the absorber plate and tube (Bergene and Lovvik, 1995) and tank size in the case of thermosiphon systems (Agarwal and Garg, 1994, Huang et al., 2001). Sandnes and Rekstad (2000) experimentally compared a PV/T collector operating at zero current output with a conventional thermal collector, and showed that the lower optical absorption of solar cells combined with increased heat transfer resistance resulted in reduced thermal efficiency.
2.5.2 Air cooled PV/T collectors

![Diagram of Air cooled PV/T collectors](image)

*Figure 2-13. Side view of a PV/T air collector.*

Figure 2-13 shows one configuration of a PV/T collector that uses air as the working fluid. Various other configurations have been investigated, such as double pass collectors (Hegazy, 2000, Sopian et al., 1997, Sopian et al., 1996) and collectors with additional aluminium absorption baffles (Tripanagnostopoulos et al., 2001a). Studies at MIT (Cox and Raghuraman, 1985) suggested that PV/T air collectors were less thermally efficient than PV/T collectors using water as the heat transfer fluid. A key reason was the low absorber to air heat transfer coefficient, which is mainly due to the fact that air has much lower conductivity than water. Other reasons cited, such as high infrared emittance from the cells and low PV cell packing factor, are common to PV/T water collectors. A novel suggestion by Cox was to use a gridded metal back contact for cells mounted on glass to allow light with long wavelength to pass to the back absorber plate. Various options for combining PV/T collectors with both air and water collecting thermal energy have been proposed (Elazari, 2000, Prakash, 1994, Tripanagnostopoulos et al., 2001b). PV/T air collectors have been installed in a number of showcase projects such as the PV-Vent houses in Copenhagen (Pedersen, 2000) and the Brockshill Environment Centre in Leicester (Cartmell et al., 2001).

2.5.3 Concentrating PV/T collectors

![Diagram of Concentrating PV/T collectors](image)

*Figure 2-14. Low concentration non-tracking PV/T collectors*
Most concentrating PV/T collectors studied to date have been designed for domestic applications, and therefore the focus has been on low concentration non-tracking PV/T collectors that require little maintenance. Garg analysed the use of non-tracking plane reflectors with concentration ratio around 2x for a PV/T air collector (Garg et al., 1991) and a thermosiphon system (Garg et al., 1994). Sharan and Kandpal (1992) tested a single-axis tracking PV/T collector that used a Fresnel reflector of 5x concentration. The use of CPC troughs for PV/T concentrating collectors has received some interest recently (Brogren, 2001, Brogren and Karlsson, 2002, Brogren et al., 2000, Garg and Adhikari, 1999). Garg analysed a 3x concentration CPC with air as the working fluid, and concluded that the concentrator can improve performance if higher operation temperatures are required. In Sweden, Brogren is exploring the use of CPCs for PV/T applications that require water for space heating. In particular, Brogren is investigating the optical properties of CPCs and the impact on PV cell performance due to effects such as non-uniform illumination, high temperature and high light intensity.

![Cross-section of a concentrating PV/T collector similar to the CHAPS collector.](image)

The CHAPS system, at a concentration ratio of around 30-40x, is one of the few medium concentration PV/T systems that has been built and tested. Previously ENTECH have tested two demonstration systems using linear Fresnel lenses with combined PV/T (O’Neill, 2003). The first was a 25x concentration system at the Hyatt hotel at Dallas / Fort Worth International Airport. This 24 kW electrical and 120 kW thermal system was in operation during 1982 - 1992, and supplied electricity and hot water to the central utility plant at the hotel. The second system at Sandia-Albuquerque was a 40x concentration system, with 22 kW electrical and 70 kW thermal.

High concentration of sunlight will cause solar cells to operate at high temperatures unless there is very good heat transfer from the cells to the heat sink. As discussed previously in section 2.2, most of the light absorbed by the solar cells is dissipated as heat. Some heat generation is unavoidable due to the fact that only the bandgap energy can be used for electrical current generation. One concept that reduces the heat flux on the cells, and allows high temperature applications, is spectral beam splitting. Using either a bandpass or bandstop filter, radiation in the useful range for photovoltaic conversion is directed to the solar cells,
and the remainder is directed to a thermal receiver. This method is known as spectral beam splitting, and is described in an excellent review by Imenes and Mills (2002). Many researchers have been attracted by the apparent simplicity of the concept, and the potential of very high electrical conversion efficiencies by combining efficient photovoltaic conversion with thermal electricity generation by heat engines or thermophotovoltaic systems operating at elevated temperatures. Hamdy et al. (1988) carried out a TRNSYS simulation comparing a 50x concentration PV/T collector with beam splitting to 50x and 22x concentration PV-only collectors. They found that the system with beam splitting had higher conversion efficiency due to the removal of the unwanted part of the spectrum, but lower overall electrical output, and concluded that the merit of beam splitting depend on an economic evaluation of the extra cost of the filters and the usefulness of the thermal energy. A Japanese consortium (Yang et al., 1997) proposed a similar system using a selective parabolic mirror. Spectral beam splitting has been proposed for two concentrator systems in Australia, the Multi Tower Solar Array (Mills et al., 2002) and the Solar Systems SS20 dish collector (Lasich, 2001).

To the author’s knowledge, the CHAPS system is the only concentrator PV/T system using a reflective linear concentrator with concentration ratios in the range 20–40x. As such, most of the challenges faced to optimise the CHAPS system are new challenges and require new solutions.

2.6 Introduction to TRNSYS

TRNSYS is a software program used mainly for system simulation using annual weather data, and is particularly useful in the design phase of a project (Solar Energy Laboratory, 2000). A TRNSYS model of the CHAPS collector is outlined in chapter 7 in this thesis. The model was used extensively during the design phase for the Bruce Hall project. TRNSYS relies on a modular approach to solve large systems of equations described by Fortran subroutines. Each Fortran subroutine (called a Type) contains a model for a system component. TRNSYS has been under development at the University of Wisconsin-Madison since the 1970s, and was released commercially in 1975. TRNSYS is informally supported by a large world wide user group, and anyone is able to write their own model and interface it with other models available in the standard TRNSYS library. In each model, the relationship between the input and output are defined using algebraic and first order differential equations. The models are linked together to form a system, for example, a solar hot water system, which would have separate Types for handling weather data, thermal storage, solar collection, water pumping and control. TRNSYS solves the set of equations created by the system at each time step. The user defines parameters of each subcomponent, and the links between components. This information is contained in a text file known as a Deck. A number of graphical interfaces are available for creating the deck files. The graphical interface IISiBat 3 was used in this thesis.
2.7 Heat transfer theory

Analysis of the performance of a solar thermal collector requires an understanding of the various modes of heat transfer. Some basic heat transfer theory is presented below, and referred to later in chapter 5 with particular reference to the CHAPS receiver.

2.7.1 Convective heat transfer for internal flow

The heat transfer for internal flow within a passage is strongly dependent on whether the flow is laminar or turbulent. The Reynolds number for flow in a circular tube is given by:

\[ Re_D = \frac{u_m D}{\nu} \]  \hspace{1cm} (2-5)

where \( u_m \) is the mean fluid velocity over the tube cross section, \( D \) is the tube diameter, and \( \nu \) is the kinematic viscosity of the fluid. For fully developed flow, the onset of turbulence is somewhere between \( Re_D \approx 2300 \) and 10,000, depending on factors such as the smoothness of the tube.

For non-circular tubes, it is common for a term known as *hydraulic diameter* to be used for the calculation of Reynolds number. Hydraulic diameter \( D_h \) is based on the relationship between the flow cross-sectional area \( A_{xs} \) and the wetted perimeter \( P \):

\[ D_h \equiv \frac{4A_{xs}}{P} \]  \hspace{1cm} (2-6)

The Nusselt number \( Nu \) provides a dimensionless measure of the convective heat transfer from the inside surface of a tube:

\[ Nu_D = \frac{h_c D}{k} \]  \hspace{1cm} (2-7)

where \( h_c \) is the heat transfer coefficient for convection, \( k \) is the thermal conductivity of the fluid, and \( D \) the tube diameter. As for the Reynolds number, the hydraulic diameter \( D_h \) is often used for non-circular tubes. Generally for *turbulent* flow there is emphasis on empirical correlations in the calculation of the Nusselt number, because of the complexity of the flow.
A commonly used empirical expression is the Dittus-Boelter equation (as stated by Incropera and DeWitt (1990)):

\[ \text{Nu}_D = 0.023 \cdot \text{Re}^{0.8} \cdot \text{Pr}^{0.4} \]  

(2-8)

where Pr is the Prandtl number, a property of the fluid\(^1\). An empirical relation for fully developed laminar flow in tubes was proposed by Siedler and Tate (as stated by Holman and White (1992)):

\[ \text{Nu}_D = 1.86 \left( \frac{\text{Re}_D \cdot \text{Pr}}{\text{Pr}^{1/3} \left( \frac{\mu}{\mu_w} \right)^{0.14}} \right) \]  

(2-9)

where \(L\) is the length of the tube, and \(\mu\) dynamic viscosity. All properties are evaluated at the mean bulk temperature of the fluid, except \(\mu_w\), which is evaluated at the wall temperature.

2.7.2 Convective heat transfer for external flow

A solar collector has heat losses due to both free and forced convection. Free convection is driven by the natural buoyancy forces of air heated near the collector surface. Forced convection is due to wind. Empirical correlations for free convection heat transfer (assuming laminar flow) are typically of the form:

\[ \overline{\text{Nu}}_L = C(G_{RL} \cdot \text{Pr})^m \]  

(2-10)

where the average Nusselt number \(\text{Nu}\) over characteristic length \(L\) is given by a function of the Grashof number \(G_{RL}\) and the Prandtl number \(\text{Pr}\). For a vertical isothermal plate, the constants are \(C = 0.59\) and \(m = 0.25\) for \(10^1 < G_{RL} \cdot \text{Pr} < 10^5\). The Grashof number gives the ratio of buoyancy to viscous forces, and is calculated using the following equation based on the characteristic length:

\[ \text{Gr}_L = \frac{g \beta \Delta T L^3}{v^2} \]

---

\(^1\) Note that the exponent for the Prandtl number has value 0.4 for heating and 0.3 for cooling, but for the present application water is always being heated, and therefore the value is always 0.4.
Gr_L = g \beta (T_s - T_\infty) L^3 / v^2 \quad (2-11)

where $g$ is the acceleration due to gravity, $\beta$ is the volumetric thermal expansion coefficient (which for a perfect gas is $T_i^{-1}$), $T_s$ is the surface temperature, $T_\infty$ is the fluid temperature and $\nu$ the kinematic viscosity of the fluid. All properties are evaluated at the film temperature, $T_f = (T_s + T_\infty)/2$. Churchill and Chu (1975) proposed a more accurate correlation for free convection on a vertical isothermal plate of the form:

$$\overline{Nu}_L = 0.68 + \frac{0.670(Gr_L Pr)^{1/4}}{1 + (0.492 / Pr)^9/16} \quad (2-12)$$

for $0 < Gr_L Pr < 10^9$. Empirical correlations for forced convection heat transfer tend to take on a form similar to that of free convection (equation 2-10) as shown below:

$$\overline{Nu}_L = C \, Re_L^m \, Pr^n \quad (2-13)$$

where the Reynolds number $Re_L$ gives the ratio of inertial to viscous forces for forces (analogous to the Grashof number above) given by:

$$Re_L = \frac{VL}{\nu} \quad (2-14)$$

where $V$ is the velocity of the fluid. Using the Blasius similarity solution outlined in Incropera and DeWitt (1990), the coefficients in equation 2-13 can be shown to be $C = 0.664$, $m = 1/2$ and $n = 1/3$ for a flat plate in parallel flow.
3.1 Introduction

When considering the design of a PV/Thermal collector, determination of the ratio of the values of the electrical and thermal output from the system allows a rational approach to optimising such a system. Important decisions that physically impact upon the design of a PV/Thermal are made based on prioritising the electrical and thermal output. An energy value ratio helps determine the appropriate priority given to the electrical or thermal components of a PV/Thermal collector. The concept of Equivalent Electrical Energy is introduced, defined as the sum of the thermal energy output divided by the energy value ratio, and the electrical energy output. This chapter discusses three different methodologies for determining this energy value ratio: thermodynamic valuation, market based valuation and environmental valuation.

The thermodynamic section addresses the common misconception that a Joule of energy in the form of heat and a Joule of energy in the form of electricity are equally valuable. Electrical energy and thermal energy are not equally useful. The concept of usefulness of energy can be described roughly using examples, or a precise definition can be adopted using the thermodynamic concept of exergy. We are familiar with the fact that thermal electric power plants have energy conversion efficiencies, around 40%, and therefore electricity that comes out of a power plant is clearly more precious than an equivalent amount of thermal energy that goes in. However, if the thermal energy is in the form of domestic hot water for washing and cleaning, then the link between the value of the hot water and the value of the electricity is less clear. Therefore a market based methodology is proposed, that uses levelised energy costs of conventional PV and solar hot water systems to determine an appropriate energy value ratio. The final methodology takes an environmental approach, comparing the greenhouse gas intensity of conventional PV and solar hot water systems. This chapter focuses on domestic sized PV/Thermal systems, where the thermal energy is stored for use as domestic hot water. The methods are demonstrated using data appropriate for Australian conditions.
3.2 Thermodynamic valuation

3.2.1 Energy

The first law of thermodynamics tells us that work and heat are both forms of energy. Thus a simplistic first law approach is to consider them to be of equal value. A number of studies of PV/Thermal collectors have used a simple one-to-one energy approach to express the combined efficiency of the electrical and thermal output (Abdalla and Wilson, 2001, Hegazy, 2000, Sharan and Kandpal, 1992, Sopian et al., 1997, Sopian et al., 1996). However both the second law of thermodynamics and common sense tell us that electricity is not the same as heat, and is more useful.

3.2.2 Primary-energy saving

To improve on the simple first law approach, one method that has been proposed is to use an energy efficiency ratio derived from conventional power plants to give a weighted overall efficiency measure (Huang et al., 2001). In this approach, a so-called primary-energy saving efficiency $\eta_{pes}$, is defined, to give a favourable weighting to the electrical output:

$$\eta_{pes} = \frac{\eta_{elec}}{\eta_{power}} + \eta_{th}$$  \hspace{1cm} (3.1)

where $\eta_{elec}$ is the PV conversion efficiency, $\eta_{power}$ is the conversion efficiency of a conventional thermal power plant, and $\eta_{th}$ is the thermal efficiency of the PV/Thermal system. Assuming a power plant is 40% efficient, this method is equivalent to assuming an electrical / thermal value ratio of 2.5. While an improvement on a direct energy comparison, this technique does not consider the temperature or the pressure of the thermal output from the PV/Thermal system. Power stations use high-pressure steam suitable for driving turbines, which would not be generated by PV/Thermal collectors. It is self-evident that low temperature hot water from a PV/Thermal system is not as thermodynamically useful as high temperature steam from a coal-fired boiler.
3.2.3 Exergy

The second law of thermodynamics addresses the fundamental limits that apply in the efficiency of conversion of heat to work. The predictions and formulations of the second law all follow from the simple observation that heat flows naturally from hot to cold objects but never the other way. Since work (e.g., electricity) can always be done on systems no matter how hot they are, it has a status equivalent to heat flow from a body of infinite temperature. The second law leads to a quantification of the “relative value” of different energy streams via Exergy analysis. Exergy (sometimes called Availability) is defined as the maximum theoretical useful work obtainable from a system as it returns to equilibrium with the environment. Bejan et al. (1996) strongly advocate determining the electrical / thermal value ratio for system optimization, using exergy analysis. Researchers in Japan have applied a similar idea to annual results from a flat plate PV/Thermal collector (Fujisawa and Tani, 1997, Takashima et al., 1994).

For a control volume, the rate of delivery of exergy from a flow ($\Delta \hat{A}_{21}$) if there is a single inlet and single exit denoted by 1 and 2, respectively, is given by (see, for example, Moran and Shapiro (1998))

$$\Delta \hat{A}_{21} = \dot{m} \left( h_2 - h_1 - T_0 (s_2 - s_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1) \right)$$  \hspace{1cm} (3-2)

where $\dot{m}$ is the mass flow, $h$ and $s$ are specific enthalpy and entropy respectively, $T_0$ the environmental temperature, $V$ the fluid velocity, $z$ the height of the inlet or outlet, and $g$ the acceleration due to gravity. The energy delivered to the control volume by the same flow ($\Delta \hat{Q}_{21}$) is:

$$\Delta \hat{Q}_{21} = \dot{m} \left( h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1) \right)$$  \hspace{1cm} (3-3)

Table 4-1 summarises the energetic and exergettic value of the energy flows for the typical conditions of a PV/T system.
Table 3-1. Exergetic comparison of electrical and thermal output.

<table>
<thead>
<tr>
<th>Energy stream</th>
<th>Energy**</th>
<th>Exergy equation</th>
<th>Exergy</th>
<th>Energy/Exergy ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electrical</td>
<td>1000 W</td>
<td>$\Delta \hat{A}<em>{\text{elec}} = \Delta \hat{Q}</em>{\text{elec}} = \dot{m}(h_2 - h_1)$</td>
<td>1000 W</td>
<td>1</td>
</tr>
<tr>
<td>Thermal *</td>
<td>1000 W</td>
<td>$\Delta \hat{A}_{\text{th}} = \dot{m}(h_2 - h_1 - T_0(s_2 - s_1))$</td>
<td>59.5 W</td>
<td>16.8</td>
</tr>
</tbody>
</table>

* Assuming: $p = 500$ kPa, $T_1 = T_0 = 25^\circ$C, $T_2 = 65^\circ$C, $\dot{m} = 5.98$ g/s, $V_1 = V_2$ and $z_1 = z_2$
  which gives $h_1 = 105$ kJ/kg, $h_2 = 272$ kJ/kg, $s_1 = 0.366$ kJ.kg$^{-1}$.K$^{-1}$ and $s_2 = 0.893$ kJ.kg$^{-1}$.K$^{-1}$.

** Energy is expressed here as a rate of energy delivered, or ‘power’.

For this example, the ratio of electrical-to-thermal exergy is 16.8. However, the significance of an exergy comparison is not clear if electrical or mechanical work is not the only desired output from the system, such as when the thermal output is hot water used directly for showers and washing.

3.3 Economic valuation

3.3.1 Open Market Approach

In the open market approach an electricity / thermal value ratio is given by the ratio typically achieved by a household using conventional electricity and hot water generating sources available on the open market. The price of electricity is reasonably straightforward to determine, as the lowest price offered by the retailers to deliver electricity to the consumer. It is more difficult to attach a price to thermal energy based on market value, as usually it is not thermal energy that is delivered to the consumer, but energy in the form of electricity or gas. This is then converted to thermal energy in the form of hot water, and stored in a tank or circulated around the house for heating. The main exception is district heating, where thermal energy is delivered directly to customers through hot water or steam in pipes.

The price of the thermal energy must therefore incorporate the cost of a hot water tank with its electric elements or gas burners. Even heat from a district heating system has an additional installation cost, as a heat exchanger is required to extract heat from the primary hot water loop. Thermal storage systems have heat losses during the day, which increase the cost of the hot water delivered. Gas boosted storage systems are not perfectly efficient, further increasing the cost of hot water delivered.
The cost of a gas or electric hot water system can be expressed as a levelised cost in $/kWh by applying discounted cash flow methods. The net present value (NPV) is equal to the difference between the present value of the net cash flows generated by a project and the initial cash outlay. This can be expressed as:

$$NPV = \sum_{t=1}^{n_p} \frac{C_t}{(1 + k_d \Delta t)^t} - C_0$$

(3-4)

where $C_0$ is the capital cost, $C_t$ is the net cash flow generated at time $t$, $n_p$ is the life of the project, $k_d$ is the discount rate and $\Delta t$ is the compounding interval. The NPV equation is often presented with the $\Delta t$ omitted because it is assumed to be one year, but this is not strictly correct. The levelised energy cost for a system is the unit price of energy output that would result in the system having a zero NPV over its lifetime. Table 3-2 sets out an example of levelised cost calculations for gas and electric hot water services under Australian conditions.

<table>
<thead>
<tr>
<th>Table 3-2. Example of thermal energy cost calculations.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unit</td>
</tr>
<tr>
<td>Capital cost of HWS</td>
</tr>
<tr>
<td>Gas or Electricity consumed</td>
</tr>
<tr>
<td>Primary energy price</td>
</tr>
<tr>
<td>Effective energy cost*</td>
</tr>
<tr>
<td>Levelised cost of HWS</td>
</tr>
<tr>
<td>Total cost of hot water</td>
</tr>
</tbody>
</table>

* Note: higher than the primary energy price due to the inefficiencies in heating and storing the water

Common assumptions:
Cost data is in ¥2001 Australian Dollars
Installation cost is $200
Hot water energy demand is 3490 kWh/year
Life of the HWS is 15 years
Inflation rate is 3.0%
Discount rate 7.82%
Capital cost and installation prices come from Rheem (Feb, 2001). Hot water demand was taken from Australian Standards AS4234-1994, based on a large domestic system based in Canberra. Energy consumption was calculated using the simulation software TRNSYS (Klein, 1976a, Klein, 1976b), with gas burner efficiency and thermostat settings also based on Australian Standards. 50 mm insulation was assumed for the electric off-peak and high efficiency gas tanks, and 25 mm for standard tanks. System lifetimes were estimated, based on a typical warranty period of 10 years, and data from a survey of the United States by ASHRAE Technical Committee TC 1.8 (Akalin, 1978). The discount rate is a nominal rate based on the variable loans rate for personal mortgages with the major Australian banks. Both gas and electricity energy prices were obtained from the Australian Gas Association (2000) adjusted to 2001 dollars, with off-peak rates assumed to be half normal rates. The energy inflation rate was assumed to be 3%. Operation and maintenance costs were not included for the sake of simplicity, system availability was assumed to be 100%, and a HWS was assumed to have no residual value at the end of its lifetime.

The example in table 3-2 shows that the true cost of thermal energy (i.e. the hot water delivered to the customer) is significantly more than the cost of the primary energy source. As well as the calculated annual plant cost, the cost of thermal losses, and gas boiler inefficiencies significantly contribute to the overall total cost of the hot water. To develop a ratio between electrical and thermal energy cost, it is reasonable to assume the lowest cost option for the thermal energy (option b in table 3-2). Dividing the average standard tariff for electricity of 10.9 c/kWh (Australian Gas Association, 2000) by this value yields an electrical-to-thermal value ratio of 1.33. Although this analysis is specific to Australia in 2001, it is expected that the ratio is likely to remain fairly constant unless there is a major difference in the basic electricity / gas value ratio.

### 3.3.2 Renewable Energy Market Approach

The market that a PV/Thermal system operates in is not an open market. Most countries have policy measures designed to promote the use of renewable energy. Some of the more common policy instruments are buy back schemes, capital subsidies, tax exemptions, competitive bidding procedures for a specified market share, and renewable energy targets (International Energy Agency, 1998). The impact of these schemes is the creation of a separate market for renewable energy. In some cases the market can be further subdivided by generation type (eg. solar, wind, biomass, etc) as certain renewable energy policies are selective about what generation types are eligible.

It is possible to develop levelised energy costs for both electrical and thermal energy from renewable sources using the discounted cash flow method. This has been done by examining
grid connected flat plate photovoltaic systems and flat plate domestic hot water systems, which represent the alternative to a combined PV/Thermal system. Due to the wider availability of financial data for the US market, US financial data has been used in both cases, but in conjunction with Australian (Canberra) weather data.

### 3.3.2.1 Grid-connected photovoltaics levelised energy cost

PV modules prices offered online by six U.S. suppliers were surveyed in April 2001, with the results shown in Figure 3-1.

![Figure 3-1: Photovoltaic module prices surveyed in the U.S. April 2001.](image)

Table 3-3 shows a levelised cost calculation for a grid connected PV system using the mean price of modules larger than 40 W of $5.36/Wp. The levelised cost is calculated only once, as all parameters except module cost are assumed to be the same for each system.

Balance-of-system costs, cell and inverter efficiency were taken from year 2000 estimates by the U.S. Department of Energy (DeMeo and Galdo, 1997). The DOE also suggest an additional efficiency factor of 0.9 to account from operation away from standard rating conditions. PV module efficiency was estimated for crystalline silicon cells. However, the LEC is not sensitive to this figure. Canberra weather data for a typical year was used, assuming modules face north and are tilted at the latitude angle of 35.3°. It is assumed that the cost of energy will increase each year by 3%. The nominal savings for each year were estimated by inflating the base savings by this value. The 3% inflation value is an estimate
based on projected electricity price increases in the United States (DeMeo and Galdo, 1997). The system lifetime of 20 years is commonly used as a realistic value for PV. Other financial assumptions are the same as for the previous example. A levelised energy cost of US $0.367 is calculated for the PV system.

**Table 3-3. Levelised energy cost calculations for a grid-connected residential photovoltaic system**

<table>
<thead>
<tr>
<th>Item</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specifications</td>
<td></td>
<td></td>
</tr>
<tr>
<td>d.c. PV module efficiency</td>
<td>%</td>
<td>14.0%</td>
</tr>
<tr>
<td>Inverter efficiency</td>
<td>%</td>
<td>90%</td>
</tr>
<tr>
<td>Factor accounting for non-standard rating conditions</td>
<td>%</td>
<td>90%</td>
</tr>
<tr>
<td>a.c. System efficiency</td>
<td>%</td>
<td>11.3%</td>
</tr>
<tr>
<td>Performance</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Yearly insolation</td>
<td>kWh/m²</td>
<td>1990</td>
</tr>
<tr>
<td>Yearly ac electricity produced</td>
<td>kWh/m²</td>
<td>226</td>
</tr>
<tr>
<td>Cost</td>
<td></td>
<td></td>
</tr>
<tr>
<td>dc power PV module cost</td>
<td>USD/Wp</td>
<td>$5.36</td>
</tr>
<tr>
<td>Power-related BOS</td>
<td>USD/Wp</td>
<td>$1.22</td>
</tr>
<tr>
<td>Area-related BOS</td>
<td>USD/m²</td>
<td>$138</td>
</tr>
<tr>
<td>System Total</td>
<td>USD/m²</td>
<td>$1059</td>
</tr>
<tr>
<td>LEC calculation</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Energy inflation</td>
<td>%</td>
<td>3.0%</td>
</tr>
<tr>
<td>System Lifetime</td>
<td>Years</td>
<td>20</td>
</tr>
<tr>
<td>Discount Rate</td>
<td>%</td>
<td>7.82%</td>
</tr>
<tr>
<td>Levelised Energy Cost</td>
<td>USD/kWh</td>
<td>$0.367</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Year</th>
<th>Cashflow (USD)</th>
<th>Sum of Present Value (USD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>-1059</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>85</td>
<td>-980</td>
</tr>
<tr>
<td>2</td>
<td>88</td>
<td>-905</td>
</tr>
<tr>
<td>3</td>
<td>90</td>
<td>-833</td>
</tr>
<tr>
<td>4</td>
<td>93</td>
<td>-764</td>
</tr>
<tr>
<td>5</td>
<td>96</td>
<td>-698</td>
</tr>
<tr>
<td>6</td>
<td>99</td>
<td>-635</td>
</tr>
<tr>
<td>7</td>
<td>102</td>
<td>-575</td>
</tr>
<tr>
<td>8</td>
<td>105</td>
<td>-518</td>
</tr>
<tr>
<td>9</td>
<td>108</td>
<td>-463</td>
</tr>
<tr>
<td>10</td>
<td>111</td>
<td>-411</td>
</tr>
<tr>
<td>11</td>
<td>114</td>
<td>-361</td>
</tr>
<tr>
<td>12</td>
<td>118</td>
<td>-313</td>
</tr>
<tr>
<td>13</td>
<td>121</td>
<td>-267</td>
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<tr>
<td>14</td>
<td>125</td>
<td>-224</td>
</tr>
<tr>
<td>15</td>
<td>129</td>
<td>-182</td>
</tr>
<tr>
<td>16</td>
<td>133</td>
<td>-142</td>
</tr>
<tr>
<td>17</td>
<td>137</td>
<td>-104</td>
</tr>
<tr>
<td>18</td>
<td>141</td>
<td>-68</td>
</tr>
<tr>
<td>19</td>
<td>145</td>
<td>-33</td>
</tr>
<tr>
<td>20</td>
<td>149</td>
<td>0</td>
</tr>
</tbody>
</table>

**3.3.2.2 Solar hot water levelised energy cost**

Solar HWS prices are full system prices, and include collectors, tanks, fittings and a provision for installation. Thus, in contrast to the PV case, LEC must be calculated for each individual system, as each has different parameters such as collector area, efficiency, tank size, etc. Table 3-4 shows a specific example of a levelised energy cost calculation for a solar hot water heater.
Table 3-4. Example of levelised energy cost calculation for a solar hot water system

<table>
<thead>
<tr>
<th>Item</th>
<th>Unit</th>
<th>Value</th>
<th>Year</th>
<th>Cashflow (USD)</th>
<th>Sum of Present Value (USD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specifications</td>
<td></td>
<td></td>
<td>0</td>
<td>-1714</td>
<td></td>
</tr>
<tr>
<td>Collector area</td>
<td>m(^2)</td>
<td>3.47</td>
<td>1</td>
<td>138</td>
<td>-1587</td>
</tr>
<tr>
<td>Volume of tank</td>
<td>L</td>
<td>303</td>
<td>2</td>
<td>142</td>
<td>-1464</td>
</tr>
<tr>
<td>Volume of equiv. conventional HWS</td>
<td>L</td>
<td>156</td>
<td>3</td>
<td>146</td>
<td>-1348</td>
</tr>
<tr>
<td>Yearly insolation</td>
<td>kWh/m(^2)</td>
<td>1990</td>
<td>4</td>
<td>151</td>
<td>-1236</td>
</tr>
<tr>
<td>Performance</td>
<td></td>
<td></td>
<td>5</td>
<td>155</td>
<td>-1130</td>
</tr>
<tr>
<td>Hot water energy demand</td>
<td>kWh</td>
<td>3517</td>
<td>6</td>
<td>160</td>
<td>-1028</td>
</tr>
<tr>
<td>Tank auxiliary energy used</td>
<td>kWh</td>
<td>1660</td>
<td>7</td>
<td>165</td>
<td>-931</td>
</tr>
<tr>
<td>Solar energy used</td>
<td>kWh</td>
<td>1857</td>
<td>8</td>
<td>170</td>
<td>-838</td>
</tr>
<tr>
<td>Cost</td>
<td></td>
<td></td>
<td>9</td>
<td>175</td>
<td>-749</td>
</tr>
<tr>
<td>Cost of solar HWS including</td>
<td>USD</td>
<td>$2240</td>
<td>10</td>
<td>180</td>
<td>-665</td>
</tr>
<tr>
<td>installation</td>
<td></td>
<td></td>
<td>11</td>
<td>185</td>
<td>-584</td>
</tr>
<tr>
<td>Cost of equivalent conventional</td>
<td>USD</td>
<td>$530</td>
<td>12</td>
<td>191</td>
<td>-506</td>
</tr>
<tr>
<td>HWS</td>
<td></td>
<td></td>
<td>13</td>
<td>197</td>
<td>-432</td>
</tr>
<tr>
<td>Cost of solar component of solar</td>
<td>USD</td>
<td>$1710</td>
<td>14</td>
<td>202</td>
<td>-362</td>
</tr>
<tr>
<td>HWS</td>
<td></td>
<td></td>
<td>15</td>
<td>209</td>
<td>-294</td>
</tr>
<tr>
<td>LEC Calculation</td>
<td></td>
<td></td>
<td>16</td>
<td>215</td>
<td>-230</td>
</tr>
<tr>
<td>Energy Inflation</td>
<td>%</td>
<td>3.0%</td>
<td>17</td>
<td>221</td>
<td>-169</td>
</tr>
<tr>
<td>System lifetime</td>
<td>Years</td>
<td>20</td>
<td>18</td>
<td>228</td>
<td>-110</td>
</tr>
<tr>
<td>Discount Rate</td>
<td>%</td>
<td>7.82%</td>
<td>19</td>
<td>235</td>
<td>-54</td>
</tr>
<tr>
<td>Levelised energy cost</td>
<td>USD/kWh</td>
<td>$0.072</td>
<td>20</td>
<td>242</td>
<td>0</td>
</tr>
</tbody>
</table>

Solar collectors surveyed came from four major U.S. manufacturers (SunEarth, Heliodyne, ACR Solar, and AET) with prices obtained from five retailers. Data is from the first half of the year 2001. System performance was simulated using TRNSYS. Collector performance parameters came from an independent directory compiled by the Solar Rating and Certification Corporation (2000) in the U.S. Mass flow was assumed to be 10ml/s for each 1 m\(^2\) of collector area. A hot water demand profile was used, based on an energy draw profile set out in the Australian Standards AS4234-1994. The magnitude of the demand was adjusted depending on the size of the tank. Canberra weather data was as used, as above for the PV systems. The TRNSYS model used simple differential temperature control, a Type 1 collector, a Type 140 tank with 10 thermal zones, a thermostat set to 60°C, and a 3.6 kW auxiliary heating element, in the second and third zone from the top of the tank respectively. The over-temperature cutoff for the controller was set to 95°C. The base UA-value was 2.27 W.K\(^{-1}\) for a 300L tank, and this was adjusted according to tank size. The price of a conventional HWS capable of meeting the same demand was deducted from each solar HWS
such that net capital cost is a ‘solar only’ cost. The conventional HWS has a volume based on 0.75 times the daily hot water demand, as recommended by the European Committee for Standardization (1997). The price of the conventional HW system was calculated using normalized Y2001 price data from a range of systems produced by the two large U.S. manufacturers, Marathon and Bradford White. The lifetime of 20 years for the solar HW systems was based on advice from the SRCC, assuming specific components with shorter expected service life such as pumps, controls, and mixing valves are replaced as necessary during the system life. However, such maintenance costs are excluded for the sake of simplicity, and other financial assumptions are as previously.

Figure 3-2 shows the levelised energy cost calculated for each of the solar hot water systems examined, using the same method as outlined in table 3-4. The mean value of levelised energy cost obtained was USD $0.087/kWh.

![Graph showing levelised energy cost vs collector output](image)

**Figure 3-2. Levelised energy cost of hot water from a variety of solar hot water systems.**

The levelised cost calculations above give a ratio between electrical to thermal value from renewable sources of 4.24. This ratio of 4.24 favours electrical energy far more than the ratio of 1.33 developed in section 3.3.1. An optimisation of a PV/Thermal system using this ratio would therefore treat the electricity from the system as 4.24 times more valuable than the hot water output. Note that when ‘low end’ prices for both the PV system (USD $0.325/kWh) and the SHWS (USD $0.075/kWh) are used, this ratio does not change significantly, and has a value of 4.33 for the data surveyed.
3.4 Environmental valuation

One of the main drivers of renewable energy is its use as a means of reducing greenhouse gas (GHG) emissions associated with energy generation from fossil fuels. It is conceivable that the reduction of greenhouse gas emissions may become linked to strong enough financial incentives, such as carbon taxes and carbon credits, that their reduction will become a key design criterion. In this case, optimisation of a PV/Thermal system on a greenhouse gas savings basis would be important, and a ratio between the thermal and electrical output would need to be used based on greenhouse gas emissions.

3.4.1 Avoided emissions

The simplest way to calculate such a ratio is to look at emissions avoided by the use of the PV/Thermal system.

\[
\frac{\text{Ratio of electrical to thermal value}}{\text{Avoided CO}_2 \text{ by PV electricity generated}} = \frac{\text{Avoided CO}_2 \text{ by solar hot water generated}}{(3-5)}
\]

Assuming that emissions due to the PV electricity and the solar hot water are zero, this ratio simply becomes:

\[
\frac{\text{Ratio of electrical to thermal value}}{\text{Emissions due to conventional electricity}} = \frac{\text{Emissions due to conventional HWS}}{(3-6)}
\]

It is assumed that a new solar HWS replaces a gas HWS rather than an electric HWS. Natural gas in Australia has an average carbon dioxide emission factor of 183.2 g-CO\(_2\) per kWh of gas (Australian Gas Association, 2001). A hot water system with a high efficiency burner will have conversion efficiency around 85% (Australian Standard AS4234-1994), resulting in carbon dioxide emission levels of 215.6 g-CO\(_2\) per kWh of hot water. Emissions due to electricity vary according to the fuel source used for generation, and vary substantially between different regions. For example, in Australia in 1999, emissions were 1467 g-CO\(_2\)/kWh in Victoria, due to the extensive use of brown coal; 701 g-CO\(_2\)/kWh in the Northern Territory, which uses mainly natural gas; and only 1 g-CO\(_2\)/kWh in Tasmania due to almost total use of hydro power (George Wilkenfeld and Associates, 2000). On average, emissions were 1034 g-CO\(_2\)/kWh for electricity delivered. Using this average value, the ratio of electrical to thermal avoided CO\(_2\) emissions is 4.8.
### 3.4.2 Life cycle emissions

Another way to make a comparison on an environmental basis between electrical and thermal output from PV/Thermal collector is to consider the life cycle GHG emissions. Life cycle GHG emissions include the emissions that originate from processes used to make a product, as well as embodied emissions from the materials from which the product is manufactured. Using this methodology, emissions attributed to renewable energy are not zero. Instead, emissions for thermal energy in the form of hot water take into account the life cycle emissions of the solar hot water system, and include the embodied emissions of the tank, collector and other components, and the emissions produced in its manufacture. Similarly, life cycle emissions for a domestic PV system include emissions from all materials and fabrication steps (eg. MG-Si production, wafer production, cell manufacture, and panel fabrication), plus emissions due to the BOS components.

Following on from equation 3-5, the following expression will give the ratio of electrical to thermal CO\(_2\) emissions based on a life cycle analysis:

\[
\begin{bmatrix}
\text{Ratio of electrical to thermal value} \\
\end{bmatrix} =
\begin{bmatrix}
\text{life cycle emissions of conventional electricity} \\
\text{life cycle emissions of conventional hot water}
\end{bmatrix}
\begin{bmatrix}
\text{life cycle emissions of PV electricity} \\
\text{life cycle emissions of solar hot water}
\end{bmatrix}
\]  

(3-7)

Using life-cycle GHG emission intensities for conventional electricity generation methods (Dones and Frischknecht, 1998), the average life-cycle greenhouse gas emission in Australia was calculated as 1042 g-CO\(_2\)/kWh based on the fuel mix for electricity generation in 2001 (Electricity Supply Association of Australia, 2001). Embodied energy requirements for manufacturing a 3 kWp rooftop PV installation were calculated by Dones and Frischknecht for Swiss conditions. Their results were adjusted for Australian emissions intensity and Canberra insolation to give lifetime embodied energy values of 118 g-CO\(_2\)/kW and 201 g-CO\(_2\)/kWh for monocrystalline and polycrystalline PV respectively.

Life-cycle emissions for solar and conventional hot water systems in Australia were included in a study by Crawford (2000), which uses a hybrid process/input-output methodology. Using embodied emissions results from Crawford, life-cycle emissions were derived as 119 g-CO\(_2\)/kWh and 230 g-CO\(_2\)/kWh for solar and conventional hot water heaters respectively. These results are substituted into equation 3-7 to give a ratio of 8.32 or 7.58 when mSi or pSi cells are used respectively.
The ratio is noticeably higher than when embodied emissions are not included, and favours renewable electricity generation to a greater extent. The reason is that the embodied emissions associated with a solar hot water system are high relative to the alternative, a conventional gas hot water system.

3.5 Comparison of methods

As figure 3-3 demonstrates, the range of energy value ratios that could be used varies widely, from a lower value of 1, which could be used if the energy of the electricity and hot water was considered equally useful, to an upper value of 17, which could be used if the hot water output was to be used to produce electrical or mechanical work. In a commercial context, issues of cost cannot be ignored and therefore it is suggested that the most suitable method to provide a realistic energy value ratio for a PV/Thermal system is the renewable energy market method, which, for the case study described, gave an energy value ratio of 4.2. However this method is also most subject to variability, due to continued improvements in manufacturing technology leading to rapid price reductions, particularly for PV modules. Energy value ratios based on open market energy cost would make sense if there were absolutely no subsidies available for renewable energy, and if an appropriate penalty for GHG emissions were incorporated into fossil fuel prices. Energy value ratios based on displaced GHG emissions favour electricity even more than the other methods, due mainly to Australia’s GHG intensive electricity production. These ratios will vary substantially according to the fuel mix for electricity production in the particular location.

The conclusion is that there is no simple answer for determining what energy value ratio should be used, rather the ratio should be a parameter selected for the circumstance applicable to a particular installation. The methodologies presented are intended to show how to work out the energy value ratio and provide guidance about what methodology to use for a particular application.
3.6 Optimisation methodology

Establishing an electrical / thermal energy value ratio allows rational design optimization to be carried out. With such a ratio available, the dual outputs of a PV/T system can be considered as a single stream of Equivalent Electrical Energy. The concept of Equivalent Electrical Energy \( Q_{eq,\text{Elec}} \) is introduced to relate thermal energy \( Q_{th} \) to electrical energy \( Q_{\text{elec}} \) via a ratio:

\[
Q_{eq,\text{elec}} = \frac{Q_{th}}{[\text{Energy value ratio}]} + Q_{\text{elec}} \tag{3-8}
\]

The design goal then becomes the production of equivalent electrical energy for the minimum possible ‘equivalent electrical levelised energy cost’ (EELEC). This is a generalization of the approach known as ‘thermoeconomic optimisation’. For simple systems or parts of systems, the EELEC can be determined from the unit cost of energy input plus the annualized cost of the capital invested. If the dependence of overall efficiency on capital investment is known then a simple optimum design choice can be made.

For a complex system such as a combined PV/Thermal system that incorporates storage tanks and re-circulation of cooling water, major dynamic variations in performance are introduced. In this case simple thermoeconomic optimization concepts will assist with the creative design process, but a full simulation using a package like TRNSYS will be needed to optimize a design parameter quantitatively.

The field of thermoeconomic optimization has been taken a step further with the development of ‘exergoeconomic optimisation’. The long history of exergoeconomic analysis is outlined by Tsatsaronis (1996) and the method described comprehensively in Bejan et al. (1996). A major motivation for the extension of thermoeconomics into the exergy domain was primarily to deal with multiple energy output streams of differing value. The analysis of electrical / thermal energy value ratios presented here suggests that an exergy based analysis is simply one special case. If it is recognized that when multiple useful outputs are encountered, an appropriate ratio must be selected for the circumstances, then ‘thermoeconomics’ and exergoeconomics’ are seen to become examples of a single methodology.
The CHAPS system

The early stages of CHAPS system development at ANU focused on a domestic style collector, designed for house roofs. This collector consisted of two individual mirrors that tracked the sun on dual axes. A prototype of this system, called CHAPS 1 (top left in figure 4-1), was constructed and operational for a period of about a year. Due to concerns about marketability of the system and the effect of wind loading on domestic roofs, further development of CHAPS 1 style systems has ceased, and effort is now focused on long single-axis tracking CHAPS systems for large buildings. The long trough CHAPS prototype (top
right in figure 4-1) is composed of 10 individual mirrors. The first commercial installation of the CHAPS technology is the Bruce Hall CHAPS system (bottom right in figure 4-1). The 300 m² system will provide electricity and domestic and heating hot water for Bruce Hall, a residential college at the ANU. There are eight collectors, each 24.5 m long consisting of 17 individual mirrors. The Bruce Hall CHAPS system is currently under construction and is due for completion in late 2004. Experimental work on the radiation flux profile in this thesis was carried out on the Long Trough CHAPS prototype. Experimental work on the thermal and electrical performance of the CHAPS system was carried out on the CHAPS test rig (pictured bottom left in figure 4-1). The term ‘CHAPS receiver’ refers in this thesis to receivers common to the long trough, the test rig and the Bruce Hall CHAPS systems.

4.1 Solar cells

![Monocrystalline silicon solar cells produced at ANU; the wafer (left) and finished cell (right).](image)

Monocrystalline silicon solar cells are produced at the ANU for the CHAPS systems. Low series resistance is achieved in the ANU concentrator cells by: a) narrow spacing of the conductive fingers (0.3mm compared to around 3 mm for one sun cells), which reduces the distance electrons travel through the silicon; b) heavy phosphorous doping beneath the fingers to reduce the contact resistance; and c) electroplated silver fingers that are thin but high, to increase the cross sectional area of the metal while minimising finger shading.

4.2 Mirrors

The parabolic mirrors were developed at the ANU and follow on from similar development of three-dimensional curved mirrors for dishes (Johnston et al., 2001). The reflective surface is a glass-on-metal laminate (GOML) composed of a 1 mm thick silver backed mirror,
laminated to a sheet metal substrate (figure 4-3). The mirror is held in its parabolic shape by stamped tab ribs at either end of the mirror. The mirrors are around 93.5% reflective, which compares well with other reflective surfaces such as anodised aluminium at 81% (Brogren et al., 2000). The glass surface is highly scratch resistant when compared to some plastic film concentrators. The mirrors have been subjected to a number of years of outdoor and accelerated lifetime testing without significant deterioration. The ribs are trimmed just above the stamped tabs so that they cause minimal shading, and fixed from below to receiver support brackets. The mirrors are butted together end-to-end to form the trough. The dimensions of the troughs for the various CHAPS systems are given in table 4-1.

![Figure 4-3. Schematic diagram of the GOML mirror troughs showing the stamped tab ribs.](image)

**Table 4-1. Dimensions of the CHAPS systems.**

<table>
<thead>
<tr>
<th>Name of system</th>
<th>Mirror dimensions</th>
<th>Number of mirrors</th>
</tr>
</thead>
<tbody>
<tr>
<td>Long trough CHAPS prototype</td>
<td>1.55 wide x 1.5 long</td>
<td>1 row of 10 mirrors</td>
</tr>
<tr>
<td>CHAPS test rig</td>
<td>1.25 wide x 1.5 long</td>
<td>1 mirror</td>
</tr>
<tr>
<td>Bruce Hall CHAPS system</td>
<td>1.55 wide x 1.42 long</td>
<td>8 rows of 17 mirrors</td>
</tr>
</tbody>
</table>

The optical accuracy of the system is influenced by the mirror accuracy, the torsional rigidity of the collector, and the tracking tolerance. The system has been designed to have close to 90% accuracy most of the time, and better than 85% optical accuracy all of the time. There is a detailed discussion of the mirror accuracy in section 6.4.3, and a discussion of the tracking accuracy and torsional rigidity in section 4.4.2.

While the geometrical concentration ratio is around 37x, the illumination flux intensity is far from even across the width of the solar cell. Under good sunlight conditions, the centre of the focal beam may have flux intensities upwards of 100 suns. The resulting uneven radiation
and temperature distribution affects the operation of the solar cells. This is discussed in some detail in section 6.3.

4.3 Receivers

4.3.1 Evolution of the design

The development of the CHAPS systems was preceded by PV trough technology development at the ANU since the mid-1990s, culminating in the commissioning of a 20kW two-axis tracking passively cooled PV trough array at Rockingham, Western Australia (Smeltink et al., 2000). The Rockingham receivers were passively cooled via fins attached to the back of the cell tray. The CHAPS 1 prototype used the same cell mounting tray design as for Rockingham, but a section of aluminium was screwed on to the back, with a heat transfer paste sandwiched between (figure 4-4).

The CHAPS 1 prototype was in operation for a period of about a year, after which time there was noticeable pitting of the aluminium, believed to be due to galvanic corrosion. Aluminium is a very active metal and copper a quite noble one. As both materials are in contact with the cooling fluid, allowing an ionic path, the aluminium corrodes at an accelerated rate. It was decided at this point to investigate direct copper-aluminium bonding of the receiver, so that copper pipework could be used throughout the system. In the absence of an electrolyte such as the water, it was hoped corrosion would not be an issue. The narrower fluid conduit was desirable in order to attain turbulent fluid flow and hence better heat transfer. A new extrusion was designed to allow mounting of a copper tube (figure 4-5).
Various options were considered for bonding the pipe, including soldering, adhesives and mechanical clamping. In each case, an intimate thermal contact was necessary. Inter-metallic bonding between copper and aluminium is known to be technically difficult. Results from soldering tests on a small section of receiver using a very active flux indicated that an intimate bond could be achieved between the copper and aluminium. However, there was no success scaling up to a full receiver. An aluminium metal filled epoxy called Duralco 132 was trialed. However, heat transfer experiments indicated significant degradation in the heat transfer rate when the test sample was subjected to a period of accelerated thermal cycling. The mismatch in thermal expansion coefficients of aluminium (24 x 10^{-6}/°C) and copper (17 x 10^{-6}/°C) means that the bond would experience cyclical stress every day the collector is in use. It was thought that the epoxy would be too brittle to withstand these conditions. Mechanical clamping was never trialed due to concerns about achieving intimate contact between the pipe and the extrusion for the full receiver length.

4.3.2 Summary of the current design

Figure 4-6. Cross-section of a CHAPS receiver aluminium extrusion.
The current fully extruded receiver design (figure 4-6) was made possible by the development of purpose-designed anti-corrosive additive for the heat transfer fluid, made by Atherton Chemicals, a chemical manufacturing company in Sydney. The additive consists of corrosion inhibitors sodium tolytriazole and sodium mercaptobenzotriazole (0.25% by volume) and works by inhibiting the formation of copper ions in the pipework. For locations where freezing of the water is of concern, the fluid also contains 30% propylene glycol. The heat transfer fluid is pumped through the extruded aluminium receiver spine to cool the cells and collect thermal energy. Internal fins have been incorporated in the fluid conduit to increase the heat transfer surface in order to minimise the operating temperature difference between the cells and the fluid.

![Figure 4-7. The optical system for a CHAPS receiver.](image)

The solar cells are bonded to an aluminium receiver with a thermally conductive, electrically insulating tape. They are connected in series and encapsulated with silicone and low-iron glass. Schottky bypass diodes are used to protect the cells against going into reverse bias in case of partial shading of the receiver. The back and sides of the receiver are insulated with 20 mm thick rockwool encased by an aluminium cover.

### 4.3.3 Receiver sub-components

Silicon and aluminium are very good thermal conductors. However, it is difficult to achieve a bond with good heat transfer while maintaining electrical isolation. A double-sided adhesive tape made by Chomerics has been used for the CHAPS receivers to date. The tape is not very conductive (thermal conductivity of 0.37 W.m⁻¹.K⁻¹) but is very thin (0.127 mm), and therefore was thought to have adequately low thermal impedance. However, as is discussed in section 5.6.6.3, modelling using Strand7 indicates that the Chomerics tape contributes
significantly to the large temperature difference between the heat transfer fluid and the solar cell at the middle of the cell.

The solar cells are encapsulated on the front surface with a layer of transparent silicone rubber (Wacker Silgel 612) and an outer layer of low-iron glass (PPG Starphire). Low iron (Fe$_2$O$_3$) content is important, as Fe$_2$O$_3$ will absorb the infrared portion of the solar spectrum. The silicone isolates the solar cells electrically, and protects them from potential weather damage. The optics of an air-glass-silicone cover are better than for an air-glass-air cover (as discussed in section 4.5.1), which is a typical arrangement for a solar thermal collector. While there is a gain in optical efficiency due to the silicone layer, the good thermal insulation afforded by an air gap is sacrificed, with a reduction in thermal impedance of at least an order of magnitude.

4.4 Sun tracking

Solar concentrators are required to track the sun to collect direct radiation. The earth moves about the sun in an elliptical orbit. The earth rotates each day about the polar axis, which is inclined at 23.45° from the normal to the plane of the earth’s orbit. Therefore, the position of the sun in the sky relative to a known point on earth can be approximated by a function of two variables, the day of the year and the time of the day. The position of the sun from a known point on earth is described by two variables, commonly called the azimuth $\gamma$ and the zenith $\theta_z$ angles, as shown in figure 4-8. Azimuth angle describes the angle between the direction to the equator (north or south) and the projection of the sun’s position on the horizontal plane. The zenith angle describes the angle between the sun’s position and vertical.

![Figure 4-8. Reference system for description of sun position (southern hemisphere).](image)

The majority of tracking concentrator systems are mechanically actuated and controlled by a microprocessor. A range of algorithms has been developed to calculate the position of the sun, as reviewed by Blanco-Muriel et al. (2001). It is generally accepted that the PSA algorithm, which is given by Blanco-Muriel et al. is the most accurate algorithm currently
available. For the period 1999 to 2015, the PSA algorithm enables the direction of the solar vector to be determined with an error of less than 0.5 minutes of arc.

4.4.1 Single-axis versus two-axis tracking

Two-axis tracking allows the position of the sun to be closely followed, and therefore direct radiation will be incident perpendicular to the plane of the aperture of the collector. This tracking technique is necessary if the collector concentrates light in three dimensions, such as for a parabolic dish, or if the collector is sensitive to variation in angle, as is the case for a refractive system using a Fresnel lens.

Single-axis tracking allows the sun to be followed in one plane only. This is the minimum tracking requirement for a collector that concentrates light in two dimensions, such as a parabolic trough collector. The exceptions are low-concentration systems such as compound parabolic collectors and plane reflectors. Due to the varying angle of incidence of sunlight along the axis of a single-axis tracking collector, the area of direct light “seen” by the collector is reduced when compared to a two-axis tracking collector (figure 4-9 on the left). This effect is known as cosine loss, where the incident flux intensity $\hat{G}$ is given by the relationship:

$$\hat{G} = \hat{G}_{direct} \cos(\theta) \quad (4-1)$$

where $\theta$ is the angle of incidence and $\hat{G}_{direct}$ is the direct radiation flux intensity. Single-axis tracking collectors may also have end losses, due to light that misses the end of the collector because of the angle of incidence (figure 4-9 on the right).

![Figure 4-9. Cosine losses due to a reduced area of light flux ‘seen’ by collector (left) and end losses due to light missing the receiver (right).](image-url)
Tracking concentrator systems are only justified in regions where there is a reasonable amount of direct radiation. Much of Australia is well suited for concentrator applications. As shown in figure 4-10, central Australia averages above 9 hours of sunshine a day – exceptional conditions for concentrators. Much of the eastern seaboard, where the majority of the population is, averages more than 6 hours of sunshine a day. This compares to, say, Paris with 5.1 hours, Hong Kong with 5.3 hours and Bombay with 7.4 hours of sunshine a day.

![Figure 4-10. Average daily sunshine hours (annual) provided by the Bureau of Meteorology.](image)

Canberra receives an average 1780 kWh/m² per year total radiation (direct and diffuse) on a horizontal plane, 1300 kWh/m² of which is direct radiation (based on data from Morrison and Litvak (1988)). A fixed collector surface inclined at the latitude angle of 35.3° in Canberra receives a total of 1990 kWh/m², as shown below in figure 4-11. In comparison, a two-axis tracking concentrator receives 2100 kWh/m² per year direct radiation. For a single-axis tracking concentrator, this reduces to 1860 kWh/m² per year due to cosine losses, and if end losses are included for a 15 m long CHAPS trough, there is a further 2.5% reduction.
A single-axis tracking system with the long central axis oriented north-south is compared to one with an axis oriented east-west in figure 4-11. While a north-south oriented system is optimal for a full year, during the winter months an east-west oriented system actually performs better. Figure 4-12 shows incident radiation for sunny summer and winter days in Canberra for the two orientations. The cosine losses in the middle of the day for a north-south oriented system mean that an east-west oriented system performs about 20% better in winter, which for a thermal system is usually the period of highest hot water demand.

Figure 4-11. Comparison of tracking techniques using Canberra weather data. All data is direct radiation except where indicated. The impact of end losses for a 15m long trough is also shown.

Figure 4-12. Comparison direct radiation incident on north-south and east-west oriented single-axis tracking collectors in summer (left) and winter (right).
The CHAPS 1 prototype system was a two-axis tracking system based on a ‘daisy wheel’ arrangement, shown in figure 4-13. The azimuth tracking was achieved by rotation on a ring mounted flush to a roof. The second axis of rotation was along the edge of the mirror, allowing each trough to roll to the correct zenith angle. The long trough CHAPS prototype and Bruce Hall CHAPS systems track the sun on a single axis, with the roll axis located just below the centre of the mirror. Figure 4-13 shows a view from underneath the trough of the tracking mechanism. A linear actuator causes rotation of the main beam through cables and a pulley.

Figure 4-13. Tracking arrangements for the CHAPS 1 (left) and long trough CHAPS prototypes seen from below (right).

4.4.2 Tracking tolerance

Ideally, tracking of a concentrating collector should be continuous to achieve optimum optical performance. However, the actuator motor would require very high gearing and a more complex motor drive, and therefore most systems incorporate periodic movement of the collector. The frequency and precision of the movement determines the accuracy of the tracking. Figure 4-14 shows the number of stops and starts the actuator on the CHAPS prototype system would have to do over a 20 year lifetime for a range of tracking tolerances. A lower tracking tolerance causes fewer stops and starts, and it is likely to cause less wear on components such as motor brushes. Stepper motors have been considered as an alternative to the current dc motors. The advantage of a stepper motor is that there is no need for position encoders for feedback in order to move the motor a precise amount.
To determine how accurate the tracking needs to be, it is helpful to look at the percentage capture of light on a 38 mm wide target (equivalent to the width of an ANU solar cell). Figure 4-15 shows how the percentage captured changes when the target is moved lateral to the focal beam. The peak of the curve corresponds to the position where the centre of the target is located at the centre of the focal beam. The data shows the average capture from the combination of all 10 mirrors on the CHAPS long trough. The incidence angle (i.e. the angle the sun makes with vertical along the longitudinal axis of the trough) was 55°, and at this angle the peak capture is 90.3%. At lower incidence angles the percentage capture is marginally higher, peaking around 95%, and at higher incidence angles, there is very little energy available (6.6% of annual radiation for the orientation of the prototype system). Based on this measured data, it was decided that 85% capture is a realistic target. From figure 4-15 it can be seen that the focal beam must be kept within an 18 mm band to achieve this target.

A 38 mm wide solar cell subtends an angle of 1.81° on the long trough CHAPS prototype, from the pivot point located underneath the mirror. The 18 mm target zone subtends an angle of 0.86°. Therefore the sum of the angular displacement in the collector and the tracking tolerance should not exceed 0.86°. It was decided that the trough would be designed to have no greater than 0.5° angular displacement from the actuator to the furthest end (and therefore an average of 0.25° angular displacement). It was also decided that the tracking tolerance would be 0.35°. In midsummer, this means around 420 individual trough movements throughout the day, and over the 20 year life of the collectors, 2.8 million individual movements.
Figure 4-15. Percent capture of light on the solar cells as they are moved laterally through the focal beam. Data averaged from the long trough CHAPS prototype mirrors, with the light incident at 55° (figure from Glen Johnston).

4.5 Spectral dependency

To fully understand both the thermal and photovoltaic performance of the CHAPS system, it is necessary to understand the spectral dependency of the transparent media (the glass and silicone), the absorbing surface (the silicon solar cell) and the reflecting surface (the silver backed glass mirror).

4.5.1 Transmission through the cover

When the sun is directly normal to the aperture of the long trough CHAPS collector, light is reflected back to the receiver at angles from 2.7° to 48.1° relative to the surface normal of the cover glass.

Figure 4-16. Maximum and minimum angles of incidence for light incident normal to the aperture of the collector.
The reflectivity of light incident at an interface between two mediums (such as air to glass) is expressed using the Fresnel equation (see for example Duffie and Beckman (1974)):

$$\rho = \frac{G_\rho}{G_0} = \frac{1}{2} \left[ \frac{\sin^2(\theta_2 - \theta_1)}{\sin^2(\theta_2 + \theta_1)} + \frac{\tan^2(\theta_2 - \theta_1)}{\tan^2(\theta_2 + \theta_1)} \right]$$  \hspace{1cm} (4-2)

where $\theta_1$ and $\theta_2$ are the angles of incidence and refraction, as shown in figure 4-17.

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{figure4-17.png}
\caption{Reflection and refraction in media with refractive indices $n_1$ and $n_2$.}
\end{figure}

The angles $\theta_1$ and $\theta_2$ are related to the refractive indices of the materials $n_1$ and $n_2$ by Snell’s law:

$$\frac{n_1}{n_2} = \frac{\sin \theta_2}{\sin \theta_1}$$  \hspace{1cm} (4-3)

Using equations 4-2 and 4-3 it is possible to calculate the reflection at an air-glass interface for a range of angles. As shown in figure 4-18, for angles above about 40°, the reflection increases significantly.
Figure 4-18. Reflection at an air-glass interface for a range of incidence angle. The refractive index of glass $n_{\text{glass}}$ is 1.52.

The rim angle of the long trough CHAPS prototype is 48.1°. The initial reflection from the cover glass for light originating from this part of the trough is 5.74%, compared with 4.26% for light originating from near the centre of the trough. Figure 4-19 shows the initial air-glass reflection for light originating from different positions across the width of the trough. The average value is 4.53%, a little more than the minimum which occurs at normal incidence angles. If the rim angle was to be increased, for example for a wider trough, a shorter focal length or both, then a receiver with a flat glass plate would rapidly start to cause significant reflection losses. For the present design, the impact of the rim angle is small, and therefore all further analysis of reflection and transmission through the cover materials is based on normal incidence to simplify the experimental process.

Figure 4-19. Initial reflection from the air-glass interface on the receiver for light originating from different parts of the long trough CHAPS mirrors. The refractive index of glass $n_{\text{glass}}$ is 1.52.
A typical thermal flat plate collector has an air gap between the glass and the absorber to provide some thermal insulation. The total light transmission through such a cover, ignoring absorption effects, can be derived as follows:

\[
\tau_{\text{air-glass-air}} = \frac{1 - \rho_{\text{glass-air}}}{1 + \rho_{\text{glass-air}}} \quad (4-4)
\]

The CHAPS receiver has no air gap. Transmission through the air-glass-silicone interface can be derived thus:

\[
\tau_{\text{air-glass-silicone}} = \frac{(1 - \rho_{\text{glass-air}})(1 - \rho_{\text{glass-silicone}})}{(1 - \rho_{\text{glass-air}} \cdot \rho_{\text{glass-silicone}})} \quad (4-5)
\]

Substituting in refractive index values of 1.52, 1.40 and 1.00 for glass, silicone and air respectively, gives transmissivity of 95.6% for a glass-silicone cover compared to 91.8% for a glass-air cover. For the CHAPS receiver, the better optical performance without an air gap outweighs the gain in thermal performance, and also improves the electrical output. The calculated values give an estimate of the expected maximum transmission for the cover. However it ignores absorption and the fact that refractive index is also wavelength dependent. It is therefore important to experimentally investigate spectral dependencies.

4.5.2 Measurement of transmission, reflection and absorption

A Cary 5 spectrophotometer was used for spectral measurements of transmission, reflection and absorption for the various optical components of the CHAPS collector. The spectrophotometer consists of a 110 mm diameter integrating sphere that collects most reflected or transmitted radiation, removes directional preferences, and presents an integrated signal to a detector. The measurement is compared to 0% and 100% baseline measurements to determine the percentage reflected or transmitted. The spectrophotometer was used in the spectral range 300 – 2500 nm.

4.5.3 Results of spectrophotometer measurements

The physical geometry of the optics of a receiver was shown previously in figure 4-7. The spectral dependence of the mirror, glass, silicone and silicon solar cell, has been measured and the results presented in figure 4-20. Average figures for reflectance, transmission and
absorptance are weighted by the wavelength dependence of the global irradiance at AM1.5. Therefore optical properties in the visible region (approx. 400-700 nm) are more heavily weighted than in the ultraviolet region or near infrared.

![Graphs of various optical components including mirror reflection, glass and silicone transmission, and solar cell absorption as a function of wavelength.](image)

*Figure 4-20. Spectrophotometer measurements of the key optical components of the CHAPS system.*

### 4.5.4 Spectral dependency of the mirror reflection

The mirror is made of low iron glass with a silver backing. As can be seen in the glass transmission curve in figure 4-20, glass absorbs light below about 400 nm, and therefore the
mirror also absorbs light in this spectral region. The mirror has a weighted average reflectivity of 93.5%. Reflectivity actually peaks at 96.5% at 1700 nm, but there is minimal solar radiation in this region.

### 4.5.5 Spectral dependency of the glass transmission

The glass is quite transparent above 400 nm, with a weighted average transmissivity around 90% for the thickness (2 mm) used on the receivers. However this figure includes reflection losses at both the front and back air-glass interfaces, as well as absorption. Using equation 4-4 and the assumption of a constant refractive index of 1.52, the transmissivity could be expected to be 91.8. The reason that the actual transmissivity is lower is due to a combination of absorption, and extra reflection due to the fact that refractive index is wavelength dependent. Determining exactly how much absorption there is in the glass (and silicone) is important for the CHAPS collector, as there is a very high radiation flux at the centre of the receiver. The heat flux due to absorption in the cover materials at the centre of the focal beam is significant, and contributes to raising the temperature and hence lowering the efficiency of the solar cells. Excessive absorption could cause physical damage such as delamination. In order to determine the amount of absorption there is in the glass, transmission for various thicknesses of glass was measured, shown in figure 4-21. Weighted averages were calculated based on the global irradiance at AM1.5, and the results for the various thicknesses are shown in figure 4-22.

![Figure 4-21. Transmission measurements for glass of various thicknesses.](image-url)
Figure 4-22. Weighted average transmission for various glass thicknesses.

Extrapolation of this data with an exponential curve indicates the weighted average transmissivity when there is zero absorption is around 91.6, a value that incorporates the variation in refractive index over different wavelengths. An average reflectivity $\rho_{\text{glass,air}}$ of 4.4% is calculated based on equation 4-4. To quantify the absorption in the glass, a modified version of equation 4-4 that includes absorption $\alpha$ is used:

$$
\tau_{\text{air-glass-air}} = \frac{(1 - \rho_{\text{glass,air}})^2(1 - \alpha)}{1 - (1 - \alpha)^2 \rho_{\text{glass,air}}^2}
$$

Bouger’s law describes absorption of radiation in a partially transparent medium, assuming absorbed radiation is proportional to the local radiation intensity $I$ in the medium and the distance the radiation travels in the medium $x$:

$$
dI = IK \, dx
$$

where $K$ is the extinction coefficient, and is assumed to be constant in the solar spectrum. Integrating equation 4-7 between thickness limits of 0 and $\delta$ gives:
\[
\frac{G_\delta}{G_0} = e^{-K\delta} = 1 - \alpha
\]  

(4-8)

Therefore, by measuring the relationship between the glass sample thickness and the absorption, the value of \( K \) is determined. For the Starphire glass, this value is 0.0046/mm. It is therefore possible to calculate both the reflection and absorption for any thickness of glass. The absorption integrated over the AM1.5 spectrum that could be expected in the 3.3 mm thick Starphire glass is 1.5%.

**4.5.6 Spectral dependency of the silicone transmission**

The transmissivity of silicone is very good between about 400 – 1100 nm. At higher wavelengths, distinct absorption bands can be seen. These bands correspond to excitation frequencies for the covalent bonds within the silicone compound. To determine the weighted average absorption of the silicone, transmissivity measurements were carried out on silicone samples of various thicknesses, shown in figure 4-23. Weighted averages were calculated based on the global irradiance at AM1.5, and the results for twelve different thicknesses of silicone are shown in figure 4-24.

![Figure 4-23. Transmission measurements for silicone of various thicknesses.](image)
Figure 4.24. Weighted average transmission for silicone samples of varying thickness.

Using the same procedure described in section 4.5.5 for glass, the extinction coefficient for absorption was calculated to be 0.010/mm. Therefore, the absorption expected in the 2 mm layer of silicone in a CHAPS receiver is 2.0%.

4.5.7 Glass-silicone cover reflection

Measurements on the spectrophotometer of the transmission of glass and silicone include reflections from both front and back surfaces of the sample. Radiation incident upon an encapsulated solar cell suffers reflection losses from one glass-air interface, and one glass-silicone interface. Using the weighted transmissivity values determined above, it is possible to calculate weighted refractive indices for glass (1.53) and silicone (1.5). It is therefore possible to calculate weighted reflectivity for both the air-glass and glass-silicone interfaces. However, using weighted refractive indices to calculate the overall reflection from the cover is not strictly correct. For higher accuracy it is necessary to determine the refractive index for each wavelength, and hence calculate the reflectivity at each wavelength, before applying a weighting based on the solar spectrum. Based on approximate calculations of the change in refractive index of silicone and glass, the error in the calculated overall transmission is estimated to be around 0.2% if weighted averages are used. Determination of the final transmission through the cover depends on the absorption of the solar cells, as some of the light reflected from the cells will be captured by further reflection from the air-glass and
glass-silicone interfaces (approximately 0.5%). Ignoring this, and using weighted values for refractive indices, the transmission through the cover is calculated as 92.3%. The calculation is done numerically using the ray tracing software Opticad, in order to account for multiple internal reflections within the cover. With zero absorption in the glass and silicone the transmission would increase to 95.6%.

4.5.8 Spectral dependency of the solar cell absorption

As discussed in the background section 2.2 on silicon solar cells, silicon absorbs light strongly below about 1100 nm, as is apparent in figure 4-25.

![Figure 4-25. Reflection for a cell before and after encapsulation.](image)

Silicon has a relatively high refractive index, averaging around 3.6 for longer wavelengths, but rising strongly below about 500 nm to a peak of 6.5 at 380 nm. Assuming the refractive index is 3.6, the Fresnel equation 4-2 is used to calculate the reflectivity in air as 32%. The peak in reflectivity at short wavelengths apparent in figure 4-25 is due to the increase in refractive index. Figure 4-25 also shows that the reflectivity of a finished solar cell is significantly less than what would be expected from bare flat silicon. To improve the light trapping, cells are textured to allow the reflected light a second opportunity to enter the cell, as is illustrated in figure 4-26.
Figure 4-26. An illustration of the principal of trapping incoming light by surface texturing, and an image of a textured wafer from a scanning electron microscope.

Furthermore, the texturing of the wafer allows light to be retained within the cell through total internal reflection (TIR). The angle for TIR is given by Snell’s Law:

$$\theta_{TIR} = \sin^{-1} \left( \frac{n_{ext}}{n_{Si}} \right)$$  \hspace{1cm} (4-9)$$

where $n_{Si}$ and $n_{ext}$ are the refractive indices of silicon (3.6) and an external medium (1.0 for air). For a ray of light within the silicon to escape, it must strike the surface at an angle less than $\theta_{TIR}$ (16° for silicon and air) from perpendicular. At the back of the solar cell there is another oxide layer and then a layer of silver, so nearly all light that is not absorbed by the cell is reflected back through the wafer. Light returning to the top surface of the cell has a small chance of hitting the surface within the ‘escape cone’ of angle 16°, and therefore is likely to reflect back into the wafer, further increasing the probability of absorption. For wavelengths below 1100 nm, the primary absorption mode is the creation of an electron-hole pair, which contributes to the photocurrent of the cell. For wavelengths above 1100 nm, the primary absorption modes are by absorption in the rear metal reflector and by free carrier absorption (absorption by the free holes and electrons in the silicon), which does not contributed to the cell photocurrent. Although these absorption modes are weak, light will pass multiple times through the wafer and therefore the probability of absorption is increased. So the overall absorption in this region remains as high as 70%.

Generally the surface of the silicon has an oxide layer, which is required for a number of reasons, most importantly surface passivation, and protection of the silicon. The silicon
dioxide layer has a refractive index of about 1.46 (although this too is variable with wavelength). However, because the refractive index of the oxide is similar to that of the silicone potant, the oxide has negligible effect on light absorption. The cells do not have an antireflection coating optimised for encapsulation. The reason for this is that the antireflection coating (e.g. of silicon nitride) would reduce the reflection losses by only 2-3%, but would substantially add to cell fabrication complexity and cost.

Silver contact fingers also contribute to the reflection from the front surface. The coverage of the figures on the ANU concentrator cells is around 11%. However, the fingers are rounded and therefore much of the light that hits the fingers is reflected back at an angle and re-enters the cell due to the total internal reflection from the air-glass surface. The ‘effective’ coverage of the fingers can be calculated to be around 4.5%. Figure 4-25 shows the reflectivity of a cell both before and after encapsulation. The weighted reflectivity of the cell is reduced from 15.9% to 11.4% when it is encapsulated. The complement of this weighted reflectivity is what is known as the transmission-absorption product (Duffie and Beckman, 1974), and this is used later in the thermal analysis in chapters 5 and 6.
Thermal Performance

Chapter 5

5.1 Experimental method

The thermal performance of a CHAPS receiver was characterised using a custom built outdoor testing unit (figure 5-1) known as the CHAPS test rig.

![Outdoor testing unit](image1)

*Figure 5-1. Outdoor testing unit.*

5.1.1 The mirror

The 1.25 m wide trough on the CHAPS test rig is narrower than the mirrors on the long trough CHAPS prototype. It is also a little longer than the receiver, to ensure that the radiation flux is as consistent as possible right to the ends of the receiver. The length of the mirror is 1.604 m, and the length of the cell string is 1.422 m. Figure 5-2 shows the flux profile on the cells, where position zero is at the edge of the first cell. The perturbations in the profile are within ±2.3% from the mean, which is quite good uniformity for a mirrored trough concentrator. This data was obtained by moving a single cell along the length of the receiver while monitoring cell current.
5.1.2 The receiver

The test receiver was made up of 28 cells connected in series, one of which was bypassed due to failure upon installation. The cells average 16.8% efficiency at 65 degrees under 30 suns flux intensity. The cells were chosen to have matched maximum power point currents.

5.1.3 Data logging equipment

5.1.3.1 Data Logger

A dataTaker DT600 was used for the data logging. Data was logged every 10 seconds, and a running average was stored each minute for most data points. The data logger has 15 bit resolution.

5.1.3.2 Direct beam radiation

Direct beam radiation was measured with a Kipp & Zonen CH1 normal incidence angle pyrheliometer, which is a first-class pyrheliometer under the ISO9060 standard. The instrument measures radiation in the spectral range 200 – 4000 nm. Keogh (2001) reviewed a number of papers discussing the accuracy of first-class pyrheliometers, and the consensus suggests an uncertainty of about 2%. The pyrheliometer was mounted to the end of the test rig trough, which was tracked manually to face the sun directly at all times.

5.1.3.3 Ambient temperature

Ambient temperature was measured with a platinum resistive device (PT100) in a probe mounted in a Stevenson’s screen, which shades the probe from direct sunlight and rain, but allows the circulation of air. The calibration of the PT100 sensor was fine tuned using ice
slurry (0.0°C) and boiling water (98.0°C)\(^2\). The uncertainty is estimated to be <0.1°C based on the calibration results.

### 5.1.3.4 Wind speed

The wind speed and direction was measured using an anemometer and wind vane manufactured by Davis Instruments. The equipment was mounted horizontally around 1.5 m from the ground adjacent to the collector. The wind cups give a pulsed output (3578 pulses/hour = 1 m/s wind speed) which is logged over a period of a minute to give an average value. The uncertainties of the wind speed and direction measurements given by the manufacturer are 5% and 7% respectively.

### 5.1.3.5 Inlet water temperature

The inlet temperature of the water was measured with a platinum resistive device (PT100) mounted in a probe inserted into the flow. The calibration technique for the probe and its accuracy were the same as for the ambient temperature probe. The inlet temperature was held reasonably constant by mixing hot and cold water from the nearby engineering building with a mechanical tempering valve. For inlet water temperatures higher than 55°C, a booster heater with PID control was used to regulate the inlet water temperature. The uncertainty is estimated to be <0.1°C based on the calibration results.

### 5.1.3.6 Temperature difference across the receiver

The temperature difference across the collector was measured using a differential thermocouple arrangement with type K thermocouples mounted in probes inserted directly into the flow at either end of the receiver. Calibration of the differential thermocouples was carried out by measuring the temperature difference between an ice slurry at 0.0°C and boiling water at 98.0°C. Additionally, observation was made of any difference in temperature when both probes were in the ice slurry or the boiling water. It is estimated that for the range of differential temperature measured, uncertainty is around 2%.

### 5.1.3.7 Volumetric flow

Volumetric flow was measured with a calibrated turbine flow meter from RS Components. It was found that linearity for flows greater than 25 ml/s was within ±1%, and within ±3% for

\(^2\) Note that the elevation of Canberra is around 600 m, hence the low boiling point for water.
flows down to about 15 ml/s. Lower flow rates could not be measured with suitable accuracy with the flow meters used. Tests were carried out to assess whether viscosity changes at different temperatures had an effect on accuracy, and the result was that no systematic error was found. Rough calibration of the flow was carried out during each test with a bucket and stopwatch to ensure there were no gross errors due to blockages in the meter.

5.1.3.8 Current and voltage

A resistive load with voltage regulation\(^3\) was held across the receiver at the maximum power point. The load was set at a defined voltage, and adjusted to the maximum power voltage by regular comparison with IV curves measured from the receiver. The voltage was measured across the receiver, and the current measured across a 1 m\(\Omega\) shunt resistor. The voltage signals were measured using the data logger. The measurement points for voltage were at either end of the receiver, close to the last cell, thereby avoiding any voltage drop in the high current cables. Uncertainty in the measurement of current and voltage is very small compared to uncertainty in the measurement of the radiation, and is therefore neglected. However, there is some uncertainty associated with the accuracy of the manual tracking, which has a direct effect on the electrical output. The position of the light beam never remains exactly in the middle of the cells, and therefore the amount of light spillage from the cells is always changing. The error due to this effect is estimated to be around 1%.

5.1.3.9 Temperature of the receiver body

Four type K thermocouples were mounted on the receiver in the positions shown diagrammatically in figure 5-3.

---

\(^3\) Thanks to Bruce Condon for designing and building the voltage controlled resistive load.
Thermocouples TK1 to TK3 were mounted between just behind the cell, and thermocouple TK4 was mounted on the back of the extrusion. The thermocouple terminations were mounted to an insulated isothermal block to maintain the same temperature, shown diagrammatically in figure 5-4. The reference temperature of the block was measured with a 4-wire PT100 probe.

![Figure 5-4. Isothermal block arrangement for thermocouples.](image)

The thermocouples were calibrated relative to the inlet temperature by heavily insulating the receiver, passing water through, and adjusting the offset to account for the minor temperature drop along the receiver. The uncertainty for thermocouple measurements using an isothermal block is around 0.3°C.

### 5.2 Measured efficiency

The thermal output $\dot{Q}_{th}$ and electrical output $\dot{Q}_{elec}$ of the CHAPS collector are calculated from the measured data as follows:

\begin{align}
\dot{Q}_{th} &= c_p \cdot \dot{m} \cdot (T_{out} - T_{in}) \\
\dot{Q}_{elec} &= I_{mp} \cdot V_{mp}
\end{align}

where $c_p$ is the specific heat at the average fluid temperature, $\dot{m}$ is the mass flow, and $T_{out}$ and $T_{in}$ are the outlet and inlet temperatures respectively of the fluid. $I_{mp}$ and $V_{mp}$ are the current and voltage of the receiver at the maximum power point. The thermal efficiency $\eta_{th}$ and electrical efficiency $\eta_{elec}$ are calculated based on the following definitions:
\[
\eta_{th} = \frac{\dot{Q}_{th}}{G_d \times A_m}
\] (5-12)

\[
\eta_{elec} = \frac{\dot{Q}_{elec}}{G_d \times A_m}
\] (5-13)

where \(\dot{Q}_d\) is the direct beam radiation and \(A_m\) is the product of the mirror width of 125 cm, and length of 150 cm. Note that the receiver aperture is actually only 143.5 cm long, so the efficiency measure includes losses due to hydraulic connections at the ends of the receiver, as well as losses due to the shading caused by the receiver itself.

Figure 5-5 shows the efficiency results for the receiver when it is operating both with and without an electrical load. \(T_f\) and \(T_{amb}\) are the mean fluid temperature and ambient temperature respectively. \(G_d\) is the direct beam radiation. The curves are fitted to the data using a least squares linear fit.

\[\text{Figure 5-5. Efficiency curves for the CHAPS receiver.}\]
5.2.1 Range of conditions

The average wind speed across all tests was 0.21 m/s – quite low because the tests were conducted in a sheltered courtyard. The peak wind speed was 1 m/s. The water flow rate varied between about 37.5-42.5 ml/s between different runs, but usually within a 1 ml/s range for a given run. While flow rate influences the heat transfer coefficient within the receiver, and hence the efficiency, calculations show that the corresponding uncertainty in efficiency for this range of flows is around 0.3%

Similarly, the inlet temperature was normally very stable for a given run, within range of 0.3°C. Direct beam radiation ranged between 829-928 W.m² and ambient temperature between 14.8-27.5°C.

5.2.2 Error

The uncertainty in the calculation of thermal output (equation 5-10) depends primarily on the uncertainty in the measurement of mass flow and temperature difference, and is calculated to be 2.2%. The specific heat of the fluid also depends on a measured quantity (absolute temperature of the fluid, with uncertainty ~0.15°C), but this is a higher order effect and can be ignored.

As discussed later in the chapter, the wind conditions have a reasonably significant effect on the thermal losses. The variation in wind speed was minimised during this experiment due to the sheltered location. However, maximum variation in the thermal output for the range of wind conditions experienced could be expected to be in the order of 40 W or around 4%. For runs at lower end of the range of temperatures, the variation in thermal output due to wind conditions is likely to be substantially lower.

Efficiency measurements include the uncertainty in the measurement of the direct beam radiation (equations 5-12 and 5-13 below). For the thermal efficiency plots, the measurement uncertainty combined with the variable wind conditions give uncertainty around 5.0%. The uncertainty in the electrical efficiency calculations is lower, around 2.2%.

5.2.3 Discussion

The electrical efficiency is plotted on the same horizontal axis in figure 5-5 in order to show the data points corresponding to the thermal efficiency data points. However, the electrical efficiency depends on absolute temperature rather than temperature difference. To illustrate
this point, the same data set plotted against the average fluid temperature shows a reasonably linear relationship (figure 5-6). The actual temperature of the solar cells has not been measured (or simulated) as part of this experiment. Addition of the electrical efficiency data to the thermal efficiency data (plotted in figure 5-5) yields a combined efficiency trend very similar to that of the thermal efficiency when there is no electrical load.

![Figure 5-6. Electrical efficiency for various fluid temperatures.](image)

To make a direct comparison between the efficiency curves of the CHAPS collector (figure 5-5) with those of a typical flat plate collector, it is necessary to make an assumption regarding the ratio of direct beam radiation $G_D$ to total global radiation $G_T$. Figure 5-7 shows a comparison between the CHAPS collector tested above (using the curve fit for thermal efficiency with concurrent electrical generation) and a Solahart Oyster Ko collector. There is some discrepancy between the efficiency coefficients based on aperture area from Swiss tests under the European standard EN 12975 (Solartechnik Prüfung Fororschung, 2002), and those from the Solar Rating and Certification Corporation (2000) in the US. Both results are shown. It is assumed for the sake of comparison that the direct radiation is 90% of the total radiation, which is a typical instantaneous value on a clear sunny day.
Figure 5-7. Thermal efficiency of the CHAPs collector using direct radiation (left) and a comparison with a flat plate collector using total (direct and diffuse) radiation (right).

The comparison in figure 5-7 shows that at lower operating temperatures, a flat plate collector has a higher efficiency than the CHAPs collector, but that as the operating temperature rises, the gap in performance is reduced. Clearly the thermal losses increase more rapidly for a flat plate collector due to the larger surface area. However the comparison is only valid when the two collectors are oriented directly towards the sun – a concentrating collector must track the sun, and therefore the efficiency does not suffer to the same extent as a flat plate collector due to reflection losses and cosine losses at higher incidence angles. To better understand how a CHAPS system performs in comparison to a flat plate collector, annual energy output should be calculated. This is the subject of a case study using TRNSYS in chapter 7.

5.3 Heat transfer between the receiver and the fluid

A measure of the convective heat transfer from the inside of a tube is given by the dimensionless Nusselt number $Nu$. The CHAPS receivers have been sized to operate with fully turbulent flow. The Nusselt number for turbulent flow is given by equation 2-8. However, due to the short length of the CHAPS test rig receiver relative to the full trough length, most flows measured are in the transition or laminar regions. The Nusselt number for laminar flow is given by equation 2-9.

To increase the heat transfer of a given length of a tube, there needs to be an increase in the product of the internal surface area $A$ and the heat transfer coefficient $h_c$, which is related to the Nusselt number as per equation 2-7. In typical industrial applications, various methods to
increase one or both are employed, particularly for heat-exchanger applications. Inserts such as twisted tape increase the convection coefficient by introducing a swirl in the flow. Roughening the inside surface of the tube is another way to increase the heat transfer coefficient for convection. To increase the surface area, internal fins can be incorporated. If the fins are helical, then they will also increase the mixing and augment the heat transfer coefficient. The trade-off for all heat transfer augmentation techniques is that there is an increase in the friction factor, and hence a higher pressure drop and correspondingly higher pumping power requirement.

The fluid conduit and solar cell mounting tray for the CHAPS receivers is made of extruded aluminium. Longitudinal fins have been incorporated to increase the surface area.

5.3.1 Determination of the rate of heat transfer in a receiver

Thermal resistance for convection $R_{conv}$ is used to express the convective heat transfer from a surface. It is defined as the inverse of the product of the wetted surface area $A$ and the convective heat transfer coefficient $h_c$. Thermal resistance was determined experimentally using the following relationship:

$$R_{conv} = \frac{T_i - T_f}{Q} = \frac{1}{h_c A}$$ (5-14)

where $T_i$ and $T_f$ are the average temperatures of the heat transfer tube surface and the bulk temperature of the fluid respectively, and $\dot{Q}$ is the rate of heat transfer to the fluid. The rate of heat transfer is not strictly linear along the receiver, due to various non-linear factors such as radiation losses and thermal entry length; however, it is assumed that the temperature difference $T_i - T_f$ near the centre of the tube is close to the average. Therefore, the tube temperature was estimated using the average of measured temperature from thermocouples TK2 and TK4 (which were mounted on the receiver as described in section 5.1.3.9), and the fluid temperature was linearly interpolated with position based on the known inlet and outlet temperatures. The rate of heat transfer was calculated from measured fluid temperature difference and flow rate using equation 5-10. Due to the relatively high uncertainty in the measurements from the thermocouples, and the magnitude of the temperature difference between the tube surface and the fluid temperature, the total error in thermal resistance is high, from 5.5% at the lowest flow rate to 15% at the high flow rate.
5.3.2 Results and correlations for internal fins

The thermal resistance and uncertainty for a range of flow rates is plotted in figure 5-8. As may be expected from the Dittus-Boelter (equation 2-8), the heat transfer is improved by increased flow. It should be noted that the flow rate for a 24 m long trough is likely to be around 250 ml/s. However, at such high flow rates the uncertainty in the measurement of the temperature change on the test rig is too high to obtain meaningful results.

![Figure 5-8. Measured and predicted thermal resistance for various flow rates.](image)

Determining a correlation for the calculation of the Nusselt number allows the thermal resistance to be predicted for any flow rate. Typically for tubes with internal fins, the hydraulic diameter $D_H$ is used for calculation of the Reynolds number (equation 2-5) and the heat transfer co-efficient $h$ (equation 2-7). The hydraulic diameter is 11.1 mm, which compares to the nominal diameter, without fins, of 23.5 mm. The Reynolds numbers that correspond to the data points in figure 5-8 range from 1073 to 6520.

Figure 5-8 shows the predicted thermal resistance for the various measured flow rates. The thermal resistance predicted when the hydraulic diameter is substituted into the Seidler and Tate equation for laminar flow (equation 2-9) substantially over-predicts the measured thermal resistance. A better fit for thermal resistance is given by the Dittus-Boelter equation (equation 2-8), also using hydraulic diameter. A reasonably good fit is observed with the measured data for flow rates up to 66 ml/s. This is perhaps a little surprising, as the use of the hydraulic diameter is known to over-predict the Nusselt number when there are internal fins.
(see, for example, Scott and Webb (1981) and Jensen and Vlakancic (1999)). Various empirically derived correction factors have been proposed to account for the effects of fins of different heights, widths, and helix angles. The most commonly correction factor $F_H$ is that proposed by Carnavos, which is as follows (as stated by Scott and Webb (1981)):

$$F_H = \frac{Nu}{Nu_{D_h}} = \left( \frac{A_{xs}}{A_n} \right)^{0.1} \left( \frac{P_n}{P} \right)^{0.5} \sec^3 \alpha_H$$

(5-15)

where $A_{xs}$ is the flow cross-sectional area, $P$ is the wetted perimeter, $A_n$ is the cross-sectional area if there were no fins, $P_n$ is the wetted perimeter if there were no fins, and $\alpha_H$ is the helix angle of the fins (which in this case is zero). For the particular fin geometry of the CHAPS receiver, the correction factor is calculated as 0.74.

The use of this correction factor would improve the correlation between the measured and calculated results only for the higher flow rates measured. Possibly this is because the Dittus-Boelter equation and the Carnavos correction factor are intended for use with fully turbulent flow. Unfortunately, while the flow would be fully turbulent for a full length trough, the flows measured on the test trough are either in the transition or laminar region. The Nusselt number calculated using a standard laminar expression for a plain wall tube with hydraulic diameter over-predicts thermal resistance. The determination of more accurate Nusselt numbers specific for laminar flow in an internally finned tube has been investigated both experimentally (Shome and Jensen, 1996a, Shome and Jensen, 1996b) and analytically (Campo and Chang, 1997) but is reasonably complex and specific to particular geometries, and therefore beyond the scope of this thesis.

Another area of uncertainty is the appropriateness of using hydraulic diameter as the characteristic length. Previous experimental work by Edwards et al. (1996) showed that the overall turbulence structure in a longitudinally finned tube was governed by two regions; the core and the interfin region. Jensen and Vlakancic (1999) developed their own length scale to account more explicitly for these two flow regions.

However, the most important result is that the trend in thermal resistance predicted by the Dittus-Boelter equation matches the measured results. Regardless of which method is used to calculate the characteristic diameter, most correlations developed use a simple scaling factor based only on geometry. In this case it was found that the Dittus-Boelter equation with a unity scaling factor gave a reasonable prediction of thermal resistance for the lower flow rates (up to 66 ml/s) and that the Carnavos correction factor seems to give a good fit with the
higher flows. Further investigation at higher flow rates on a full length trough would be necessary to confirm this result.

5.4 Heat transfer from the surface of the receiver

Both convection and radiation are significant modes of heat loss from the cover glass of the receiver. Radiation is reasonably straightforward to calculate if the surface temperature of the glass is known. Convection is much more difficult to determine, both theoretically and experimentally.

5.4.1 Radiative heat transfer

The radiative surface of the receiver is the glass. Any thermal radiation emitted by the solar cells, being in the infrared, is almost immediately absorbed by the silicone potant, and therefore conduction is the only mode of heat transfer within the receiver materials. The receiver ‘sees’ both the mirror and the surrounds, as shown in figure 5-9.

![Figure 5-9. Field of view for the cover glass on the receiver.](image)

Radiation is emitted from the glass at long wavelengths – mostly greater than 5 μm – to which the glass of the mirrors is also completely opaque. In other words, radiation that is not reflected is absorbed in the glass. Similarly, surrounding materials will tend to be very absorbent at long wavelengths, whether the collectors are roof or ground mounted. Therefore, it is reasonable to calculate the radiation loss \( \dot{Q}_{\text{rad}} \) for the simple case of grey body radiation from the glass surface:

\[
\dot{Q}_{\text{rad}} = \epsilon_g \cdot \sigma \cdot A_g \cdot \left( T_g^4 - T_{\text{amb}}^4 \right)
\]  

(5-16)
where \( \varepsilon_g \) is the emissivity of the glass (\( \varepsilon_g = 0.88 \)), \( \sigma \) is the Stefan-Boltzmann constant (\( \sigma = 5.67 \times 10^{-8} \text{ W.m}^{-2}.\text{K}^{-4} \)), \( A_g \) is the area of the glass, and \( T_g \) and \( T_{\text{amb}} \) the glass and surrounding ambient temperatures respectively.

### 5.4.1.1 Measurement of glass temperature

Glass temperature has been measured using a thermal camera. Thermal images of the receiver cover glass were taken using a FLIR ThermaCAM® SC 2000 Infrared Camera. Radiation measured by the camera depends not only on the temperature of the glass, but is also a function of its emissivity. The emissivity can be determined experimentally prior to on-sun tests by measuring the surface temperature with a temperature sensor. An image is taken, and the ThermCAM software adjusts the emissivity until it gives the same temperature. The temperature has to be far above the ambient temperature, and therefore a sample piece of glass with a surface mounted thermocouple was placed in an oven and allowed to stabilise thermally at about 80°C. The door of the oven was then opened and an image immediately taken. The emissivity of the glass was determined to be 0.88.

Images of the receiver near the inlet and outlet under typical steady state conditions are shown in figure 5-10.

![Thermal images near the inlet and outlet of a CHAPS receiver.](image)

**Figure 5-10. Thermal images near the inlet and outlet of a CHAPS receiver.**

### 5.4.1.2 Calculation of radiation losses

The temperature profile varies across the width of the cover glass by as much as 30°C. Therefore, because the emission is dependent on the fourth power of temperature, some error would be introduced by calculating the total radiation losses based on a mean temperature. To examine this analytically, the receiver was divided up into 50 segments along both the length and width, as shown in figure 5-11.
Radiation loss from each little segment was calculated, and added to give total loss. Typical radiative losses for a receiver on the test rig are about 20-30 W, as shown in figure 5-12. Using the same assumptions for aperture area as in section 5.2, this represents 1.1% - 1.6% absolute loss due to radiation from the glass. The calculations also show that using the mean temperature to calculate the loss introduces an error of only 1.7% for a typical temperature profile. This translates to less than half a watt in a system that produces more than a kilowatt, and therefore the error in using this approximation is negligible.
5.4.2 Convective heat transfer

Calculation of convective heat loss from the surface of the receiver under operating conditions is subject to much uncertainty due to rapid variation in wind speed and direction, the changing angle of inclination of the receiver, and the approximate nature of the empirical correlations used to calculate the heat transfer.

5.4.2.1 Free convection

The underside of the receiver can be treated as an inclined heated plate. An empirical correlation for the Nusselt number for free convection on a vertical isothermal plate was given previously in equation 2-12. Clearly the receiver is rarely vertical, and therefore the buoyancy force has a component both normal and parallel to the plate surface. For the bottom side of a heated plate, the gravitational constant $g$ used to calculate the Grashof number (equation 2-11), and hence the Nusselt number, can simply be replaced by $g \cos(\theta_{inc})$, where $\theta_{inc}$ is the angle of inclination from vertical (Fujii and Imura, 1972). When the plate is near horizontal, this relationship no longer applies, as convection is strongly impeded by the plate and flow must move horizontally before it can ascend. Some error is introduced by the assumption of an isothermal plate for the glass surface at the mean temperature, but this is likely to be minor. The characteristic length for calculating free convection is measured in the transverse direction to the axis of rotation for the collector, i.e. the $y$ direction in figure 5-13 or the width of the receiver (8 cm). In the longitudinal direction, the receiver is always horizontal.

![Figure 5-13. Characteristic length for free convection](image)

5.4.2.2 Forced convection

Forced convection is due to wind. Wind speed is usually measured by wind cups mounted horizontally, and therefore only the horizontal component of the wind speed is measured, and usually the measurement is made some distance away from obstructions such as the mirrors.
Therefore the wind speed in the plane of the receiver is likely to be different to that measured by the anemometer. While the uncertainty in the measurement of wind speed at the anemometer is ~5%, the uncertainty for wind speed in the plane of the receiver can only be estimated, and for the purposes of the later analysis, it is estimated as 25%. In the following discussion and analysis, the term ‘wind speed’ refers to air movement in the plane of the receiver. The term ‘wind direction’ is also referenced to this plane. The characteristic length $L_{\text{forced}}$ describes the length of the path of the air moving across the receiver glass, and is equal to $W / \cos(\theta_e)$ where $W$ is the width of the receiver and $\theta_e$ is the angle of the wind in the plane of the glass where $0^\circ$ is perpendicular to the receiver. An empirical correlation for the Nusselt number for forced convection was given previously by equation 2-13. However, in this equation the characteristic length $L$ will different to the length used for calculation of free convection. The wind direction may have a component in both the x and y direction in figure 5-13, and therefore the characteristic length will be longer than the width of the receiver.

### 5.4.2.3 Mixed convection

Both free and forced convection contribute to the heat transfer. A typical method of representing mixed convection, as stated by Incropera and DeWitt (1990), is:

$$Nu^n = Nu^F_n \pm Nu^N_n$$  \hspace{1cm} (5-17)

where the Nusselt numbers $Nu_F$ and $Nu_N$ are determined from correlations for pure forced and natural (or free) convection respectively. The plus sign applies for natural convection assisting the flow in the same direction as the forced flow. Similarly, the minus sign applies for opposing flow. The value of $n$ is around 3.5 for a horizontal plate. This is considered to be rough, and only a first approximation. Clearly, whether or not the natural convection is in the same direction as the forced flow (wind) depends on the orientation and inclination of the receiver, and the wind direction.

### 5.4.2.4 Convection calculations for a CHAPS receiver

Figures 5-14 and 5-15 show Nusselt numbers calculated for free and forced convection for a CHAPS receiver. The Nusselt number for free convection is dominated by the glass surface temperature and the inclination of the receiver, with typical values in the order of 12 to 17. The Nusselt number for forced convection is dominated by wind speed and direction, with typical values in the order of 40 to 75. Therefore, unless it is very still, forced convection due to wind will dominate the heat loss. For example, assuming $Nu_F = 50$ and $Nu_N = 15$, equation
Figure 5-14. Calculated Nusselt number for free convection from a CHAPS receiver for various glass surface temperatures and receiver inclinations. Assumes ambient temperature is 25°C.

Figure 5-15. Calculated Nusselt number for forced convection due to wind at various speeds and directions relative to a CHAPS receiver. Assumes the film temperature is 47.5°C.
5.17 gives a combined Nusselt number of $\text{Nu} = 50.2$. The larger term for forced convection dominates the equation.

Despite the increase in Nusselt number due to the longer characteristic length when the wind direction is away from perpendicular to the receiver, there is a net decrease in the heat loss, as the convection coefficient $h_c$ is also calculated using the longer characteristic length in equation 2.7. Figure 5-16 shows the calculated convective heat loss from a 1.5 m long CHAPS receiver on the test rig for a range of Nusselt numbers and glass temperatures, and two different wind directions. The magnitude of convection losses is typically around 30-75 W (or 1.6 – 4.0% absolute efficiency).

![Figure 5-16. Calculated convective heat loss from a CHAPS receiver assuming the ambient temperature is 25°C, for different wind directions. The circles show typical operating conditions.](image)

In summary, the convection coefficient is mostly by the forced convection. The absolute losses therefore depend primarily on the wind speed and the temperature of the glass. Typical losses have been calculated, but it is very difficult to verify the calculations experimentally. One commonly employed method to determine convection losses is to carefully measure all other inputs and losses to the system, and back-calculate the convection using the difference. Convection losses are estimated to be in the order of 30-75 W on the test rig. The uncertainty estimated for the two key components, solar input and thermal energy collected, is around 37 W and 25 W respectively. The convection losses are of a similar magnitude to the probable error, and hence it is considered that in this case experimental determination of convection loss does not have sufficient accuracy to yield meaningful results.
5.5 Heat transfer within the receiver materials

Conduction is the mode of heat transfer through materials within the receiver. In order to minimise the temperature of the cells and therefore maximise electrical efficiency, the aim is to have good heat transfer in the material joining the cells to the extruded aluminium channel. Additionally, it is necessary to maintain electrical isolation between the cells (which have a metalised rear contact over the whole cell) and the aluminium. A thin (0.13 mm) double sided tape is currently used for attaching the cells to the receiver. While the conductivity of the tape is stated by the manufacturer, experience has shown that heat transfer can be significantly affected by the pressure applied in the join. Most electrically isolating, thermally conductive materials are designed to attach electrical components to heat sinks, and are usually mechanically fastened. In contrast, while some pressure is temporarily applied as the cells are laid on the tape, there is no permanent fastening. Therefore, an experiment has been carried out to determine the conductivity that could be expected between the cells and the aluminium extrusion.

5.5.1 Measurement of thermal resistance tests for various heat sinking tapes

A testing device was designed to accurately measure thermal resistance for various heat sinking tapes\(^4\). A sample piece of thermal tape was placed between two aluminium blocks. A 50 W, 10 Ω resistor was mounted to the top block to provide a constant heat source. The bottom block was heat sunk through fins to an ice bath. Thermocouples measured the temperature difference between the top and bottom block. The entire setup was heavily insulated with polystyrene foam. The test setup is shown below in figures 5-17 and 5-18.

\(^4\) Thanks to James Cotsell for implementing these tests and for his contribution to the design.
Figure 5-17. Schematic representation of the thermal transfer testing station.

Figure 5-18. Photos of the thermal transfer testing station.

A power supply delivered 40 W (20 V and 2 A) and temperature readings were taken at 1 minute intervals for 25 minutes. Steady state was usually reached after about 10-15 minutes. Four different materials were tested, as described in table 5-1.

Table 5-1. Description of some materials that could be used to bond cells to the receiver.

<table>
<thead>
<tr>
<th>Name of material</th>
<th>Material characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chomerics T404</td>
<td>Kapton MT Thermally conductive polyimide film coated with aluminium oxide loaded acrylic pressure sensitive adhesive.</td>
</tr>
<tr>
<td>Bergquist Bondply 660</td>
<td>Kaladex 2000 polyethylene naphthalate (PEN) film coated with acrylic adhesive.</td>
</tr>
<tr>
<td>Saint Gobain C675</td>
<td>2 mil aluminium foil coated with acrylic pressure sensitive adhesive (note: this is not an electrically isolating foil).</td>
</tr>
<tr>
<td>AIT Coolbond CB7208-E</td>
<td>Aluminium Nitrite filled thermoplastic film adhesive.</td>
</tr>
</tbody>
</table>
Results for four different material tested are shown in figure 5-19 below. Twenty minutes were allowed for the system to stabilise thermally before the temperatures of the blocks were measured. *Thermal resistance for conduction* is a useful measure for the rate of heat transfer, defined by:

\[
 R_{\text{cond}} = \frac{(T_1 - T_2)}{\dot{Q}} = \frac{t}{kA} 
\]

(5-18)

where \( \dot{Q} = 40 \text{ W} \) is the power delivered through the resistor, \( T_1 \) and \( T_2 \) are measured temperatures in the top and bottom aluminium block respectively, and \( k \) and \( t \) are the conductivity and thickness of the tape.

![Temperature difference across tape](image)

<table>
<thead>
<tr>
<th>Material</th>
<th>Measured Conductivity ((\text{W.m}^{-1}.\text{K}^{-1}))</th>
<th>Stated * conductivity ((\text{W.m}^{-1}.\text{K}^{-1}))</th>
<th>U-value ((\text{W.m}^2.\text{K}^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chomerics</td>
<td>0.209</td>
<td>0.37</td>
<td>1646</td>
</tr>
<tr>
<td>Bergquist</td>
<td>0.243</td>
<td>0.4</td>
<td>1733</td>
</tr>
<tr>
<td>Saint Gobain</td>
<td>0.476</td>
<td>1.1</td>
<td>3121</td>
</tr>
<tr>
<td>AIT Coolbond</td>
<td>0.632</td>
<td>3.6</td>
<td>4169</td>
</tr>
</tbody>
</table>

* Value quoted by the manufacturer

Figure 5-19. Results of the conductivity measurements.

Note that the measured conductivity is in each case significantly lower than that stated by the manufacturer. This is not necessarily because false information has been provided, but possibly shows the sensitivity of the conductivity to pressure across the joint. The tapes are mostly used for heat sinking electrical components to printed circuit boards, and usually there is a positive pressure applied by mechanical fastening. In the case of the solar cells, it is not possible to apply mechanical pressure for a variety of reasons such as allowance for thermal expansion of the silicone potant, and avoidance of pressure points on the glass cover that can lead to cracking. The Chomerics and Bergquist tapes have more than double the thermal resistance than the AIT Coolbond. However, they are far less expensive (at the time of writing, AUD $16 per receiver, compared to AUD $44 for the AIT Coolbond). The cost of the adhesive is an important factor in choosing between different products. Early CHAPS receivers employed the Chomerics tape. However, in recent development the Bergquist tape has been used. The main reason is that under long-term testing at high concentrations of light,
the Chomerics tape seems to char, whereas the Bergquist tape has not yet shown signs of any deterioration. The UV tolerance of the AIT Coolbond is unknown.

5.5.2 Other materials

Table 5-2. Thermal conductivity of other materials

<table>
<thead>
<tr>
<th>Material</th>
<th>Thermal conductivity (W.m(^{-1}.K(^{-1}))</th>
<th>Thickness (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminium</td>
<td>209</td>
<td>Variable</td>
</tr>
<tr>
<td>Silicon</td>
<td>157</td>
<td>0.35</td>
</tr>
<tr>
<td>Glass</td>
<td>0.937</td>
<td>3.3</td>
</tr>
<tr>
<td>Silicone</td>
<td>0.2</td>
<td>2.0</td>
</tr>
<tr>
<td>Rockwool</td>
<td>0.041</td>
<td>Variable</td>
</tr>
</tbody>
</table>

The thermal conductivities of other materials within the receiver are well known. The aluminium and glass conductivity values are the default values for aluminium alloy 6063-T5 and window glass in the Strand7 finite element analysis program introduced later in this chapter. The thermal conductivity of silicon is given by Jones and Jones (2004) for silicon at 300 K. The conductivity of the silicone potant (Wacker-Chemie GmbH, 2001) and rockwool (Bradford Insulation, 2004) are given by the manufacturers. The bonding is intimate between the silicone potant and the glass on one side, and the cell on the other.

5.5.3 Measured losses through the insulation

The heat loss through the receiver insulation was calculated by a simple experiment where hot water was passed through a receiver with heavy insulation on the front surface. The hydraulic fittings at the end of the receiver were also heavily insulated for the test.

Figure 5-20. Set-up for the UA-value tests (left) and hydraulic fittings arrangement (right).
It was assumed that there was negligible heat transfer through the insulated front surface of the receiver due to the very thick insulation. Therefore heat transfer from the fluid to the surroundings has thermal resistance due to heat transfer between the fluid and receiver (which can be estimated using the results from section 5.3), thermal resistance due to the insulation between the receiver and the insulation cover, and thermal resistance between the cover and surroundings due to convection and radiation (which can be estimated from the results in section 5.4). Solving the resulting set of simultaneous equations allows the UA-value for the insulation to be determined. However, the result is quite sensitive to error in the calculated coefficient of convection at the surface of the cover. Additionally, the temperature difference across the receiver, measured using the differential thermocouple arrangement, is small (~0.2°C), and hence the uncertainty for this measurement is higher than for the previous measurements (when the receiver was on sun, the measured temperature difference was between 2.5-15°C, depending primarily on the flow rate), and estimated to be around 10%. The UA-value was calculated to be 1.85 W.K\(^{-1}\). However, because the calculation is sensitive to uncertainty in the value of heat transfer to the surroundings, the uncertainty in the UA-value is estimated to be ±40%. The experiment was repeated with the heavy insulation around the hydraulic fittings at the end of the receiver removed, exposing the insulated pipework around the temperature probes, and the bare hydraulic fittings at the ends of the receiver. This region should perhaps have been better insulated during the experiments; however, by comparing the heat loss before and after the heavy insulation was removed, heat loss due to the fittings was determined. The UA-value for the fittings was calculated to be approximately 0.15 W.K\(^{-1}\).

There is a trade-off between the thickness of insulation and shading of the mirror due to the width of the receiver. The current insulation thickness around the sides is sized to be flush with the edge of the solar cell tray. Although thicker insulation would result in lower losses through the insulation, this is outweighed by concurrent losses due to the reduction in incident light because of shading (as shown by simulation later in section 7.3.5.1)

### 5.6 Simulation of the conduction using Strand7

Strand7 is a finite element analysis program that is capable of heat transfer analysis for simple systems. A thermal model has been created that examines a 2-D slice through centre of the receiver. Each individual finite element has an assigned property that includes heat transfer details.
Coefficients for convection and radiation are defined at the boundary of the modelled receiver (i.e. the outer elements), as are the surrounding ambient conditions. The software has no capability for computational fluid dynamics analysis, and therefore convection coefficients must be carefully determined for the steady state conditions particular to the simulation. For each result presented below, the convection and radiation coefficients are estimated from the measurements and calculations described above in sections 5.3 and 5.4. Energy can be introduced to the model in two ways: as a heat flux or a heat source. A heat flux is used to model the energy incident upon the solar cells, and is modelled as a gaussian distribution across the cell. The energy absorbed by the glass and silicone is modelled by assuming there is a heat source within each element.

5.6.1 Energy input

The parameters for the gaussian curve representing radiation flux were determined by comparison with a flux profile curve measured using videographic flux mapping (Johnston, 1998). The magnitude of the curve was adjusted to match the total energy (determined by integration of the gaussian curve) to that expected for the GOML mirror with known dimensions, optical accuracy and reflectivity. The thermal energy flux on the cells was determined by scaling the incident radiation to account for the receiver reflectivity and absorption of the cover materials and the cells. Finally the energy converted directly to electricity was removed, as it does not contribute thermally. Figure 5-22 shows these curves
for typical conditions using the test rig mirror, with data from 11:45am, 3 November 2003. The measured direct beam radiation was 1054 W.m\(^{-2}\) for the 1.25 m wide mirror, with 8.25 cm shading from the receiver.

![Figure 5-22. Radiation flux intensity used for the Strand7 model.](image)

The base case model assumes 1.5% absorption in the 3.3 mm glass, and 2.0% absorption in the ~2 mm layer of silicone. This is based on absorption results outlined in section 4.5.5 and 4.5.6. The distribution of the energy absorbed is modelled with a gaussian curve to match the intensity of light transmitting through the cover.

The back of the receiver cover also absorbs a little energy from the sun. The absorption (weighted by the solar spectrum) of a typical sample of aluminium sheet was measured to be around 32%. Therefore the flux intensity is 32% of global radiation.

### 5.6.2 Energy loss

Each element in the Strand7 model can have both a coefficient defined for convection losses, and an emissivity value for radiation losses. Additionally, the surrounding temperature for convection or radiation is defined for each individual element. In the case of convection from the walls of the conduit, this allows the fluid temperature to be set as the surrounding temperature in the model.

As outlined in section 5.4.1.1, the emissivity of the glass was measured as 0.88. The convection coefficient for the glass is variable, as discussed in section 5.4.2, and results below in section 5.6.6.1 show the impact of several different conditions. The insulation cover
is made of a sheet of commercial unpolished aluminium. The emissivity of the cover was assumed to be around 0.10 at typical wavelengths for emission. As forced convection due to wind dominates the heat transfer, the coefficient for convection for the top and side surfaces was approximated using the same value as for the lower heated surface. As shown in the analysis in section 5.4.2, the impact of natural convection (which would be quite different between the top, bottom and sides of the receiver) is very small compared to the forced convection, unless there is virtually no wind.

5.6.3 Base case

The temperature plot from modelling (using the boundary conditions from measurements on 3 November 2003) for the base case model is shown in figure 5-23.

![Temperature plot](image)

*Figure 5-23. Strand 7 temperature plot from 3 November 2003 for the base case model.*

Table 5-3 gives the operating conditions and key assumptions that were used for the base case model.
Table 5-3. Operating conditions and modelling assumptions for the base case Strand7 model.

<table>
<thead>
<tr>
<th></th>
<th>3 November 2003</th>
</tr>
</thead>
<tbody>
<tr>
<td>Date</td>
<td>3 November 2003</td>
</tr>
<tr>
<td>Time</td>
<td>11.45 am</td>
</tr>
<tr>
<td>Ambient temperature</td>
<td>19.9°C</td>
</tr>
<tr>
<td>Direct radiation</td>
<td>1054 W.m(^{-2})</td>
</tr>
<tr>
<td>Wind speed</td>
<td>0.68 m.s(^{-1})</td>
</tr>
<tr>
<td>Flow rate</td>
<td>122.4 ml.s(^{-1})</td>
</tr>
<tr>
<td>Bulk water temperature*</td>
<td>48.0 °C</td>
</tr>
<tr>
<td>Thermocouple 2</td>
<td>52.9 °C</td>
</tr>
<tr>
<td>Thermocouple 4</td>
<td>49.1 °C</td>
</tr>
<tr>
<td>Reflectivity of mirror</td>
<td>93.5%</td>
</tr>
<tr>
<td>Mirror shape accuracy</td>
<td>95.0%</td>
</tr>
<tr>
<td>Reflectivity of the receiver</td>
<td>11.4%</td>
</tr>
<tr>
<td>Absorption in the glass cover</td>
<td>1.5%</td>
</tr>
<tr>
<td>Absorption in the silicone potant</td>
<td>2.0%</td>
</tr>
<tr>
<td>Electrical conversion efficiency**</td>
<td>13.6%</td>
</tr>
<tr>
<td>Reflectivity of the aluminium cover</td>
<td>68%</td>
</tr>
<tr>
<td>Emissivity of the glass</td>
<td>0.88</td>
</tr>
<tr>
<td>Emissivity of the aluminium cover</td>
<td>0.10</td>
</tr>
<tr>
<td>Convection coefficient – cover and glass</td>
<td>11.0 W.m(^{-2}).K(^{-1})</td>
</tr>
<tr>
<td>Convection coefficient – fluid</td>
<td>2.074 W.m(^{-2}).K(^{-1})</td>
</tr>
<tr>
<td>Thermal tape conductivity</td>
<td>0.209 W.m(^{-1}).K(^{-1})</td>
</tr>
</tbody>
</table>

* Linearly interpolated for the position of thermocouples 2 and 4 based on measured inlet and outlet temperatures.
** Defined as the amount of electrical energy generated divided by the light incident on the cells.

5.6.4 Validation

The temperature profiles generated using Strand7 can be compared to the temperatures measured by the thermocouples mounted on the receiver and the glass temperature measured by the thermal camera to tune the model and validate the results. Figure 5-24 shows a plot of the difference between the measured and predicted temperatures of the receiver at the back of the channel and near the front. Temperatures from the receiver were recorded by the thermocouples 2 and 4 (locations given in figure 5-3), and the predicted temperatures were determined from the corresponding positions in the Strand7 model.
The trend clearly shows that the model better predicts the temperature of the aluminium channel at higher flows. At lower flow rates, it was found that the temperature difference predicted by the Strand7 calculations between thermocouple 2 (positioned between cell and channel) and thermocouple 4 (positioned at the top of the channel) is greater than that measured. An overestimate of the assumed value of conductivity of the aluminium extrusion might cause such a discrepancy, as heat is transferred from the illuminated side of the receiver to the back. However, it is unlikely that the conductivity varies significantly from values in literature. More probable is that at lower flows there is a higher degree of stratification of the flow within the channel, with the hotter, more buoyant fluid at the top of the channel. It has been assumed in the model that the water temperature in the channel is uniform at all surfaces, which in reality would only be the case in fully turbulent flow. As discussed in section 5.3.1, the flows measured are mostly in the transition zone between laminar and turbulent flow, and the two lowest flow runs are laminar. It was also suggested that the heat transfer seemed to approach turbulent flow for the two highest flow runs, based on the thermal resistance values calculated using the Carnavos correction factor. This seems to support the hypothesis that the discrepancy between the measured and simulated thermocouple temperatures at the lower flow rates in figure 5-24 is due to stratified flow.

The other method of validation available is to compare the glass temperatures measured using the thermal camera with the Strand7 results. The glass surface temperature was measured using the thermal camera, as described in section 5.4.1.1. As figure 5-25 shows, measured glass temperatures are consistently higher than those obtained from the model, particularly at
the peak of the illumination. In order to determine the most likely reason, a sensitivity analysis was carried out examining the key parameters that affect glass temperature. Note that the precise positions of the peaks of the curves do not always line up because the focal line was not always exactly in the middle of the cell at the time the image was taken.

![Graph showing comparison between measured and simulated glass surface temperatures.](image)

*Figure 5.25. Comparison between measured and simulated glass surface temperatures.*

### 5.6.5 Sensitivity analysis

The sensitivity of the model to error in the parameters in table 5-3 is examined in figure 5-26. The arrows indicate an increase in the value of the parameter. The bold red lines indicate the default parameter values. Note that these base case values are a little different to those in table 5-3. However, the model is the same and therefore the sensitivity analysis is valid.
Figure 5-26a. Sensitivity to the heat transfer coefficient for convection to the surrounding air.

Range of $h_{\text{c-air}}$ values (default is $4.6 \text{ W.m}^{-2}\text{K}^{-1}$)

$[2, 5, 10, 15] \text{ (W.m}^{-2}\text{K}^{-1})$

Figure 5-26b. Sensitivity to the heat transfer coefficient for convection to the water.

Range of $h_{\text{c-water}}$ values (default is $40 \text{ W.m}^{-2}\text{K}^{-1}$)

$[200, 300, 400, 500, 750, 1000, 1500, 2000] \text{ (W.m}^{-2}\text{K}^{-1})$

Figure 5-26c. Sensitivity to the peak radiation flux.

Range of values for magnitude of gaussian curve (default is $95,600 \text{ W.m}^{-2}$)

$[80, 90, 100, 110 \times 10^3 \text{ (W.m}^{-2})$

Figure 5-26d. Sensitivity to the conductivity of the tape bonding cells to the receiver.

Range of $k_{\text{tape}}$ values (default is $0.37 \text{ W.m}^{-1}\text{K}^{-1}$)

$[0.1, 0.2, 0.3, 0.4, 0.6, 1.0] \text{ (W.m}^{-1}\text{K}^{-1})$

Figure 5-26e. Sensitivity to the absorption in the cover glass and silicone potant.

Range for magnitude of $\alpha_{\text{glass}} + \alpha_{\text{silicone}}$ (default is 3%)

$[0.5\%, 1.0\%, 2.0\%, 3.0\%, 4.0\%]$
Taking into account uncertainty in the values of the various parameters, the two parameters that most affect the temperature at the centre of the glass relative to the temperature at the edge are the conductivity of the cell-receiver bond and the absorption of the glass and silicone potant. The conductivity has been measured with reasonable accuracy, as discussed in section 5.5.1.

The base case values for absorption in the silicone and glass assume that light passes only once through the material, i.e. the path length for absorption is equal to the thickness. However, as was discussed in some detail in section 4.5.8, the surface of the solar cell is textured to increase light trapping. In addition, 11% of the surface is covered by highly reflective silver fingers. Therefore, the fraction of light that is reflected from the cell tends to bounce around between the cell and the inside surface of the glass, with path lengths significantly longer than the thickness of the cover materials. This light may eventually be absorbed or escape the glass, but in either case there is likely to be quite a high amount of absorption in the glass and potant.

![Figure 5-27. An illustration that the path length through the cover materials for some light can be longer than the thickness of the materials.](image)

More accurate estimation of the amount of absorption in the cover materials could be carried out with a detailed optical model of the cell and encapsulating materials, which includes the textured surface and accounts for the variability of refractive index with wavelength. However, this is outside the scope of the present work. Instead the amount of absorption has been estimated by comparing the measured glass temperature with the results of the model with increased absorption. It was found that an absorption value around 1.8 times that of a ‘single pass’ gives a reasonable fit with the experimental data (i.e. $\alpha_{\text{glass}} = 2.7\%$ and $\alpha_{\text{silicone}} = 3.6\%$), as shown in figure 5-28.
Figure 5-28. Glass temperature - Strand7 model with absorption in the cover materials increased by 100% compared to the base case in table 5-3.

While it is useful to know the temperature of the glass for calculation of thermal losses and for validating the model, the most interesting temperature profile is that of the solar cell. As is discussed in detail in the following chapter on electrical performance, the efficiency of the cells is reduced as their temperature increases. Figure 5-29 shows the cell temperature profiles predicted for the two absorption cases shown in figure 5-28.

Figure 5-29. Solar cell temperature – Strand7 base case compared to increased absorption in the cover materials.

Interestingly, there is virtually no difference in cell temperature, despite the significant difference in the temperature of the glass. The cell temperature is mostly dictated by the conductivity of the bond to the aluminium heat sink, as most of the energy is absorbed by the cell and removed by the heat sink. The increase in thermal losses due to the increased glass temperature is not high compared to the energy dissipated by the heat sink. There is some incentive to reduce glass temperature, mostly to do with alleviating thermal stresses in the
receiver. The simplest solution is to minimise the thickness of the silicone and glass as much as possible. However, the present cover arrangement has had significant life time testing, and the use of thinner glass is unlikely to be adopted until extensive testing of durability has been carried out. Note also that while the amount of light available for electrical conversion is reduced by the absorption, much of the absorption is in the infrared and therefore does not affect the solar cell performance.

5.6.6 Results of the Strand7 modelling

5.6.6.1 Wind speed and direction

The Strand7 model has been used to compare different wind speeds and directions relative to the receiver. Figure 5-30 shows the predicted thermal output in watts, including the useful thermal energy collected in the water, the thermal losses from the glass surface (split up into the radiation and convection components), and losses from the insulation cover. In each case the thermal energy input conditions are kept the same (a total of 1249 W, including the heat flux on the cells and the back of the receiver, and absorption in the glass and silicone). The base case (shown on the left in figure 5-30) is based on the data in table 5-3 but includes 1.8x higher absorption in the glass and silicone. For the purposes of the thermal analysis, it is assumed that the amount of electrical energy removed is the same, although calculations based on the cell temperature show that there is 1-2 W variation.

Figure 5-30. Thermal energy outputs and losses for various wind speeds and directions (where 0° is perpendicular to the receiver).
The trend shows convection losses increasing with wind speed, but decreasing as the wind direction shifts away from perpendicular. Radiation losses decrease a little with increased wind speed, as the temperature of the glass is marginally cooler in stronger winds.

### 5.6.6.2 Fluid temperature and flow rate

The model has been used to compare the effect of various fluid flow rates and temperatures. The heat transfer coefficient for different flow rates is calculated from the empirical expression for Nusselt number using the Dittus-Boelter equation and the Carnavos correction factor, discussed in section 5.3. The base case in figure 5-31 also has a calculated heat transfer coefficient, rather than the experimental result used previously, which makes a small difference to the balance of energy outputs.

![Figure 5-31. Thermal energy outputs for various fluid flow rates (left) and temperatures (right).](image)

Higher flow rates result in a small increase in thermal output. At a flow rate of 250 ml/s, which is typical for a full length trough, the thermal efficiency improves by around 0.45% absolute compared to the base case. The effect of fluid temperature is far more pronounced. At the higher end of the range of operating temperatures, radiation begins to form a significant part of the thermal losses.

### 5.6.6.3 Conductivity of the thermal tape

One of the main reasons for simulating the receiver in Strand7 was to examine the impact of different bonding materials to attach the solar cells to the receiver. Currently, the temperature of the solar cells at the peak of the beam is significantly higher than the temperature of the heat sink. Figure 5-32 shows the temperature profile across the solar cell
for the measured base case conductivity of 0.209 W.m\(^{-1}\).K\(^{-1}\), compared to theoretical cases with higher conductivity. The values can be compared to the conductivity values in table 5-4 for the four alternative materials discussed above in section 5.5.1. Using the best material from those tested, the AIT Coolbond, would result in a drop in the peak cell temperature of 12.7°C compared to the base case. This has a significant effect on the electrical efficiency, particularly as most of the electrical generation is in the region of the highest radiation flux. The improvement in electrical efficiency due to the reduced cell temperature is calculated to be around 4.5% absolute, or around 10 W for a receiver on the test rig. The thermal efficiency stays reasonably constant, as the improvement due to lower thermal resistance is offset by the reduction in energy available because of the greater electrical conversion. There is a strong incentive to find a more conductive (but affordable) tape, epoxy or another method of cell fastening, as significant gains in electrical performance are available.

![Figure 5-32. Solar cell temperature for tapes of various thermal conductivity.](image)

**Table 5-4.** Conductivity, adjusted for the material thickness in the Strand7 model.

<table>
<thead>
<tr>
<th>Material</th>
<th>Equivalent conductivity (W.m(^{-1}).K(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chomerics T404</td>
<td>0.209</td>
</tr>
<tr>
<td>Bergquist Bondply 660</td>
<td>0.220</td>
</tr>
<tr>
<td>AIT Cool-Bond CB7208-E</td>
<td>0.527</td>
</tr>
<tr>
<td>Saint Gobain C675</td>
<td>0.396</td>
</tr>
</tbody>
</table>
The efficiencies of the individual solar cells that make up a photovoltaic receiver dictate its maximum efficiency. Under operating conditions, the receiver efficiency is always less than its theoretical maximum. The cells operate at elevated temperatures, hence the solar cell efficiency is reduced. Furthermore, the distribution of light on the receiver is neither uniform across the cells or along the length of the receiver. This also has a detrimental impact on receiver efficiency. This chapter examines the magnitude of the efficiency reduction of the CHAP receivers due to these effects.

6.1 Temperature dependency

6.1.1 Measurement of I-V curves using the flash tester

A constant voltage flash tester (figure 6-1) has been used for solar cell testing at high concentration ratios. The flash tester was built at the ANU by Keogh (2001). Most conventional flash testing systems maintain constant light intensity while rapidly sweeping an I-V curve of the cell. This method has two disadvantages – it requires a flash that is specially engineered to produce constant light output, and the rapid change in cell bias-voltage causes transient errors. The approach developed by Keogh essentially does the reverse – it maintains a constant bias voltage on the cell while allowing the light level to vary. This allows the use of a commercial xenon flash, and reduces sensitivity to transient errors. While the spectrum of the flash is not known accurately, experimental measurements on a group of ANU concentrator cells showed that spectral mismatch is less than 2.5% for any of these cells. The light uniformity at 30 suns is ±2% over an area ~10 cm x 10 cm. The solar cell is held to an aluminium plate by a vacuum. Mounted along the sides of this plate are electrical contacts to the cell. Temperature control of the plate is achieved by use of a thermoelectric cooler (Peltier device) and a digital process controller. For the following experiments, the temperature range was 25-85°C and the light concentration ratio was 30,000 W.m⁻². Uncertainty in the measurement is given by Keogh as 5.5% for short circuit current, 1.3% for open circuit voltage and 1.5% for fill factor. The cell-testing block keeps the test or reference cell within ±3°C of the nominal set point.
6.1.2 Temperature dependency results from flash tester measurements

Figure 6-2 shows the sensitivity of open circuit voltage, fill factor and efficiency to temperature, for four randomly selected ANU solar cells, measured on the flash tester. All measurements are for an even illumination of 30 suns. As is later discussed in section 6.3, the results are a little different when the cells have non-uniform illumination and temperature.

While the cells range in quality, the most dominant reason for the efficiency reduction is because of a drop in open circuit voltage with temperatures of around 1.9 mV/°C. The reason for the sensitivity of voltage to temperature is that the reverse saturation current $J_0$ increases rapidly with temperature. The short circuit current actually increases a little with temperature. This is because the diffusion length increases slightly, which means there is a higher probability of carriers reaching the pn junction. The fill factor is lowered with temperature. The resulting drop in efficiency varies a little between cells, but is typically between 0.3 - 0.4%/°C relative.
6.1.3 Temperature dependency results from a full receiver

To verify that the relationship of \( V_{oc} \) with temperature holds for a full receiver, tests were carried out on a receiver on (a) the ‘20 sun tester’ and (b) the outdoor CHAPS test rig. The
20 sun tester is an indoor testing rig consisting of a row of ELH 120V, 300W lights located near to the receiver to produce an intensity (contra to what the name suggests) around 15 suns. The lights were tuned to have even illumination on each cell of the receiver with a tolerance of ±5%.

For both the 20 sun tester and the CHAPS outdoor test rig, water was passed through the receiver at a known temperature and at a mass flow that ensured little temperature variation along the length of the receiver. Both the illumination and temperature distribution could be expected to be quite even across the cells on the 20 sun tester, conditions similar to those inside the flash tester. However, in the CHAPS outdoor test rig, both the illumination and temperature profiles could be expected to be quite uneven across the cells. An average temperature of the receiver was measured by a thermocouple located directly behind the middle of the central cell in the receiver. The resulting temperature dependency curves are shown below in figure 6-4.

![Figure 6-4. $V_{oc}$ dependency on temperature for a full receiver.](image)

Both lines on figure 6-4 indicate a temperature dependency of around 55 mV/°C, which when normalized by dividing by the number of cells (31) is approximately 1.8 mV/°C. This is only marginally lower than the value of 1.9 mV/°C found previously for a single cell tested on the flash tester. It is therefore apparent that the dominant trend with increasing temperature, the reduction in open circuit voltage, is consistent between individual cells and full receivers. The effect of the temperature and illumination profile across the cells on open circuit voltage is investigated in further detail in section 6.3.

### 6.2 Illumination profile

The illumination profile has little effect on the thermal performance of the receivers, but a large impact on electrical performance. The impact on electrical performance of
non-uniform illumination in the transverse and longitudinal direction is discussed in turn in the following sections, but first the source and magnitude of the non-uniformities are investigated.

6.2.1 The sun shape

The sun is not a point light source, and therefore even if there existed a device that was a perfect solar concentrating trough, the flux distribution at the focus would not be a line, but would have a shape similar to (but not the same as) a gaussian distribution, known in solar energy applications as sunshape. Incident radiation for concentrators comes directly from the solar disc, which has a half-angle of 4.65 mrad (Puliaev et al., 2000), and from the annular circumsolar region, which has its inner limit at the edge of the solar disc, and its outer limit generally set at the acceptance angle of a pyrheliometer (half-angle typically 2.5° or 43.6 mrad). The circumsolar ratio (CSR) is defined as the radiation within the circumsolar region, divided by the radiation from the direct beam and circumsolar regions combined. Figure 6-5 shows the sunshape measured at various CSR values (Neumann et al., 2002) and plotted on a logarithmic scale. It also shows the resulting reflected beam shape calculated assuming a perfect 1.55 m wide parabolic trough with focal length 840 mm. In designing a solar concentrator, there is a trade-off between the concentration ratio and the percentage of incident radiation. A high concentration ratio usually means a lower cost receiver relative to the mirror area, but higher optical losses due to radiation from the circumsolar region missing the receiver.

![Figure 6-5. Sunshape for various circumsolar ratios.](image)

Buie et al. (2002) developed a relationship between the acceptance angle and amount of incident radiation. The relationship depends on the circumsolar ratio, which in turn depends on the amount of scattering of the solar radiation passing through the Earth’s atmosphere.
Measurement of about 2300 CSR values in Germany, France and Spain by Neumann et al. (2002) showed that the CSR value is mostly less than 5%. 71.5% of the measurements were in the 0-5% CSR zone, and a further 13.9% were in the 5-10% CSR zone. The proportion of low CSR values is likely to be even higher for most locations in Australia, where the air is generally clearer. Buie’s relationship shows that (assuming 5% CSR) around 95% of radiation comes from within a region of angular extent 5 mrad, but after that there is a rapid increase in the angle for only modest gains in incident radiation. For example, for 98% incident radiation, the angular extent of the solar region more than doubles to 13 mrad. Therefore the sun shape often defines maximum concentration ratio of a collector. When higher concentration ratios are required, a secondary concentrator is used at the focus of the mirror.

6.3 Non-uniform illumination in the transverse direction

The illumination profile across a cell has been determined experimentally by Johnston (1998) using videographic flux mapping. A typical mean flux cross-section for the 15 m CHAPS prototype (at a 3.8° angle of incidence) is shown in figure 6-10. At near normal angles of incidence the maximum flux intensity exceeds 100 suns in localised regions.

![Figure 6-6. Mean flux profile cross-section on the CHAPS prototype system at 3.8° angle of incidence. Figure from Johnston.](image)

6.3.1 Modelling the effect of a non-uniform illumination profile

Cells under concentrated sunlight have large currents, and therefore particular attention must be paid to series resistance, which is distributed in nature. The main sources of resistance
within a conventional front-illuminated solar cell are the metal grid, the finger/emitter contacts, the shallow emitter region, the bulk region, the rear contacts and the rear metal. Since it is possible to build a relatively thick rear metal layer with large area coverage, the rear metal resistance can be made negligible. The rear semiconductor/contact resistance can also be kept small through the use of localized rear surface doping. The effect of highly non-uniform light on solar cells is modelled and discussed in detail by Franklin and Coventry (2002). The first author, Franklin, developed a detailed two-dimensional finite element model of a solar cell that allows different illumination and temperature profiles to be simulated. Details of the model and associated assumptions can be found in the paper, which is attached as appendix C. A solar cell can be considered to comprise a number of identical ‘quarter finger-space’ units, shown in figure 6-7. Within this space, the cell is divided into many sub-elements, shown in figure 6-7 at position (x, y), and each sub-element is modelled using the single-diode solar cell equation 2-4. In the case of the ANU concentrator cells, there are 170 fingers, and hence 680 quarter finger-space units. The resistive network is solved using a controlled iteration procedure, with the overall cell voltage held at a constant.

Figure 6-7. Quarter finger-space cell unit with element Δx by Δy at position (x,y). Figure from Franklin.

Figure 6-8 shows the results of the modeling. The I-V curves correspond to three cases: uniform illumination and uniform temperature, distributed illumination and uniform temperature, and distributed illumination and distributed temperature. In each case the cell
absorbs the same amount of light and the average cell temperature is identical. Assumptions regarding the precise shape of the distributed illumination and temperature curve are given in Franklin and Coventry (2002), but essentially the distributions are modelled with a gaussian curve and correspond reasonably well to conditions that exist at the focus of a CHAPS mirror. When the distributed illumination is introduced, there is a reduction in open circuit voltage. When the cell is modelled to include both distributed illumination and temperature, there is a further reduction. The ‘knee’ of the I-V curve can also be seen to soften slightly, indicating the increased effect of series resistance. The difference between open-circuit voltage for the case of uniform illumination and temperature and for the case of distributed illumination and uniform temperature is around 5 mV, with a further reduction of 2 mV when distributed temperature is included.

Figure 6-8. Simulated I-V curves: average concentration ratio =22.7, average temperature = 69°C. Figure from Franklin.

Under open circuit conditions, there is no net current flowing into or out of the cell. However there are in fact currents flowing within the cell. The dashed line in figure 6-9 shows the current flow in the fingers under open circuit conditions, based on distributed illumination and temperature profiles. In the region furthest from the bus-bar, where the cell is most heavily illuminated, a current is leaving the finger (due to the generated photocurrent in that region of the cell), whilst in the region near the bus-bar, current is entering the finger. Therefore there exists significant current flow from the illuminated region to the dark region, shown by the solid line. The net current is zero at the bus-bar, since the entire cell is operating at open circuit, but increases to a maximum of about 4.5 amps (significant
considering the short circuit current is in the order of 15 amps) at roughly two-thirds of the distance to the centre of the cell.

![Figure 6-9](image)

*Figure 6-9. Simulated current distributions at open-circuit. Figure from Franklin.*

### 6.3.2 Experimental comparison with the model

To simulate the effect of non-uniform flux distribution across a cell for a linear concentrator, an I-V curve was measured in the flash tester for a single solar cell under two scenarios: 30 suns concentration over the whole cell, and 90 suns concentration over the middle third of the cell (shown by the shaded blocks in figure 6-10).

![Figure 6-10](image)

*Figure 6-10. Flux profiles tested in the flash tester.*
An aluminium mask shading all bar the middle third of the cell was clamped to the surface of the cell in the flash tester. The distance between the lamp and the cell was decreased until it was possible to achieve the same short circuit current for the masked cell as for the unmasked cell at 30 suns concentration. Tests were carried out with the solar cell held to a heat sink block at 25°C. The results (figure 6-11) show a reduction in open circuit voltage of 6.5 mV, and a softening of the I-V curve, resulting in an efficiency drop from 20.6% for uniform illumination to 19.4% for centralised illumination.

![Image](image.png)

*Figure 6-11. I-V curves for uniform and centralised illumination of a solar cell.*

The cell parameters are not identical to those of the modelled cell (which are typical values rather than specific to the tested cell). In addition, a step illumination profile is applied (due to the aluminium mask) rather than the gaussian profile that is used in the model, and so no exact quantitative comparison can be made. However, in qualitative terms, the measured curve shows very similar characteristics to the modelled curves in figure 6-8. The fill factor lowered, and a significant drop in open circuit voltage is observed, similar to that predicted by the modelling.

Figure 6-12 shows flash tester results comparing a cell that has evenly distributed illumination, and the same cell with the ‘top hat’ illumination profile, for a range of temperatures. The magnitude of the voltage drop due to the non-uniform illumination increases a little when the cell is operating at elevated temperatures. This would be expected, as the internal current flows under open circuit conditions would increase a little, as the diffusion length increases with temperature. The magnitude of the voltage drop due to non-uniform illumination becomes more significant in relative terms when the open circuit voltage is reduced by temperature.
Overall, in the absence of a secondary flux modifier, the efficiency losses due to the non-uniform illumination flux profile across the cell are unavoidable, and for good cells, are in the order of 5-15% relative, depending on the temperature of the cells. As shown in figure 6-12, the efficiency drops and the effect of the non-uniform illumination increases as the cell temperature increases, hence the relative drop in efficiency is magnified at higher temperatures.

### 6.4 Non-uniform illumination in the longitudinal direction

Photovoltaic systems, whether flat plate or concentrating, normally have groups of solar cells connected in series in order to increase voltage and limit current. Low current means cable sizes for transmission can be reduced (and hence cost reduced) without significantly increasing voltage losses due to series resistance. Efficient dc-ac conversion is favoured by high voltage and low current. If solar cells are connected in series, the current passing through each cell is the same. Because current is almost linearly dependent on the incident light, the current in a string of identical solar cells will be limited by the cell with the least illumination. For an unshaded flat plate PV system, the illumination across a panel comes directly from the sun and is hence very consistent. However, for a linear concentrator such as the CHAPS system, the longitudinal radiation flux profile along the string of cells is affected by the shape of the mirror, shading due to receiver supports and gaps in the illumination due
to gaps between mirrors. It is quite likely that a single cell will have lower illumination than other cells in the string, and hence limit the current and performance of all cells in the series. Ensuring an even flux profile for all cells is perhaps the largest challenge for the successful design of PV concentrator systems. The following work examines in detail the radiation flux profile of a linear concentrator, both experimentally and with ray tracing simulations, based on a detailed analysis of the optical system.

6.4.1 The ‘Skywalker’ module - measurement of the longitudinal flux profile

A device, known as the ‘Skywalker’ module, was designed and built\(^5\) for measuring the illumination flux profile along the focus of a receiver. The Skywalker module consists of a calibrated concentrator solar cell mounted on an aluminium block and encapsulated with silicone and glass. The short circuit current of the solar cell is measured across a resistor, mounted on the back of the block. Water flows through channels milled into the block to keep the cell at a consistent temperature (figure 6-13), which is measured by a thermocouple mounted in the aluminium behind the cell. Using results from the solar cell calibration, the radiation flux intensity at the cell can be calculated.

![Figure 6-13. Skywalker cell mounting block, showing the 5mm deep cooling channels.](image)

The block is mounted on a trolley that is moved along the focal line of the collector by a motor and pulley system (figure 6-14). A potentiometer measures the position of the trolley.

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\(^5\) The device was built by Jaap den Hartog to drawings developed by the author.
LABVIEW™ is used to control movement of the trolley, collect position and short circuit current data, and process and display the results.

![Figure 6-14. The ‘Skywalker’ module (left), mounted on a trolley at the focus (right).](image)

### 6.4.2 Results from the skywalker module

Measurements of the flux profile were taken for a range of mirrors and a range of incidence angles. The first study looks at trends for all mirrors on the long trough CHAPS prototype.

#### 6.4.2.1 Comparison of mirrors

The measurements were taken on different days at the time when the sun was normal to the trough surface. All 10 mirrors on the long trough CHAPS prototype were measured, with the results shown in figure 6-15. The mirrors were measured with the adjacent mirrors covered and with the adjacent mirrors uncovered.
Figure 6-15. Flux profiles for all mirrors on the long trough CHAPS prototype. The results on the left have adjacent mirrors covered, and on the right all mirrors are uncovered.

A steeper drop-off in light intensity at either end of each mirror is apparent when the adjacent mirrors are covered. Note that at the ends of mirrors 1 and 10 there is no adjacent mirror, so the results are the same between the plots on the left and right in figure 6-15. Clearly each mirror contributes some light to its neighbouring mirror. This effect is discussed in detail in the following sections.

A prominent ‘hump’ is a feature of seven of the ten mirrors, pointing to a systematic problem in the manufacture, rather than a one-off glitch. The hump always occurs at the same relative position (the mirrors are not all the same way around), and has a distinctive shape, with a ‘dip’ in light intensity on either side. It was thought that slight ridges across the mirror in the
transverse direction (analysed experimentally using photogrammetry later in section 6.4.5) might contribute to this hump. It was also thought that the ridges were most likely due to the placement of five ribs used to support the mirror mould. The mirror mould has since been rebuilt with many more ribs and thicker metal. The resulting flux profile when the sun is normal to the mirror is shown below in figure 6-16.

![Flux profile for a mirror from the ‘new’ mould.](image)

The flux profile shows some unusual closely spaced perturbations (believed to be due to ripple in a new batch of glass), but the large ‘hump’ no longer exists. However, all the following analysis is based on measurements of mirrors from the old production process.

### 6.4.2.2 Results from a single mirror for a range of incidence angles

In the case where radiation is incident upon the mirror at an angle away from the surface normal, the effect of the gap between mirrors and the receiver support arms becomes significant. This has been investigated for a range of incidence angles using the ‘skywalker’ device. The results are shown in summary in figure 6-17.
Figure 6-17. Flux profile at the focal line for a range of angles of incidence.

The salient features of the flux profile are seen to move along the focal line as the angle of incidence changes. This is one of the most difficult aspects of single-axis tracking linear PV concentrators. It is not possible to isolate the problem areas and treat them specifically. For example, if the region of low illumination were always at the end of a receiver, then it is likely that overall efficiency would be improved by the absence of solar cells in this region. However, while the deepest dips in the flux profile occur near the ends of the receiver when the sun is near perpendicular, at other times the dips are near the middle of the receiver, and solar cells at the ends do contribute significantly, as can be seen in figure 6-17. The deepest dip in the measured data occurs at an incidence angle of 4.7°, where the minimum flux intensity is 27% lower than the median. Even at 22.1°, where the impact of the gap between mirrors and the receiver supports is being ‘smoothed out’, the dip is 11% lower than the median. Typically the minimum illumination intensity is between 10-20% lower than the median, depending on the incidence angle of light. Given that these regions of low illumination affect the entire receiver performance, it is clear that further investigation is required to understand the precise cause. This is the motivation behind the detailed analysis of the mirror shape, and the ray tracing studies that follow in this chapter.
6.4.2.3  Attenuation of peaks and troughs

The magnitude of peaks and troughs in the illumination profile at the focus could be expected to be attenuated somewhat due to the single short-circuit current reading for the 50 mm long solar cell effectively averaging the illumination. This was investigated experimentally by masking the solar cell at the focus of the ‘skywalker’ device, all bar a narrow slit of 1.5 mm.

Figure 6-18. Comparison between ‘skywalker’ results with a 50 mm long cell and a 1.5 mm long cell.

Figure 6-18 shows the close similarity between the full cell data and the masked cell data with a moving average over 50 mm, and confirms that the flux profile is the same for the two measurements. Significant variation in the illumination profile is observed, particularly in the region under the influence of the mirror gap and support post shading (seen close-up in the bottom graph in figure 6-18). The peaks and troughs in illumination are deep, and it is
therefore fortunate that the 50 mm long cells used in the receiver do provide some illumination ‘smoothing’.

### 6.4.3 Shape error of the mirror

The GOML mirrors are not perfect parabolic troughs. The surface shape of the 4th and 5th mirror panels from the end of the long trough CHAPS prototype system were measured using the photogrammetric method developed by Johnston (1998), with accuracy in displacement estimated to be 20-40 microns. The photogrammetric method involves taking a series of photographs of target points on a mirror from different positions, establishment of reference coordinate system with known target points on the mirror, and complex trigonometric calculations using custom developed software to create 3-dimensional coordinates for the target points. Figure 6-19 shows the shape error of one of the troughs, where the error is the difference between the measured z-coordinates and those calculated for a parabolic trough fitted to the data using a least squares technique.

Maximum deviation from the ideal shape is in the order of 1 mm, and the majority of the mirror surface is within 0.4 mm. The unusual ‘spikes’ in the profile that can be seen on both troughs correspond to places on the mirrors where the mirror glass has delaminated from the backing sheet metal. The delamination occurs only when ‘Galvabond®’ instead of ‘Colorbond®’ is used for the backing sheet metal, and therefore does not occur on the production ready mirrors. However it serves as a graphic example of the accuracy of the photogrammetry process, as the bulges caused by the delamination, while small in magnitude, can be clearly seen on the mirror and show up graphically in the results of the photogrammetry.

---

6 Galvabond® steel is a hot-dipped zinc-coated commercial forming steel, and Colorbond® is a similar steel, but with a painted finish. Both products are manufactured by BlueScope Steel, formerly BHP Steel. Colorbond® steel was not used in pre-production runs as appropriately sized sheets were not available in small quantities.
Figure 6-19. Shape error for adjacent troughs 4 & 5 on the Long CHAPS system.
To examine the impact of shape error of this magnitude, it is instructive to look at the slope error, shown for the two troughs below in figure 6-20.

Figure 6-20. Deviation from a perfect parabolic trough.

The slope error is seen to be high around the perimeter of the trough, where there are peaks around 15 mrad. The distinct ridges in the longitudinal direction that can be seen in figure 6-
19 fall away sharply at the edges of the mirror, and cause a band of transverse slope error on the other side of the ridge, about 200 mm in from the edge. The majority of the mirror surface has a slope error less than 4 mrad.

6.4.4 The effect of slope error on the reflected flux profile

If a mirror was a perfect parabolic trough, and direct light from the sun was perfectly collimated, then the concentrated beam would be a line at the focus. Direct light is not perfectly collimated, as discussed in section 6.2.1. Neither is a real trough a perfect parabola. The following analysis quantifies how slope error in different regions of a trough is manifested at the focus. For the purpose of the analysis, it is assumed that light from the sun is collimated.

The term transverse slope error refers to the slope error oriented across a trough, as shown in figure 6-21. Transverse slope error causes transverse displacement of the light on the focal plane. When light is incident at an angle to the tracking plane, as it is for a single axis tracking system, transverse slope error also causes longitudinal displacement of light at the focal plane.

![Diagram of how transverse slope error relates to transverse displacement of the beam at the focus of a trough.](image)

Longitudinal slope error is slope error oriented in the direction of the trough axis of rotation. Longitudinal slope error causes the greatest displacement error in the longitudinal direction, but also some transverse displacement error. In practice, real slope error is a combination of longitudinal and transverse slope error. Figure 6-22 is a reference chart with the contour lines giving the displacement error. The x-axis gives the position that the light hits the mirror relative to the centre of the mirror, and the y-axis gives the magnitude of the slope error.
Individual graphs show either the longitudinal or transverse displacement error resulting from either longitudinal or transverse slope error. Results for three different angles of incidence of incidence are plotted to show how the problems associated with slope error worsen as the angle of incidence is increased.

The solar cells are 40 mm wide and the mirror width on the long trough CHAPS prototype is 1550 mm with focal length 840 mm. As shown in section 6.2.1, for typical circumsolar ratio values, around 95% of incident radiation comes from a region with half-angle 5 mrad. Figure 6-22 can be used to translate this angle to a 20 mm wide region at the centre of the solar cells.

The mirror slope error at the edges of the mirror (marked ‘A’ on the slope error plots in figure 6-20) is predominantly in the transverse direction. The magnitude of the error is relatively high in this region, often more than 8 mrad, with peaks at 16 mrad. Referring to figure 6-22, the transverse displacement due to a transverse slope error of 8 mrad is around 20 mm, and for a slope of 16 mrad, around 50mm. This means most of the radiation reflecting from the outer edges of the mirror will miss the solar cells.

Longitudinal slope error at the ends of the mirror (marked ‘B’ in figure 6-20) causes large longitudinal displacement of the focal point, between 20 and 50 mm, depending on the transverse position light strikes the mirror and incidence angle. There is also some transverse displacement due to the slope error at the ends, although, with the exception of when light is incident on the corners of the mirror at large angles, most radiation will hit the cells.

The longitudinal bands of transverse slope error located about 550 mm on either side of the centre (marked ‘C’ in figure 6-20) have slope error peaking at around 4 - 8 mrad, which corresponds to a transverse focus error of 10-20 mm. This could be expected to cause a broadening of the flux profile at the focus and some further radiation losses.

Finally, there are a number of transverse bands of longitudinal slope error across the centre of the trough (marked ‘D’ in figure 6-20). These are perhaps a little more visible in the shape error plots in figure 6-19, and typically have slope error usually peaking at around 2 – 4 mrad, and sometimes as much as 6 mrad. These could cause longitudinal focus errors ranging as high as 30 mm, although typically 5 to 15 mm. The cumulative effect of these errors is not simple to interpret, which is the part of the motivation behind the ray tracing in the next section.
Figure 6-22. Displacement error at the focus of a trough resulting from slope error.
6.4.5 Ray tracing – simulation of the longitudinal flux profile

Geometric ray tracing can be used to simulate an optical system such as a parabolic trough collector. OptiCAD, a commercial computer program for ray tracing, has been used to simulate the flux profile at the focus of the two adjacent GOML mirrors analysed in the previous section (OptiCAD Corporation, 2001).

6.4.5.1 Preliminaries

The sun source was modelled in OptiCAD as a ‘pillbox’ shape, with half-angle set to 4.65 mrad. This means that all radiation coming from within the solar disc is assumed to be of equal intensity, and that no radiation originates from outside the solar disc. Figure 6-23 shows a comparison between the flux profile at the focus of a trough from a ‘pillbox’ sunshape and a real sunshape at a circumsolr ratio of 5%. The OptiCAD studies focus on the flux profile longitudinal to the receiver, and for these purposes the pillbox profile is adequate.

![Figure 6-23. The flux image at the focus of a trough is compared for a ‘pillbox’ sunshape and a sunshape with circumsolr ratio (CSR) of 5%.

OptiCAD allows three-dimensional surfaces to be defined then positioned and oriented in a global coordinate system. Predefined shapes such as parabolic troughs and rectangular films can be used to simulate a geometrically ideal concentrator, as shown in figure 6-24. Individual objects may have surface properties defined, such as the degree of reflectivity or the amount of radiation scattering. A non-dimensional study has been carried out, and therefore properties that affect just the magnitude of the reflected flux profile are not important, and are set to the OptiCAD default values.
To simulate the measured mirror shape, a multifaceted mirror (called a polynet in OptiCAD) has been defined. The polynet is made up of continuous groupings of individual triangular polygon facets. The number of facets is determined by a trade-off between the desired optical accuracy and the processing time and software limitations of the program. Figure 6-25 shows the radiation flux profile for a range of faceted mirrors. At least 150-200 facets across the width of the mirror are required to give a flux profile that matches the ideal case with suitable accuracy. For 200 facets, 80% of radiation flux falls underneath the pillbox shaped flux profile and over 90% within the ±7 mm width of the pillbox shape. The remainder of radiation falls within ±12 mm, which is well inside the ±20 mm width of a cell.

Due to constraints in the photogrammetry process, the mirror was measured at around 70 points across the width and 60 points along the length, and therefore some interpolation is
required. Two interpolation schemes were assessed: linear interpolation (first interpolating across the width of the trough, then across the length) and a cubic interpolation, performed using Matlab. Figure 6-26 shows a comparison of the shape error using the two interpolation schemes. The plot shows both a transverse and longitudinal slice of mirror 4 at 600 mm from the edge and end respectively. The difference between interpolation schemes throughout the bulk of the troughs is minimal, and therefore the simpler linear interpolation is employed.

Another constraint in the photogrammetry process meant that it was not possible to measure a row of points right at the very end of the mirror. Unfortunately, the ends of the mirror have the largest slope error, and it was found that the last 50 mm at either end of the mirror has a significant effect on the flux profile in the critical area of lowest flux. Data points were extrapolated using a linear scheme right to the ends of the trough, as shown in blue in figure 6-26. Raw shape data was available for a limited number of intermittent points very close to the end of the trough, and it was found that a linear extrapolation gave a closer fit than a cubic scheme, which tended to overestimate the magnitude of the shape error near the ends.

![Figure 6-26](image_url)

*Figure 6-26. Comparison between interpolation schemes for the shape error on trough 4.*

Whether or not a smoothing scheme should also be used depends very much on the accuracy of the photogrammetry technique. Johnston (1998) estimates the measurement error to be
20–40 μm. Figure 6-27 shows the measured data with error bars of ±30 μm. The plot indicates that the salient features, specifically the jagged troughs and peaks, are outside the error bars, and therefore can not be attributed to measurement error. Therefore no smoothing scheme is employed for the mirror shape data.

![Section at 600mm: Range of uncertainty ±30 μm](Image)

*Figure 6-27. Shape error showing the estimated magnitude of the measurement error.*

In the longitudinal direction, the interval between points is split into three in both the longitudinal and the transverse directions. Therefore the trough is modelled by a ‘polynet’ with around 210 points across the width and 180 points across the length. A cell target 40 mm wide is placed at the focal point, and a further sheet simulating the receiver cover is placed slightly above the focal point. The gap between the mirrors on the CHAPS troughs has been kept as small as possible, but there remains a 19 mm gap between the mirror glass. The receiver support arms are made from 10 mm x 25 mm steel bar, supported from outside the mirror, as shown in figure 6-28.

![Shading sheet Target Receiver support Mirror](Image)

*Figure 6-28. Support arm geometry.*

Computational time on a Pentium4 computer was around a day for each run. At least 5 million rays were required to achieve suitable resolution from the results of each ray trace.
However, the main factor that increased computational time was size of the many-faceted ‘polynet’ that made up each mirror. For this reason, only half of each adjacent mirror was modelled (resulting in around 37,000 individual triangular facets per trough), and the extent of the rays was limited such that a flux profile was obtained at the target between 0 mm and 600 mm from the end of the trough for a range of sun angles between 0° and 20°. This part of the receiver is by far the most interesting as it nearly always contains the region of lowest illumination.

### 6.4.5.2 Validation of the ray tracing

A comparison is made between the flux profile predicted by ray tracing and the measured flux profile for a ‘real’ GOML mirror shape in figure 6-29. Also plotted in figure 6-29 is the flux profile resulting from a ‘perfect’ parabolic mirror shape.

![Comparison between the measured and predicted radiation flux profiles.](image)
The flux profile created by ray tracing using the ‘real’ mirrors shows reasonable agreement with the measured profile. The magnitude of the peaks are not perfectly matched. However, given the sensitivity of this area to the mirror shape at the ends of the mirror, the error is to be expected. Importantly, the peaks and dips coincide in position for the simulated and measured data, and the magnitude of the deepest dip is reasonably well predicted. It can be concluded that the ray tracing is a useful tool for predicting flux profile if the measured mirror shape is used, and that the observed variations in the flux profiles are indeed a result of mirror shape imperfections.

6.4.5.3 Analysis

The results show a significant difference between ray tracing from a perfect parabolic mirror and from a ‘real’ mirror. Figure 6-29 shows that where there is a dip in the ‘perfect’ trough profile, there is a peak in the ‘real’ trough profile. The dip in the ‘perfect’ trough profile corresponds to the position where one would expect a dip based on the position of the sun and the gap between mirrors. However, as both the measured data and the modelled data using the ‘real’ trough shape show, there is a peak. The reason for the peak is that the significant slope error in a narrow margin at the ends of the troughs (seen previously in figure 6-20) creates a kind of ‘pseudo-focus’ in the longitudinal direction, shown diagrammatically in figure 6-30.

![Diagram of pseudo-focus](image)

Figure 6-30. Pseudo-focus in the region between adjacent mirrors.

The aim of the ray tracing simulation was to identify the cause of the non-uniformities in the measured flux profile, and in particular, to determine what causes the deepest dip in the profile, as this has the most effect on the overall electrical performance of a PV concentrator. Figures 6-31a-d show the results of comparisons between a perfect parabolic trough (the dotted lines) and the simulated ‘real’ trough for all combinations of receiver supports and gaps between mirrors.
Figure 6-31a. The flux profile without a gap between mirrors and without receiver supports.

Figure 6-31b. The flux profile without a gap between mirrors and with a receiver support.

Figure 6-31a shows the flux profile that could be expected if there was no gap between the mirrors and no receiver support. For all angles, a pronounced hump can be seen in the flux profile for the position corresponding to the end of the mirror (i.e. for 0°, 4°, 8°, 12°, 16°, 20° the positions are 0, 59, 119, 180, 242 and 308 mm from the end of the receiver respectively), further demonstrating the effect of the pseudo-focus due to the slope error at the ends of the mirror. However, an interesting corollary can be drawn by direct comparison of figure 6-31a and 6-31c, as shown in figure 6-33. In all cases the illumination at the point corresponding to the gap between the mirrors is not the minimum. Moving the mirrors apart has the effect of reducing the peak at this point. Eventually, of course, if the gap is large enough then there will be a corresponding gap in the flux profile.
Modelling indicates that for the mirrors measured, a further 10 mm gap would be possible before the concentrating effect of the pseudo-focus due to the sloped mirror ends is negated (figure 6-32). However, while the general shape of the curve on either side of the hump in figure 6-33 is similar for the gap and no gap cases, there is a slight decrease in the magnitude of the lowest dip when there is a gap. Therefore there remains some advantage in further minimising the gap.
The effect of the receiver support can be seen in figures 6-31b and 6-31d, but is a little obscured by the interference due to the mirror end effects. Unlike for a ‘perfect’ mirror, the shading due to the support for ‘real’ mirrors is significant, particularly for incidence angles around 4° to 8°. An unavoidable consequence of the pseudo-focus at the mirror ends is that there will be a region on either side of the hump that has lower illumination than average. The radiation has to be ‘stolen’ from somewhere, and in effect light that is incident on this curved section at the end of the mirror is spread out along a greater length of the receiver, and hence the concentration ratio is reduced where there is no superposition of light from the adjacent mirror. When there is no receiver support, the dip is larger on the side nearest the mirror end (figs 6-31a and 6-31c). When there is a receiver support, for incidence angles up to about 10°, the coincidence of the position of the dip on the side of the hump furthest from the mirror end and the shade due to the receiver support causes the minimum radiation in the flux profile (figs 6-31b and 6-31d). For larger angles, the impact of the receiver support shading ‘smears’ along the focus and becomes less significant, and the largest dip occurs on the other side of the hump closer to the mirror end.
6.4.6 Improvement of the flux profile

The results of the ray tracing indicate that further reduction in the gap between mirrors would marginally improve the electrical performance of the CHAPS collector. However, the mirrors are held in shape at their ends by tabs stamped into sheet metal, and it is therefore not possible to reduce the distance between mirrors with the current design. The EUCLIDES™ system supports the mirrors from behind, and is therefore able to achieve an average of 4 mm spacing between mirrors (Anton et al., 2000). However, the stamped tab rib design is integral to the low cost and high optical accuracy design of the mirrors for the CHAPS collectors. The results above show that some convex curvature in the mirror shape at the ends improves the flux profile around the mirror gap (when compared to the perfect mirror), but that the convexity is a little too severe. Investigations continue into methods of optimising the mirror shape in the manufacturing process to more precisely compensate for the mirror spacing.

The potential of secondary flux modifiers to improve the homogeneity of the flux profile near the focus has yet to be investigated. Secondary flux modifiers can also reduce light spillage where the tolerance of tracking or mirror alignment is an issue. It is suggested that a simple
secondary flux modifier could be fabricated from the insulation cover (figure 6-1). Various
designs for secondary flux modifiers could be simulated prior to fabrication using the ray
tracing model described in this thesis. It is suggested that this be the subject of future work.

![Figure 6-1. A suggestion for a simple secondary flux modifier fabricated from the aluminium insulation cover.](image)

### 6.5 Mitigation of the effect of an uneven flux profile

One possibility for mitigating the effect on electrical performance of regions of low
illumination is to use higher quality solar cells in these areas. Figure 6-31 shows that the
lowest dips in illumination occur at small incidence angles in the region close to the end of
the receiver. Therefore, cells with higher than average maximum power currents should be
installed in these regions. In the early production runs of the ANU solar cells it has been
found that the maximum power current at 30 suns and 65°C has roughly a normal distribution
with an average current of 18.6 A and a standard deviation of 0.35 A. The distribution is not
very wide. However, the best cells at the ends of the receivers are compared to the worst cells
in the centre, and therefore there should be a few cells at either end of the receiver around 5%
better than the worst cell in the centre.

### 6.6 Bypass diodes

Bypass diodes are usually applied to protect shaded solar cells from breakdown under reverse
bias. Woyte et al. (2003) present a literature review of work in the area of optimising PV
module design and determining the maximum number of solar cells per bypass diode to avoid
the formation of hot spots. A receiver is made up of a string of cells in series, and therefore
all cells must carry the same current. If one cell in the string is partially shaded, then both
the maximum power point current and the short circuit current of this cell will be reduced. If
the complete string continues to operate at the maximum power point, then current will reduce to a value close to the short circuit current of the partially shaded cell. If the current increases just slightly above this point, then the voltage of the partially shaded cell will move into reverse bias, potentially reducing the power output of the string to nearly zero, and risking destruction of the reverse biased cell. For example, consider the I-V curves in figure 6-2. If there are ten cells in series, nine illuminated at 30 suns, and one at 20 suns, then at open circuit the reverse bias on the partially shaded cell will be approximately equal to the open circuit voltages of the other nine cells combined.

![Figure 6-2. I-V curves for cells illuminated at 30 suns and 20 suns.](image)

If there are enough cells in series, for example 26 in a typical CHAPS receiver, then the reverse bias is as high as 17 V. Testing of the ANU concentrator cells shows that a sustained reverse bias of that order causes hot spots and irreversible damage to the cell. A conservative approach would be to limit the maximum reverse bias to around 10 V. Bypass diodes can be employed to shunt current around a shaded cell. A bypass diode made of a silicon pn junction drops about 0.7 V in order to carry a current equal to the maximum power current of a receiver, which is equivalent to the voltage gain of a solar cell. Therefore, it is often better to group solar cells into mini-strings with a single bypass diode, rather than lose 0.7 V for each cell that is shaded. Schottky diodes, which are made by bonding a metal such as aluminium or platinum to n-type silicon, have a turn-on voltage of 0.3 V and are therefore a better option as bypass diodes. However, the use of bypass diodes is not simply for protection of cells. As discussed in section 6.4, non-uniform illumination for single axis tracking systems is virtually unavoidable. Bypass diodes may be used to optimise the annual output of a system by shunting out sections of an array that are being shaded and limiting the output of the full array.
6.6.1 Bypass diodes for two axis tracking systems

For a two-axis tracking system such as the domestic CHAPS system, there is no shading that affects one cell more than another. However due to the shape error at the ends of the mirror there is some stretching of the flux profile near the ends, which, if the receiver is the same length as the mirror, results in a drop in light intensity on the end few cells. Investigations were carried out comparing the performance of a receiver with groups of diodes to one with a single diode per cell. The tests were carried out with a 1.6 m long receiver on a 1.6 m long and 1.25 m wide mirror. One of the receivers had four diodes across four strings in groups of 9, 8, 8 and 6 cells. Figure 6-3 shows the individual I-V curves for the four diode sections when they are separated.

![Figure 6-3. IV curves for diode sections of a CHAPS receiver.](image)

The open circuit voltages are different due to the number of cells in series in each string. However, if the illumination was consistent for the entire receiver the short circuit current should be the same for all strings. Strings 1 and 4 show significantly lower short circuit current, which demonstrates the effect of the spillage of light off the ends of the receiver. When the four cell strings are connected in series, they form an I-V curve with ‘kinks’, as shown on the left hand side in figure 6-4. The kinks correspond to points on the curve where the bypass diodes turn on. There are three distinct kinks, which means that three diodes have switched on and that the short circuit current is equivalent to the short circuit current of the only remaining and most strongly illuminated string. Note that the same receiver was tested using the 20 sun tester and observed to have no kinks. Therefore it can be concluded that they are due entirely to the uneven illumination profile caused by the mirror. Removal of a couple of cells from either end of the receiver proves almost neutral in terms of performance (because the drop in total voltage is offset by the increase in the receiver current) and has the advantage of a reduction in receiver cost.
Figure 6-4. IV curves of receivers with 4 diode strings (left) and a diode per cell (right).

On the right hand side of figure 6-4 is an IV curve for the receiver that has a bypass diode for each cell. Note that open circuit voltage is much lower for this receiver as the operating temperature was higher, at 80°C compared to 32°C. For this receiver, the kinks in the curve are still apparent, but because only one cell is dropping out for each kink, the step in current at each kink is much smaller. Some diodes switch on in the flat, almost horizontal, part of the curve but do not have much impact on the overall shape. The least illuminated cells at the ends of the receiver cause the most noticeable kinks near the ‘knee’ of the curve, and therefore have the most effect on the maximum power, as can be seen on the corresponding power curve.

6.6.1 Bypass diodes for single axis tracking systems

For a single axis tracking system, the region of low illumination shifts as the sun angle changes, and therefore design of a diode arrangement to optimise the overall performance is difficult. However, some minor gains are possible. As was shown in figure 6-31, the region of a receiver with the lowest illumination tends to be near the ends. Therefore diodes have been incorporated across three cells at either end of the CHAPS receivers to allow this region to be bypassed if the overall performance is significantly affected. A maximum power point tracker (MPPT) determines the operating current and voltage for an array. If the end cells have significantly lower than average current, then it is possible that the maximum power current will be closer to the current of the unshaded cells, and hence the end diode will switch on. However, while there is an increase in current, there is a reduction in voltage due to the missing cells and the bypass diode voltage drop, and therefore in practice, the diode only comes on if the minimum illumination on these cells is significantly lower (~12.5%) than the least illuminated cell in the remainder of the receiver. Exclusion of cells from the regions near the end of mirrors has no positive effect, and in fact the mirrors would be better utilised if
there were cells in the positions corresponding to the gaps between the mirrors. Receivers with double normal length have been investigated, but at this stage the improvement in system performance does not justify the extra manufacturing complexity and cost.

Due to the geometry of a single axis tracking concentrator, for much of the day, the receivers at the end of the array will be partially illuminated. Rather than lose the output of a whole receiver, it is better to have bypass diodes positioned across smaller groups of cells. A diode plan with five diodes is used for the CHAPS receivers (3-6-8-6-3 across the 26 cells), which allows a standard manufacturing technique for each receiver, and a reasonable balance between performance and cost.


7.1 The new PV/T collector component

The standard TRNSYS library contains a PV/Thermal component (Type 50). The model is based on modifications to the Hottel-Whillier-Bliss equations that are used for the standard Type 1 flat plate solar collector (Florschuetz, 1979). However, it does not account for radiation losses and has no thermal capacitance, and is therefore not considered to be detailed enough to simulate a CHAPS collector. It is also an empirical model and therefore it is difficult to model variations in physical sub-components of the PV/T collector. According to personal communication with Miroslav Bosonac from the Danish Teknologisk Institut (2/10/2001), a flat plate PV/T model has been developed for use by consultants in Denmark designing buildings incorporating PV/T systems, such as the PV/VENT projects (Pedersen, 2000). However, there are no existing components that can adequately simulate the CHAPS collector, and consequently a detailed analytical PV/Thermal TRNSYS component has been written (Type 262), based closely on the analytical equations outlined in this thesis.

7.1.1 Theoretical formulation

The following section describes the theoretical formulation of the TRNSYS PV/T component. Much of the detail is based on equations and assumptions described in detail in chapters 2 and 5. The Fortran code for the model is attached in section 1 of appendix A. Equation 7-1 describes the change in temperature of the fluid in the receiver element with respect to time,

\[
C_{p-col} \frac{dT}{dt} + \dot{m}_f c_{p-f} (T - T_{inlet}) = \dot{Q}_{th}(T)
\]

(7-1)

where \( c_{p-f} \) is the specific heat of the fluid, \( \dot{m}_f \) is the fluid mass flow, \( T \) and \( T_{inlet} \) are the outlet and inlet fluid temperatures, and \( \dot{Q}_{th}(T) \) is the net energy flow. Because the receiver is made up of many materials and contains water, the specific heat term for the collector \( C_{p-col} \)
is an average term, weighted by the masses of each material, as per equation 7-2, where $m_i$ and $c_i$ are the mass and specific heat terms for $i$ elements that make up the receiver.

$$C_{p\text{-col}} = \sum_{\text{col}} m_i c_i \tag{7-2}$$

An inherent assumption with this simplified first order equation is that the change in outlet temperature is independent of position. In other words, there is no allowance for the delay in fluid transport from one end of the receiver to the other. This assumption is discussed later in section 7.2.2.

### 7.1.2 Discrete element model

The model allows the collector to be divided up into a series of elements along its length. The primary reason is because electrical output is dependent on temperature, and for certain installations there may be a significant temperature difference between the inlet and outlet. As electrical efficiency reduces along the length of the trough, there is more energy available for thermal conversion. It is assumed that each element can be characterized by a single temperature $T$, which in the model, is taken as the average of the inlet and outlet temperatures of the element at the previous time step. Equation 7-1 is derived from the energy balance of a control element, taking into account:

- The change in energy content of the element
- The energy transfer by the fluid flow
- The temperature dependent energy flow between the element and surrounding
- A line heat source.

Although the thermal energy flow $\dot{Q}_{th}(T)$ is a function of temperature, which changes with time, a numerical approach to the solution of the equation is to treat $\dot{Q}_{th}(T)$ as a constant. The value of $\dot{Q}_{th}$ is calculated based on using the average temperature of the element a short time earlier. Therefore, equation 7-1 can be rearranged to form a first order differential equation. This method of solution of the energy balance equation is suggested in the TRNSYS reference manual (Solar Energy Laboratory, 2000) for components with a temperature response dependent on time.
\[
\frac{dT}{dt} + AT = B \quad \text{where} \quad A = \frac{\dot{m} f c_p}{c_{p-col}} \quad \text{and} \quad B = \frac{\dot{Q}_{th} + \dot{m} f c_p T_{inlet}}{c_{p-col}}
\] (7-3)

Solving T with respect to time t gives:

\[
T = T_{initial} e^{At} + \frac{B}{A} e^{At} - \frac{B}{A}
\] (7-4)

where \(T_{initial}\) is the outlet temperature at a starting time t. At some small time interval later \(t+\Delta t\), the outlet temperature \(T_{final}\) can be expressed:

\[
T_{final} = T_{initial} e^{A\Delta t} + \frac{B}{A} e^{A\Delta t} - \frac{B}{A}
\] (7-5)

The average outlet temperature of the element over the period of t to \(t+\Delta t\) is calculated in order to determine the inlet temperature of the next element. This can be obtained by integrating equation 7-5 as follows:

\[
\overline{T} = \frac{1}{\Delta t} \int_{t}^{t+\Delta t} T_{inlet} e^{At} + \frac{B}{A} e^{At} - \frac{B}{A}
\] (7-6)

which has solution:

\[
\overline{T} = \frac{1}{A\Delta t} \left( \frac{B}{A} + T_{initial} \right) \left( e^{A\Delta t} - 1 \right) - \frac{B}{A}
\] (7-7)

### 7.1.3 Energy balance

The value of \(\dot{Q}_{th}(T)\) can be calculated by solving a set of non-linear equations that physically describe the temperature dependent energy flow between the element and the surroundings. These equations are often represented as an equivalent thermal network, for example the combined PV/thermal networks described by Chow (2003) and Mattei et al. (1998). The thermal network shown in figure 7-1 is used to describe energy flow in the CHAPS receiver. To simplify the calculation, it is assumed that the network has discrete layers of constant
temperature, that is the tube, plate, solar cells and insulation cover. The glass layer (which also incorporates the silicone potant) is divided into two layers so that energy input due to absorption of light through the optical material can be accounted for. The thermal network can be used to describe the two main receiver designs discussed in chapter 4, the copper pipe bonded to an aluminium extrusion, and the full aluminium extrusion. For a full aluminium extrusion, the absorber plate and tube are a single component and can be represented as a single node in the network below. In this case the thermal resistance between the two nodes in the model is set to a negligible value.

![Thermal network diagram]

Figure 7-1. Thermal network describing a PV/T concentrating collector.

### 7.1.4 Inputs

Solar radiation is the only energy input. The radiation is divided up into the fraction that is absorbed in the cover materials, and the remaining fraction that is absorbed by the solar cells. It is assumed that the radiation flux is evenly distributed across the receiver. A simplifying assumption is made regarding the energy absorbed in the cover glass and silicone potant. It is assumed that all radiation is absorbed at a point half-way through the cover, rather than gradually throughout. The solar radiation incident on the receiver $\dot{Q}_{\text{sun}}$ is given by:
\[
\dot{Q}_{\text{sun}} = \dot{G}_D \cdot A_m \cdot \rho_m \cdot F_{\text{shape}} \cdot F_{\text{dirt}} \cdot F_{\text{shade}}
\]  

(7-8)

where \( \dot{G}_D \) is the total direct beam radiation over the non-shaded mirror area \( A_m \) with reflectivity \( \rho_m \). Scaling factors are introduced to account for the shape accuracy \( F_{\text{shape}} \) and dirtiness \( F_{\text{dirt}} \) of the mirrors. In a collector field, there is often shading from other mirrors, which can be introduced as an input \( F_{\text{shade}} \). The shape error is estimated based on flux mapping of a similar mirror to that on the CHAPS test rig. From the flux mapping, it is calculated that the percentage of light captured within a 38 mm wide target that is moved from one side of the focal beam to the other (figure 7-2) is around 99%.

![Figure 7-2. Percentage of light captured within a 38mm wide target that is moved from one side of the focal beam to the other. Raw data for graph from Greg Burgess.](image)

The radiation absorbed by the cells \( \dot{Q}_{\text{abs-cells}} \) accounts for transmission and absorption by the transmission-absorption product (\( \tau \alpha \)). The component of radiation absorbed in the cover is removed (\( 1-\alpha_g \)).

\[
\dot{Q}_{\text{abs-cells}} = \dot{Q}_{\text{sun}} \cdot (\tau \alpha) \cdot (1-\alpha_g)
\]  

(7-9)

Similarly, the absorption in the cover materials \( \dot{Q}_{\text{abs-glass}} \) accounts for the remaining solar radiation not reflected from the cover:

\[
\dot{Q}_{\text{abs-glass}} = \dot{Q}_{\text{sun}} \cdot (\tau \alpha) \cdot \alpha_g
\]  

(7-10)
To simplify the model, light incident on the back of the receiver is assumed not to affect the surface temperature. Simulations using the Strand7 model show that this introduces error of around 0.6% to the overall thermal output.

### 7.1.5 Outputs

The thermal output $\dot{Q}_{th}$ is the energy transferred into the water (not including the effect of thermal capacitance):

$$
\dot{Q}_{th} = h_{c-w} A_f [T_{tube} - T_f]
$$

(7-11)

where $T_{tube}$ and $T_f$ are the temperatures of the tube and fluid respectively, and $A_f$ the surface area of the inside of the tube. The empirical correlation that defines the coefficient of convection $h_{c-w}$ is defined in chapter 5, as is all the nomenclature. The key equations are repeated below:

$$
h_{c-w} = \frac{F_H Nu D_h k}{D_h}
$$

(7-12)

where the Nusselt number is given by the Dittus-Boelter equation:

$$
Nu_D = 0.023 \cdot Re^{0.8} Pr^{0.4}
$$

(7-13)

and modified to account for the internal fins according to the Carnavos relation:

$$
F_H = \left( \frac{A_{xs}}{A_n} \right)^{0.1} \left( \frac{P_n}{P} \right)^{0.5} = 0.74
$$

(7-14)

Electrical output $\dot{Q}_{elec}$ is derived from the simplified maximum power output expression given in Wenham et al. (1994):

$$
\dot{Q}_{elec} = \dot{Q}_{sun} \cdot \eta_{ref} \cdot \exp\left(\beta(T_{cells} - T_{ref})\right) \cdot F_{uniformity}
$$

(7-15)
where $\beta$ is the temperature coefficient giving the relationship between solar cell efficiency and temperature, and $T_{\text{cells}}$ the temperature of the solar cells. The scaling factor $F_{\text{uniformity}}$ combines the effect of non-uniformities both transverse and longitudinal to the receiver, as discussed below. This scaling factor is more valid when simulations are made over a range of operating conditions.

Sometimes a simpler linearised relationship is used; however, this becomes more inaccurate at the higher temperatures possible with a concentrating PV/T collector. The reference efficiency $\eta_{\text{ref}}$ is measured at a reference temperature $T_{\text{ref}}$, usually 25°C.

It is assumed in the TRNSYS model that the illumination is even across the width of the receiver. The Strand7 modelling in figure 7-3 compares the receiver temperature profile of a receiver under this assumption with the base case model from section 5.6, and figure 7-4 shows the temperature of the glass surface for both cases.

![Figure 7-3. Temperature profiles for gaussian (left) and even (right) illumination and absorption profiles.](image)

![Figure 7-4. Glass surface temperatures for gaussian (left) and even (right) illumination and absorption profiles.](image)
The assumption of uniform flux distribution causes some error in the calculation of electrical output. As was shown in section 6.3, non-uniform illumination across the cell causes a reduction of the fill factor and the open-circuit voltage. The increased high temperature in the region of maximum current generation (the centre of the cell) also contributes to reduction in efficiency. Overall, the reduction in electrical efficiency due to non-uniform illumination across the cell is around 5-15% relative depending on the temperature of the cells. A simple scaling factor on electrical efficiency is used to account for non-uniform illumination. The value of the scaling factor is assumed to be 0.9, which accounts for a typical range of operating temperatures.

As discussed in section 6.4, flux non-uniformities in the longitudinal direction cause variation in the electrical performance at different times of the day dependent on the sun angle. For the purposes of the model it is assumed that the losses can be represented by a single scaling factor. This is only valid when a range of sun angles occur in the period of the simulation. The scaling factor is estimated to be 0.85.

### 7.1.6 Thermal losses

Radiation losses $\dot{Q}_{rad}$ from the glass surface were discussed in section 5.4.1. Equation 5-16 (repeated below) is used in the TRNSYS component to determine the losses based on the calculated mean glass temperature.

$$\dot{Q}_{rad} = \varepsilon_g \cdot \sigma \cdot A_g \cdot \left(T_g^4 - T_{amb}^4\right) \tag{7-16}$$

As can be seen in figure 7-4, the actual glass temperature profile varies significantly from the mean glass temperature. It was calculated previously (section 5.4.1.2) that using the average glass temperature rather than the real distributed temperature in the calculation of radiation loss results in error around 1.7%, or less than half a watt.

Convection loss from the cover $\dot{Q}_{conv}$ can be calculated analytically, as per the analysis in section 5.4.2. However, a simpler empirical approach is used in the TRNSYS component, where the coefficient of convection $h_{c-a}$ is estimated based on the wind speed $u_{wind}$ using a second order polynomial fit of the calculated analytical data:

$$\dot{Q}_{conv} = h_{c-a} A_g \left(T_g - T_{amb}\right) \tag{7-17}$$
where: 
\[ h_{c-a} = c_0 + c_1 u_{\text{wind}} + c_2 u_{\text{wind}}^2 \] (7-18)

Initial estimates of the coefficients \( c_0, c_1 \) and \( c_2 \) are based on calculations using equations 2-8 to 2-13, and 5-17, and the assumptions that the wind blows directly across the receiver angled at 50° from horizontal, the glass temperature is 70°C and the ambient temperature is 25°C.

Heat losses through the insulation \( \dot{Q}_{\text{ins}} \) are given by:

\[ \dot{Q}_{\text{ins}} = U_{\text{ins}} A_{\text{ins}} (T_{\text{plate}} - T_{\text{cover}}) \] (7-19)

where it is assumed that the heat transfer coefficient \( U_{\text{ins}} \) remains reasonably constant within the range of operating temperature, and that the insulation thickness is uniform. \( A_{\text{ins}} \) is the area of insulation, and \( T_{\text{plate}} \) is the temperature of the absorber plate surrounded by the insulation.

Losses from the insulation cover are by means of convection and radiation. To simplify the model it is assumed that convection losses from the cover have the same coefficient of convection as for the glass (equation 7-18). While this is not true for natural convection due to buoyancy forces, it is a valid assumption for forced convection due to wind, which tends to dominate the heat transfer.

\[ \dot{Q}_{\text{conv}}^{*} = h_{c-a} A_{\text{cover}} (T_{\text{cover}} - T_{\text{amb}}) \] (7-20)

The radiation losses from the insulation cover are calculated using a linear expression, where equation 7-16 is simplified to the form:

\[ \dot{Q}_{\text{rad}}^{*} = \varepsilon_{\text{cover}} \cdot \sigma \cdot A_{\text{cover}} \cdot (T_{\text{cover}} - T_{\text{amb}}) (T_{\text{cover}}^2 + T_{\text{amb}}^2) \] (7-21)

which is approximated by:

\[ \dot{Q}_{\text{rad}}^{*} = 4 \cdot \varepsilon_{\text{cover}} \cdot \sigma \cdot A_{\text{cover}} \cdot \overline{T}^3 (T_{\text{cover}} - T_{\text{amb}}) \] (7-22)

where the mean of the ambient and insulation cover temperatures \( \overline{T} \) can be estimated with reasonable accuracy as the temperature difference is never large. This simplification allows
the thermal network to be solved much more easily, as both modes of heat loss from the cover are expressed as a function of the temperature difference between the cover and ambient.

7.1.7 Other relationships

Other relationships required to determine $\dot{Q}_{th}$ are:

$$\dot{Q}_{th} = U_{pt} A_{pt} (T_p - T_i) \quad (7-23)$$

where $U_{pt}$ and $A_{pt}$ are the heat transfer coefficient and contact area of the joint between the plate and tube in a receiver. The heat transfer coefficient would be set to a very high value if the there is no separate plate and tube, such as for the CHAPS extruded receiver.

The TRNSYS model works by solving the energy balance at each node in figure 7-1. Energy transfer through the glass is given by the following equations:

$$\dot{Q}_{cg'} = U_{cg'} A_{cg} (T_{cells} - T_{g-mid}) \quad (7-24)$$

$$\dot{Q}_{cg''} = U_{cg''} A_{cg} (T_{g-mid} - T_g) \quad (7-25)$$

$$U_{cg'} = U_{cg''} = 2U_{cg} \quad (7-26)$$

where $\dot{Q}_{cg'}$ and $\dot{Q}_{cg''}$ are the energy transfers through the glass (from the cells to the mid-point, and the mid-point to the surface respectively). The heat transfer coefficients $U_{cg'}$ and $U_{cg''}$ are set to twice the value of the heat transfer coefficient through all the cover materials $U_{cg}$. Therefore the artificial ‘mid-point’ for absorption is thermally but not necessarily geometrically in the middle of the cover when multiple cover materials are used. The magnitude of the absorption at the midpoint $\dot{Q}_{abs}$ is calculated to be the same as the total of distributed absorption throughout the glass-silicone cover.
7.1.8 Incidence angle modifiers

The PV/T model does not account for changes in reflectivity at different incidence angles of light. Figure 7-5 shows the annual losses due to reflection from a CHAPS receiver, compared to the light incident upon the receiver. The chart is divided into bins of 5°, where 0° is normal to the trough. It is assumed that the single-axis tracking trough is located in Canberra, and that it is oriented with the long axis running east-west. The losses are calculated by multiplying the energy incident on the receiver for a particular range of angles by the average reflectivity in that range.

![Graph showing annual losses due to reflection compared to total incident light, for a range of incidence angles on an east-west oriented CHAPS collector located in Canberra.](image)

Figure 7-5. Annual losses due to reflection compared to total incident light, for a range of incidence angles on an east-west oriented CHAPS collector located in Canberra.

It was shown previously in figure 4-18 that reflection at an air-glass interface begins to increase dramatically at incidence angles more than about 45°. Figure 7-5 shows that while the amount of radiation incident upon the receiver declines at higher incidence angles, the steep increase in reflection causes significant reflection losses at these higher angles. The error in total absorbed radiation caused by neglecting the changing value of reflection and using the value of reflection normal to the glass is around 1.4% for the location and orientation described above. This is significant, given that reflection losses (including the effect of incidence angle) are around 5.9% annually. It is recommended that the inclusion of incidence angle modifiers in the TRNSYS model be the subject of future work. However, for the purposes of validation of the model, incidence angle modifiers were not required as radiation was incident normal to the trough.
Table 7-1. TRNSYS parameters used in the validations.

<table>
<thead>
<tr>
<th>Parameter name in TRNSYS</th>
<th>Parameter description</th>
<th>Value (SI units)</th>
<th>Value (TRNSYS units)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Mode</strong></td>
<td>Coefficient of convection is:</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td>1 = user specified; 2 = calculated explicitly</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Cells</strong></td>
<td>Divide receiver into this many nodes</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td><strong>Reff</strong></td>
<td>$\eta_{\text{ref}}$ Reference PV efficiency</td>
<td>0.1553</td>
<td>0.1553</td>
</tr>
<tr>
<td><strong>Reftemp</strong></td>
<td>$T_{\text{ref}}$ Reference temperature for PV efficiency</td>
<td>65°C</td>
<td>65°C</td>
</tr>
<tr>
<td><strong>Beta</strong></td>
<td>$\beta$ Temperature coefficient for PV efficiency</td>
<td>-0.0035</td>
<td>-0.0035</td>
</tr>
<tr>
<td><strong>Uniformity</strong></td>
<td>$F_{\text{uniform}}$ Scaling factor to adjust electrical efficiency</td>
<td>0.845</td>
<td>0.845</td>
</tr>
<tr>
<td><strong>Length</strong></td>
<td>Length of the troughs</td>
<td>1.465 m</td>
<td>1.465 m</td>
</tr>
<tr>
<td><strong>Width</strong></td>
<td>Width of the mirror (unshaded)</td>
<td>1.17 m</td>
<td>1.17 m</td>
</tr>
<tr>
<td><strong>Rfm</strong></td>
<td>$\rho_m$ Reflectivity of the mirror</td>
<td>0.935</td>
<td>0.935</td>
</tr>
<tr>
<td><strong>Shape</strong></td>
<td>$F_{\text{shape}}$ Optical accuracy of the mirror</td>
<td>0.99</td>
<td>0.99</td>
</tr>
<tr>
<td><strong>Transabs</strong></td>
<td>$(\alpha)$ Transmission-absorption coefficient</td>
<td>0.886</td>
<td>0.886</td>
</tr>
<tr>
<td><strong>Coverabs</strong></td>
<td>$\alpha_g$ Fraction of light absorbed in the cover</td>
<td>0.063</td>
<td>0.063</td>
</tr>
<tr>
<td><strong>Emis</strong></td>
<td>$\varepsilon_g$ Emissivity of the glass surface</td>
<td>0.88</td>
<td>0.88</td>
</tr>
<tr>
<td><strong>Mass</strong></td>
<td>Mass of the receiver/liquid</td>
<td>7.7 kg</td>
<td>7.7 kg</td>
</tr>
<tr>
<td><strong>Cp</strong></td>
<td>Specific heat of the receiver/liquid</td>
<td>1.08 kJ.kg$^{-1}$.K$^{-1}$</td>
<td>1.08 kJ.kg$^{-1}$.K$^{-1}$</td>
</tr>
<tr>
<td><strong>Tstart</strong></td>
<td>Initial temperature of the receiver/liquid</td>
<td>37.7°C</td>
<td>37.7°C</td>
</tr>
<tr>
<td><strong>Wglass</strong></td>
<td>Width of the glass cover</td>
<td>0.08 m</td>
<td>0.08 m</td>
</tr>
<tr>
<td><strong>Ucg</strong></td>
<td>$U_{cg}$ U value between cells and glass</td>
<td>90.8 W.m$^{-2}$.K$^{-1}$</td>
<td>327 kJ.hr$^{-1}$.m$^{-2}$.K$^{-1}$</td>
</tr>
<tr>
<td><strong>Wcg</strong></td>
<td>Width of interface between cells and glass</td>
<td>0.07 m</td>
<td>0.07 m</td>
</tr>
<tr>
<td><strong>Ucp</strong></td>
<td>$U_{cp}$ U value between cells and plate</td>
<td>1646 W.m$^{-2}$.K$^{-1}$</td>
<td>5926 kJ.hr$^{-1}$.m$^{-2}$.K$^{-1}$</td>
</tr>
<tr>
<td><strong>Wcp</strong></td>
<td>Width of interface between cells and plate</td>
<td>0.04 m</td>
<td>0.04 m</td>
</tr>
<tr>
<td><strong>Upt</strong></td>
<td>$U_{pt}$ U value between plate and tube</td>
<td>277,777 W.m$^{-2}$.K$^{-1}$</td>
<td>1,000,000 kJ.hr$^{-1}$.m$^{-2}$.K$^{-1}$</td>
</tr>
<tr>
<td><strong>Wpt</strong></td>
<td>Width of interface between plate and tube</td>
<td>1 m</td>
<td>1 m</td>
</tr>
<tr>
<td><strong>Uinsul</strong></td>
<td>$U_{\text{insul}}$ U value for the insulation</td>
<td>8.62 W.m$^{-2}$.K$^{-1}$</td>
<td>31 kJ.hr$^{-1}$.m$^{-2}$.K$^{-1}$</td>
</tr>
<tr>
<td><strong>Winsul</strong></td>
<td>Perimeter length for cal. insulation loss</td>
<td>0.147 m</td>
<td>0.147 m</td>
</tr>
<tr>
<td><strong>Wcover</strong></td>
<td>Perimeter length for cal. convection and radiation losses from the cover</td>
<td>0.2 m</td>
<td>0.2 m</td>
</tr>
<tr>
<td><strong>Emiscover</strong></td>
<td>$\varepsilon_{\text{cover}}$ Emissivity of the insulation cover</td>
<td>0.1</td>
<td>0.1</td>
</tr>
<tr>
<td><strong>Cw0</strong></td>
<td>$c_0$ Coefficients for the calculation of convective heat loss from all outer surfaces of the receiver. Coefficients as per equation 7-18.</td>
<td>6.1 W.m$^{-2}$.K$^{-1}$</td>
<td>22 kJ.hr$^{-1}$.m$^{-2}$.K$^{-1}$</td>
</tr>
<tr>
<td><strong>Cw1</strong></td>
<td>$c_1$</td>
<td>7.6</td>
<td>27.4</td>
</tr>
<tr>
<td></td>
<td>$(W.m^2.K^{-1})/(m.s^{-1})$</td>
<td>$(kJ.hr^{-1}.m^2.K^{-1})/(m.s^{-1})$</td>
<td></td>
</tr>
<tr>
<td><strong>Cw2</strong></td>
<td>$c_2$</td>
<td>-0.55</td>
<td>-2.0</td>
</tr>
<tr>
<td></td>
<td>$(W.m^2.K^{-1})/(m.s^{-1})^2$</td>
<td>$(kJ.hr^{-1}.m^2.K^{-1})/(m.s^{-1})^2$</td>
<td></td>
</tr>
</tbody>
</table>
Table 7-1 continued.

<table>
<thead>
<tr>
<th>Parameter name in TRNSYS</th>
<th>Parameter description</th>
<th>Value SI units</th>
<th>Value (TRNSYS units)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tol</td>
<td>Tolerance for the iterative calculation of $Q_m$</td>
<td>0.1</td>
<td>0.1</td>
</tr>
<tr>
<td>Perim</td>
<td>Wetted perimeter of the fluid conduit</td>
<td>0.1298 m</td>
<td>0.1298 m</td>
</tr>
<tr>
<td>XSarea</td>
<td>Cross-sectional area of the fluid conduit</td>
<td>0.000359 m²</td>
<td>0.000359 m²</td>
</tr>
<tr>
<td>Fh</td>
<td>Carnavos correction factor for Nusselt number to account for the internal fins</td>
<td>0.74</td>
<td>0.74</td>
</tr>
</tbody>
</table>

1 Measured, as stated in chapter 4.
2 Measured, as stated in chapter 5.
3 Measured, as stated in chapter 6.
4 Measured, as stated in this chapter.
5 Estimate, based on flux mapping of a similar trough.
6 Calculated based on the conductivities and thickness in table 5-2.
7 Calculated, as described in section 7.1.6.
8 Estimated.

7.2 Validation of the PV/T collector component

7.2.1 Parameters

The values of all parameters (corresponding to the CHAPS test rig) for the new PV/T ‘Type 262’ component are given in table 7-1. Some of the parameters are physical parameters such as mirror and receiver dimensions. Assumptions regarding optical properties are the same as used in chapter 5 for the Strand7 thermal modelling. The receiver electrical efficiency was adjusted using the uniformity scaling factor so that the output from the model matched the measured output. Heat transfer coefficients for conduction within the receiver use the same conductivity values given previously in section 5.5.2. The UA-value was determined experimentally as 1.85 W.K⁻¹ (section 5.5.3), giving a U-value of 8.6 W.m⁻².K⁻¹ for the receiver geometry. The UA-value for the two end fittings combined was also determined experimentally as 0.15 W.K⁻¹. A pipe component (Type 31) has been included at each end of the receiver in the TRNSYS model to allow for the losses through these fittings.
7.2.2 Dynamic effects

To examine dynamic effects, the CHAPS test rig has been used to collect experimental data at short time intervals of 4 seconds. The tests measure the response to an almost instantaneous input of solar radiation by moving the mirror quickly onto sun. Tests were carried out on two days with different inlet temperatures. On each day two different flow rates were measured. The results are compared to the modelled output from the TRNSYS PV/T component. The results are shown in figure 7-6.

The mass of the test receiver is 7.7 kg, distributed throughout the receiver as shown in table 7-1. Adjusting the mass parameter in the TRNSYS PV/T component allows the rate of response of the outlet water temperature to the step in radiation input to be tuned. As shown in figure 7-7, a mass of 7.7 kg gives simulation results very close to the measured rate of response for approximately 75% of the temperature rise. After this point, the model continues to predict a sharp increase in outlet temperature, levelling off quickly as the new equilibrium temperature is reached. It fails to predict the more gradual increase in outlet water temperature measured from the collector. The same phenomenon can be seen in other results in figure 7-6.

Table 7-1. Mass of the test receiver (1.5 m long).

<table>
<thead>
<tr>
<th>Material</th>
<th>Mass (kg)</th>
<th>Heat capacity (kJ kg⁻¹ K⁻¹)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminium extrusion</td>
<td>3.3</td>
<td>0.88</td>
</tr>
<tr>
<td>Glass</td>
<td>0.85</td>
<td>0.75</td>
</tr>
<tr>
<td>Silicone potant</td>
<td>0.6</td>
<td>2.0</td>
</tr>
<tr>
<td>Fittings</td>
<td>0.35</td>
<td>0.88</td>
</tr>
<tr>
<td>Insulation</td>
<td>0.25</td>
<td>0.84</td>
</tr>
<tr>
<td>Insulation cover</td>
<td>1.8</td>
<td>0.44</td>
</tr>
<tr>
<td>Water</td>
<td>0.54</td>
<td>4.18</td>
</tr>
<tr>
<td>Total</td>
<td>7.7</td>
<td>1.08</td>
</tr>
</tbody>
</table>
Figure 7-6. Response of the CHAPS test rig to rapid changes in solar input.
Figure 7-7. Comparison of the response of the Type 262 TRNSYS component with various thermal masses to a rapid increase in solar input. The time step is 4 seconds.

One reason why the TRNSYS PV/T component does not better predict the more gradual increase in outlet water temperature, is that the thermal capacitance $C_{p-cop}$ is modelled as a ‘lumped’ parameter. In other words, while in reality the thermal capacitance is distributed about the cross-section of a receiver, equation 7-1 is based on a one-dimensional model, with heat capacitance lumped in the water. The consequence of the assumption of lumped capacitance is that the lag in thermal response of materials distributed throughout the receiver cross-section is not accounted for. For example, the insulation cover contributes about 10% to the weighted thermal capacitance, but heat transfer through the insulation is slow, and therefore there is a significant lag time before the cover reaches its new equilibrium temperature after a step input.

Another reason why the simulated output temperature responds more quickly than in reality, is because of the lack of a ‘fluid transport’ aspect to the model. Equation 7-1 describes the change in outlet temperature with respect to time, but is simplified for the change in temperature with respect to position. The outlet temperature for an element in the model is calculated at the end of each time step, and the inlet temperature to the next element is taken as average outlet temperature of the previous element over the whole time step. Therefore, any change in the first element affects all elements in the collector at each time step. In reality, fluid takes some time to propagate along the receiver. The time constant for a receiver is given by the product of thermal resistance and thermal capacitance, and for the flow rate of 45 ml/s for the data in figure 7-7, is 44 seconds. For the same flow rate, it takes 12 seconds for the bulk of fluid to travel from one end of the receiver to the other. The
assumption that any element is influenced by any preceding element at each time step means that for short time steps, the simulation tends to exaggerate effects resulting from changes to the inlet temperature. The more elements specified in the TRNSYS model, the greater this effect, as shown in figure 7-8.

![Graph](image)

*Figure 7-8. Comparison of the response of the Type 262 TRNSYS component divided into various numbers of elements. Mass of the receiver is set to 7.7 kg. Time step is 4 seconds.*

For longer time steps, the difference in thermal response due to the number of elements diminishes. Figure 7-9 shows the same data set as above reduced to 1 minute time intervals. Insolation data was averaged over the minute from the higher resolution measured data.
Figure 7-9. Comparison of the response of the Type 262 TRNSYS component divided into various numbers of elements. Mass of the receiver is set to 7.7 kg. Time step is 1 minute.

The main reason why a TRNSYS component was developed to simulate the CHAPS collector was to allow annual energy output predictions. As shown above, the simplifying assumptions that allow the model to run efficiently for an annual simulation also cause some problems if the model is to be used as a true ‘dynamic’ model to measure thermal response with very short time steps. However, for annual simulations, it is most important that energy balances are correct. Therefore the results above were used to tune the mass of the model to give the best fit with the measured data for the rate of response. The aim is to avoid any consistent error in the energy output associated with dynamic effects. It is preferable in an annual simulation to use a model with more than a single element, as the output along the length of the collector (particularly the electrical output) should be adjusted when there are large differences in the operating temperature at the inlet and outlet. Therefore, for the modelling that follows in this chapter, the PV/T receiver is divided into multiple elements (10 elements for all the following validations).

All data was collected at 1 minute time intervals, and the mass in the model was ‘tuned’ to give accurate energy output for a 10 element model with one minute time steps. Figure 7-10 shows the thermal response for a 1 minute time step for both a sudden increase and a sudden decrease in solar input. In both cases, a mass increase is required to give a neutral energy difference between the measured and simulated response. However, the PV/T component better simulates the response to a sudden radiation increase, and much more mass, or artificial thermal capacitance, is required to tune the model for a sudden decrease in radiation. As increases and decreases in radiation tend to occur in pairs (for example, a cloud passing over
the collector), the mass chosen to ‘tune’ the model should minimise the sum of errors for these two cases. A somewhat surprising outcome is that decreasing the mass reduces the combined error. For low mass, the positive difference between the simulated and measured energy during the temperature rise in part offsets the negative difference when the temperature falls. For simplicity, it is recommended that the thermal capacitance should be based on the actual mass of the collector.

![Figure 7-10. Comparison of the response of the Type 262 TRNSYS component with various thermal masses to a rapid increase in solar input. Time step is 1 minute.](image)

### 7.2.3 Optical efficiency

The optical efficiency predicted using the analytical data is 75%, as shown in table 7-2.

<table>
<thead>
<tr>
<th></th>
<th>Scaling factor</th>
<th>Optical efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>Direct radiation</td>
<td>1</td>
<td>100 %</td>
</tr>
<tr>
<td>Hydraulic fittings (Receiver length 1.465 m, mirror length 1.5 m)</td>
<td>0.977</td>
<td>97.7 %</td>
</tr>
<tr>
<td>Shading by the receiver (Receiver width 0.08 m, mirror width 1.25 m)</td>
<td>0.936</td>
<td>91.4%</td>
</tr>
<tr>
<td>Mirror reflectivity</td>
<td>0.935</td>
<td>85.5%</td>
</tr>
<tr>
<td>Shape error (light that misses the cells)</td>
<td>0.99</td>
<td>84.6%</td>
</tr>
<tr>
<td>Transmission-absorption of the receiver</td>
<td>0.886</td>
<td>75.0%</td>
</tr>
</tbody>
</table>
The measured optical efficiency is typically determined by examining the combined thermal and electrical output when there are minimal thermal losses. For a flat plate collector, this is usually when the average fluid temperature is similar to the ambient temperature. However, as discussed in chapter 6, for a concentrating collector, there is a far higher radiation flux concentrated on a small area and the average temperature of the glass surface may be around 15-20°C higher than the fluid temperature. Therefore, despite the water temperature being around ambient, there are heat losses due to convection and radiation from the glass surface.

The efficiency curves given previously in figure 5-5 show the combined electrical and thermal efficiency was measured to be 76.2% at the intercept. This is somewhat higher than expected based on the analytical data in table 7-2, particularly as the optical efficiency data does not include heat losses. There are a number of factors that may contribute to the discrepancy:

- Light incident at either end of the receiver on the aluminium covers shading the hydraulic connections may contribute a little thermally. The covers were in thermal contact with the hydraulic fittings through metal clips.
- Light absorbed by the cover on the back of the receiver contributes to the thermal efficiency (by an estimated 0.6% absolute based on a comparison using STRAND7 both with and without the absorption).
- Error in the flow measurements. The flow meter is sensitive to any debris in the flow, and although it was calibrated regularly, it is possible there was some systematic error between calibrations.

### 7.2.4 Calibration at different operating temperatures

Predictions of thermal efficiency from the TRNSYS PV/T component are compared with the efficiency measurements at various operating temperatures. The data is taken from four separate days of measurement using the CHAPS test rig. The raw data is presented in detail in appendix B. The experimental method was discussed in detail in chapter 5. Figure 7-11 shows the efficiency plots using unadjusted ‘first estimate’ values for all parameters, using the values in table 7-1. Trend lines are calculated using a ‘least squares’ linear curve fit. The plot shows the discrepancy in efficiency at the intercept, including thermal losses, is around 2.6%. Likely causes of the discrepancy are discussed in the previous section.
Figure 7-11. Comparison of measured and simulated thermal efficiency for different operating temperature – initial estimate of parameters.

The scatter for the simulated data is almost entirely due to variability in the wind speed. In figure 7-11 the calculated data points shows the extent of scatter which is directly attributable to wind conditions. As may be expected, the simulated points fall on a line if the wind speed is set to a constant. While the experiment was carried out in a sheltered courtyard, wind speeds up to 0.5 m/s were experienced. For clarity, further validation results will use a linear fit of the simulated data.

Comparison between the slope of the measured and simulated trends in figure 7-11 allow the parameters that affect the thermal losses to be tuned. The first parameter to be tuned is the temperature dependency of electrical efficiency, which in turn affects the thermal efficiency. Figure 7-12 shows the electrical efficiency plotted against the average fluid temperature for both the measured and simulated cases. The original temperature dependency estimate was based on the value of -0.35%/°C relative, obtained by testing individual solar cells (section 5.5.1). The data in figure 7-12 indicates that for a full receiver, the efficiency is more sensitive to temperature. It was found that a value of -0.40%/°C relative gives very good agreement between the measured and simulated results.
The slope of the curve for the simulated data is not as steep as for the measured data, as shown on the left in figure 7-13. Therefore, actual thermal losses due to convection and radiation are higher than predicted by the model using the initial estimated parameters. Parameters that contribute to heat loss are the value for emissivity (for radiation losses) and the coefficient of heat transfer for convection. The value for emissivity from the glass surface has been measured with reasonable precision, and error within the bounds of the uncertainty in measurement does not significantly affect the heat loss. The value of emissivity from the insulation cover is estimated as 0.1. Error in this value has a minor effect on the slope of the line (for example, an increase in emissivity 0.1 to 0.5 causes the slope to change from -1.14 to -1.18 for the base case below in figure 7-13). The UA-value for the insulation was calculated based on measurements, but the uncertainty was high (~40%), as discussed in section 5.5.1. Error is expected in the prediction of thermal loss from the receiver due to convection from the outer surface. The estimates of the coefficient of convection are based on empirical equations that can result in errors as large as 25% (Incropera and DeWitt, 1990). Additionally, uncertainty in the measured wind speed is estimated to be around 25% due mainly to the difference in orientation of the receiver with respect to the anemometer. As an indication of how this uncertainty affects the slope of the efficiency curves, increasing the heat transfer due to both convection and conduction by 35% gives heat loss results similar to the experimental data (plot on the right in figure 7-13).
The offset due to error in optical efficiency remains. Overall, the PV/T component has been developed as a detailed analytical model to allow the user to predict the impact of changes to individual subcomponents. This functionality is demonstrated in the following section, where the impact of physical changes to the receiver is examined through annual simulations. Where possible, individual subcomponents have been tested experimentally to determine parameters feeding into the model, and otherwise they have been calculated. While the accuracy of the model compared to the experimental data is not perfect based on the measured and calculated parameters, it is considered to be sufficiently accurate for the purposes of comparative annual simulations, and long term energy output predictions. In addition, the model behaves consistently and in a stable manner, and the solution time is reasonable.

### 7.3 Annual simulations

The performance of an individual TRNSYS component is often strongly influenced by the system it is part of, and by the performance measure used to assess the collector. For example, a domestic hot water system sized such that most of the winter load is met by solar energy will have excess energy in summer. As a result, the ‘annual efficiency’ of the collector in such a system would be lower than if the system were sized to meet the peak summer load, and operated all year round. However, clearly a larger collector area relative to a given hot water load will deliver more solar energy and hence reduce auxiliary energy use.

A simple simulation has been developed in order to conduct analysis of the CHAPS collector. The system models a single 24 m CHAPS trough the same dimensions as those in the Bruce
Hall CHAPS system. The collector is coupled to thermal storage, a hot water load and to a finned tube heat exchanger to dump excess thermal energy to the surrounding air (figure 7-14).

![Figure 7-14. Schematic layout of the CHAPS system simulation.](image)

### 7.3.1 Performance measures

Solar hot water systems commonly use either solar energy fraction (SEF) or fractional energy saving (FES) as measures of annual performance. The solar energy fraction is the fraction of annual energy used by a system that comes from the sun, and is described mathematically as follows:

\[
SEF = 1 - \frac{Q_{\text{aux,solar}}}{Q_{\text{demand}}} \tag{7-27}
\]

where \(Q_{\text{aux,solar}}\) is the auxiliary energy used by the solar hot water system and \(Q_{\text{demand}}\) is the total energy demand for the year. Note that some definitions of solar energy fraction include the energy lost from the tank. This is the case for the ratings used by the Solar Rating and Certification Corporation (2000) in the USA. Another performance measure that is often used is fractional energy saving, which is defined as follows:

\[
FES = 1 - \frac{Q_{\text{aux,solar}}}{Q_{\text{aux,conventional}}} \tag{7-28}
\]

where \(Q_{\text{aux,conventional}}\) is the energy used by a conventional hot water system under the same conditions as the solar hot water system. The advantage of this method is that a measure is obtained that people can relate to due to the comparison to a conventional hot water system. The disadvantage is that a reference conventional system must be defined, and hence the
value of Fractional Energy Saving will depend on this definition. The test standards in Europe have both definitions. The Solar Energy Fraction will be used as a performance measure in the following analysis. Note that the auxiliary energy demand often includes pumping power requirements, but this is excluded from the present analysis.

### 7.3.2 System inputs and parameters

The weather data used for Canberra is a compilation of months from different years, chosen such that they reflect long-term averages for the particular month (Morrison and Litvak, 1988). The weather data is passed into the radiation processor (Type 16), which interpolates the data so that time intervals less than an hour can be used in the simulation. The radiation processor also does sun-earth geometry calculations for single axis tracking to calculate the direct radiation on the trough and the incidence angle of the light. This accounts for cosine losses. It is assumed that the trough is unshaded by other objects, and is located on a flat surface with the long axis aligned east-west. End losses are calculated separately at each time step based on the geometry of the trough and the incidence angle of light on the collector. The calculation of end losses is carried out by a simple custom-built component (the code is included in section 2 of appendix A).

Thermal storage is via a Type 38 plug flow tank, modelled as a vertical cylinder divided into 10 segments of equal volume. Key parameters are given in table 7-3.

**Table 7-3. Key parameters for the Type 38 Tank component.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>2. Tank volume</td>
<td>m$^3$</td>
<td>1.5</td>
</tr>
<tr>
<td>3. Tank height</td>
<td>m</td>
<td>1.53</td>
</tr>
<tr>
<td>4. Height of collector return</td>
<td>m</td>
<td>1.06</td>
</tr>
<tr>
<td>7. Thermal conductivity</td>
<td>kJ.hr$^{-1}$.m$^{-1}$.K$^{-1}$</td>
<td>7.2</td>
</tr>
<tr>
<td>9. Overall Loss Coefficient</td>
<td>kJ.hr$^{-1}$.K$^{-1}$</td>
<td>26.4</td>
</tr>
<tr>
<td>12. Maximum heating rate</td>
<td>kJ.hr$^{-1}$</td>
<td>64800</td>
</tr>
<tr>
<td>13. Auxiliary height</td>
<td>m</td>
<td>1.2</td>
</tr>
<tr>
<td>14. Thermostat height</td>
<td>m</td>
<td>1.25</td>
</tr>
<tr>
<td>15. Set point temperature</td>
<td>ºC</td>
<td>65</td>
</tr>
<tr>
<td>16. Temperature deadband</td>
<td>ºC</td>
<td>8</td>
</tr>
</tbody>
</table>

When the total volume is varied it is assumed that the height remains unchanged, and the losses are adjusted proportional to the change in surface area. The heating capacity of the auxiliary heating element is also adjusted proportional to the tank volume. The heating element is located in the third segment from the top of the tank. The thermostat that controls
the heating element is located in the segment above, and for the base case model is set to 65°C, a typical value for a domestic hot water system. Hot water is drawn from the top of the tank according to the demand profile given in Australian Standard AS 4234-1994, and cold water is made up at the bottom of the tank. A mixing valve is employed to vary the flow rate such that the appropriate energy quantity is drawn off regardless of the outlet temperature of the tank. The base case tank volume is 1.5 m³ and the hot water load is 380 MJ/day.

At times, the solar collectors produce more thermal energy than required to meet the load. Flat plate collectors are typically designed to withstand the stagnation temperature (i.e. the temperature reached with no flow and full sun), which may be as high as 140°C. However, a concentrating collector stagnates at far higher temperatures, and therefore it is necessary to either move the collector away from the sun or to shed the excess energy by some means. Both the Bruce Hall CHAPS system and the long trough CHAPS prototype employ a finned tube to remove heat by natural convection. The TRNSYS simulation is controlled by monitoring the average tank temperature, and once a predefined set point temperature is reached, flow is diverted to the cooling fins.

The finned tube (pictured below) is made of aluminium fins (60 mm x 70 mm) press fitted to a ½” copper tube. The design is very similar to the standard cooling or heating coil design commonly used in HVAC applications, but the fins have increased spacing to allow for better natural convection. The finned tube is manufactured by Hurlcon Pty. Ltd.

![Figure 7-15. A short section of the finned tube heat exchanger used to shed heat from the CHAPS system (right) and the finned tube installed at Bruce Hall (left).](image)

The finned tube is simulated by a custom built component (Type 273), which calculates losses from the fins via natural convection and forced convection due to wind. The Fortran code for this component is included as section 3 of appendix A. The method of solution is similar to that of the Type 262 PV/T component described previously in this chapter. The fin-tube is used only intermittently for the simulations in this chapter, and experimental validation of the fin-tube model is beyond the scope of this thesis. The base case value for
the length of the fin-tube is 60 m, chosen to keep the equilibrium temperature of the outlet from the collector on a hot still summer day at around 85°C.

The set point temperature for diversion of flow to the fins affects both thermal and electrical performance. A high set point means the system will store more energy in the tanks during hotter periods and hence reduce the auxiliary energy use. The downside is that the collector will spend more time operating at elevated temperatures, and hence electrical efficiency is reduced. This tradeoff is explored in more detail in section 7.3.5.3. The bypass temperature used in the base case for the TRNSYS system analysis is 70°C. A controller monitors the temperature at the bottom of the tank, and the temperature at the outlet of the CHAPS receiver. If the temperature difference is above a defined upper dead band setting, then the pump is switched on. Similarly, if the temperature difference falls below a lower dead band setting, then the pump is turned off. This is a commonly employed scheme for pumping control for solar hot water systems, known as ‘delta T’ control. A simple controller has been written specifically for the CHAPS system that incorporates the delta T control as well as the ability to bypass the tank to the cooling fins when the tank temperature is above the set point. The Fortran code is included as section 4 of appendix A.

The flow delivered by the pump largely determines the maximum temperature difference between the inlet and outlet of the collector. The base case flow rate is set to 1100 kg/hr.

The deck file which contains full detail of all parameters used in the base case model is attached as section 5 of appendix A.

7.3.3 Sample output from the system model

The following graphs show typical output results from the TRNSYS system model, using historical TMY weather data (from the first four days in April 1984). The 1st - 3rd April are sunny days, and the 4th April is a cloudy day, as can be seen by the plot of direct beam radiation in figure 7-16.
Figure 7-16. Sample output for a CHAPS collector supplying a DHW system.
7.3.4 Sensitivity of performance to key system parameters

For systems that use water at a lower temperature, for example, an in-floor hydronic heating system, there is an advantage in setting the tank thermostat at a lower temperature as shown in figure 7-17. The results are based on a simulation with base case system parameters as described in section 7.3.2, and base case CHAPS collector parameters as given in table 7-1. The base case setting is 65°C.

Figure 7-17. Solar energy fraction for various tank thermostat settings.

Figure 7-18 shows the solar energy fraction for various daily mid-winter energy draws for a range of tank volumes. To put the hot water load in context, the peak mid-winter load for a large domestic hot water system in Canberra according to AS 4234-1994 is 38 MJ/day.

Figure 7-18. Solar energy fraction and annual electrical efficiency for various tank volumes and hot water loads.
The base case tank volume chosen for the system optimisation runs was 1.5 m$^3$, as this is close to optimal for a range of loads. The base case hot water load selected was 380 MJ/day.

While higher flows generally give better instantaneous performance for the collector (as more turbulence promotes better heat transfer), a lower flow rate gives better system performance over a year. The reason is because the tank will have a higher degree of temperature stratification when the flow rate is low and the collector outlet temperature is closer to the water temperature at the top of the tank. Good stratification aids collector efficiency (as cooler water is input to the collector) and reduces the amount of auxiliary boosting at the top of the tank to achieve the thermostat set point. However, if the flow rate is very low then the fluid will be very hot at the outlet, and may boil. In practice, an over-temperature sensor, incorporated to protect the collector, would be activated and the collector would park. Figure 7-19 shows the solar energy fraction increasing as the flow decreases. However, at very low flows the performance falls away as the collector spends more time off sun. The electrical performance output is reduced under the hotter operating conditions at low flows. The base case flow rate is set to 1100 kg/hr for the purposes of further simulation.

![Figure 7-19](image)

**Figure 7-19.** Solar energy fraction and annual electrical efficiency for various flow rates.

Finally, the tolerance for convergence of the TRNSYS simulation affects the output. The impact of different convergence tolerances varies for different models. Tolerance of 0.001 was chosen for the present simulation based on a compromise between accuracy and simulation speed.
7.3.5 Analysis of the CHAPS collector through annual simulations

Annual simulation of the performance of a CHAPS collector in a ‘realistic’ system model allows the impact of changes to individual components of the CHAPS collector to be investigated. This is the primary advantage of an analytical model rather than an empirical model of the collector.

7.3.5.1 Insulation thickness

Increasing the insulation thickness around the receiver reduces heat loss from the collector. However, the extra width also means extra shading and hence reduced solar input. In practice, the cell tray on the CHAPS receiver is 80 mm wide to allow for the cell tabs and wiring. Insulation is packed on either side of the finned fluid conduit to a total width also of 80 mm. Therefore, at its present thickness of 24 mm, the insulation on the sides of the receiver causes no additional shading. Annual simulations were carried out for a range of insulation thicknesses. Figure 7-21 shows both thermal and electrical performance plotted against the thickness of the insulation on the side of the receiver. The black line shows the result based on the current receiver geometry. The blue line shows the result assuming that reducing the insulation thickness also reduces the shading (i.e. excluding the shading due to the cell tray). It is assumed for all simulation runs that the insulation thickness at the top is 7 mm thicker than the sides (as is the case for the CHAPS receivers). The optimal thickness for thermal performance is shown in figure 7-21 to be around 15 mm if there was no cell tray. The current thickness of insulation on the sides of the CHAPS receivers of 24 mm is optimal given the present geometry of the receiver extrusion.
Also apparent in figure 7-21 is the continuous improvement in electrical performance as the insulation is reduced. The first reason is that there is increased solar input due to reduced shading. The difference between the slopes of the two lines in figure 7-21 indicates this is the dominant reason for improved overall efficiency. In addition, there are higher thermal losses with reduced insulation, resulting in a lower operating temperature and hence improved electrical efficiency (seen by the slope of the black line in figure 7-21). An interesting question is whether the optimal insulation thickness for the thermal output is also optimal for the overall output – that is, electrical and thermal output combined. This depends very much on the ‘value’ of the thermal output relative to the electrical output. Chapter 3 contains a discussion on methods of comparing the value of electrical and thermal output, based on a case study comparing a solar domestic hot water system with a grid connected flat plate PV panel. The concept of equivalent electrical energy introduced in chapter 3 relates thermal energy to electrical energy via an energy value ratio (equation 3-8). In other words, if the electrical output is considered equally valuable as the thermal output, the energy value ratio is 1. However, generally the electrical output is considered more valuable, whether for thermodynamic, economic or environmental reasons, and the energy value ratio is more than 1. The value of the energy value ratio is different for every system, and depends on the criteria by which the concept of ‘value’ is assessed. For the case study examined in chapter 3, the energy value ratios ranged from 1 to 17 depending on the method of comparison. It is suggested in chapter 3 that the method of determining the energy value ratio most applicable for a PV/T system is based on levelised energy cost calculations in a renewable energy market. The energy value ratio particular to the case study using this technique was 4.2.

The present study of insulation thickness shows the importance of having an approach to compare the value of the electrical and thermal output. Figure 7-22 shows the ‘equivalent
electrical energy' for a range of energy value ratios. It is assumed for these graphs that the useful electrical output is given by the dc output calculated by annual simulations in TRNSYS, multiplied by scaling factors of 90% for dc-ac conversion, and 85% for operation away from ideal conditions (i.e. accounting for the effect of non-uniform light in the longitudinal direction). Useful thermal output is calculated by subtracting the auxiliary boost energy required over a year from the annual hot water load.

![Graphs showing equivalent electrical energy for different insulation thicknesses and energy value ratios.](image)

**Figure 7-22. Equivalent electrical energy for a range of insulation thicknesses and energy value ratios.**

The graphs show that in all cases where the cell tray is included, the optimum insulation thickness is the same or a little less than exists in the current design. However, if the cell tray is narrowed, then the optimum insulation thickness varies, depending on the value of the electrical energy relative to the thermal energy. If the electrical energy is highly valued (corresponding to a higher energy value ratio), then the improvement in electrical efficiency (driven by reduced shading) dominates, and little to no insulation is preferable. However, for energy value ratios below 3, thermal performance has an influence, and the optimal insulation thickness is about 10-15 mm.
7.3.5.2 Conductivity of the tape bonding the cells to the receiver

As was identified previously in chapter 5, the conductivity of the tape bonding the cells to the receiver is not as high as that of other thermally conductive, electrically isolating products on the market. The U-value for the Chomerics tape used to date for CHAPS receivers is 1646 W.m\(^{-2}\).K\(^{-1}\), compared to the sample of AIT Coolbond tape, measured at 4169 W.m\(^{-2}\).K\(^{-1}\). Figure 7-23 shows that both electrical and thermal performances improve with a higher conductivity interface between the cells and the aluminium. The red circles in figure 7-23 correspond to the U-value for Chomerics tape, and the blue circles correspond to the AIT Coolbond. Electrical performance benefits most from improved heat transfer from the cells to the heat sink, with improvement of 2.1% absolute comparing the AIT Coolbond to the Chomerics tape.

![Figure 7-23. Solar energy fraction and annual electrical efficiency for U-values for the interface between the solar cells and the aluminium heat sink.](image)

The curves show gradually diminishing return for improved heat transfer. However, from the range of materials available for heat sinking the solar cells, it is clear that some significant improvement in electrical performance is possible. A cost-benefit analysis is required to determine whether or not a better and more expensive heat sinking material is warranted. Using the cost figures for the thermal tapes given previously in chapter 4, and assuming photovoltaic electricity costs as per table 3-3 in chapter 3, the simple payback time based on savings from the electrical output alone is around 4.5 years. Therefore, pending reliability testing, the use of the higher conductivity tape seems justified.

Note also that the TRNSYS component is one-dimensional regarding heat transfer to the fluid. The model assumes an even radiation flux distribution across the receiver and does not account for the higher flux and higher temperature at the centre of the cell. It was calculated previously in section 5.6.6.3 that instantaneous efficiency improvement in the order of 4.5%
absolute could be expected comparing AIT Coolbond to Chomerics tape. Hence, in reality the improvement is likely to be a little more than predicted by the TRNSYS PV/T component.

Most point focus PV concentrator systems require significantly better thermal conductivity in their heat sinking due to the much higher radiation flux concentrations, and ceramic materials are commonly used. However, given the decreasing returns for better heat transfer apparent in figure 7-23, it is unlikely that the additional cost of ceramic materials justifies the expense and added complexity for a CHAPS receiver.

7.3.5.3 Optimal use of the cooling fins

The CHAPS system is designed to be used in conjunction with systems requiring low to mid temperature hot water. The present case study has a domestic hot water load requiring 65°C water. A fundamental premise in the design of the CHAPS collector is that it is economically worthwhile sacrificing some electrical efficiency to producing hot water at useful temperatures. This assumption can be tested using the system model by varying the temperature setting that controls the bypass of fluid from the tank to the cooling fins. Figure 7-24 shows the thermal and electrical output for a range of bypass set points.

![Figure 7-24](image)

*Figure 7-24. Solar energy fraction and annual electrical efficiency for various bypass temperature settings for when fluid is directed away from the tank to the fin-tube heat exchanger.*

When the bypass temperature is set to the lower extreme, the average water temperature in the tank is always hotter than the set point, and therefore the water from the collector is always directed to the passive cooling fins. In other words, the tank operates independently from the solar collector. The solar fraction is in this case slightly negative, as the auxiliary boosting
over a year is more than the load due to thermal losses from the tank. When the bypass temperature is at the upper extreme, the tank is able to store more solar energy before the flow is diverted to the cooling fins. Therefore, the solar fraction is close to optimum, given the size of the collectors relative to the load.

The electrical efficiency displays an interesting peak at bypass temperatures around 40°C. Intuitively it may be expected that the electrical efficiency would continually decrease as the bypass temperature increases, because hotter water would be diverted to the receivers. However, at times there is an advantage in drawing water from the bottom of the tank. For example, during or soon after periods of external water draw off, cold water from the mains water supply is made up into the bottom of the tank. During these periods, it is likely that the water temperature at the bottom of the tank is lower than the temperature returned at the outlet of the cooling fins, particularly on a hot day. If the bypass temperature is at the lower extreme, then the tank is effectively isolated from the collector, and the advantage gained by the periods of cold water draw from mains is lost. If there is no over-temperature protection, then as the bypass temperature rises, the electrical efficiency falls until an equilibrium point is reached where the water temperature can rise no higher based on the water draw off schedule and the solar input, as shown by the blue line in figure 7-24. However, prior to this point the over-temperature protection, which is monitoring the outlet temperature to prevent boiling, has an effect. The black line in figure 7-24 demonstrates that there is a rapid drop off in electrical efficiency due to the collector parking off sun, prior to the equilibrium point.

The optimal bypass temperature for electrical efficiency is far from optimal for the overall system. Figure 7-25 shows the ‘equivalent electrical energy’ for a range of energy value ratios, calculated as discussed previously in section 7.3.5.1. The optimal bypass temperature depends on the value of the electrical energy relative to the thermal energy. However, for typical values of the energy value ratio the optimal bypass temperature is in the range 75-85°C.

It is shown that if there is value in the thermal energy (i.e. if there is an application that can directly use the thermal energy to offset other auxiliary heating costs) then it is worth sacrificing a little electrical output by operating at higher temperatures suitable for the thermal application.
Figure 7-25. Equivalent electrical energy for a range of bypass temperatures and energy value ratios.

7.3.5.4 Comparison to a flat plate collector

The CHAPS collector can be compared directly with a flat plate solar hot water collector through a system model. A Solahart Oyster K collector is used for the comparison, with efficiency coefficients based on aperture area from tests under the Solar Rating and Certification Corporation (2000). The system model is identical to the CHAPS base case model except that:

- It is assumed that the flat plate collector is fixed, oriented towards north and tilted at the latitude angle of 35.3 degrees.
- The flat plate collector accepts both direct and diffuse radiation.

The results for an annual simulation are summarised in table 7-4.
Table 7-4. Annual energy output comparing the CHAPS base case model with a flat plate model.

<table>
<thead>
<tr>
<th>Unit</th>
<th>Flat plate collector</th>
<th>CHAPS collector</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Q_{\text{solar}}$</td>
<td>kWh/day 204.1</td>
<td>kWh/day 166.4</td>
</tr>
<tr>
<td>$Q_{\text{load}}$</td>
<td>kWh/day 95.7</td>
<td>kWh/day 95.7</td>
</tr>
<tr>
<td>$Q_{\text{aux}}$</td>
<td>kWh/day 25.8</td>
<td>kWh/day 40.3</td>
</tr>
<tr>
<td>$Q_{\text{elec}}$</td>
<td>kWh/day -</td>
<td>kWh/day 19.6</td>
</tr>
<tr>
<td>$Q_{\text{therm}}$</td>
<td>kWh/day 69.9</td>
<td>kWh/day 55.5</td>
</tr>
<tr>
<td>S.E.F.</td>
<td>0.73</td>
<td>0.58</td>
</tr>
</tbody>
</table>

* $Q_{\text{therm}} = Q_{\text{load}} - Q_{\text{aux}}$

Despite the deficit in the solar energy available to the CHAPS collector due to the non-use of diffuse light, the annual output considering electrical and thermal energy combined is greater than for a flat plate collector. Figure 7-26 shows a comparison of performance of the CHAPS collector compared to the flat plate collector based on equivalent electrical energy for a range of energy value ratios. For the energy value ratio of 4.2 (from the case study in chapter 3), the CHAPS system has equivalent electrical output close to twice that of a flat plate system. It is useful to compare annual output in this way when establishing cost targets for the CHAPS system.

\[ \text{Relative performance based on equivalent electrical energy} \]

\[ \text{Energy value ratio} \]

Figure 7-26. Relative performance of the CHAPS collector compared to a flat plate collector based on equivalent electrical energy for a range of energy value ratios.

7.3.6 Further case studies

Clearly certain thermal applications are better suited to CHAPS collectors than others. The example of a domestic hot water system shows the balance between thermal performance and electrical output for a reasonably low temperature application. It is difficult to assess other
applications without carrying out specific case studies. An interesting case study would be to assess the suitability of the CHAPS system for solar cooling applications. By varying the flow rate the CHAPS collector has the ability to produce hot water at a consistent outlet temperature above 80-85°C, which is a useful temperature for a single-effect absorption chiller (Chinnappa, 1992). Furthermore, it would be possible to produce hot water (under pressure) at temperatures around 150°C, useful for a double-effect absorption chiller, with the advantage of a superior coefficient of performance (1.43 compared to 0.79 for a single stage cycle). This advantage would be offset by the reduction in both thermal and electrical efficiency of the CHAPS at the higher temperature. It is suggested that such a case study, designed to explore the upper operating temperature range suitable for a CHAPS collector, would be interesting and worthwhile further work.
Conclusions and recommendations for further work

Chapter 8

8.1 Conclusions

Surveying recent advances in the field of concentrator photovoltaics has shown that most new concentrator PV projects are point focus systems using refractive optics, with increasing use of very high efficiency (>30%) multi-junction solar cells. At the time of writing, the CHAPS system is found to be the only mid-range concentration PV/T collector using reflective optics.

A number of methodologies are discussed that can be followed to assess the concept of energy value, or ‘usefulness’. Often the simplistic approach of assigning equal value to electrical and thermal energy is taken, but both the second law of thermodynamics and common sense tell us that electricity is not the same as heat, and is more useful. Thermodynamic methods of assessing energy value are compared to an economic valuation, through levelised energy costs, and an environmental evaluation, through embodied greenhouse gas emissions. It is proposed that the value of electrical energy and thermal energy is best compared using a simple ratio that depends on the circumstances of an individual system. The value of the ratio ranges from a lower bound of 1, using a direct energy comparison, to an upper bound of 17, using an exergetic comparison. It is recommended that the ratio is best determined using an economic evaluation of the cost of renewable electricity and hot water, which gives an energy value ratio of 4.2 for the case study examined.

The CHAPS collector is described in detail, including analysis of system sub-components. The optical performance of the mirror, glass, silicone potant and solar cells is determined experimentally for a range of wavelengths using a spectrophotometer.

Experimental measurement of the thermal and electrical efficiency of a receiver was carried out for fluid temperatures ranging from close to ambient temperature to around 60°C above
ambient. In this range, the electrical system efficiency was found to be around 10-12%, while thermal efficiency was around 54-64%. The results were used to calibrate an analytical model of the collector developed in the TRNSYS software environment. Tests were also carried out at a range of fluid flow rates in order to determine the effectiveness of internal fins, and to test various empirically derived expressions for determining the heat transfer from the receiver to the fluid. Use of the Carnavos correction factor (with a value of 0.74 for the CHAPS receiver geometry) with the standard Dittus-Boelter equation for Nusselt number was found to be adequate. Measurement of the glass surface temperature during the testing was carried out using an infrared camera, to assist with calculation of radiation and convection losses from the glass, and to help determine the extent of radiation absorption in the glass and silicone cover materials. It was found that total absorption in the cover materials is around 6.3%. Finite element analysis was carried out using the software package Strand7 to examine the conductive heat transfer within the receiver materials, and to obtain a temperature profile of the receiver under operating conditions. The results were used to test the impact on electrical performance of the use of different materials to bond the solar cells to the aluminium receiver. It was demonstrated that further improvement is possible without prohibitive extra cost.

The efficiency of solar conversion is reduced when solar cells operate at an elevated temperature. The temperature dependency was quantified for the ANU concentrator cells by measuring I-V curves in a flash tester for a range of temperatures, and was found to be around 0.3 - 0.4%/°C relative. Furthermore, the results for a single cell were compared to those of a full receiver tested on a constant illumination indoor test rig, showing a drop in open circuit voltage of 1.8 mV/°C, which was consistent with the flash tester results.

The reflected radiation flux profile from a parabolic trough across a receiver is highly non-uniform. It can vary from almost zero radiation at the edge of a cell to more than 100 suns at the centre of the focal beam. The impact on cell efficiency of the non-uniform flux profile was investigated experimentally using a partially masked cell on the flash tester. It was found that efficiency is reduced by about 10% relative because of reduced fill factor and a drop in cell voltage due to the significant internal current flows within the cell.

Ideally, the radiation flux profile longitudinal to the receivers would be perfectly uniform. However, due to practical constraints such as supports for the receivers and gaps between mirrors, there are regions in the flux profile where the illumination intensity is lower. Compounding this effect is the fact that mirrors are not perfect parabolas, and small shape errors can produce significant variations in the flux profile at the focus. A flux scanning device was designed and built to accurately measure the longitudinal flux profile. Measurements were carried out for a range of mirrors and for different sun angles. At worst,
the minimum flux intensity was measured to be 27% lower than the median, but it is typically 10-20% lower, depending on the incidence angle of the light. A single cell with low flux intensity limits the current and performance of all cells in series. Therefore, measuring and understanding the cause of variations in the longitudinal flux profile is of fundamental importance.

To develop a deeper understanding of the causes of flux non-uniformities, the precise shape of two mirrors was measured using photogrammetry techniques. The mirrors were then simulated in Opticad as composite mirrors made up of hundreds of tiny facets incorporating the measured shape error. Ray tracing was carried out to produce a flux profile at the focus, and good agreement was achieved between the measured and simulated results. Some unexpected results were found. In particular, there was a local peak in the flux intensity of the focal beam in the position where, based on geometry, a dip would be expected due to the gap between mirrors. It was found that a small region of high slope error at the end of each mirror caused a kind of ‘pseudo focus’ in the longitudinal direction that partly compensated for the gap between the mirrors. The ray tracing also allowed the effect of the receiver supports, the gap between mirrors, and the shape imperfections, to be examined individually. The results led to some improvement in the trough design to give smaller gaps between mirrors, and a change to the mirror manufacturing process to eliminate a prominent ‘hump’ in the flux profile near the centre of the mirror. Methods of mitigating the effects of the uneven flux profile are proposed. One technique is to sort the solar cells from the production process so that cells with a higher maximum power current are positioned in regions that are more frequently affected by low flux distributions. Another technique is to strategically place bypass diodes over small groups of cells, although this only helps at times when the flux profile is particularly badly affected.

A new TRNSYS component was developed to simulate the CHAPS collector. The component contains a detailed analytical model of the collector, including thermal capacitance effects, and the ability to change parameters in the model that correspond to properties of physical sub-components. For example, it is possible to adjust optical properties like mirror reflectivity and glass transmissivity, or heat transfer properties like the conductivity of materials within the receiver. The accuracy of the model was examined by comparison with experimental data. Despite some error in optical efficiency, the model was found to have sufficient accuracy for long term simulations. A case study was carried out using the new CHAPS component as part of a domestic hot water system in an annual simulation. It was then possible to test what effect changes to the CHAPS design have over a typical year. For example, the appropriateness of the current insulation thickness was tested, given the trade-off between shading and heat loss. The analysis also demonstrated the
practical importance of having an approach to compare the value of the thermal output to the electrical output.

Overall, it can be concluded that the CHAPS collector is a promising solar technology with the potential for low cost hot water and PV electricity generation. From this thesis the following conclusions can be drawn:

• the physics of all components of the CHAPS system is well understood;
• the electrical and thermal performance of the collector under realistic operating conditions is well understood, and consistent with theoretical predictions;
• there are opportunities for further optimisation of performance, particularly associated with improvement of the uniformity of the radiation flux profile;
• a proven TRNSYS tool has been established that is essential for system design.

8.2 Recommendations for further work

This thesis provides a detailed description of all facets of the CHAPS collector, from design to operation, and gives a solid basis for further development and improvement in performance.

Simulation

An interesting study would be to explore the breadth of applications suitable for the use of CHAPS collectors. Such a study could be carried out in the TRNSYS simulation environment using the new CHAPS component. The following examples suggest a range of possibilities:

• Solar cooling. By using variable speed pumping, the CHAPS collector is able to supply water at temperatures suitable to drive absorption chillers, both single effect (80-85°C) and double effect (150°C). However, a key question is whether it is worthwhile sacrificing electrical efficiency to supply hot water at these temperatures.
• Space heating. The new Bruce Hall project incorporates eight CHAPS collectors supplying hot water for an in-floor heating system as well as domestic hot water. There are many design issues associated with such a combsystem, such as optimal hot water storage volumes, and a range of control issues.
• Electrical only. For applications where the thermal output has no practical use, there may be some advantage in investigating alternative methods of shedding heat. For example, the solar concentrating PV dishes manufactured by Australian company Solar Systems are located in outback Australia, away from any township. Heat is shed via a fan-forced water-to-air heat exchanger; however, the parasitic energy use due to the fans is quite high. Solar Systems proposed that future systems will tap into underground aquifers to shed heat. An interesting study would be to quantify the relative merits of the various passive and active cooling methods.

**Experimental**

When the Bruce Hall CHAPS system is commissioned, the electrical output of eight full CHAPS systems will be monitored. An interesting exercise would be to examine the electrical output for a range of incidence angle, and compare the results to predictions based on the measured dip in the flux profile at the corresponding incidence angles. It may be possible to develop a correlation for between solar incidence angle and the relative reduction in electrical output. The thermal efficiency could also be examined for the Bruce Hall CHAPS troughs to assess whether heat transfer increases as is expected for the higher flow rates.

Tolerance to tracking error and the effect of mirror misalignment and torsion in the trough is an important issue that has yet to receive only theoretical consideration. It is suggested that this be examined in detail for the Bruce Hall CHAPS troughs.

**Design**

There are a number of possible avenues for further improvement of the performance of the CHAPS collectors.

• Secondary flux modifiers. Secondary flux modifiers can be incorporated to improve the homogeneity of the flux profile near the focus. They can also reduce light spillage where the tolerance of tracking or mirror alignment is an issue. It is suggested that a simple secondary flux modifier could be fabricated from the insulation cover. Various designs for secondary flux modifiers could be simulated prior to fabrication using the ray tracing model described in this thesis.

• Sliver® cells. Thin, single crystal silicon solar cells, called Sliver® cells, have been manufactured at the ANU. Current research by Franklin and Blakers (2004) shows
promising results for the use of Sliver® cells in concentrator systems. The cells are very thin, and under concentrated light of 30 suns, typical current output is around 300mA, with efficiency expected to be above 20%. Cells would be connected in series in strings that would build voltage at a rate of around 5-10V per linear cm. The possibility of rapidly building voltage opens up interesting possibilities for parallel connection of cell strings, potentially removing many of the problems associated with flux non-uniformities along the length of a receiver. However, much research remains to practically realise and test the design of a receiver incorporating Sliver® cells.
Bibliography


Appendix A

A1 Fortran code for the PV/T TRNSYS component

SUBROUTINE TYPE262 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
C*******************************************************************************
C PV/Thermal Collector SUBROUTINE
C
C Author: Joe Coventry
C Date commenced: 9/7/2001 last modified: 10/2/2004
C
C This component is a detailed model of a concentrating PV/thermal collector.
C The model considers capacitance effects. The model is a detailed
C model, where the various quantities of heat transfer between cover,
C cells, plate, tube and fluid are calculated by iteratively solving the
C equations that physically describe the modes of heat transfer.
C
C*******************************************************************************
C STANDARD TRNSYS DECLARATIONS
C DOUBLE PRECISION XIN,OUT
PARAMETER (NIMAX=7,NPMAX=34,NO=13,ND=0)
INTEGER*4 INFO,ICNTRL,NP
REAL T,DTDT,PAR,TIME
DIMENSION XIN(NIMAX),OUT(NO),PAR(NPMAX),INFO(15)
CHARACTER*3 YCHECK(NIMAX),OCHECK(NO)
C Declaration specific to this model
C
C TRNSYS common variables needed for using the store S and the
C simulation time step DELT
C
INCLUDE 'D:\INCLUDE\PARAM.INC'
COMMON /SIM/ TIME0,TIMEF,DELT,IWARN
COMMON /STORE/ NSTORE,IAV,S(NUMSTR)
COMMON/LUNITS/LUR,LUW,IFORM,LUK
C
C Maximum number of discrete elements (this can be changed)
PARAMETER (MAXCELLS=100)
C
C External functions
EXTERNAL QTHFUNC1, CPWATER

C Some constants used in the model (temp in K at 0 DegC, Stefan-Boltzmann constant)
DATA IUNIT/0/, TZERO/273.15/, SBC/20.41E-8/

C Declaration of parameters (commented below)
   INTEGER MODE
   INTEGER CELLS
   REAL REFEFF
   REAL REFTEMP
   REAL BETA
   REAL UNIFORMITY
   REAL LENGTH
   REAL WIDTH
   REAL REFLM
   REAL SHAPE
   REAL TRANSABS
   REAL COVERABS
   REAL EMIS
   REAL MASS
   REAL CP
   REAL TSTART
   REAL WGLASS
   REAL UCG
   REAL WCG
   REAL UCP
   REAL WCP
   REAL UPT
   REAL WPT
   REAL UINSUL
   REAL WINSUL
   REAL WCOVER
   REAL EMISCOV
   REAL CW0,CW1,CW2
   REAL TOL
   REAL HCTF
   REAL PERIM
   REAL XSAREA
   REAL FH

C Counters
   INTEGER J
Inputs used in the model (commented below)

- REAL TFI
- REAL FLOW
- REAL ID
- REAL TAMBI
- REAL WIND
- REAL SHADE
- REAL DIRT

Variables used within the model (in order of appearance)

- Capacitance of each node
  - REAL CAPNODE
- Sum of thermal output
  - REAL QTHSSUM
- Sum of electrical output
  - REAL QELECSUM
- Sum of losses through insulation
  - REAL QINSULSUM
- Sum of convection losses from the glass
  - REAL QCONVSSUM
- Sum of radiation losses from the glass
  - REAL QRADSSUM
- Sum of cell temperatures (used to give an average cell temp)
  - REAL TCSUM
- Sum of glass temperatures (used to give an average glass temp)
  - REAL TGSUM
- Sum of plate temperatures (used to give an average plate temp)
  - REAL TPSUM
- Sum of cell temperatures (used to give an average tube temp)
  - REAL TTSUM
- Sum of mid glass temperatures (used to give an average mid glass temp)
  - REAL TGMIDSUM
- Sum of insulation cover temperatures (used to give an average ins. cover temp)
  - REAL TCOSUM
- Temperature of the inlet of the node
  - REAL TFIN(MAXCELLS)
- Average temperature of the fluid in the node
  - REAL TBAR(MAXCELLS)
- Temperature of the fluid in the node at the previous time step
  - REAL TFINIT(MAXCELLS)
Temperature at the inlet of the node at the previous time step
REAL TFINPREV(MAXCELLS)

Temperature of the fluid at the outlet of the node
REAL TFINAL(MAXCELLS)

Upper and lower guesses for the bisection
REAL QTHUPPER,QTHLOWER

Boolean flag to indicate if the solution is in the guessed bounds
LOGICAL SUCCESS

Thermal output
REAL QTH

Array of thermal outputs
REAL QTHARRAY(MAXCELLS)

Electrical output
REAL QELEC

Array of electrical output
REAL QELECARRAY(MAXCELLS)

Convective heat loss from glass
REAL QCONV

Array of convective heat loss from glass
REAL QCONVARRAY(MAXCELLS)

Radiative heat loss from glass
REAL QRAD

Array of radiative heat loss from glass
REAL QRADARRAY(MAXCELLS)

Heat loss through the insulation
REAL QINS

Array of heat loss through the insulation
REAL QINSARRAY(MAXCELLS)

Tube temperature
REAL TT

Array of tube temperatures
REAL TTARRAY(MAXCELLS)

Plate temperature
REAL TP

Array of plate temperatures
REAL TPARRAY(MAXCELLS)

Cell temperature
REAL TC

Array of cell temperatures
REAL TCARRAY(MAXCELLS)

Glass surface temperature
REAL TG
Array of glass surface temperatures
REAL TGARRAY(MAXCELLS)

Mid glass/silicone temperature
REAL TGID

Array of mid glass/silicone temperatures
REAL TGIDARRAY(MAXCELLS)

Insulation cover temperature
REAL TCOVER

Array of insulation cover temperatures
REAL TCOVERARRAY(MAXCELLS)

Coefficients for the differential equation \( \frac{dt}{dt} = AT + B \)
REAL AA, BB

Heat capacity of the water
REAL CPF

Common block variables used in functions
COMMON/PVTPARMS/ID,WIDTH,LENGTH,CELLS,REFLM,SHAPE,SHADE,DIRT,CW0,
1 CW1,CW2,WIND,MODE,TF,XSAREA,PERIM,FLOW,FH,HCTF,TT,TP,UPT,
1 WPT,INSUL,WINSUL,TAMB,SBC,EMISCOV,WCOVER,TCOVER,QINS,
1 QCP,UCP,WCP,QUELEC,REFEFF,BETA,REFTEMP,TC,QABSCELLS,
1 TRANSABS,QCG1,TGID,UCG,WCG,QABSGlass,COVERABS,QCG2,TG,
1 QCONV,WGLASS,QRAD,EMIS,UNIFORMITY

If it is the very first call of the simulation then continue,
with these initialisation things otherwise go down to parameters.

IF (INFO(7).GE.0) GO TO 10

Set this to number of outputs
INFO(6)=NO

Set to 1 as routine depends on passage of time
INFO(9)=1

Set the common store size to the maximum number of cells
INFO(10)=2*MAXCELLS

Check that the user has provided the right number of inputs, outputs
and derivatives

CALL TYPECK(1,INFO,NIMAX,NPMAK,ND)
C Set the input and output variable types
DATA YCHECK,'TE1','MF1','IR1','TE1','VE1','DM1','DM1'/
DATA OCHECK,'TE1','MF1','PW1','PW1','PW1','PW1','PW1','PW1'/
1 'TE1','TE1','TE1','TE1','TE1','TE1',CALL RCHECK(INFO, YCHECK, OCHECK)

C Set the first storage place in the middle of the allocated variables
ISTORE = INFO(10)

C Get the values of the parameter for this component (only once)
10 IF(INFO(1) .EQ. UNIT) GO TO 30
IUUNIT = INFO(1)

C First do common parameters

C MODE - whether or not the fluid convection coefficient is calculated
C explicitly 1 = no, 2 = yes
MODE = INT(PAR(1) + 0.1)

C CELLS - number of nodes in the receiver is divided into along its length
CELLS = INT(PAR(2) + 0.1)

C REFEFF - the reference efficiency of an encapsulated solar cell
REFEFF = PAR(3)

C REFTEMP - the reference temperature for calculation of solar cell efficiency
REFTEMP = PAR(4) + TZERO

C BETA - the coefficient relating cell efficiency and temperature
BETA = PAR(5)

C UNIFORMITY - Scaling factor to account for the drop in electrical performance of the
cell due to both longitudinal and transverse non-uniform radiation and temperature

UNIFORMITY = PAR(6)

C LENGTH - Length of the troughs
LENGTH = PAR(7)

C WIDTH - Width of the mirror (i.e., unshaded bit)
WIDTH=PAR(8)

C REFLM - Reflectivity of the mirror
REFLM=PAR(9)

C SHAPE - Optical accuracy of the mirror (1 = perfect)
SHAPE=PAR(10)

C TRANSABS - the transmission absorption coefficient for the cells
TRANSABS = PAR(11)

C COVERABS - the fraction of light passing through the cover that is absorbed
COVERABS = PAR(12)

C EMIS - the emissivity of the surface of the cell encapsulation
EMIS=PAR(13)

C MASS - mass of the receiver and fluid combined
MASS=PAR(14)+0.01

C CP - thermal capacitance of the receiver and fluid combined
CP=PAR(15)

C TSTART - initial temperature of the receiver and fluid
TSTART=PAR(16)+TZERO

C WGLASS - Width of glass cover for the purposes of calculating convection
and radiation losses
WGLASS=PAR(17)

C UCG - U value between the cells and the glass
UCG=PAR(18)

C WCG - Width of the connection between cells and glass for heat transfer purposes
WCG = PAR(19)

C UCP - U value between the cells and the plate
UCP=PAR(20)

C WCP - Width of interface between the cells and the plate
WCP = PAR(21)
C UPT - U value between the plate and tube
   UPT = PAR(22)

C WPT - Width of the interface between plate and tube
   WPT = PAR(23)

C UINSUL - U value for the insulation
   UINSUL = PAR(24)

C WINSUL - Perimeter length for calculating insulation losses
   WINSUL = PAR(25)

C WCOVER - Perimeter length for calculating convection and radiation losses
   from the insulation cover
   WCOVER = PAR(26)

C EMISCOV - Emissivity of the insulation cover
   EMISCOV = PAR(27)

C CW0, CW1, CW2 - Coefficients for calculation of convective heat loss from
   all outer surfaces of the receiver. hc = CW0 + CW1 * Wind speed + CW2 * Wind speed^2
   CW0 = PAR(28)
   CW1 = PAR(29)
   CW2 = PAR(30)

C TOL - tolerance of the iterative calculation of Qth
   TOL = PAR(31)

C Now do parameters specific to modes of operation

C Firstly, the input mode determines whether the convection
   coefficient should be calculated explicitly or entered
   as a parameter

IF (MODE.EQ.1) THEN

C HCTF - coefficient of convection for the fluid set as a parameter
   HCTF = PAR(32)/CELLS

ELSEIF (MODE.EQ.2) THEN
C PERIM - Wetted perimeter of the flow passage in the tube
PERIM = PAR(32)

C XSAREA - cross-sectional area of the flow passage in the tube
XSAREA = PAR(33)

C FH - Correction for Nusselt number to account for the internal fins
(i.e. Using the Carnavos relation gives 0.74 for the CHAPS receiver)
FH = PAR(34)

ENDIF

C---------------------------------------------------------------
C Set the initial values for fluid temperature in the receiver

IF (INFO(7).EQ.-1) THEN
    DO 20 J=1,CELLS
    S (ISTORE+(J-1)) = TSTART
    20    CONTINUE
    RETURN
ENDIF
C---------------------------------------------------------------

30 CONTINUE

C Set the storage place for this particular component
ISTORE = INFO(10)

C Get the common values for the inputs for this component

C TFI - Inlet fluid temperature (degC)
TFI = XIN(1) + TZERO

C FLOW - Flow rate (kg/hr)
FLOW = XIN(2)

C ID - Direct beam radiation (kJ/hr.m2)
ID = XIN(3)

C TAMB - Ambient temperature (degC)
TAMB = XIN(4) + TZERO
C WIND - Wind speed (m/s)
   WIND=XIN(5)

C SHADE - fraction of the mirrors that are unshaded by adjacent mirrors
C 1 = no shading, 0 = full shading
   SHADE=XIN(6)

C DIRT - Measure of cleanliness of the mirrors (1 = perfectly clean)
   DIRT=XIN(7)

C--------------------------------------------------------------------------
C Retrieve the outlet temps from the previous time step
DO 40 J=1,CELLS
   TFINIT(J)= S(ISTORE+(J-1))
40 CONTINUE

C Retrieve inlet temps from the previous time step
DO 45 J=1,CELLS
   TFINPREV(J)= S(ISTORE+(J-1)+CELLS)
45 CONTINUE

C--------------------------------------------------------------------------
C Thermal performance
C--------------------------------------------------------------------------

C Set the capacitance for each cell node

   CAPNODE=(MASS*CP)/CELLS

C Initialise energy sums for this timestep
   QTHSUM=0
   QELECSUM=0
   QINSULSUM=0
   QCONVSUM=0
   QRADSUM=0

C Initialise temp sums (to be used to calculate mean temperature)
   TCSUM=0
   TGSUM=0
   TFSUM=0
   TTSUM=0
   TGMSUM=0
TCOVSUM=0

C Set up the loop for the cells

50 DO 60 J=1,CELLS

C Set the inlet fluid temp

   IF (J.EQ.1) THEN
      TFIN(J)=TFI
   ELSE
      TFIN(J)=TBAR(J-1)
   ENDF

C Set fluid temp as average temp in the element at the previous time step
   TF=(TFINPREV(J)+TFINIT(J))/2

C The following uses the Bisection Algorithm routine to solve the simultaneous equations describing this system.

C For a concentrating collector

C Set the lower limit for the bisection (zero heat transfer)
   QTHLOWER=0

C Set the upper limit for the bisection (extraterrestrial radiation)
   QTHUPPER=4760*WIDTH*LENGTH/CELLS

C Check if the limits bound the solution by using a bracketing algorithm
   CALL ZBRAC(QTHFUNC1,QTHLOWER,QTHUPPER,SUCCESS)

C If the solution is bounded, then proceed to calculate the thermal heat transfer, Qth

   IF (SUCCESS) THEN CALL BISECTION(QTHFUNC1,QTHLOWER,QTHUPPER,TOL,QTH)

C Otherwise there is a problem
END IF

C Store energy transfer values in arrays
   QTHARRAY(J)=QTH
   QINSARRAY(J)=QINS
QLEGARRAY(J)=QLEG
QRADARRAY(J)=QRAD
QCONVARRAY(J)=QCONV

C Store various temperatures in arrays
TTARRAY(J)=TT
TPARRAY(J)=TP
TCARRAY(J)=TC
TGARRAY(J)=TG
TGMIDARRAY(J)=TGMID
TCOVARRAY(J)=TCOVER

C Find heat capacity of the fluid
CPF = CPWATER(TF)/1000

C Set up the differential equation for the collector in the form dT/dt=AT+B
BB=QTH/CAPNODE+FLOW*CPF*TFIN(J)/CAPNODE
AA=-FLOW*CPF/CAPNODE

C Solve the differential equation analytically.
IF(AA.EQ.0.) THEN
   TFINAL(J)=TFINIT(J)+BB*DELT
   TBAR(J)=TFINIT(J)+BB*DELT/2.
ELSE
   TFINAL(J)=TFINIT(J)*(EXP(AA*DELT))
   TBAR(J)=1./AA*DELT*(TFINIT(J)+BB/AA)*
   1
   ((EXP(AA*DELT))-1.)-BB/AA
ENDIF

C Update energy sums and temp sums
QTHSUN=QTHSUN+QTH
QLEGCSUN=QLEGCSUN+QLEG
QINSULSUN=QINSULSUN+QINS
QCONVSUN=QCONVSUN+QCONV
QRADSUN=QRADSUN+QRAD
TGSUM=TGSUM+TG
TTSUM = TTSUM + TT
TPSUM = TPSUM + TP
TCSUM = TCSUM + TC
TGMIDSUM = TGMIDSUM + TGMID
TCOVSUM = TCOVSUM + TCOVER

CONTINUE

SET THE OUTPUTS

C     Outlet fluid temperature (degC)
     OUT(1) = TBAR(CELLS) - TZERO
C     Outlet flow rate (kg/hr)
     OUT(2) = FLOW
C     Electrical output (kJ/hr)
     OUT(3) = QELECSUM
C     Thermal output (kJ/hr)
     OUT(4) = QTHSUM
C     Insulation losses (kJ/hr)
     OUT(5) = QINSULSUM
C     Convection losses (kJ/hr)
     OUT(6) = QCONVSUM
C     Radiation losses (kJ/hr)
     OUT(7) = QRADSUM
C     Mean cell temperature (degC)
     OUT(8) = TCSUM/CELLS - TZERO
C     Mean glass temperature (degC)
     OUT(9) = TGSUM/CELLS - TZERO
C     Mean plate temperature (degC)
     OUT(10) = TPSUM/CELLS - TZERO
C     Mean tube temperature (degC)
     OUT(11) = TTSUM/CELLS - TZERO
C     Mean mid glass/silicone temperature (degC)
     OUT(12) = TGMIDSUM/CELLS - TZERO
C     Mean insulation cover temperature (degC)
     OUT(13) = TCOVSUM/CELLS - TZERO

Enter final temp data into store for next timestep
DO 80 J=1,CELLS
     STORE(J) = TFINAL(J)
CONTINUE
C Enter inlet temp data into store for next timestep
   DO 90 J=1,CELLS
       S(ISTORE+(J-1)+CELLS)=TFIN(J)
90    CONTINUE

   RETURN 1
   END

C----------------------------------------

    REAL FUNCTION QTHFUNC1(QTHERMAL)

C This function calculates the equations that
C describe the collector.
C The physically meaningless objective function
C is returned.
C QTHERMAL is the guess,
C TF is the known fluid temp that varies for each node

c   External functions

        EXTERNAL VISCOSITY, CONDUCTIVITY, CPWATER, DENSITY

C   Internal variables

C Radiation incident on the receiver
    REAL QSUN
C Coefficient of convection for air past the receiver
    REAL HCCONV
C Viscosity of water at atmospheric pressure
    REAL MU_W
C Conductivity of water at atmospheric pressure
    REAL K_W
C CP value of water at atmospheric pressure
    REAL CP_W
C Density of water at atmospheric pressure
    REAL DENSITY_W
C Prandtl number of water at atmospheric pressure
    REAL PR
C Hydraulic diameter of receiver
    REAL DH
C  Bulk fluid velocity through receiver
    REAL VF
C  Reynolds number
    REAL RE
C  Nusselt number
    REAL NU
C  Coefficient of convection for water in the tube
    REAL HC
C  Short term variable for heat transfer
    REAL H1, H2, H3
C  Estimate of average of insulation cover temp and ambient temp
    REAL TMEAN

C  Common block variables used in functions
    COMMON/PVTPARAMS/ID,WIDTH,LENGTH,CELLS,REFLM,SHAPE,SHADE,DIRT,CW0,
    CW1,CW2,WIND,MODE,TF,XSAREA,PERIM,FLOW,FH,HCTF,TT,TP,UP1,
    WPT,UISUL,WINSUL,TAMB,SBC,EMISCOV,WCOVER,TCOVER,QINS,
    QCP,UCP,WCP,QUELC,REFEFF,BETA,REFTEMP,TC,QABSCELLS,
    TRANSABS,QCG1,TGMD,UCG,WCG,QABSGLAS,COVERABS,QCG2,TG,
    QConv,WGLASS,QRAD,EMIS,UNIFORMITY

C  Re-declaration of parameters (from main program)
    INTEGER MODE
    INTEGER CELLS
    REAL REFFE
    REAL REFFTEMP
    REAL BETA
    REAL UNIFORMITY
    REAL LENGTH
    REAL WIDTH
    REAL REFLM
    REAL SHAPE
    REAL TRANSABS
    REAL COVERABS
    REAL EMIS
    REAL MASS
    REAL CP
    REAL TSTART
    REAL WGLASS
    REAL UCG
    REAL WCG
    REAL UCP
REAL WCP
REAL UPT
REAL WPT
REAL UINSUL
REAL WINSUL
REAL WCOVER
REAL EMISCOV
REAL CW0,CW1,CW2
REAL HCTF
REAL PERIM
REAL XSAREA
REAL FH

C Inputs used in the model
REAL TFI
REAL FLOW
REAL ID
REAL TAMB
REAL WIND
REAL SHADE
REAL DIRT

C Variables used within the model

C Electrical output
REAL QELEC
C Convective heat loss from glass
REAL QCONV
C Radiative heat loss from glass
REAL QRAD
C Heat loss through the insulation
REAL QINS
C Tube temperature
REAL TT
C Plate temperature
REAL TP
C Cell temperature
REAL TC
C Glass surface temperature
REAL TG
C Mid glass/silicone temperature
REAL TGMID

C Insulation cover temperature
REAL TCOVER

C Calculate the radiation incident on the cells
QSUN=ID*WIDTH*LENGTH/CELLS*REFL*SHAPE*SHADE*DIRT

C Calculate the convection coefficient due to wind
HCCONV=CW0+C*WIND+C*WIND*WIND

C Calculate convection coeff in the pipe if required
IF (MODE.EQ.2) THEN

C Calculate saturated water properties at atmospheric pressure
based on curve fits from Incropera and De Witt, 'Fundamentals
of Heat and Mass Transfer'

MU_W=VISCOSITY(TF)
K_W=CONDUCTIVITY(TF)
CP_W=CPWATER(TF)
DENSITY_W = DENSITY(TF)

C Prandtl number
PR = CP_W*MU_W/K_W

C DH - Hydraulic diameter - defined as 4 x Flow cross sectional area / wetted perimeter
DH = 4*XSAREA/PERIM

C VF - Fluid velocity (m/s) (also convert kg/hr to m3/s)
VF = FLOW/3600000/XSAREA

C Reynolds number (based on hydraulic diameter)
RE = DENSITY_W*VF*DH/MU_W

C The accuracy of this method is not so good for Reynolds numbers below about 4000
C Experimentally, it was found that using unmodified Dittus-Boelter seemed to produce
C a decent fit down to about Re = 1000 (reasons given in thesis). Below 1000 the
C Nusselt is set to 5 but this is only so the model doesn't crash. Not recommended.

IF (RE.LT.1000) THEN

NU = 10 ! Rough value to stop model crashing when there is no flow
ELSEIF ((RE.GE.1000).AND.(RE.LE.4000)) THEN

C Use Dittus - Boelter to calculate Nu
NU = 0.023*RE**0.8*PR**0.4

ELSE

C Use Dittus - Boelter to calculate Nu
NU = 0.023*RE**0.8*PR**0.4

C Adjust to account for the internal fins
NU = NU * FH

ENDIF

C Calculate coefficient of convection based on the hydraulic diameter
HC = NU*K_W/DH

C Convert units back from W/m2.K to kJ/hr.m2.K
HC = HC * 3.6

C Multiply HC through but the wetted surface area
HCTF = HC*PERIM*LENGTH/CELLS

ENDIF

C Write equations that describe system without air gap

C Calculate tube temperature
TT=QThermal/HCTF+TF

C Calculate plate temperature
TP=QThermal/(UPT*LENGTH/CELLS*WPT)+TT

C Calculate cover temperature by working out heat transfer coeffs for
C a) conduction through insulation
H1 = UINSUL*LENGTH/CELLS*WINSUL
C b) radiation loss from the cover
C estimate of mean temp between cover and ambient
TMEAN = TAMB + 5
H2 = 4*SBC*EMISCOV*WCOVER*LENGTH/CELLS*TMEAN**3
C c) convection loss from the cover
H3 = HCCONV*WCOVER*LENGTH/CELLS
C Now put it together to calculate temperature of the cover
TCOVER = (H1*TP+(H2+H3)*TAMB)/(H1+H2+H3)

C Calculate losses through the insulation
QINS = H1 * (TP-TCOVER)

C Calculate heat transfer from the cells to the plate
QCP = QTHERMAL + QINS

C Calculate the temperature of the cells
TC = QCP/(UCP*LENGTH/CELLS*WCP)+TP

C Calculate the electrical output
QELEC=QSUN*REFEFF*EXP(BETA*(TC-REFTEMP))*UNIFORMITY

C Calculate the solar absorption by the cells
QABSCELLS = QSUN*TRANSABS*(1-COVERABS)

C Calculate QCG1
QCG1=QABSCELLS-QELEC-QCP

C Calculate temperature at glass midpoint
TGMID = TC-QCG1/(2*UCG*WCG*LENGTH/CELLS)

C Calculate the solar absorption in the cover glass/silicone
QABSGLASS = QSUN*TRANSABS*COVERABS

C Calculate QCG2
QCG2=QCG1+QABSGLASS

C Calculate the glass temperature
TG = TGMID - QCG2/(2*UCG*WCG*LENGTH/CELLS)

C Calculate convection losses from the glass cover
QCONV=HCCONV*WGLASS*LENGTH/CELLS*(TG-TAMB)

C Calculate radiation losses from the glass cover
QRAD=SBC*EMIS*WGLASS*LENGTH/CELLS*(TG**4-TAMB**4)

C Calculate objective function that is to be solved
QTHFUNC1=QCG2-(QRAD+QCONV)

END
Bisection method

RECURSIVE SUBROUTINE BISECTION(FUNC,X1,X2,XACC,XROOT)

Using bisection, finds the root of a function FUNC known to lie between
X1 and X2. The root, returned as XROOT, will be refined until its accuracy
is +/- XACC

COMMON/LUNITS/LUR,LUW,IFORM,LUK
REAL FMID,F,XROOT,DX,XMID
PARAMETER (JMAX=50)

FMID=FUNC(X2)
F=FUNC(X1)

IF(F*FMID.GE.0.) THEN
   WRITE(LUW,'(A,I3,A,I3,A)') 'ERROR at Type 262 (Unit',unit,
1       '): Bisection method out of range'
   CALL MYSTOP(1001)
   ENDIF

IF(F.LT.0.) THEN
   XROOT=X1
   DX=X2-X1
ELSE
   XROOT=X2
   DX=X1-X2
ENDIF

DO 15000 J=1,JMAX
   DX=DX*.5
   XMID=XROOT+DX
   FMID=FUNC(XMID)
   IF(FMID.LT.0.) XROOT=XMID
   IF(ABS(DX).LT.XACC .OR. FMID.EQ.0.) RETURN
15000 CONTINUE
END
SUBROUTINE ZBRA CloZ(FUNC,X1,X2,SUCCESS)

C Given a function FUNC and an initial guessed range X1 to X2, the routine
C expands the range geometrically until a root is bracketed by the returned
C values of X1 and X2.

PARAMETER (FACTOR=1.6,NTRY=50)
LOGICAL SUCCESS
IF(X1.EQ.X2)PAUSE 'You have to guess an initial range'
F1=FUNC(X1)
F2=FUNC(X2)
SUCCESS=.TRUE.
DO 16000 J=1,NTRY
   IF (F1*F2.LT.0.)RETURN
   IF(ABS(F1).LT.ABS(F2))THEN
      X1=X1+FACTOR*(X1-X2)
      F1=FUNC(X1)
   ELSE
      X2=X2+FACTOR*(X2-X1)
      F2=FUNC(X2)
   END IF
16000 CONTINUE
SUCCESS=.FALSE.
RETURN
END

C ---------------------------------------------------------------
C
C WATER PROPERTIES - from curve fits to Incropera and De Witt, Saturated water
C at atmospheric pressure
C
C Real FUNCTION CPWATER(T)
real T
CPWATER=0.0000032759702*T**4 - 0.0043807394*T**3 + 2.2005092*T**2
1 - 491.60617*T + 45358.904
end

Real FUNCTION CONDUCTIVITY(T)
real T
CONDUCTIVITY=-0.48064+0.0058471*T-0.000073317*T**2
end
REAL FUNCTION VISCOSITY(T)
real T
VISCOSITY=0.00000000028665*T**4 - 0.00000039376*T**3 + 0.00020328*T**2 - 0.046803*T + 0.40639
END

REAL FUNCTION DENSITY(T)
real T
T = T - 273.15
DENSITY=0.0000149*T**3 - 0.0057637*T**2 + 0.0063843*T + 1000.2418616
T = T + 273.15
END
A2 Fortran code for the End Loss TRNSYS component

SUBROUTINE TYPE123 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
C**********************************************************************
C Object: End Loss Modifier
C IISiBat Model: EndLoss
C
C Author: Joe Coventry
C Editor: Joe Coventry
C Date: 20/6/2000 last modified: 20/6/2000
C
C Calculates the end losses from a parabolic mirror
C**********************************************************************
C
C STANDARD TRNSYS DECLARATIONS
DOUBLE PRECISION XIN,OUT
INTEGER NI,NP,ND,NO
PARAMETER (NI=1,NP=3,NO=1,ND=0)
INTEGER*4 INFO,ICNTRL
REAL T,DTDT,PAR,TIME
DIMENSION XIN(NI),OUT(NO),PAR(NP),INFO(15)
CHARACTER*3 YCHECK(NI),OCHECK(NO)
DATA RDCONV/0.017453/
C**********************************************************************
C IF ITS THE FIRST CALL TO THIS UNIT, DO SOME BOOKKEEPING
IF (INFO(7).GE.0) GO TO 100
C
C FIRST CALL OF SIMULATION, CALL THE TYPECK SUBROUTINE TO CHECK THAT THE
C USER HAS PROVIDED THE CORRECT NUMBER OF INPUTS, PARAMETERS, AND
C DERIVS

INFO(6)=NO
INFO(9)=1
CALL TYPECK(1,INFO,NI,NP,ND)
RETURN 1

C END OF THE FIRST ITERATION BOOKKEEPING
C**********************************************************************
GET THE VALUES OF THE PARAMETERS FOR THIS COMPONENT

Trough_Length=PAR(1)
Trough_Width=PAR(2)
Focal_Length=PAR(3)

GET THE VALUES OF THE INPUTS TO THIS COMPONENT

Incidence_Angle=XIN(1)

End losses calculated based on the geometry of the trough and the incidence angle of light on a single axis tracking trough

AAA = Focal_Length/Trough_Length
BBB = (Trough_Width**2)/(48*Focal_Length**2)

End_Loss_Factor = 1-AAA*(1+BBB)*tan(Incidence_Angle*RDCNV)

SET THE OUTPUTS

End Loss Factor
OUT(1)=End_Loss_Factor

RETURN 1

END
A3 Fortran code for the Fin-tube TRNSYS component

SUBROUTINE TYPE273 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
C********************************************************************
C Object: Improved fin tube model
C IISiBat Model: Type 273
C
C Author: Joe Coventry
C Editor:
C
C STANDARD TRNSYS DECLARATIONS
DOUBLE PRECISION XIN,OUT
INTEGER NIMAX,NPMAX,ND,NO
PARAMETER (NIMAX=4,NPMAX=11,NO=2,ND=0)
INTEGER*4 INFO,ICNTRL
REAL T,DTDT,PAR,TIME
DIMENSION XIN(NIMAX),OUT(NO),PAR(NPMAX),INFO(15)
CHARACTER*3 YCHECK(NIMAX),OCHECK(NO)
C
C My declarations
C
C Some constants used in the model
DATA CPF/4.18/IUNIT/0/

C PARAMETERS
C
C Length of the finned tube
REAL L
C Number of fins
REAL NFINS
C Area of the fins
REAL AFINS
C Diameter of the tube
REAL DTUBE
C Thickness of the fins
REAL TFINS
C Conductivity of the fins
REAL KFINS
C Number of discrete elements
REAL NODES
Heat capacity of the fluid
REAL CPF

Heat capacity of the pipe
REAL CPP

Mass of the fluid
REAL MASSF

Mass of the pipe
REAL MASSP

INPUTS

Inlet temperature to the fin-tube
REAL TIN

Ambient temperature
REAL TAMB

Flow rate (kg/hr)
REAL FLOW

Wind speed (m/s)
REAL WS

VARIABLES IN CALCULATION

Energy transfer through an element of the fin-tube
EXTERNAL QTHFIN

Maximum number of discrete elements
PARAMETER (MAXNODES=100)

Sum of thermal energy transfer through the fin-tube
REAL QTHSUM

Fluid temperature
REAL TF

Temperature of the inlet of the node
REAL TFIN(MAXNODES)

Average temperature of the fluid in the node
REAL TBAR(MAXNODES)

Temperature of the fluid in the node at the previous time step
REAL TFINIT(MAXNODES)

Temperature of the fluid at the outlet of the node
REAL TFINAL(MAXNODES)

Boolean indicator of success of bisection method
LOGICAL SUCCESS

Need to include this file
INCLUDE '..\INCLUDE\PARAM.INC'
COMMON variables to be used in other subroutines
COMMON/FINPARAMS/ L, NFINS, AFINS, DTUBE, TFINS, KFINS,
   TIN, TAMB, FLOW, WS, TF, NODES
COMMON /SIM/ TIME0,TIMEF,DELT,IWARN
   COMMON /STORE/ NSTORE,IAV,S(NUMSTR)
COMMON/LUNITS/LUR,LUW,IFORM,LUK

If it is the very first call of the simulation then continue,
with these initialisation things otherwise go down to parameters.

IF (INFO(7).GE.0) GO TO 10

Set this to number of outputs
INFO(6)=NO

Set to 1 as routine depends on passage of time
INFO(9)=1

Set the common store size to the maximum number of cells
INFO(10)=MAXNODES

Check that the user has provided the right number of inputs, outputs
and derivatives

CALL TYPECK(1,INFO,NIMAX,NMAX,ND)

Set the first storage place in the middle of the allocated variables
ISTORE=INFO(10)

Get the values of the parameter for this component (only once)
10  IF(INFO(1).EQ. IUNIT) GO TO 30
   IUNIT=INFO(1)

   L=PAR(1)
   NFINS=PAR(2)
   AFINS=PAR(3)
   DTUBE=PAR(4)
   TFINS=PAR(5)
   KFINS=PAR(6)
   NODES=INT(PAR(7)+0.001)
CPP = PAR(8)
MASSP = PAR(9) + 0.01
CPF = PAR(10)
MASSF = PAR(11) + 0.01

C Set the initial values for fluid temperature, equivalent the average temp
C if the collector was left for a while with no losses after initial conditions.
C Also set initial values of QLD to zero
IF (INFO(7),EQ.-1) THEN
   TSTART = 300
   DO 20 J = 1, NODES
       S(ISTORE+(J-1)) = TSTART
   CONTINUE
   Return 1
END IF

C-----------------------------------------------------------------------

30 CONTINUE

C Set the storage place for this particular component
ISTORE = INFO(10)

C GET THE VALUES OF THE INPUTS TO THIS COMPONENT
TIN = XIN(1) + 273.13
FLOW = XIN(2)
TAMB = XIN(3) + 273.13
WS = XIN(4)

C Retrieve initial fluid temps and QLD from the s-array
DO 40 J = 1, NODES
    TFINIT(J) = S(ISTORE+(J-1))
40 CONTINUE
C-----------------------------------------------------------------------
C Thermal performance
C-----------------------------------------------------------------------

IF (FLOW.EQ.0.) GO TO 70

C Set the capacitance for each cell node

CAPNODE = (MASSF*CPF + MASSP*CPP)/NODES
QTHSUM = 0
DO 60 J=1,NODES

C Set the inlet fluid temp
   IF (J.EQ.1) THEN
      TFIN(J)=TIN
   ELSE
      TFIN(J)=TBAR(J-1)
   ENDIF

C Set fluid temp to original fluid temp of the node (at prev. time step)
   TF=TFINIT(J)

C Note: because of this assumption the model may be unstable in the initial time steps if mass flow is low, or the heat exchanger is a lot longer than it needs to be.

C The following uses the Bisection Algorithm routine to solve the simultaneous equations describing this system.

C Make guess of values
   QTHLOWER=0
   QTHUPPER=4760*L/NODES
   CALL ZBRAC3(QTHFIN,QTHLOWER,QTHUPPER,SUCCESS)
   CALL BISECTION3(QTHFIN,QTHLOWER,QTHUPPER,0.1,QTH)

C Set up the differential equation for the collector in the form dT/dt=AT+B
   BB=-QTH/CAPNODE+FLOW*CPF*TFIN(J)/CAPNODE
   AA=-FLOW*CPF/CAPNODE

C Solve the differential equation analytically.

   IF(AA.EQ.0.) THEN
      TFINAL(J)=TFINIT(J)+BB*DELT
      TBAR(J)=TFINIT(J)+BB*DELT/2.
   ELSE
      TFINAL(J)=TFINIT(J)*((EXP(AA*DELT))
      +BB/AA*(EXP(AA*DELT))-BB/AA
      TBAR(J)=1./AA/DELT*(TFINIT(J)+BB/AA)*
      ((EXP(AA*DELT))-.1.)*BB/AA
ENDIF

C Update energy sums and temp sums
    QTHSUM=QTHSUM+QTH

60    CONTINUE

C SET THE OUTPUTS
70    CONTINUE
C Outlet fluid temperature
    IF (FLOW.EQ.0) THEN
        OUT(1) = TAMB-273.13
        DO 75 J=1,NODES
            S(ISTORE+(J-1))=TFINIT(J)
        75    CONTINUE
    ELSE
        OUT(1)=TBAR(NODES)-273.13
        DO 80 J=1,NODES
            S(ISTORE+(J-1))=TFINAL(J)
        80    CONTINUE
    ENDIF

C Outlet flow rate
    OUT(2)=FLOW

RETURN 1
END

C*************************************************************************************************************************
C
C THIS FUNCTION CALCULATES THE FIN EFFICIENCY (EFFECTIVENESS) OF AN ANNULAR FIN OF CONSTANT THICKNESS.
C
C ALPHA = RADIUS AT FIN BASE / RADIUS AT FIN TIP
C BETA = RADIUS AT FIN TIP * (SQRT (2 * CONVECTION COEFFICIENT / FIN CONDUCTIVITY * FIN THICKNESS))
C
FUNCTION FINEFF(ALPHA,BETA)
    REAL I0,I1,K0,K1
    ALPBET = ALPHA * BETA
    CALL BESSEL2(ALPBET,I0,I1,K0,K1)
XI0 = I0
XI1 = I1
XK0 = K0
XK1 = K1
CALL BESSEL2(BETA,I0,I1,K0,K1)
YI0 = I0
YI1 = I1
YK0 = K0
YK1 = K1
FINEFF = 2*ALPHA/BETA/(1. - ALPHA**2)*(XK1*YI1 - XI1*YK1)/
        (XK0*YI1 + XI0*YK1)
RETURN
END

C
C*---------------------------------------------------------------
C*---------------------------------------------------------------
C
C THIS SUBROUTINE USES POLYNOMIAL APPROXIMATIONS TO EVALUATE
C THE BESSEL FUNCTIONS. THE APPROXIMATIONS ARE FROM ABRAMOWITZ
C AND STEGUN, HANDBOOK OF MATHEMATICAL FUNCTIONS, DOVER
C PUBLICATIONS, INC., NEW YORK, NY.
C
SUBROUTINE BESSEL2(X,I0,I1,K0,K1)
COMMON /UNIT5/ LUR,LUW,IFORM,LUK
REAL X,I0,I1,K0,K1,IT
C
C THE FOLLOWING DATA STATEMENTS CONTAIN THE COEFFICIENTS TO
C THE POLYNOMIALS.
C
C I0
DATA A0/1.0/,A1/3.5156229/,A2/3.0899424/,A3/1.2067492/
DATA A4/0.2659732/,A5/0.0360768/,A6/0.0045813/
C I0
DATA B0/0.39894228/,B1/0.01328592/,B2/0.00225319/
DATA B3/-0.00157565/,B4/0.00916281/,B5/-0.02057706/
DATA B6/0.02635537/,B7/-0.01647633/,B8/0.00392377/
C I1
DATA C0/0.5/,C1/0.87890594/,C2/0.51498869/,C3/0.15084934/
DATA C4/0.02658733/,C5/0.00301532/,C6/0.00032411/
C I1
DATA D0/0.39894228/,D1/-0.03988024/,D2/-0.00362018/
DATA D3/0.00163801/,D4/-0.01031555/,D5/0.02282967/
DATA D6/-0.02895312/,D7/0.01787654/,D8/-0.00420059/
C K0
DATA E0/-0.57721566/,E1/0.4227842/,E2/0.23069756/
DATA E3/0.0348859/,E4/0.00262698/,E5/0.0001075/,E6/0.0000074/
C K0
DATA F0/1.25331414/,F1/-0.07832358/,F2/0.02189568/
DATA F3/-0.01062446/,F4/0.00587872/,F5/-0.0025154/
DATA F6/0.00053208/
C K1
DATA G0/1.0/,G1/0.15443144/,G2/-0.67278579/,G3/-0.18156897/
DATA G4/-0.01919402/,G5/-0.00110404/,G6/-0.00004686/
C K1
DATA H0/1.25331414/,H1/0.23498619/,H2/-0.0365562/
DATA H3/0.01504268/,H4/-0.00780353/,H5/0.00325614/
DATA H6/-0.00068245/
C
IF (X.LT. -3.75) THEN
WRITE(LUW,100) 164,52,52,X
CALL MYSTOP(164)
RETURN
ENDIF
T=X/3.75
TT=T**T
C
C I0
C
IF (X.LE. 3.75) THEN
I0=A0+TT*(A1+TT*(A2+TT*(A3+TT*(A4+TT*(A5+TT*A6))))))
ELSE
IT=1/T
I0=(B0+IT*B1+IT*B2+IT*B3+IT*B4+IT*B5+IT*B6+IT* .(B7+IT*B8)))/(SQRT(X)*EXP(-X))
ENDIF
C
C I1
C
IF (X.LE. 3.75) THEN
I1=(C0+TT*(C1+TT*(C2+TT*(C3+TT*(C4+TT*(C5+TT*C6)))))))*X
ELSE
IT=1/T
I1=(D0+IT*D1+IT*D2+IT*D3+IT*D4+IT*D5+IT*D6+IT* .(D7+IT*D8)))/(SQRT(X)*EXP(-X))
END IF
C
C K0
C
IF (X .LE. 0.0) THEN
   WRITE(LUW,100) X
   CALL MYSTOP(1001)
   RETURN
END IF
X1 = (X/2)**2
X2 = 2./X
IF (X .LE. 2.0) THEN
   K0=ALOG(X/2)**10+E0+X1**(E1+X1**(E2+X1**(E3+X1**(E4+X1**
   .           (E5+X1*E6))))))
ELSE
   K0=(F0+X2**(F1+X2**(F2+X2**(F3+X2**(F4+X2**(F5+X2*F6)))))))
     /(SQRT(X)*EXP(X))
END IF
RETURN
C
C K1
C
IF (X .LE. 2.0) THEN
   K1=(X*ALOG(X/2)**11+G0+X1**(G1+X1**(G2+X1**(G3+X1**(G4+X1**
   .           (G5+X1*G6))))))/X
ELSE
   K1=(H0+X2**(H1+X2**(H2+X2**(H3+X2**(H4+X2**(H5+X2*H6)))))))
     /(SQRT(X)*EXP(X))
END IF
RETURN
C
C FORMATS
C
100 FORMAT(//,1X;;;; ERROR;;;;,8X,'TRNSYS ERROR # ',I3,/1X,
   .UNIT ',I3,' TYPE ',I3,' COOLING COIL',/1X,
   .THE BESSEL FUNCTION CALLED FROM THE COOLING COIL SUBROUTINE COULD
   .NOT BE',I1X,'EVALUATED AT THE GIVEN VALUE OF ',F5.2,',')
END
C
C--------------------------------------------------------------------------
Function that calculates the heat lost from the fins

REAL FUNCTION QTHFIN(QGUESS)

COMMON/FINPARAMS/ L, NFINS, AFINS, DTUBE, TFINS, KFIN,
1 TIN, TAMB, FLOW, WS, TF, NODES

C Redecclare the common variables
REAL L, NFINS, AFINS, DTUBE, TFINS, KFIN, QTHSUM
REAL TIN, TAMB, FLOW, WS, TF, NODES

C Nusselt number, forced convection in air
REAL NUF

C Nusselt number, natural convection in air
REAL NUN

C Reynolds number, forced convection in air
REAL REF

C Equivalent diameter of the fins
REAL DFIN

C Temperature at the base of the fins
REAL TBASE

C Rayleigh number
REAL RA

C Maximum of forced and natural convection nusselt numbers
REAL NU

C Nusselt number for the fluid in the pipe
REAL NUFLUID

C Coefficient of convection between fin and ambient air
REAL HCFA

C Reynolds number for fluid
REAL REFLUID

C Prandtl number for fluid
REAL PRFLUID

C Viscosity for water
REAL MU_W

C Conductivity of water
REAL K_W

C Specific heat, water
REAL CP_W

C Coefficient of convection between the fin base and the water
REAL HCFF

C Variables used in calculation
REAL A,B, Q1, Q2
Note: these are in SI units from the back of Holman
DATA PRAIR/0.708/, CONDAIR/0.02624/,PI/3.14159/
DATA VISAIR/15.69E-6/, ALPHA/0.22160E-4/

C Reynolds number and prandtl number for the fluid in the pipe

MU_W=VISCOSITY2(TF)
K_W=CONDUCTIVITY2(TF)
CP_W=CPWATER2(TF)

REFLUID = 4.*FLOW/(3600.*PI*DTUBE*MU_W)
PRFLUID = CP_W*MU_W/K_W

C Nusselt number for the fluid in the pipe

IF (REFLUID .GE.2300) THEN ! then turbulent
  NUFLUID = 0.023*REFLUID**0.8*PRFLUID**0.4
ELSE ! else laminar
  NUFLUID = 3.66 + (0.0668*(DTUBE/L)*REFLUID*PRFLUID)/
         1 +0.04*((DTUBE/L)*REFLUID*PRFLUID)**0.666)
ENDIF

C Calculate convection coefficient between finbase and fluid

HCFF = NUFLUID * K_W / DTUBE *3.6

C Area of the tube inner surface

AREAT = L/NODES * DTUBE*PI

C Calculate the temperature at the base of the fin by assuming a
guessed energy transfer

TBASE = TF - QGUESS/(HCFF*AREAT)

C Equivalent diameter for fins

DFIN = (4.*AFINS/PI)**0.5

C Nusselt for forced convection of air
REF = WS*DFIN/2/VISAIR
NUF = 0.332*REF**0.5*PRAIR**0.33

C Nusselt for natural convection of air (use TF instead of TBASE)
RA = 9.81*(1/TAMB)*ABS(TF-TAMB)*(DFIN/2)**3/(VISAIR*ALPHA)
NUN = 0.68 + 0.670*RA**0.25/(1+(0.492/PRAIR)**(9/16))**(4/9)

C Take whichever nusselt number is largest
NU = MAX(NUN,NUF)

C Calculate convection coefficient between fin and ambient air
HCFA = NU * CONDAIR / (DFIN/2) * 3.6

C Calculate the fin efficiency
C
C   ALPHA = RADIUS AT FIN BASE / RADIUS AT FIN TIP
C   BETA = RADIUS AT FIN TIP *
C   (SQRT (2 * CONVECTION COEFFICIENT /
C   FIN CONDUCTIVITY / FIN THICKNESS))
A = DTUBE/DFIN
B = DFIN/2*(SQRT(2*HCFA/KFINS/TFINS))
FEFF = FINEFF(A,B)

C Calculate the area of the fin (x2 to include both sides)
AREAF = NFINS*L/NODES*AFINS*2

C Find the value of Q using the base temp and calculate error
Q1 = FEFF*HCFA*(TBASE-TAMB)*AREAF
QTHFIN = Q1-QGUESS

END
WATER PROPERTIES, taken from the Type 60 tank model

Real FUNCTION CPWATER2(T)
real T
CPWATER2 = 45359.491.6*T+2.2005*T*T-.0043807*T**3+
& 3.276d-6*T**4
end

Real FUNCTION CONDUCTIVITY2(T)
real T
CONDUCTIVITY2 =-.48064+.0058471*T-7.3317d-6*T*T
end

REAL FUNCTION VISCOSITY2(T)
real T
VISCOSITY2 =0.23873-0.26422e-02*T+1.1062e-05*T**2
& -2.0705e-08*T**3 + 1.4593e-11*T**4
END

C---------------------------------------------------------------
C  Bisection method
C
C  RECURSIVE SUBROUTINE BISECTION3(FUNC,X1,X2,XACC,XROOT)
C
C  Using bisection, finds the root of a function FUNC known to lie between
C  X1 and X2. The root, returned as XROOT, will be refined until its accuracy
C  is +/- XACC
C
COMMON/LUNITS/LUR,LUW,IFORM,LUK
REAL FMID,F,XROOT,DX,XMID
PARAMETER (JMAX=50)

FMID=FUNC(X2)
F=FUNC(X1)

IF(F*FMID.GE.0.) THEN
  WRITE(LUW,['(A,I3,A,I3,A)'] 'ERROR at Type 262 (Unit',unit,
1 '!): Bisection method out of range'
CALL MYSTOP(1001)
ENDIF
IF (F .LT. 0.) THEN
   XROOT = X1
   DX = X2 - X1
ELSE
   XROOT = X2
   DX = X1 - X2
ENDIF

DO 15000 J = 1, JMAX
   DX = DX * .5
   XMID = XROOT + DX
   FMID = FUNC(XMID)
   IF (FMID .LT. 0.) XROOT = XMID
   IF (ABS(DX) .LT. XACC .OR. FMID .EQ. 0.) RETURN
15000 CONTINUE
END

C-----------------------------------------------

SUBROUTINE ZBRAC3 (FUNC,X1,X2,SUCCESS)
C
C Given a function FUNC and an initial guessed range X1 to X2, the routine
C expands the range geometrically until a root is bracketed by the returned
C values of X1 and X2.

PARAMETER (FACTOR=1.6,NTRY=50)
LOGICAL SUCCESS
IF (X1.EQ.X2) PAUSE 'You have to guess an initial range'
F1 = FUNC(X1)
F2 = FUNC(X2)
SUCCESS = .TRUE.
DO 16000 J = 1, NTRY
   IF (F1 * F2 .LT. 0.) RETURN
   IF (ABS(F1) .LT. ABS(F2)) THEN
      X1 = X1 + FACTOR * (X1 - X2)
      F1 = FUNC(X1)
   ELSE
      X2 = X2 + FACTOR * (X2 - X1)
      F2 = FUNC(X2)
   END IF
16000 CONTINUE
SUCCESS = .FALSE.
RETURN
END
A4 Fortran code for the controller component

SUBROUTINE TYPE275 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
C***************************************************************************
C Object: Bruce Hall Controller
C IISiBat Model: Bruce Hall controller
C
C Author: Joe Coventry
C Editor:
C Date:  2/9/2002 last modified: 25/5/2002
C
C Controller for Bruce Hall project. Note that this component is also useful
C for CHAPS systems with a single pump. The second 'tank pump' can simply be
C ignored.

C STANDARD TRNSYS DECLARATIONS
DOUBLE PRECISION XIN,OUT
INTEGER NI,NP,ND,NO
PARAMETER (NI=6,NP=9,NO=5,ND=0)
INTEGER*4 INFO,ICNTRL
REAL T,DTDT,PAR,TIME
DIMENSION XIN(NI),OUT(NO),PAR(NP),INFO(15)
CHARACTER*3 YCHECK(NI),OCHECK(NO)

C PARAMETERS

C Controller mode 1: Delta T.  2: Fixed outlet temp (not yet available)
INTEGER ControllerMode
C Bypass setpoint for tank temperature for diverting flow to the cooling fins
REAL TTset
C Upper deadband for the collector pump (pump 1)
REAL DBupperC
C Lower deadband for the collector pump (pump 1)
REAL DBlowerC
C Upper deadband for the tank pump (pump 2)
REAL DBupperT
C Lower deadband for the tank pump (pump 2)
REAL DBlowerT
C Upper deadband for the bypass valve operation.
REAL DBupperV
C Lower deadband for the bypass valve operation
REAL DBlowerV
C Temperature difference between ambient and the control setpoint for
C operation of the pumps when the valve is bypassing fluid to the cooling fins
REAL TVDT

C INPUTS
C Temperature at the outlet of the collectors
REAL TCout
C Monitoring temperature for the tank
REAL TTout
C Control signal for pump 1 at the previous iteration
INTEGER CFpump1old
C Control signal for pump 2 at the previous iteration
INTEGER CFpump2old
C Control signal for the bypass valve at the previous iteration
INTEGER CFvalveold
C Ambient temperature
REAL Tamb

C OUTPUTS
C Control signal for pump 1
INTEGER CFpump1
C Control signal for pump 2
INTEGER CFpump2
C Control signal for the bypass valve
INTEGER CFvalve

C OTHER VARIABLES
C Setpoint for control of the bypass valve
REAL TVSet

C IF ITS THE FIRST CALL TO THIS UNIT, DO SOME BOOKKEEPING
IF (INFO(7).GE.0) GO TO 100

C FIRST CALL OF SIMULATION, CALL THE TYPECK SUBROUTINE TO CHECK THAT THE
C USER HAS PROVIDED THE CORRECT NUMBER OF INPUTS,PARAMETERS, AND DERIVS
INFO(6)=NO
INFO(9)=1
CALL TYPECK(1,INFO,NI,NP,ND)

RETURN 1

C END OF THE FIRST ITERATION BOOKKEEPING
C---------------------------------------------------------------
C GET THE VALUES OF THE PARAMETERS FOR THIS COMPONENT
100 CONTINUE

C Mode 2 not enabled
ControllerMode=INT(PAR(1)+0.01)
TTset=PAR(2)
DBupperC=PAR(3)
DBlowerC=PAR(4)
DBupperV=PAR(5)
DBlowerV=PAR(6)
DBupperT=PAR(7)
DBlowerT=PAR(8)
TVDT=PAR(9)

C GET THE VALUES OF THE INPUTS TO THIS COMPONENT
TCout=XIN(1)
TTout=XIN(2)
CFpump1old=INT(XIN(3)+0.01)
CFvalveold=INT(XIN(4)+0.01)
CFpump2old=INT(XIN(5)+0.01)
Tamb = XIN(6)
C---------------------------------------------------------------
C Set the control temp for the bypass valve
TVset = Tamb + TVDT
C---------------------------------------------------------------
C Put in something to stop the controller sticking

IF (INFO(7).EQ.0) THEN
  OUT(4)=0
  OUT(5)=0
END IF

C Main logic
C Mode 1 - Delta T
IF (ControllerMode.EQ.1) THEN

C Bypass solenoid valve control
IF ( (CFvalveold.EQ.1).AND.(TTout-TTset).GE.DBlowerV) THEN
    CFvalve=1
ELSEIF ( (CFvalveold.EQ.1).AND.(TTout-TTset).LT.DBlowerV) THEN
    CFvalve=0
ELSEIF ( (CFvalveold.EQ.0).AND.(TTout-TTset).GE.DBupperV) THEN
    CFvalve=1
ELSEIF ( (CFvalveold.EQ.0).AND.(TTout-TTset).LT.DBupperV) THEN
    CFvalve=0
ENDIF

C Case 1 - If valve is off
IF (CFvalve.EQ.0) THEN

    IF ( (CFpump2old.EQ.1).AND.(TCout-TTout).GE.DBlowerT) THEN
        CFpump2=1
        CFpump1=1
    ELSEIF ( (CFpump2old.EQ.1).AND.(TCout-TTout).LT.DBlowerT) THEN
        CFpump2=0
        CFpump1=0
    ELSEIF ( (CFpump2old.EQ.0).AND.(TCout-TTout).GE.DBupperT) THEN
        CFpump2=1
        CFpump1=1
    ELSEIF ( (CFpump2old.EQ.0).AND.(TCout-TTout).LT.DBupperT) THEN
        CFpump2=0
        CFpump1=0
    ENDIF

    C Case 2 - when the control valve is on
    ELSEIF (CFvalve.EQ.1) THEN
        CFpump2=0

    C Collector pump control

        IF ( (CFpump1old.EQ.1).AND.(TCout-TVset).GE.DBlowerC) THEN
            CFpump1=1
        ELSEIF ( (CFpump1old.EQ.1).AND.(TCout-TVset).LT.DBlowerC) THEN
            CFpump1=0
        ELSEIF ( (CFpump1old.EQ.0).AND.(TCout-TVset).GE.DBupperC) THEN
            CFpump1=1
        ELSEIF ( (CFpump1old.EQ.0).AND.(TCout-TVset).LT.DBupperC) THEN
            CFpump1=0
        ENDIF
BEGIN

C Has pump1 status changed?
IF (CFpump1.EQ.CFpump1old) THEN

C No
OUT(4) = INT(OUT(4)+0.1)
ELSE

C Yes
OUT(4) = INT(OUT(4)+1.1)
END IF

C Has pump2 status changed?
IF (CFpump2.EQ.CFpump2old) THEN

C No
OUT(5) = INT(OUT(5)+0.1)
ELSE

C Yes
OUT(5) = INT(OUT(5)+1.1)
END IF

C If this component is called 5 times or more in a timestep, then
simply stick to a value
IF (OUT(4).GE.5) THEN

C Fpump1=1
ENDIF

IF (OUT(5).GE.5) THEN

C Fpump2=1
ENDIF

C SET THE OUTPUTS
CONTINUE

C Collector Pump output control function
OUT(1)=CFpump1

C Bypass valve output control function
OUT(2)=CFvalve

C Tank pump output control function
OUT(3)=CFpump2

RETURN 1

END
A5 TRNSYS deck file for the system base case

VERSION 15

TRNSYS input file (deck) generated by IISiBat 3

on Wednesday, May 26, 2004 at 13:24

If you edit this file, use the File/Import TRNSYS Input File function in

IISiBat 3 to update the project.

If you have problems, questions or suggestions please contact your local

TRNSYS distributor or mailto:iisibat@cstb.fr

ASSIGN C:\trnsys15\IISiBat3\Data\Joe\modeld.LST 6

* CONTROL cards

* START, STOP and STEP

CONSTANTS 3
START=1
STOP=8760
STEP=.1

SIMULATION START STOP STEP

* User defined CONSTANTS

TOLERANCES 0.001 0.001

LIMITS 25 999 25

DFQ 1

WIDTH 80

LIST

MAP

SOLVER 0

Integration Convergence

Max iterations Max warnings Trace limit

TRNSYS numerical integration solver method

TRNSYS output file width, number of characters

NOLIST statement

MAP statement

Solver statement
* EQUATIONS "Calc. modified load"
* EQUATIONS 1
Modload = 10^7,1

* EQUATIONS "Calc. HW flow"
EQUATIONS 1
mdot = Modload/(4.18*(45-[7,2]+eq(45,[7,2])))

* EQUATIONS "Calc. solar input"
EQUATIONS 1
Qsun = 37.5[6,8]

* EQUATIONS "Otemp calc."
EQUATIONS 1
Idbout = [6,8]*[1-[17,1])

* Model "Canberra weather" (Type 9)
UNIT 5 TYPE 9  Canberra weather

PARAMETERS 36
* 1 Mode
-1
* 2 Header Lines to Skip
0
* 3 No. of values to read
10
* 4 Time interval of data
1
* 5 Interpolate or not?-1
-1
* 6 Multiplication factor-1
1.0
* 7 Addition factor-1
0
* 8 Interpolate or not?-2
-1
* 9 Multiplication factor-2
  1.0
* 10 Addition factor-2
  0
* 11 Interpolate or not?-3
  -3
* 12 Multiplication factor-3
  1.0
* 13 Addition factor-3
  0
* 14 Interpolate or not?-4
  -4
* 15 Multiplication factor-4
  10
* 16 Addition factor-4
  0
* 17 Interpolate or not?-5
  -5
* 18 Multiplication factor-5
  10
* 19 Addition factor-5
  0
* 20 Interpolate or not?-6
  6
* 21 Multiplication factor-6
  0.1
* 22 Addition factor-6
  0
* 23 Interpolate or not?-7
  7
* 24 Multiplication factor-7
  0.1
* 25 Addition factor-7
  0
* 26 Interpolate or not?-8
  8
* 27 Multiplication factor-8
  0.1
* 28 Addition factor-8
  0
* 29 Interpolate or not?-9
  -9
* 30 Multiplication factor-9
1.0
* 31 Addition factor-9
0
* 32 Interpolate or not?-10
-10
* 33 Multiplication factor-10
1.0
* 34 Addition factor-10
0
* 35 Logical unit
14
* 36 Format specification
1
(1x,3F2.0,5F3.0,F2.0,F1.0)

*** External files
ASSIGN C:\tmsys15\Weather\canberra.tmy 14

* Model "Tracking" (Type 16)
UNIT 6 TYPE 16  Tracking

PARAMETERS 9
* 1 Horiz. radiation mode
4
* 2 Tracking mode
3
* 3 Tilted surface mode
2
* 4 Starting day
1
* 5 Latitude
-35.2
* 6 Solar constant
4871
* 7 Shift in solar time
0.8
* 8 Not used
2
* 9 Solar time?
-1

INPUTS 7
* Canberra weather:Output 4 ->Total radiation on horizontal surface
5,4
* Canberra weather:Output 5 ->Direct normal beam radiation on horizontal
5,5
* Canberra weather:Time of last read ->Time of last data read
5,99
* Canberra weather:Time of next read ->Time of next data read
5,100
* [unconnected] Ground reflectance
0,0
* [unconnected] Slope of surface
0,0
* [unconnected] Azimuth of surface
0,0
*** INITIAL INPUT VALUES
0 0 0.0 1 0.2 19.18
36.03
*---------------------------------------------------------------------

* Model "AS4234 load data" (Type 9)
UNIT 7 TYPE 9 AS4234 load data

PARAMETERS 12
* 1 Mode
-1
* 2 Header Lines to Skip
2
* 3 No. of values to read
2
* 4 Time interval of data
1.0
* 5 Interpolate or not? -1
-1
* 6 Multiplication factor-1
1.0
* 7 Addition factor-1
0
* 8 Interpolate or not? -2
-2
* 9 Multiplication factor-2
1.0
* 10 Addition factor-2
* 11 Logical unit
11
* 12 Not used
-1

*** External files
ASSIGN C:\trnsys15\IIISiBat3\Data\Joe\Canload.txt 11

* Model "Tempering valve" (Type 11)
UNIT 8 TYPE 11  Tempering valve

PARAMETERS 2
* 1 Tempering valve mode
4
* 2 # of oscillations allowed
7

INPUTS 4
* AS4234 load data:Output 2 ->Inlet temperature
7,2
* Calc. HW flow:mdot ->Inlet flow rate
mdot
* Tank:Temperature to load ->Heat source temperature
13,3
* [unconnected] Set point temperature
0,0

*** INITIAL INPUT VALUES
20.0 100.0 55.0 45

* Model "Pump" (Type 3)
UNIT 9 TYPE 3  Pump

PARAMETERS 5
* 1 Maximum flow rate
1100
* 2 Fluid specific heat
4.190
* 3 Maximum power
540
* 4 Conversion coefficient
0.10
* 5 Power coefficient
0.5

INPUTS 3
* T-piece:Outlet temperature ->Inlet fluid temperature
15.1
* T-piece:Outlet flow rate ->Inlet mass flow rate
15.2
* Controller:Collector Pump output control function ->Control signal
18.1

*** INITIAL INPUT VALUES
35 36 1.0

* Model "3-way valve" (Type 11)
UNIT 10 TYPE 11 3-way valve

PARAMETERS 1
* 1 Controlled flow diverter mode
2

INPUTS 3
* CHAPS collector:TOUT ->Inlet temperature
12.1
* CHAPS collector:FLOW ->Inlet flow rate
12.2
* Controller:Bypass valve output control function ->Control signal
18.2

*** INITIAL INPUT VALUES
20.0 100.0 0.5

* Model "End loss" (Type 123)
UNIT 11 TYPE 123 End loss

PARAMETERS 3
* 1 Trough Length
24.21
* 2 Trough Width
1.55
* 3 Focal Length
.845

INPUTS 1
* Tracking:Incidence angle for surface 1 ->Incidence Angle
6,10
*** INITIAL INPUT VALUES
0
*-------------------------------------------------------------------------------------------------------------------------------------

* Model "CHAPS collector" (Type 262)
UNIT 12 TYPE 262 CHAPS collector

PARAMETERS 34
* 1 MODE
  2
* 2 CELLS
  10
* 3 REFEFF
  .161
* 4 REFTEMP
  65
* 5 BETA
  -0.004
* 6 UNIFORMITY
  .845
* 7 LENGTH
  23.19
* 8 WIDTH
  1.47
* 9 REFLM
  0.935
* 10 SHAPE
  .99
* 11 TRANSABS
  0.886
* 12 COVERABS
  0.063
* 13 EMIS
  0.88
* 14 MASS
  121.9
* 15 CP
  1.08
* 16 TSTART
  25
* 17 WGLASS
0.08
* 18 UCG
327
* 19 WCG
0.07
* 20 UCP
5787
* 21 WCP
0.04
* 22 UPT
1000000
* 23 WPT
1
* 24 UINSUL
23
* 25 WINSUL
.2
* 26 WCOVER
0.2
* 27 EMISCOV
0.1
* 28 CW0
22
* 29 CW1
27.4
* 30 CW2
-2
* 31 TOL
0.1
* 32 PERIM
0.1298
* 33 XSAREA
0.0003587
* 34 FH
0.74
INPUTS 7
* Pump:Outlet fluid temperature -> TFI
9.1
* Pump:Outlet flow rate -> FLOW
9.2
* Otemp calc.:Idbout -> ID
Idbout
* Canberra weather:Output 6 ->TAMB
  5,6
* Canberra weather:Output 7 ->WIND
  5,7
* End loss:End Loss Factor ->SHADE
  11,1
* [unconnected] DIRT
  0,0
*** INITIAL INPUT VALUES
  25 1000 1000 25 2 1
  1
* Model "Tank" (Type 38)
UNIT 13 TYPE 38 Tank

PARAMETERS 17
* 1 Inlet position mode
  1
* 2 Tank volume
  1.5
* 3 Tank height
  1.53
* 4 Height of collector return
  1.06
* 5 Fluid specific heat
  4.190
* 6 Fluid density
  1000.0
* 7 Thermal conductivity
  7.2
* 8 Tank configuration
  1
* 9 Overall Loss Coefficient
  26.365
* 10 Insulation ratio
  1.0
* 11 Initial temperature
  30.0
* 12 Maximum heating rate
  64800
* 13 Auxiliary height
1.2
  * 14 Thermostat height
1.25
  * 15 Set point temperature
65
  * 16 Temperature deadband
8
  * 17 Flue loss coefficient
0.0

INPUTS 6
  * 3-way valve: Temperature at outlet 1 -> Hot-side temperature
10,1
  * 3-way valve: Flow rate at outlet 1 -> Hot-side flowrate
10,2
  * Tempering valve: Temperature at outlet 1 -> Cold-side temperature
8,1
  * Tempering valve: Flowrate at outlet 1 -> Cold-side flowrate
8,2
  * Canberra weather: Output 6 -> Environment temperature
5,6
  * [unconnected] Control signal
0,0

*** INITIAL INPUT VALUES
45.0 100.0 20.0 100.0 22.0 1

---------------------------------------------------------------

* Model "Finned tube HE" (Type 273)
UNIT 14 TYPE 273       Finned tube HE

PARAMETERS 11
  * 1 Length of heat exchanger
60
  * 2 Fins per length
160
  * 3 Area of each fin
0.003
  * 4 Diameter of tube
0.0109
  * 5 Thickness of fins
0.00025
  * 6 Fin conductivity
* 7 NODES
10
* 8 Specific heat of fin-tube
0.4
* 9 Mass of fin tube
19
* 10 Specific heat of the fluid
4.18
* 11 Mass of fluid in the heat exchanger
60

INPUTS 4
* 3-way valve: Temperature at outlet 2 -> Temperature of inlet fluid
10,3
* 3-way valve: Flow rate at outlet 2 -> Flow rate of inlet fluid
10,4
* Canberra weather: Output 6 -> Ambient temperature
5,6
* Canberra weather: Output 7 -> Wind speed
5,7

*** INITIAL INPUT VALUES
55 625 25 1
* ---------------------------------------------------------------

* Model "T-piece" (Type 11)

UNIT 15 TYPE 11 T-piece

PARAMETERS 1
* 1 Tee piece mode
1

INPUTS 4
* Tank: Temperature to heat source -> Temperature at inlet 1
13,1
* Tank: Flow rate to heat source -> Flow rate at inlet 1
13,2
* Finned tube HE: Outlet fluid temperature -> Temperature at inlet 2
14,1
* Finned tube HE: Outlet flow rate -> Flow rate at inlet 2
14,2

*** INITIAL INPUT VALUES
20.0 100.0 20.0 100.0
* Model "T-piece2" (Type 11)
UNIT 16 TYPE 11    T-piece2

PARAMETERS 1
* 1 Tee piece mode
1

INPUTS 4
* Tank:Temperature to load ->Temperature at inlet 1
13.3
* Tank:Flow rate to load ->Flow rate at inlet 1
13.4
* Tempering valve:Temperature at outlet 2 ->Temperature at inlet 2
8.3
* Tempering valve:Flow rate at outlet 2 ->Flow rate at inlet 2
8.4
*** INITIAL INPUT VALUES
20.0 100.0 20.0 100.0

* Model "Otemp park" (Type 2)
UNIT 17 TYPE 2    Otemp park

PARAMETERS 2
* 1 No. of oscillations
5
* 2 High limit cut-out
200

INPUTS 6
* CHAPS collector:TOUT ->Upper input value
12.1
* [unconnected] Lower input value
0.0
* CHAPS collector:TOUT ->Monitoring value
12.1
* Otemp park:Output control function ->Input control function
17.1
* [unconnected] Upper dead band
0.0
* [unconnected] Lower dead band
0.0
*** INITIAL INPUT VALUES
20.0 200 20.0 0 5 0.5

*---------------------------------------------------------------------------------------------------------------
* Model "Controller" (Type 275)
UNIT 18 TYPE 275 Controller

PARAMETERS 9
* 1 Controller mode
  1
* 2 Tank upper temperature setpoint
  0
* 3 Upper deadband for pump
  5
* 4 Lower deadband for pump
  0.5
* 5 Upper deadband for bypass valve
  5
* 6 Lower deadband for bypass valve
  0.5
* 7 Upper deadband for tank pump
  5
* 8 Lower deadband for tank pump
  0.5
* 9 Delta T for bypass mode
  5

INPUTS 6
* CHAPS collector:TOUT ->Collector outlet temperature
  12,1
* Tank:Average tank temperature ->Tank monitoring temperature
  13,10
* Controller:Collector Pump output control function ->collector pump input control function
  18,1
* Controller:Bypass valve output control function ->Bypass valve control function
  18,2
* Controller:Tank pump output control function ->Tank pump input control function
  18,3
* Canberra weather:Output 6 ->Ambient temperature
  5,6

*** INITIAL INPUT VALUES
  0 0 1 0 1 0

*---------------------------------------------------------------------------------------------------------------
* Model "Output" (Type 28)

UNIT 19 TYPE 28 Output

PARAMETERS 18
* 1 Summary interval
  -1
* 2 Summary start time
  1
* 3 Summary stop time
  8760
* 4 Logical unit
  19
* 5 Output mode
  1
* 6 Operation code-1
  1
* 7 Operation code-2
  0
* 8 Operation code-3
  -4
* 9 Operation code-4
  0
* 10 Operation code-5
  -4
* 11 Operation code-6
  0
* 12 Operation code-7
  -4
* 13 Operation code-8
  0
* 14 Operation code-9
  -4
* 15 Operation code-10
  0
* 16 Operation code-11
  -4
* 17 Operation code-12
  0
* 18 Operation code-13
  -4

INPUTS 6
* Tank: Internal energy change -> Summary input-1
  13,7
* Tank: Energy rate to load -> Summary input-2
  13,6
* Tank: Auxiliary heating rate -> Summary input-3
  13,8
* CHAPS collector: QELEC -> Summary input-4
  12,3
* [unconnected] Summary input-5
  0,0
* Calc. solar input: Qsun -> Summary input-6
  Qsun

LABELS 6
DE Qload Qaux Qelec1 Qelec2 Qsun
*** External files
ASSIGN C:\trnsys15\IlSiBat3\Data\Joe\modeld.out 19

END
Appendix B

Appendix B contains the experimental raw data used for TRNSYS validations. The shaded sections show the periods of steady state measurement.

Figure B1. Measured data from 5 May 2003. The shaded areas show the steady state data used in the efficiency curves.
Figure B2. Measured and simulated data from 6 May 2003. The shaded areas show the steady state data used in the efficiency curves.
Figure B3. Measured and simulated data from 22 May 2003 (left) and the 29 May 2003 (right). The shaded areas show the steady state data used in the efficiency curves.
Appendix C


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