System modelling of the compact linear Fresnel reflector

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System Modelling of the Compact Linear Fresnel Reflector

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Doctor of Philosophy

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Abstract

The Compact Linear Fresnel Reflector is a solar thermal energy system currently at prototype stage in Australia. The system uses parallel rows of mirrors lined up underneath a long, elevated thermal absorber. The mirrors move so as to focus solar radiation onto the absorber; the absorber contains a bank of high-pressure water pipes through which water is pumped and gradually boils. The process of ‘direct steam generation’ in very long pipes, up to 300 m in a straight run, has not previously been performed at this scale; other systems use shorter pipe runs, or use other fluids such as non-boiling oil. This thesis addresses a broad range of design issues relating to the CLFR prototype and its components.

Beam solar radiation at the prototype site is estimated from available data including satellite-derived and ground-based measurements. Existing correlations for the beam component of global radiation do not apply well to Australian conditions so a new correlation is proposed.

Computational fluid dynamics simulations establish radiative heat-loss as the dominant mode for the thermal absorber. Results are gathered for a range of sizes and shapes, and heat-loss correlations are derived for use in subsequent simulation.

Two-phase flow in the absorber direct-steam-generation process is examined, and a detailed model including, pipe-friction pressure drops, flow-boiling heat transfer and cavity heat loss is presented, with validation against the experimental results of other workers. A series of ‘performance maps’ give the predicted outlet flow regime for varied inlet conditions, allowing selection of desired operating points.

A full system model is given that integrates this absorber model with ancillary components including the pump and connecting pipework; the model is used to evaluate pumping requirements and to establish expected operating conditions. The inherent pressure instability arising from the two phase flow is examined and orifice plates are sizes to stabilise this effect.

A dynamic model for the absorber pipe flow using fully implicit finite difference techniques and accurate IAPWS-IF97 steam properties gives the predicted behaviour during solar transients at both long and short time-scales.
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Finally, I would like to thank my family: my parents Ian and Jane, and my sister Katherine. Without your unfailing support I am sure it would not have been possible: thank you for everything. This thesis is dedicated to you.
Publications

The following journal papers and conference papers accompany this research:


In addition, a free open-source steam tables package freesteam, [http://freesteam.sf.net/](http://freesteam.sf.net/), was released, and extensive contributions were made to the mathematical modelling program ASCEND, [http://ascend.cheme.cmu.edu/](http://ascend.cheme.cmu.edu/).
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Chapter 1

Background

This thesis examines the heat transfer aspects of an exciting new solar thermal energy system called the Compact Linear Fresnel Reflector. The CLFR design aims to provide affordable solar-generated steam for electricity production in conventional large-scale power station steam turbines and electric generators.

This chapter will give a brief overview of the CLFR project, the various components of CLFR and the scope of the present work. Chapter 2 gives a review of related research, and outlines the areas being studied in the present work. Chapters 3 and the following give the detailed analysis and results of the present work.

1.1 The CLFR concept

The CLFR system is composed of an array of long, parallel mirrors arranged underneath a smaller number of long, parallel absorbers.

The mirrors are equipped with one-dimensional tracking, such that light from each mirror can be focussed onto one or other of the absorbers. The absorber is essentially a bank of parallel high-pressure steam pipes that allow water to be pumped in at one end; the water boils as it moves through the hot pipes, which are heated by the concentrated solar radiation focussed upon them.

The absorber is therefore designed so that it can act as a focus for solar radiation incident from a wide range of angles from the mirror rows positioned below. The steam pipes are mounted inside an insulated trapezoidal cavity. At the base of the cavity is a window that allows radiation to enter, but which prevents convective losses. The inside surface of the cavity is constructed of stainless steel and insulated on the top and sides to minimise conductive, convective and radiative losses.

To achieve its goal of affordable solar steam, the CLFR uses standard-grade materials and standard mass-produced steel and mirrors throughout. Reduced performance is accepted where final cost of generated power can be reduced.
1.2 Power conversion

Two main connection concepts are described here as ways of transforming the heat collected in the CLFR steam pipes into power for end-users.

The first concept is the ‘pure solar’ configuration. Here, the CLFR collector is used to replace the boiler in a conventional Rankine cycle power station. A hot water storage system would be used to provide a limited buffer to minimise the effect of solar transients due to passing clouds, as well as to flatten the solar radiation peak so that it lasts a few hours into the afternoon. A ‘pure solar’ configuration such as this is only relevant in the context of the larger energy grid, where other power sources can be used to provide power during other times of the day, as well as provide more controllable sources of power so that supply can be matched to demand by the grid. Although pure solar is clearly a preferred outcome, these issues make it a rather unfavourable option in the present economic climate.

The second concept is a ‘solar augmentation’ configuration. Here, heat supplied from the CLFR is used to replace feedwater heating in a conventional fuel-powered power station cycle (see Figure 1.1). In ‘un-augmented’ power stations, feedwater heating is done using ‘bleed steam’, a flow of steam taken from one of the turbine inlets. Due to the high temperature difference involved in heating feedwater with bleed steam, this process is expensive from the exergy (energy availability) point of view. So, in the solar augmentation concept, bleed steam used in the boiler feedwater heater is replaced with solar steam (or even just solar hot water). This produces savings in the cycle efficiency that are amplified beyond the simple effect of the added thermal energy input, since the higher-value bleed steam can be left to pass through the turbines. The augmentation set-up is also cheap in that it does not require the construction of new turbines and condenser; instead it permits reduced operational (coal) expenses with equal or slightly increased¹ output.

1.3 The Liddell power station project

Shortly before the start of this study, Macquarie Generation, owners of the 2 GWe Liddell power station in the Hunter Valley in New South Wales, came to an agreement with Solar Heat and Power Pty Ltd for the construction of a CLFR system to provide them with 36 MW (thermal) solar augmentation of one of the four 500 MW (electric) turbines in this coal power station. With the amplifying effects of the exergy savings in bleed steam, this augmentation was calculated to be equivalent to an overall 2% of this turbine’s final electrical output, and consequently, solar

¹The potential for increasing output depends on the rated capacity of the pumps and turbines downstream of the bleed point, due to the increased flow through that path.
1.3. THE LIDDELL POWER STATION PROJECT

Figure 1.1: A simplified and idealised Rankine cycle with bleed steam from the outlet of the high pressure turbine being used to heat the boiler feedwater.

augmentation was seen as a cost-effective way for Liddell to meet its requirements under the current Mandatory Renewable Energy Target (MRET) regulations.

The project at Liddell was planned with three stages, the first of which has been completed during the course of this thesis, and the second of which is currently underway. The location of the Liddell power station is shown in Figure 1.2.

1.3.1 Stage 1 (1400 m²)

The aim at the first stage was to produce a minimal CLFR system capable of producing hot water and steam, allowing the Solar Heat and Power company to demonstrate viability of their design, and to gain experience with the implementation of all possible aspects of their design.

The output at this stage was not intended for connection to the power plant, but instead functioned as a stand-alone test-bed. It was connected in a loop configuration with a small pump and a large demineralised water tank to allow simple steam testing to be carried out.

A photograph of the stage 1 prototype during its commissioning phase is shown in Figure 1.3.

Successful testing with the CLFR prototype was completed, which, under the agreement with Macquarie Generation, has allowed Solar Heat and Power to embark on the next stage of the project. At the time of writing, the Stage 2 prototype is approaching the commissioning phase, after which some of the first high-pressure steam results for this system will begin to be available, and Stage 3 has not yet been commenced.

Results of testing of the CLFR Stage 1 prototype were reported by Mills et al...
Figure 1.2: Map showing the location of the Liddell power station, in the Hunter Valley region of New South Wales, which extends from Newcastle on the coast up to Singleton, Muswellbrook and Scone. (Map source: NRMA website)
1.3.2 Stage 2 (27,000 m²) and Stage 3 (135,000 m²)

The Stage 2 CLFR at Liddell will increase the size of the array to 27,000 m² and will see the array supplying steam to the power station for the first time. The connection will be made upstream of the final boiler feedwater heater, in the bleed steam line. Saturated steam with a target quality of $x = 0.8$ will be fed in. Further details of the stage 2 project are given in Appendix C.

The Stage 3 project will involve the building and connection of additional modules to take the total array area to the final size of 135,000 m².

For both of the later stages, the CLFR will be connected directly to the power station, as shown in Figure 1.4. The chosen connection configuration eliminates the need for expensive (estimated 5 M AUD) new heat exchangers and pumps.

1.4 Reflector

The design of the Fresnel reflector used in the CLFR represents the major intellectual property asset of the CLFR.

The design uses a special interleaved configuration for its reflector mirrors, so as to allow maximum use of available energy incident on the land available. In other heliostat based systems, it can be seen that mirrors further away from an absorber must be spaced further apart on the ground, in order to prevent the effective area of
the mirrors being reduced by ‘shading losses’ caused by radiation incident on or reflected off the mirrors being obstructed by neighbouring mirrors. The CLFR concept overcomes this problem by providing multiple parallel absorbers, and an alternating mirror configuration, as shown in Figure 1.5. Mirrors can then be positioned much more closely to each other.

Mirrors in the reflector are arranged along north-south axes, with single-axis tracking provided in the east-west direction. Mirrors are of low-iron glass adhered to corrugated roofing steel which is in turn mounted on a space frame. The glass is elastically deflected while being mounted, to achieve the required focus. The corrugated roofing material provides a significant fraction of the overall stiffness of the mirror support structure, helping to reduce the amount of welded steelwork. At intervals along the mirror support space-frame, hoops are mounted around the structure. These hoops rest on rollers. For each mirror, one of these hoops has a drive chain welded to its circumference and a geared electric motor is mounted for the purpose of mirror tracking. The mirror can be rotated through 360°, both for following the motion of the sun as well as for turning mirrors upside down when high winds or hailstorms are predicted. This is also intended to permit mirrors to be individually micro-adjusted, so as to optimise the distribution of radiation across the bank of pipes in the absorber.
1.4. REFLECTOR

Figure 1.5: (a) Interleaved mirror rows in the CLFR design. By alternating the orientation of adjacent mirrors from left to right, the distance between mirrors can be made quite small before shading losses occur. This is because the lower edge of one mirror (A) allows the next mirror (B) to be nestled close to it while still proving a clear path between the mirror (B) and its target absorber. The next mirror (C) can again be positioned with minimal spacing (to mirror B) because of their opposite directions. Mirror spacing (D) may be dictated by the need for access for cleaning, but note also that this simplified case does not show the effects of possible shading losses early and late in the day, when the sun position is lower in the sky.

(b) A regular Fresnel reflector showing non-interleaved reflector rows. In order to avoid shading losses, the distance between reflectors has increased (A-B).
CHAPTER 1. BACKGROUND

Each mirror line is mounted on a pivot at 2.1 m above foundation level, is 2.25 m wide and 310 m long and there are 10 mirrors per absorber.

1.5 Absorber

The CLFR absorber has a cross section as shown Figure 1.6. Absorbers are mounted at 12 m above foundation level, and have an overall length of 310 m. Pipes are mounted inside the trapezoidal cavity as shown. The bottom of the cavity is covered with a transparent cover, which is intended to reduce convective losses by trapping a layer of hot air next to the hot steam pipes.

During the course of the development of the CLFR prototype, several different
1.6. THESIS SCOPE

This thesis is concerned with the following areas of the overall CLFR project:

1. To evaluate the likely radiation input that could be utilised by a CLFR installed in the Hunter Valley in New South Wales. Due to the limited number of weather-reporting ground-stations in NSW this will require some estimation using satellite-derived radiation data.

2. To model the thermal losses from the current CLFR cavity absorber design and

   (a) create a correlation for these losses that can be incorporated into a larger-scale model

   (b) provide analysis to facilitate further optimisation of cavity design

3. To perform steady-state modelling of the flow in the absorber of the Stage 2 prototype system, with the aims of

Figure 1.7: Concepts for the cavity cover for the CLFR.

concepts for cavity covers were under consideration (Figure 1.7). Firstly, a flat glass cover was considered. Next, with the aim of reducing cost, a flat plastic film was considered. When it was realised that the flat plastic film would flap in the wind and so increase convective losses, the idea of an inflated plastic membrane was considered. Finally, when it was concluded that the plastic film would likely not withstand the typical air temperatures inside the cavity, it was decided to proceed with a angled low-iron glass cavity cover. The angled glass reduces reflective losses of incident solar radiation. Low-iron glass has improved optical properties compared to standard grade window glass.

Inside the cavity, the Stage 2 collector has 12 parallel pipes (the prototype has 16 pipes), with a manifold at each end of the absorber splitting and re-joining the flow. The manifold is illustrated in Appendix C.
(a) estimating pressure drops and pumping power requirements under varying conditions,
(b) determining safe operating points to ensure no unsteady two-phase flow emerges at the absorber outlet

4. To perform transient modelling of the absorber, with the aims of

(a) modelling possible start-up and shutdown instability
(b) modelling the effects of small passing clouds
(c) quantifying the size of pressure and flow-rate overshoots arising from sudden transients

5. To perform overall system modelling of the CLFR including cycle pump(s), heat exchanger and associated pipework, with the aims of

(a) examining system pressure fluctuations and pressure relief requirements under conditions of varying pumping rates and irradiation levels, to facilitate controller design.
(b) examining the system stability under a `once through the absorber` cycle when operating in saturated steam outlet conditions.
(c) providing a system model that can be used for further optimisation work
Chapter 2

Literature Review

Following from the scope of work outlined, this literature review gives an outline of progress in commercial and prototype solar thermal collectors. Particular focus is given to direct steam generation technology and the modelling that has been used to assist in the design of such systems. The CLFR system is seen to present some special challenges that are not covered by earlier work. Two phase flow is reviewed, as well as the numerical methods of steady and dynamic system modelling. In the present work, the need to integrate detailed two-phase flow modelling with whole-system effects leads to some new problems. Finally, methods for the estimation of solar beam radiation from limited data are reviewed and some problems with current methods are seen.

2.1 Solar thermal energy systems

Solar thermal energy refers to the capture of solar irradiation as usable thermal energy. Solar ‘collectors’ perform the task of capturing this energy, and they can either be of the non-concentrating type, which includes flat-plate collectors, solar ponds, solar chimneys and evacuated tube collectors, or they can be of the concentrating type, where optical concentration is performed, normally with mirrors, so that a greater intensity of solar radiation is available at the absorbing surface [108, 3]. Concentrating collectors are able to reach higher temperatures than non-concentrating ones, and due to the fact that higher temperatures permit more thermodynamically efficient energy-conversion cycles, most large-scale solar collectors have been of the concentrating type.

Three main categories of concentrating solar collector are defined by the way in which they achieve focussing of light. The first is the dish concentrator, which provides the highest concentration ratios, at up to 1000 and higher, and hence the highest temperatures [77, 88, 90]. These concentrators require that the whole dish,
as well as the ‘receiver’ containing the absorbing surface, must track the sun about
two different axes as it passes across the sky. The second category of collector is the
parabolic trough, which has a line-focus and consequently only needs to track the sun
about one axis. By its geometry it is constrained to lower concentration ratios up to
around 85, and consequently lower temperatures. The third category is the central
receiver, which uses an array of heliostats to concentrate light at a shared absorber.
The heliostats in this case must track the sun in two directions, but there is a gain in
simplicity because the expensive thermal receiver is centralised, removing the need
for much pipework and duplication of components.

Beyond the solar collector, there are other important components in a concen-
trating solar thermal energy system. The first is the working fluid which moves
through the collector, transferring heat away from the absorber surface. Systems in
practice have used a wide range of materials here, including water, oil, air, pentane,
and molten metals and salts, in each case chosen for their thermal and transport
properties to suit the specific type of collector.

The next important component is the power block. This is where the ‘usable
energy’ is extracted from the working fluid. In the majority of systems in practice,
standard Rankine cycle steam turbines have been used, sometimes with heat ex-
changers added in order to generate steam from the working fluid if this fluid itself
is not water/steam. In dish concentrators, Stirling engines have often been used,
mounted directly at the dish focus [90], and in the case of high-temperature-air
collectors, Brayton cycle gas turbines can be used [74].

Finally, an optional component in concentrating solar thermal energy systems is
some method for energy storage. Energy storage allows the system to deliver energy
throughout intermittently cloudy periods, or, if the storage volume is large, can allow
the power block to continue operating through several days of cloudy conditions.
In practice, the capacity of the thermal storage is constrained by cost, but some
amount of storage is often found to be financially advantageous because it increases
the utilisation of the power block. Some working fluids facilitate direct storage,
such as molten salts, while others require heat exchange to another material such
as oil, rock, sand or concrete. Other available thermal storage techniques involve
phase-change materials, reversible chemical dissociation, and hot graphite [58].

2.2 SEGS

The Solar Electric Generating Systems (SEGS) in the Mojave Desert in California
are a sequence of nine large-scale parabolic trough solar thermal energy systems that
represent easily the largest and most successful solar thermal energy system in the
world to date. They were commissioned in stages from 1985 to 1991, are all still
fully operational, and have a combined peak electrical output of 354 MWe. The SEGS systems use a heat transfer oil called Sandotherm VP-1 [24] to collect heat in parabolic trough collectors with a total area of 2.5 million square metres. The collector arrays are connected through heat exchangers to a Rankine cycle power block with reheat and natural gas boosting for peak demand and low-sun periods.

Over the stages of the SEGS projects, improvements were made to the collector design and the operation and maintenance systems, and costs per energy delivered have been significantly reduced [24]. SEGS systems now have a mature design, but several limitations resulting from the use of hot oil have become apparent [180, 111].

Firstly, use of oil as the working fluid in the collector necessitates a heat exchanger for generation of steam to be passed through a turbine. Such heat exchangers are expensive and were found by Odeh to be significant in reducing the overall efficiency of the SEGS systems, where they represented a loss of 6% of the system input exergy [118]. Secondly, the heat transfer oil used in the SEGS systems needs to be periodically replaced due to thermal degradation; this is a significant operational cost, although it is likely that improvements can be made in this area. Thirdly, when the oil leaks it presents a fire danger (Mills refers to a fire at SEGS [97]) as well as an environmental contamination problem. Finally, the oil thermal degradation imposes an upper limit on the temperature at which the troughs can operate, and higher temperatures are desirable because of their ability to give higher Rankine cycle efficiencies in the power block.

As a consequence of these perceived problems with oil, the Luz company, that was developing the SEGS technology, had embarked on research into direct generation of steam in the collector. Unfortunately, the incentive situation in California that had facilitated construction of the SEGS systems expired around this time, leading Luz to bankruptcy in 1991 [180].

2.3 Direct Steam Generation

Direct Steam Generation (DSG), in the context of the field of solar thermal energy, is generation of large-scale quantities of solar-heated steam by a field of solar collectors without the use of an intermediate heat transfer fluid.

The field has grown out of an increasing awareness of limitations and operational difficulties in SEGS systems as discussed above. It is widely thought that the next substantial cost savings for line-focus solar collectors will come from a transition to direct steam generation, saving heat exchangers and removing the need for heat transfer oil, and there have been several major development projects initiated to pursue that goal.

The difference between the system configuration of SEGS-style system and a
CHAPTER 2. LITERATURE REVIEW

Figure 2.1: Reheating solar thermal energy cycles (a) with heat transfer oil and (b) with direct steam generation. The diagrams show the pump and heat exchangers that can be eliminated by replacing the heat transfer oil with water. [111]

possible DSG system are shown in Figure 2.1.

2.3.0.1 Advantages and disadvantages compared to hot oil systems

Zarza et al [180] gave the following technical advantages of DSG compared to the hot oil system used by SEGS:

- eliminates fire and pollution risk due to thermal oil,
- no need to observe the upper temperature limit of 400°C required for the use of synthetic heat transfer oil,
- improved collector heat transfer gives improves collector efficiency and permits smaller, hence cheaper collector field, and
2.3. DIRECT STEAM GENERATION

- eliminates need for expensive replacement of oil (5% of oil each year) and reduces the need for antifreeze (which, with oil, was needed below 14°C)

They meanwhile identified the following disadvantages:

- probable absorber flow instability due to two-phase flow of water/steam in the collector,
- possible process instability due to large changes in fluid volume during boiling, and possible changes in collector output steam state and flow-rate,
- if superheated steam is generated in the collector, risk of thermal stress as a result of the movement of the ‘dry-out’ point, and
- high temperature wet/dry conditions requiring higher-grade materials.

It should be emphasised that the upper temperature limit imposed upon SEGS systems due to the chemical stability of the chosen heat transfer oil also directly implies an upper limit on the temperatures in the Rankine cycle used to drive the turbines. Second-law efficiency of Rankine cycles are greatly improved when the maximum available temperature is increased, so removal of the upper temperature limit will help a great deal in improving solar thermal system efficiency, and, it is hoped, lead to significantly lower-cost solar electricity.

A major problem to be overcome in the implementation of a large-scale DSG system is the controllability of the two-phase forced convection boiling process which takes place inside the solar collector. If we assume we are concerned with a line-focus type system, then this means that subcooled water entering at one end of the pipe will enter the saturation region at a certain point along the pipe, gradually increasing in steam quality, until at some point in the pipe ‘dryout’ occurs and the steam becomes completely superheated. Depending on the flow rate and the rate of heat supplied, and the quality of the steam, a variety of different two-phase ‘flow regimes’ occur, many of which are unsteady, involving, for example, large pockets of saturated steam shooting through a slower-moving bulk of liquid water. Such flows can be tolerated inside lengths of pipe, however they could be disastrous if allowed to enter a pump or heat exchanger not designed for the large forces involved.

Another important issue relating to large-scale solar steam (although not exclusively a DSG problem) is the need to provide a steady thermal power source to the turbines in the power block. Ideally a DSG collector would give out superheated steam of a fixed temperature, at a fixed flow rate. This however will never be possible, owing to natural variations in the level of sunlight striking the collector. Seasonal and through-the-day variations act over the daily and yearly time-frames, and cloud transients act over much shorter minute and hourly timescales. System
start-up and shutdown are another concern. Efforts have been made to ensure that a DSG system be controllable and not produce wildly fluctuating steam temperatures or flow-rates, nor risk overpressure or vacuum.

Muller, 1991 [111] gives a good overview of the advantages of DSG as they were viewed at the end of the period of SEGS system installation. Muller proposes that mirrors would best be mounted along a north-south axis, and inclined slightly, to minimise cosine losses\(^1\), and raises the issues of possible two phase flow regime instability.

### 2.3.1 DISS/INDITEP

In 1992, CIEMAT\(^2\) and DLR\(^3\) made a compilation of the DSG research that had been performed by the Luz company, and in 1995 they put up a proposal for further research on direct steam generation. This eventually led to the launching of the DISS project at the Plataforma Solar de Almería, as presented by Zarza et al, 1997 [180]. The CIEMAT and DLR groups, and associated companies, have completed a large body of research of direct steam generation for their design concept, and their prototype is at present the most thoroughly studied and documented direct steam generation collector.

With regard to the direct steam generation, this work has many similarities with the CLFR project, but it must be stressed that the parabolic trough configuration, superheated steam, recirculation control strategy and relatively short collectors (48 m per straight run of absorber pipe, as opposed to 300 m in the CLFR) meaning that the design challenges are quite different in some important aspects.

#### 2.3.1.1 DISS project

The further research into parabolic trough direct steam generation, proposed by CIEMAT and DLR [180], took place at the Plataforma Solar de Almería, Spain, and was named the DDirect Solar Steam (DISS) project. The project objective was to investigate the technical and commercial feasibility of superheated direct steam generation (DSG), including development of special components, such as controllers, and improving existing components such as mirrors and tracking, and to work on power block integration. Zarza et al [180] had found that a reduction of 30\% percent in electricity generation costs, compared to previous-generation SEGS designs, was

---

\(^1\)Cosine losses result from a lower solar view factor when radiation strikes a collector at an oblique angle. End losses are another effect in linear concentrators, resulting when the orientation of the collector is such that a portion of the reflected radiation is focussed at points beyond the end of the absorber.

\(^2\)Centro de Investigaciones Energéticas, Medioambientales y Tecnológicas, a Spanish research group, part of the Spanish Ministry of Education and Science.

\(^3\)Deutsches Zentrum für Luft- und Raumfahrt (German Aerospace Centre)
Direct Steam Generation

2.3. DIRECT STEAM GENERATION

Figure 2.2: The DISS prototype system at the Plataforma Solar de Almeria, Spain [53]. The mirrors are 5.76 m wide, and inclined slightly. At the focus of each trough is a metal-in-glass absorber with an internal diameter of 50 mm. The total length of the active part of the collector is 480 m.

achievable, 65% of which would be attributable to the shift from oil heating to direct steam generation. The Direct Solar Steam (DISS) project operated in two phases, the first being a planning and design phase from 1996-1998 and the second being a testing and experimentation phase from 1998 to 2001.

A key output of the project was the 480 m prototype collector, shown in Figure 2.2, that was based on the Luz LS-3 design used in the later SEGS systems. The prototype system had nine 48 m collectors and two 24 m collectors, and an aperture area of 2765 m². Each collector was inclined at 8° from the horizontal to improve the optics. The inclined collectors were proposed by Müller [111], but instead of pivoting around the trough foci, ball joints are used and collectors pivoted closer to their centre of gravity. The test system⁴ was designed so that the three different plant layouts (Figure 2.3) could be tested, since it was uncertain how effective various control strategies would be in maintaining reliable steam production under transient conditions.

⁴An earlier and much smaller test system was also constructed, in Germany, by the DLR, called PRODISS.
Figure 2.3: Operational modes of the DISS prototype direct steam generation system [35].

Initial results of the DISS experimental work were reported by Eck and Steinmann, 2001 [35]. Partial results of steady-state and transient test results were presented, where thermo-hydraulic behaviour of the test rig was studied, and some attempt was made to establish transfer functions for the collector. Only recirculation and once-through results were presented at this time, and no temperature regulation was yet available.

For the recirculation mode, the ability to control pressure in the recirculation mode was demonstrated. Simulations on the size of thermal gradients in the recirculation mode were also confirmed with experimental results. Experiments with varying recirculation rates were performed and the effect on temperature gradient and outlet pressure were given. The Lockhart-Martinelli-Thom pressure drop model was shown to be satisfactory, slightly underestimating experimental pressure drops but without allowance made for pressure drops in the expansion u-bends.

Some results of the attempt to tune the controller parameters using transient testing were given. The effect of brief de-focussing of collector mirrors in the superheating section was studied. At this point the results were in agreement with the theoretical model, so it was considered that the system might be controllable with a standard PI controller. However, the test was then repeated with mirrors in the evaporating section of the collector: an interesting ‘spike’ effect was found. Shading a mirror in the evaporator section initially results in an increase in the collector outlet temperature, since the reduced evaporation results in a longer dwell time of the superheated steam already in the later sections of the pipe. However once the flow progresses through the pipe, the system re-stabilises at a lower temperature corresponding to the reduced overall radiation received. Clearly it is not a trivial exercise to design a controller for the once-through mode.
2.3. DIRECT STEAM GENERATION

2.3.1.2 DISS numerical modelling

Several efforts have been made to model the DISS system using computer simulation.

A key area of investigation in the DISS system is the effect of thermal stresses in the absorber pipe. This is more of a concern in the DISS system than in the CLFR, because when the sun is at low angles, the troughs are pointing near-horizontal, which means that the absorber pipe is heated from one side only. The effect can be compounded if the flow inside the absorber pipe is stratified, meaning that the water flows at the base of the pipe and the steam flows at the top. Almanza et al, 2002 [4] showed that with transient conditions and stratified flow, bending of tubes during direct steam generation was a realistic possibility. Concerns over the possible extreme thermal stresses arising from stratified flow in the DISS collector are examined by Eck et al, 2004 [41].

System transients in the DISS system were investigated initially by Eck and Steinmann, 2000 [34]. Rheinländer and Eck, 2002 [144], performed a numerical study of the pressure losses in the DISS system. Section 3 of that report gives a comparison of experimental pressure drops in the DISS project with results of empirical pressure drop correlations of Lockhardt-Martinelli-Thom and Bandel and Friedel.

Eck et al, 2003 [37] give a review of the applied research around the DSG process. With regard to pressure drops, the work of Rheinländer and Eck [144] is cited, which found that use of the Friedel correlation [50] gives results that over-predict experimental pressure drops by 5%. It is found that in order to ensure circumferential temperature variations in absorber tubes are kept within design limits, higher mass flow rates should be used, which was later confirmed by experiment [181]. Transient models of the DSG process are also mentioned: linearised models used for studying controller design, and detailed non-linear models for example in studying dryout, optimum design parameters, and placement of steam/water separators. Collector inclination is also discussed.

2.3.1.3 DISS control

Control of the direct steam generation process was identified early as a challenging aspect of the DISS project [27, 85].

Eck et al, 1999 [39] gave details of a controller suitable for configuration of a DSG system with multi-point water injection along the absorber. There were experimental results from the small PRODISS prototype collector given to confirm the numerical modelling of the system. The two-phase flow appears to have been simplified by separating the system into ‘operating modes’, with different flow models depending on superheated or saturated flow. Results for a propose adaptive controller were
‘poor’. An alternative, feed-forward, controller was proposed.

Valenzuela et al, 2003 [167] made further studies of the controllability of the DISS system. Control strategies for recirculation (with PI control) and once-through (with feed-forward control) configurations were discussed, with simple PID control was found to be unable to control the once-through system. Even with the feed-forward controller, it is evident from experimental results that the recirculation configuration is more controllable.

Valenzuela et al, 2004 [168] gave further consideration of DISS control in the once-through mode. The feed-forward controller could be used with good results in clear conditions with short transients, but during longer transients it was found not to be possible to maintain control within acceptable limits, particularly those relating to thermal stresses. For the CLFR, these limits appear to be much less of a design problem, because of the improved pipe-fluid heat transfer resulting from the downwards-facing cavity geometry, so the once-through mode may still be considered a feasible, though still unproven, operating strategy for this system.

2.3.1.4 Operational experience

Zarza et al, 2001 [183] gave a summary of experience from the 2000 hours of operation of DISS prototype system. Amongst a discussion of other operational problems involving mirror position controllers, pump seals and measurement, some interesting comments are made about start-up and shutdown issues. The large thermal mass of the prototype system, with 26 tons of steel, initially required up to 6 hours before operating conditions could be achieved. It was found to be desirable to utilise the entire collector array to collect heat during startup and to attain operating pressure, although initially the superheater section had not been planned to be used in that way. It was anticipated that this thermal mass effect would be smaller in an operational plant due to a higher ratio of collector to peripheral pipework. With the modified strategy, the achievable start-up time was reduced to 2 hours, and the shutdown took 15 minutes, but again, this might be reduced in a full scale plant.

Tuning of control systems had not been completed at this point; recirculation and once-through modes had been tuned to be resilient against short transients but the controllers were still not able to handle longer transients.

A discussion of the method of determining the peak optical efficiency of the collector was given. The average peak optical efficiency of the system was 68%. This value was calculated by measuring the absorbed heat via temperature rise in cold water, thus eliminating thermal losses. This compares to maximum values of 80%\(^5\) for the SEGS plants, and was attributed to unclean mirrors and glass, lost

\(^5\)Value as cited by Zarza et al.
aperture due to protective plates covering ‘hydrogen removing elements’, and lower-grade selective coating on the absorber tubes.

The operational experience of the ball joints installed at the end of each absorber module was good. These 400 °C, 125 bar ball joints used graphite seals, had not leaked, and had flexed with low torque, ensuring that no bending of other parts of the structure had occurred.

Concluding comments stated that pressure drops had been found in practice to be 30% lower than those predicted by Eck and Steinmann, 2001 [35], and that pumping power had been found to be 10% of that required for an oil-based collector field; this would contribute to increased system efficiency for the DSG system. A need to test with multiple parallel mirror rows was mooted (see also Natan et al, 2003 [113]). Finally, the need to attain the higher efficiencies resulting from higher temperature operation at 550 °C and 100 bar was mentioned. Temperature limits on components such as the ball joints prevented this at present.

Summaries of the DISS project were given by Zarza et al, 2004 [182] and Leon et al, 2002 [81].

### 2.3.1.5 INDITEP project

The INDITEP project is the continuation of the DISS project, with the goal of bringing the DISS concept up to the pre-commercial scale. Primarily, the goal is to produce an engineering design for a 5 MWe large-scale prototype using the technology developed in earlier work. Zarza et al, 2006 [181] give more details of the INDITEP project. Superheated operation at 65 bar and 400 °C was chosen, using the recirculation configuration for the collector field, and modelling results are shown including the expected operating point at the exit of each collector in the field. The collector array is to be the Eurotrough ‘ET-100’ design, with a north-south configuration for maximum overall energy collection. Further details of the modelling work are given by Eck at al, 2005 [36].

An aspect of the DISS design that required further study was the design of the steam separator being used in the collector field. Eck et al, 2006 [40] gave an evaluation of several different steam separators for use between the saturated and superheated sections of the DISS collector. Small-diameter vertical cylindrical cyclones (76.1 mm and 88.9 mm ID) were trialed, as well as a horizontal-flow baffle separator. The baffle separator was found to give significantly lower pressure losses. Separation efficiency is discussed, including the rather complex way in which it must be measured: the location of dry-out is found by curve-fitting the readings from thermocouples along the superheater section. Using the known rate of absorbed heat per length, and the length of saturated region, the steam quality at the separator outlet can be determined. The separation efficiency is then the ratio of the liquid
flow-rate at the liquid exit of the separator to the liquid flow-rate at the separator inlet. The experimental errors in estimating separation efficiency become quite large as the inlet steam quality rises, however it was concluded that the separators are most efficient at lower inlet qualities.

Steinmann et al. 2005 [155] described a prototype thermal storage system using pipes set in two different test materials: high temperature concrete, and a castable ceramic material including alumina and iron oxide. The storage system is shown under construction at PSA. Concepts for charging and discharging the storage are discussed.

Some new dynamic modelling of the DISS design is given by Hirsch et al. 2005 [63] and 2006 [62], including some consideration of start-up and shutdown. Some difficulty was experienced in the use of the Dymola software and it was found that bespoke steam property correlations were required for robust modelling. The modelling was confined to the recirculation mode of operation as that is the mode that has been settled upon for the future system. A satisfactory control system had not been settled upon in this work but the feed-forward strategy was again advised [63].

2.3.2 Solarmundo

The Solarmundo solar collector (Figure 2.4) is another direct steam generation project and is the CLFR’s closest competing technology because, as well as DSG, it also uses linear Fresnel reflector optics. As with CLFR, the design aims for simpler construction at lower cost than parabolic trough systems such as SEGS and Eurotrough designs. A prototype 2500 m² collector was constructed around 1999 and the project was presented by Häberle et al. 2001 [66]. At the time of writing, the Solarmundo technology has been acquired by the company Solar Power Group GmbH; they appear to have hired project engineering staff and may be looking to construct a plant in Libya.

The key difference between the Solarmundo design is the scale of the mirrors (approximately one tenth of the width of those in the CLFR) which furthermore are flat, rather than curved. The tracking system uses ganging of mirrors, possible due to their smaller size, and a secondary concentrator is used to focus light onto a single pipe for the heat transfer to the fluid. Häberle et al also show heat loss data for the absorber cavity and some thermal efficiency data for vertical radiation. Some brief costing is presented which shows that large-scale 800 MW solar-only Solarmundo systems could achieve costs as low as 0.04 EUR/kWh in Egypt. A small scale 50 MW solar-only system in Spain would achieve 0.10 EUR/kWh. Cost estimates were based on original ‘first quotations’ from suppliers, suggesting that economies of scale.

2.3. **DIRECT STEAM GENERATION**

have not been estimated.

Häberle et al, 2002 [67] gave another paper, similar to the earlier one, at a SolarPACES meeting. They gave updated estimates of efficiency and cost, and offered a comparison against the SEGS system. For a solar-only 50 MW system in Egypt, the Solarmundo electricity generation cost was 0.0750 EUR/kWh compared to 0.0845 EUR/kWh for the SEGS-based system. The advantages of absorbers being illuminated from below was mentioned: during stable stratified two phase flow, water occupies the lower half of the pipe, so it is better that heat is applied there. Tracking parabolic troughs suffer a dryout problem when illumination is from the side of the pipe.

Morin et al, 2004 [105], did modelling of the integration of the Solarmundo collector into a conventional steam-cycle power station using the tools ColSim [176, 177] and Ebsilon⁷. Their modelling was constrained to a maximum thermal input of 30 MWth, and their test case was an actual 630 MWe gas-fired power station in Egypt. Because of the 30 MWth limitation imposed, they found that a solar contribution to generated electricity of only 0.3% was optimal in terms of levelised energy cost, with a total mirror area of 48,000 m². A number of different plug-in connection types were considered; the lowest levelised energy cost resulted from connection of high-pressure hot water in series with the final boiler feedwater heater, but the fi-

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nal recommendation was for medium pressure hot water to be connected in series upstream of the penultimate feedwater heater, because of the lower risk implied by the lower-pressure equipment. Wet steam and superheated steam were both cheaper than the recommended pragmatic hot water option, although the costs of higher-grade steel to carry the two-phase steam were not considered. There was an effect observed which was that larger solar contributions decrease the solar contribution efficiency, because they decrease the average heating temperature around the entire power plant cycle. Connection of the solar array was assumed to be upstream of the last and second-last feedwater heater – a good configuration, because existing plant control mechanisms ensure that the necessary additional heating is supplied by the boiler feedwater heater. This eliminated the need for additional controller logic associated with the solar collector. The recommended configuration resulted in a net solar-to-electric efficiency of 32% at 5 MWth solar load, decreasing to about 29% at 25 MWth solar load.

2.3.3 CLFR

The CLFR concept, which is the subject of the present study, was first presented to the International Solar Energy Society in 1997 by Mills and Morrison [100]. In the earlier design concept, the absorber was to consist of evacuated tubes; this concept has since been abandoned, however the key concept of interleaving mirror rows (as described in Section 1.4) remains. Mills and Dey, 1999 [102] give an economic study of some ‘transition approaches’ to solar thermal energy, central amongst which is the idea of adding the ‘coal saver’ configuration for solar thermal boosting of a coal power station. Cost estimates against central receiver and Luz LS-3 based-systems were given.

Mills and Morrison, 2000 [101] gave some optical ray-tracing results and results of studies on mirror-packing density. The design at this stage still proposed using evacuated tubes at the focus, and they were arranged hanging side-by-side in the vertical plane along the absorber.

Odeh, Behnia and Morrison, 2000 [116] give results of steady-state modelling of the two phase flow inside direct steam generation pipes for a hypothetical system based on the SEGS LS3 collector using the Martinelli-Nelson pressure drop model and the Barnea-Taitel-Dukler flow regime model. Using the LS3 evacuated tube collector, stratified flow leads to a risk of bending stresses capable of breaking the glass envelope. It is found that stratified flow is not seen in steady-state flow in inclined absorbers. There was however a greater tendency for intermittent flow in the inclined absorbers when larger-diameter pipes were used. The operating conditions discussed by Lippke [85] and Dagan et al [27] for the Plataforma Solar de Almería project were also modelled.
Hu, Baziotopoulos and Li, 2002 [64] presented some case studies of this integration of solar thermal input into an existing power station. They found that when a power station was augmented with solar input replacing bleed steam in the feedwater heating, payback periods as short as 1.5 years could be attained. This was assuming that coal use was held constant and the solar output was used to increase the peak output levels of the power station. The study was simplified to ignore land and finance costs, and solar collector prices were estimated.

Hu et al, 2003 [65] made a specific case study of the integration of the CLFR into Liddell power station, under the assumption that the CLFR would be used to substitute bleed steam for increased output, rather than for decreased coal consumption. They found that solar heat to electrical power conversion could be done at 45% efficiency, or solar radiation to electrical power conversion at 27% peak. The overall capacity factor was 14.5%. The payback period (at zero finance cost) was less than five years and the total capital cost was estimated at 19 million AUD. Additional power generation was 45,854 MWh/y. Costing of the collector was based simply on the cost per area estimate of 143 AUD/m².

Schramek and Mills, 2003 [149] give details of the optical modelling used to determine optimal spacing of mirrors for the Multi-Tower Solar Array (MTSA) project. The compromise between ground area efficiency versus reflector area efficiency is discussed. Optical calculations using the same method but specifically for the CLFR are cited (unpublished), although it is mentioned that these showed that an optically optimised CLFR gains 3% in annual total radiation at the absorber from the use of the mirror-flipping strategy described in Section 1.4.

Some fundamental study of the nature of circumsolar radiation was performed by Buie [16, 17] and then applied to ray-trace modelling of the CLFR design that gave estimates of the effective concentration ratio that could be expected [19].

2.3.4 EURECA

Osuna et al, 2005 [120] presented a conceptual paper about a lower cost DSG thermal power station which they name EURECA. They proposed using existing trough concentrator technology to produce saturated steam (and 82% of plant power), then using a heliostat array with a superheater absorber to push steam up into the high temperature region (and provide the remaining 18% of plant power). This concept offers higher cycle efficiencies due to the higher peak temperatures, and makes optimal use of the two different collector types. The power block is also cheaper, as superheated steam turbines are smaller and cheaper, and wet-steam corrosion effects are reduced. The concept proposes biomass co-firing plus short-term thermal storage (compressed water from the outlet of the saturated steam collector) to allow constant performance during slight cloud, and also for additional capacity factor
during start-up and shutdown and possibly during evening peak demand.

2.4 A broader history

In addition to the currently-active projects reviewed above, there are some other, earlier, projects that deserve review.

The first large-scale solar thermal energy system in regular use was that of Schuman, which used 5 large parabolic troughs of 60 m length to provide 55 hp (41 kW) of power for pumping of irrigation water in Egypt in 1913 [150, 28]. This system held its place as the largest solar thermal energy system for many years, until the efforts of Francia in the 1960s and 1970s, where a series of prototypes culminated in the 1 MWe *Eurelius* central receiver system in Sicily in 1981 [151].

Several more prototype systems were constructed in the Oil Crisis years that followed, each system having different mirror, receiver and thermodynamic working fluid configurations. The major central receiver systems are shown in Table 2.1. Table 2.2 shows the history of the SEGS system as well as a number of more recent systems under current development.

A study by Kiera et al, 1991 [79] found that, on the basis of efficiency and design, different collector types are found to be preferable depending on latitude, total system size, load profile and energy cost and revenue structures. In other words, it seemed unlikely at that point that a single collector type would become ubiquitous.

2.4.1 Solar One

The large *Solar One* system, built in 1982 in Daggett, California was a major central-receiver system, rated at 10 MWe, that ran grid-connected for three years through to 1987 [134]. It used 1818 heliostats to focus light onto a 7.2 m diameter central receiver with a target height of 13.7 m. The receiver was composed of panels containing vertical single-pass-to-superheat boiler tubes spanning the full target height. The tubes had an internal diameter of 0.5 in. The system provided thermal storage using a mass of oil and rock that was heated by steam [134].

Of interest in the present work is the modelling of Ray, 1981 [136], which gives an early computational model for direct steam generation.

A problem with the Solar One system was the cracking of pipes in the receiver, attributed to thermal cycling and unsatisfactory welding techniques [134]. The need for very high-grade materials at the focus of central receiver systems is in contrast to the relatively low-grade materials that appear to suffice for line-focus systems.
Table 2.1: Selected central receiver systems [8, 147]

<table>
<thead>
<tr>
<th>Project</th>
<th>Country</th>
<th>Power (MWe)</th>
<th>Working fluid</th>
<th>Storage media</th>
<th>Operation</th>
<th>T(°C)**</th>
</tr>
</thead>
<tbody>
<tr>
<td>IEA/SSPS</td>
<td>Spain</td>
<td>0.5</td>
<td>molten sodium</td>
<td>sodium</td>
<td>1981-1984</td>
<td></td>
</tr>
<tr>
<td>EURELICS</td>
<td>Italy</td>
<td>1</td>
<td>steam</td>
<td>nitrate Salt/water</td>
<td>1981-512</td>
<td></td>
</tr>
<tr>
<td>Solar One</td>
<td>U.S.A.</td>
<td>10</td>
<td>steam</td>
<td>oil/rock</td>
<td>1982-1987</td>
<td>516</td>
</tr>
<tr>
<td>CESA-1</td>
<td>Spain</td>
<td>1</td>
<td>steam</td>
<td>nitrate Salt</td>
<td>1982-1985</td>
<td>530</td>
</tr>
<tr>
<td>MSEE/Cat B</td>
<td>U.S.A.</td>
<td>1</td>
<td>nitrate Salt</td>
<td>nitrate Salt</td>
<td>1983-1985</td>
<td>?</td>
</tr>
<tr>
<td>THEMIS</td>
<td>France</td>
<td>2.5</td>
<td>Hitce Salt</td>
<td>Hitech Salt</td>
<td>1984-1985</td>
<td>450</td>
</tr>
<tr>
<td>SPP-5</td>
<td>Russia</td>
<td>5</td>
<td>steam</td>
<td>water/steam</td>
<td>1986-1987</td>
<td></td>
</tr>
<tr>
<td>TSA</td>
<td>Spain</td>
<td>1</td>
<td>air</td>
<td>ceramic</td>
<td>1993-1995</td>
<td></td>
</tr>
<tr>
<td>Consolar</td>
<td>Israel</td>
<td>0.5 MWth</td>
<td>air</td>
<td>fossil hybrid</td>
<td>1995-2001</td>
<td></td>
</tr>
<tr>
<td>Solgate*</td>
<td>Spain</td>
<td>0.3</td>
<td>air</td>
<td>fossil hybrid</td>
<td>2002-2003</td>
<td></td>
</tr>
<tr>
<td>PS10</td>
<td>Spain</td>
<td>11</td>
<td>air</td>
<td>ceramic</td>
<td>2007-2008</td>
<td></td>
</tr>
<tr>
<td>Solar Tres*</td>
<td>Spain</td>
<td>15</td>
<td>nitrate Salt</td>
<td>nitrate Salt</td>
<td>2007-2008</td>
<td></td>
</tr>
<tr>
<td>CSIRO***</td>
<td>Australia</td>
<td>0.55 MWth</td>
<td>natural gas</td>
<td>(reformed gas)</td>
<td>2007-2008</td>
<td></td>
</tr>
</tbody>
</table>

* still under development (as of 2002)
** at receiver outlet
*** still under construction (as of Nov 2006)
<table>
<thead>
<tr>
<th>Project</th>
<th>Location</th>
<th>Power</th>
<th>Fluid</th>
<th>Technology</th>
<th>Operated</th>
<th>$T_{\text{max}}$ ($^\circ$C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Liddell CLFR 1*</td>
<td>Hunter Valley, NSW</td>
<td>1 MWth</td>
<td>water</td>
<td>CLFR, DSG</td>
<td>2005-**</td>
<td>250</td>
</tr>
<tr>
<td>Liddell CLFR 2</td>
<td>Hunter Valley, NSW</td>
<td>36 MWe</td>
<td>water</td>
<td>CLFR, DSG</td>
<td>2007***-</td>
<td></td>
</tr>
<tr>
<td>Nevada Solar One</td>
<td>Nevada, USA</td>
<td>50 MWe</td>
<td>oil</td>
<td>oil, aluminum-frame parabolic trough</td>
<td>2007****-</td>
<td>304</td>
</tr>
<tr>
<td>Andasol 1</td>
<td>Andalucía, Spain</td>
<td>50 MWe</td>
<td>oil</td>
<td>oil, Eurotrough</td>
<td>2008***-</td>
<td></td>
</tr>
<tr>
<td>Andasol 2</td>
<td>Andalucía, Spain</td>
<td>50 MWe</td>
<td>oil</td>
<td>oil, Eurotrough</td>
<td>2009***-</td>
<td></td>
</tr>
<tr>
<td>Solarmundo*</td>
<td>Liège, Belgium</td>
<td>?kW</td>
<td>water</td>
<td>DSG, linear Fresnel + secondary conc.</td>
<td>2002?**</td>
<td>~250</td>
</tr>
<tr>
<td>DISS*</td>
<td>Almería, Spain</td>
<td>?kW</td>
<td>water</td>
<td>DSG, Eurotrough</td>
<td>1999?**</td>
<td>~250</td>
</tr>
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<td>SEGSI</td>
<td>Daggett, California</td>
<td>13.8 MWe</td>
<td>oil</td>
<td>oil, therm. stor., n. gas superheat, LS-1, LS-2</td>
<td>1985-</td>
<td>307</td>
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<td>Daggett, California</td>
<td>30 MWe</td>
<td>oil</td>
<td>oil, n. gas sup'heater, LS-1, LS-2</td>
<td>1986-</td>
<td>316</td>
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<td>1987-</td>
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<tr>
<td>SEGSV</td>
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<td>30 MWe</td>
<td>oil</td>
<td>oil, natural gas boiler, LS-2</td>
<td>1988-</td>
<td>349</td>
</tr>
<tr>
<td>SEGSVI</td>
<td>Kramer Jn, Calif.</td>
<td>30 MWe</td>
<td>oil</td>
<td>oil, natural gas boiler, LS-2</td>
<td>1989-</td>
<td>390</td>
</tr>
<tr>
<td>SEGSVII</td>
<td>Kramer Jn, Calif.</td>
<td>30 MWe</td>
<td>oil</td>
<td>oil, natural gas boiler, LS-2</td>
<td>1989-</td>
<td>390</td>
</tr>
<tr>
<td>SEGSVIII</td>
<td>Harper Lake, Calif.</td>
<td>80 MWe</td>
<td>oil</td>
<td>oil, natural gas oil heater, LS-3</td>
<td>1990-</td>
<td>390</td>
</tr>
<tr>
<td>SEGSIX</td>
<td>Harper Lake, Calif.</td>
<td>80 MWe</td>
<td>oil</td>
<td>oil, natural gas oil heater, LS-3</td>
<td>1991-</td>
<td>390</td>
</tr>
</tbody>
</table>

* proof-of-concept prototype
** testing only
*** estimated
**** currently being commissioned
2.4.2 CESA-1, STEOR and the Weizmann Institute solar tower

Three central receiver systems are of special interest because they adopted a design that delivered only saturated steam, in order to reduce the engineering risk. CESA-1 was a 1 MW central tower system in Spain that followed soon after the pioneering Eurelios system. Meanwhile, the STEOR system was a solar-powered oil-reforming project trialled in California for a year in 1983 – a very direct attempt for a solar solution to the Oil Crisis. Finally, a 2 MWth saturated steam central receiver was tested in Israel in 1989. It was noted by Romero et al:

Better results regarding absorber panels lifetime and controllability have been reported for saturated steam receivers. In particular, the STEOR pilot plant Solar Thermal Enhanced Oil Recovery for oil extraction using direct injection of steam was successfully operated in Kern County, CA, during 345 days in 1983 with a high reliability. The good performance of saturated steam receivers was also qualified at the 2 MW Weizmann receiver tested in 1989 that produced steam at 15 bar for 500 hours. Even though utilizing saturated steam receivers reduces technical risks, the outlet temperatures are significantly lower than those of superheated steam making it necessary to find applications where this technology can be integrated into processes where fossil-fuel provides superheating.[147].

The testing of the Weizmann Institute receiver is described in greater depth by Epstein et al [44]. The concentrator was used for direct steam generation in Rehovot, Israel, over a period of 125 days. The receiver consisted of a set of three vertical banks of parallel tubes, positioned behind a hexagonal aperture approximately 2.5 m high and 3 m wide. The straight section of the vertical tubes was 3.5 m long, and the absorbed heat per tube was 8.34 kW, with a flow-rate of 250.8 kg/h. The exit steam quality was only 4.2%. The peak thermal efficiency was between 73 and 79 %. Because of the small size of the absorber, control using a fairly simple negative feedback system appears to have been possible, and the steam drum water level was used to control the feedwater supply valve.

Simplifications such as these are of great interest in the CLFR project, as they demonstrate that the control problems evident in the DISS project may largely be eliminated if the design can be constrained to the saturated region. Particularly given the potential of the ‘coal saver’ concept, there seems little benefit in striving to achieve direct superheated steam at the initial stages of CLFR development.
2.4.3 System-integration studies

There have been several recent studies into system integration of solar collectors in the context of large power-blocks. Studies of this type that include a detailed consideration of system transients have not yet been made, although some very useful conclusions are seen.

Odeh and Hashem, 1988 [119] performed an analysis on the integration of solar-heated air and water into an integrated solar combined-cycle system (ISCCS) based on the central receiver or ‘power tower’ design. They found that it was preferable to use solar input to heat boiler feedwater in the Rankine cycle part of the system rather than using it for air heating for the gas turbine inlet air. Three systems were compared: solar water (B), solar air (C) and solar water and air (A). All systems were tuned to give the same 62.5MW steam turbine output and 24 MW gas turbine output, totalling 86.5 MW. For the comparison, the solar input was allowed or not allowed for the various cases, and fuel use was varied to give the required total power output. Consideration was not given to varying the gas/steam ratio or optimising the sizes of the solar plants between the cases. Exergy losses in the solar air receiver and the fired boiler were attributed to the lower exergetic efficiencies of the systems A and C.

Ying and Hu, 1999 [179] put forward the advantages of integrating a solar energy plant into a Rankine cycle power plant. They showed that replacing the use of bleed steam with solar-heated steam in feedwater heater gave an effective ‘delta’ on the system that was more efficient than any stand-alone system could ever be: the work gained from the solar steam exceeded its exergy in many cases. The benefits come from the fact that the bleed steam can be used in the turbine rather than in preheating. An important issue is alluded to, that is, what should be the temperature used for the exergy calculations for the solar steam? As it turns out, this apparently academic question has a large bearing on the ability of the CLFR system to gain funding under MRET, since the displaced emissions are rather difficult to identify in this case.

Kelly et al, 2001 [78] discuss the use of a parabolic trough concentrator system (a SEGS array) in a solar-boosted combined-cycle system, which would consist of a conventional gas turbine accompanied by a Rankine cycle that uses flue gases from the gas turbine as well as solar-heated oil to provide the thermal energy for the steam turbine. They performed steady-state cycle analysis for a range of solar inputs and ambient temperatures, comparing conventional non-solar ICCS and solar-boosted ISCCS configurations. Different ways of providing the solar boosting were compared: it was found that the most efficient way to use the solar input was to use the solar-heated oil to produce high-pressure saturated steam for injection into the hottest part of the gas turbine’s flue-gas-heated steam generator (called the ‘heat
2.4. A BROADER HISTORY

Figure 2.5: Power station tie-in approaches [117]

recovery steam generator”) for superheating. In this paper, the optimum point for solar injection into the combined cycle was explained in terms of the reduced average temperature difference between flue gases passing through the heat exchanger and the temperature of the steam being heated. Solar input allows this gap to be reduced, increasing the thermal efficiency of the heat recovery steam generator. For the total 240 MWe system, the optimal solar contribution was about 100 MWth. The effect of varying the annual solar contribution between 1 to 10% was investigated. One barrier exists at the 6% contribution level; above that point it is not possible to ensure that steam temperatures are maintained. Above 9% the configuration on the various feedwater heaters and steam generators must be altered. Above around 10% is asserted that solar-only plants will be more efficient than ISCCSs.

Odeh et al, 2003 [117] presented a study on the integration of a SEGS-based DSG system with a Rankine cycle power station. Systems being compared all use the same boiler, turbine and heaters, and have a 59,400 m² collector with either evacuated-tube DSG absorbers or the original SEGS VP-I oil absorber plus associated oil-to-steam heat exchanger. Three different tie-in strategies, as shown in Figure 2.5, were evaluated, where the solar collector works in parallel with either (a) the boiler (excluding superheater), (b) the feedwater heaters, or (c) both. In all cases, the turbine inlet flow-rate was set constant, with the fuel consumption and turbine output varying in response. The specific fuel consumption (SFC, based on gross output minus pumping power) is seen to decrease as the solar radiation level rises. In the case of feedwater heater tie-in (b), the solar contribution was limited by the upper limit on the maximum temperature of the water going in to the boiler.

Nevertheless, the DSG system showed lower SFC than the oil system. An exergy analysis shows the the exergetic efficiency of the DSG system is higher than an oil system (30.5% vs 29.3%), primarily because of exergy savings due to the removal of the oil-steam heat exchanger. An annual simulation with TRNSYS shows 7% annual increase in solar steam thermal energy output when DSG is used in place of oil. Overall, the best SEGS performance in this study occurred when the collector was used to tie in at the boiler stage (subcooled water to saturated steam). However, it is likely that this is because of the choice of constant turbine flow-rate.
2.5 Two-phase flow

Two-phase flow is a field of study concerned with prediction of flow structure, pressure drop, heat transfer and dynamics in flows where two different fluid phases may be present at once. This area was of interest for many years in the design of boilers and condensers for power stations and in refrigeration systems, but became of critical importance in the design of nuclear power stations since the Second World War. As a result, a great deal of study has been conducted into computational modelling of two-phase flow for the purpose of understanding how a nuclear reactor might behave during a ‘loss of coolant accident’ [82].

The simplest two-phase flow models try to find an equivalent single fluid that approximates the combined behaviour of the liquid and gas. These are the ‘homogeneous flow’ models and they require only three conservation equations, for mass, energy and momentum, plus pressure drop and heat transfer equations for behaviour at the pipe wall, as well as fluid physical properties correlations and, optionally, transport properties such as viscosity [14].

Models of two-phase flow in use by the nuclear industry, such as TRAC, RELAP and CATHARE, however, are much more complex [110, 158, 164, 82]. They model mass, energy and momentum balances for the two fluid phases separately, allowing for gas that can move faster than liquid. They model the shape of the gas/liquid interface and its effect on heat and mass transfer. There is a wide range of correlations to allow prediction of the type of flow pattern that exists at flow conditions of liquid and gas [82]. The models include the behaviour of small bubbles and their effect on compressibility and heat transfer. Correlations for the pressure drop due to pipe friction are also used, and these show a large variation from the single-phase case. The motion of speed-of-sound pressure waves through the fluid is an important phenomena in such models; the response of a nuclear reactor in the first few milliseconds after a loss of coolant accident can be dramatic, and can lead to the most demanding design conditions [9].

In the case of the design of a solar thermal energy system of the scale of the CLFR, it is safe to say that there is not the need for a model of the complexity of those used in the nuclear industry. Heat fluxes in a line-focus solar collector are many times lower than in a nuclear power plant, so the driving force behind fast transients is reduced. The flow paths in the CLFR are also many times longer, so the role of friction can be expected to be more significant, and the transients of interest can be imagined to be of the order of seconds or minutes, rather than of milliseconds.

Despite this, the design of the CLFR does present new challenges with regard to two-phase flow. Nowhere before the CLFR have such long pipes been used in
forced-convection boiling. Very long two-phase flow pipelines are used in the oil and natural gas industry, but there is no boiling taking place. Shorter pipe lengths are used in conventional boilers. Even the other line-focus direct steam generation projects have used straight pipe runs no longer than 60 m (by Schuman in 1913, but only at atmospheric pressure [150]) and 48 m (in the DISS project [181]). Long, uninterrupted pipe runs lead to a controllability problem as discussed above, which needs to be resolved if a steady flow of steam is to be assured for power generation.

2.5.1 Flow regime transitions

When liquid and gas flow down the pipe, the nature of the flow varies widely depending on how much of each are present, on the pressure and on the pipe inclination. The widely accepted definitions of ‘flow regimes’ come from Mandhane et al, 1974 [91]. Results from a wide range of studies, including those of Taitel, Dukler and others [161, 160, 162] were combined into a unified approach by Barnea, 1987 [11], and it is the approach of Barnea that is used in the present work.

2.5.2 Pressure Drop

Pressure drops are addressed in two-phase flow with the use of empirical correlations that are reviewed by Behnia [14] and by Levy [82]. An early correlation of two-phase pressure drops was by Martinelli and Nelson, 1948 [94] and Lockhardt and Martinelli, 1949 [87]. These correlations were based on forced circulation of boiling water and isothermal conditions and apply to the wide pressure range of 1 to 220 bar and the full range of steam quality. An adaptation of these correlations was made by Thom, 1964 [163] to the case of vertical tubes. A large compilation of available data and a new correlation was proposed by Friedel [51]. Other correlations are due to Bandel, Chisholm, and Baroczy. A correlation by Müller-Steinhagen and Heck, 1986 [112], is worthy of note, as it accepts a sacrifice in accuracy in order to offer a correlation in the ‘reverse’ form of mass flow-rate as a function of pressure drop and enthalpy. As noted by Hirsch et al, 2005 [63], this can help in building dynamic flow models with the desired numerical convergence behaviour, as it facilitates de-coupling of the energy and momentum equations.

A review of pressure drop correlations by Manzano-Ruiz et al, 1987 [92] evaluated the correlations of Friedel, Chisholm, Martinelli-Nelson, Lockhardt-Martinelli and Thom, Baroczy and an annular-flow correlation of Hewitt. Over a range of 25, 50 and 97 mm diameter pipes, they evaluated these various correlations and found that the Martinelli-Nelson, Friedel and Chisholm correlations agreed best with the experimental data gathered, but that the root-mean-square error even for the best correlation was still of the order of ±22%.
It was found by Rheinländer, 2002 [144] (p. 20) that the Friedel correlation gave the best agreement with experimental results from the DISS project, giving pressure drop estimates consistently of the order of 5% too high, and with a maximum deviation of about 15%. Other models gave errors of the order of -35% (Bandel) and +13% (Thom).

It is interesting to note that while Rheinländer found the results of Bandel to be very poor in the prediction of DISS pressure drops, Müller-Steinhagen and Heck [112] found the correlation of Bandel to give the best agreement with the data they studied.

These discrepancies suggest the pressure drop correlations will need to be carefully validated by experimental measurement before major large-scale infrastructure is built that depends on their predicted results. In the present work, the pressure drops of Friedel (as used by Rheinländer) have been adopted, but are compared with the Martinelli-Nelson results (as used by Reynolds [142] and Odeh [115]). Effort has been made in the present work to check the correlation used against available data and the discrepancies between the Friedel and Martinelli-Nelson correlations have been quantified.

2.5.3 Minor losses

Minor losses in two-phase flow remains an area that has been less studied. Azzi, 2000 [6] summarises work in this field. The work of Chisholm, 1980 [22] gives some simple correlations based on a two-phase flow multiplier for pipe bends. Paliwoda, 1992 [121] treats a broader range of minor-loss components including expansions and contractions, bends, tees and valves, with reference to some experimental results. Paliwoda’s work appears to be specifically intended for use with refrigerants, rather than steam flow. Rheinländer and Eck, 2002 [144] used the correlation of Chisholm in their work, and approximated non-bend components by Chisholm-style bend equivalents and this simple approach has also been used in the present work.

2.5.4 Heat transfer

The gradual transformation of the fluid flowing through the CLFR absorber from water to steam is termed ‘flow boiling’. In this case, the heat transfer is greatly enhanced compared to the case of single-phase heat transfer, primarily because of the latent heat transfer that occurs. Macro-scale heat transfer coefficients in flow boiling are normally calculated from empirical correlations relating fluid properties, wall temperature and heat flux.8

8Models for flow boiling include some that include the effect of ‘subcooled boiling’, in which localised boiling heat transfer may occur at the pipe wall even before the bulk fluid temperature has reached boiling point. Such models have not been considered here, although they would make
Correlations for heat transfer in flow-boiling have been proposed by Gungor and Winterton, 1986 [55], and their set of correlations, in the form given by Stephan, 1992 [156] was the correlation used by both Odeh and by Reynolds in earlier UNSW work [115, 142]. Liu and Winterton, 1991 [86] gave a newer correlation that they claim is slightly better than that of Gungor and Winterton. A correlation of Kandlikar, as given by Lienhard and Lienhard, 2006 [84], is the correlation used here, as its form is simpler, and it is recommended for use by Lienhard and Lienhard. In the present work, it was observed that the flow-boiling heat transfer did not dominate, due to the greater resistance to external heat loss in the absorber cavity.

2.5.5 Transient two-phase flow

Building a model for forced convection boiling in a solar collector, and building one that is robust enough to be connected in a closed-loop configuration without other components, and then performing transient simulation with the model, is one of the important challenges addressed in this thesis.

Stewart, 1984 [158] gives some detail about the advanced transient two-phase flow models used in the nuclear industry. These models have a mathematical property that, as written, they are ‘ill posed’. This means that they cannot be solved without some modification to the problem. A variety of approaches are used, including semi-implicit integration methods, and higher-order finite difference schemes that have the result of adding enough numerical diffusion to the problem that it becomes well-posed and can then reliably be integrated.

As discussed above, the detail required in solar thermal energy applications is not as great as that required in nuclear engineering. The problem of ill-posedness is resolved by choosing the homogeneous flow model, and furthermore assuming stationary momentum. This approach has been used by a number of workers in the solar thermal field [135, 136, 85, 27, 62, 63].

2.5.5.1 Lumped parameter flow models

Lumped parameter models are models that replace complex spatially-continuous processes with simplified processes involving a small number of ‘lumped’ entities. Modelling using lumped parameters is quite effective when a process is dominated by boundary effects, such as convection or evaporation, and is significantly simpler to implement. Several examples of lumped parameter models applied to direct steam generation have been found.

An early attempt to perform computer modelling of the transient two-phase flow problem was by Adams et al, 1965 [2]. Their efforts were directed towards creating an interesting refinement. It was considered likely that subcooled boiling would occupy only a small portion of the flow path, and could therefore probably be ignored.
a linearised model for the purpose of control system design, and they achieved this using an analogue computer. Their results were validated against experimental data from a specially chosen test system.

Ray and Bowman, 1976 [135] developed a lumped-parameter transient model of a counter-flow heat exchanger and used it to simulate the response of a heat exchanger to a number of step-change perturbations in operating conditions. The model included subcooled, saturated, and superheated sections. For each region there were inlet, mid-point and outlet flow conditions in the model. Further spatial discretisation was not pursued. A significant simplification was the removal of the acceleration term in the momentum equations, which eliminated the fastest transients in the modelled system and made the numerical investigation more straightforward.

The earlier lumped-parameter work of Ray and Bowman [135] was taken further when the model was adapted by Ray, 1981 [136] for the purpose of modelling the steam generation in the Solar One central tower system (Section 2.4.1). This time the counter-current flow was replaced by a linear heat flux, minus re-radiation and convective losses. The Solar One tubes were relatively short (13 m, compared to the 300 m CLFR tubes) and were mounted vertically and connected to a steam header above. Temporal acceleration, mass holdup and compressibility were neglected, due to the small size of the tubes compared to the steam header. Friction and gravity were said to have negligible effect on the pressure. The model also included the steam header and a flow control valve. As in the earlier work, the transients in the superheated section of the pipe were shown to give very large negative eigenvalues in the Jacobian. There was confidence that these transients would be quickly damped by the steam header, so they were removed from the model. Finally, the system was linearised about computed steady-state conditions and transfer functions then calculated for the inputs of solar flux, feedwater flow and valve position, and the outputs of subcooling flow length, saturation flow length, superheating flow length, outlet steam temperature, outlet pressure and outlet flow-rate.

### 2.5.5.2 Homogeneous two-phase models

More complex than lumped-parameter models is the approach of modelling two-phase flow as a continuum composed of an equivalent single fluid, the properties of which are somehow interpolated from the properties of the two phases in the flow.

Banerjee and Hancox, 1978 [9] give a thorough study of the use of homogeneous flow equations in the problem of nuclear reactor design. They describe the simulation of loss-of-coolant accidents (LOCA) in nuclear reactors. A ‘reference numerical technique’ is described, which uses the method of characteristics to compute a transient solution for homogeneous equilibrium flow equations along characteristics and resulted in flow solver called MECA. A two-fluid model by Ferch, using the
2.5. **TWO-PHASE FLOW**

method of characteristics, is also cited there. Next, a standard explicit finite difference flow solver called RODFLOW is described, which models no-slip flow using homogeneous flow equations. Some experimental data from Edwards and O’Brien is then cited and LOCA simulations using MECA and RODFLOW and one other simulator (‘RAMA’) are compared with that data (reproduced here as Figure 2.6). The pressure undershoot seen in the experimental results is not predicted by any of the simulation models, which is ascribed to the fact that they all assume thermal equilibrium between phases. In reality, the two phases cannot achieve equilibrium in this short time, which causes the negative water-hammer effect when the first rarefaction wave reaches the closed end of the pipe. The other observation by Banerjee and Hancox is that the explicit finite difference model causes significant smoothing of the early transient (at 3 ms) but shows a good slower-scale agreement (at 300 ms) with the experimental results.

A second set of experimental results with a slightly longer time scale (~10 s) is also simulated in this same paper. This time the experimental LOCA includes flow that is initially moving, and also includes heating of the pipe walls throughout the time of the fluid expansion. Pressure agreement by the explicit finite difference technique is still in good agreement, but the method of characteristics shows a sharp pressure spike at 0.5 s that is not shown to occur experimentally. The addition of heating in this experiment leads to the critical heat flux occurring a few seconds after the loss of containment. The finite difference method also predicted this, though there was a disagreement with regard to when the critical heat flux would be reached. Again it was thought that the cause for this disagreement was non-equilibrium between phases.

A third set of experimental results was also described in this paper. Results from these experiments showed that the size of the pipe break affected the maximum temperature achieved during the LOCA. A pipe break of only ~10% of the pipe cross-sectional area was the worst case and resulted in very large (~400 °C) temperature rises over an interval of ~10 s.

The overall result of these experimental and simulation studies was that the explicit finite difference method modelling of homogeneous equilibrium flow was reasonably accurate but that it could be improved with the addition of thermal non-equilibrium effects.

Odeh, 1999 [115] constructed a transient homogeneous flow model for direct steam generation in modified SEGS collectors, pursuing the same goals as noted by Zarza [180]. Essentially the same homogeneous flow model was adopted by Reynolds, 2004 [142] with modification in studies for the earlier CLFR conceptual design. Studies by Lippke, Steinmann and Eck used stationary momentum homogeneous flow and were cited earlier with reference to the DISS project [27, 85, 34].
Figure 2.6: Comparison of predicted and experimental pressure drops from a 4 m pipe (32 mm inside diameter) at 7.0 MPa and 243 °C, for the both (a) immediate and (b) slower effects, after one end of the pipe was suddenly opened to atmospheric pressure [9].
2.5. TWO-PHASE FLOW

Natan, Barnea and Taitel, 2003 [113] performed a numerical simulation study of two-phase flow instability in parallel pipes, specifically for application to direct steam generation.

None of these studies included transient modelling of any components outside the absorber. A study that has included other components was presented recently by Hirsch et al [63]. They had success in modelling transient two-phase pressure drops using the correlation of Müller-Steinhagen [112]. This correlation has the clear advantage that it can be solved analytically and permits decoupling of momentum and energy effects in the transient two-phase model. It is noted however that the findings of Müller-Steinhagen are in contradiction with experimental findings of Rheinländer [144], saying that the most accurate two-phase flow correlation is that of Bandel.

2.5.6 Standardised steam properties correlations

The International Association for the Properties of Water and Steam produce a range of steam properties correlations, primarily in the interests of a standardised base for calculations in the energy industry. The first attempt to produce such standardised properties took place in the 1930s, and updates in accuracy and method of formulation were made in the 1960s, 1980s and 1990s [56].

There are two current ‘releases’ for the basic thermodynamic properties of ‘water substance’. The first is IAPWS-95 correlation for scientific use, which involves relatively straightforward sets of equations which are however computationally expensive to evaluate [70]. The second is the IAPWS-IF97 formulation, which is intended for industrial use, but which is significantly more complex in its form [71]. This latter correlation uses separate high-order polynomial expressions for each of four different regions (roughly corresponding to subcooled, superheated, saturated and supercritical) and even some sub-regions (for the difficult area surrounding the critical point), and covers the range from 273.15 to 1073.15 K and from 0 to 1000 bar. There is also an additional high-temperature ‘Region 5’ that covers properties at temperatures up to 2273.15 K, although over a smaller pressure range. The IAPWS-IF97 correlations are in terms of pressure and temperature for both the subcooled and superheated regions, and give saturation pressure $p_{\text{sat}}(T)$ in terms of temperature for the saturation region. In the saturation region, properties other than pressure are calculated by interpolation between values calculated at the subcooled and superheated boundaries. In the supercritical region, properties are given in terms of density and temperature.

In addition to the main IAPWS-IF97 and IAPWS-95 correlations, some further correlations for certain transport qualities including viscosity and thermal conductivity have been released, and there is also a simplified set of saturation-region
properties available from the IAPWS [73, 72, 69, 68].

Calculation with the IAPWS-IF97 steam tables was considered a desirable approach in the present work, as earlier modelling of direct steam generation had used simple linearly-interpolated steam property tables of unverified accuracy. Inaccurate steam tables can be the cause of non-trivial errors in power station calculations, where small differences in steam properties can potentially equate to hundreds of thousands of dollars per year in estimated operational costs [56]. To the author’s knowledge, other workers in the direct steam generation field have not yet adopted these industry-standard steam property correlations in their work.

2.6 Steady-state and dynamic system modelling

In the case of system-level modelling of the thermodynamics of the CLFR, a difficult modelling problem is presented. The characteristics of flow in the absorber pipes are highly non-linear: forced convection boiling, variations in pressure drop and flow rate with varying fluid properties, and the variation in the mass of water contained in the pipe as sunlight and flow-rates vary. A simplified absorber model might be sought, however the behaviour of the absorber is exactly the component in the CLFR that give it its unusual dynamics, and detail in this component needs to be retained for a full understanding of the process. Models of two-phase flow and their application to solar thermal energy systems have been described above, and these types of models are used in the present work.

The further challenges of system-level modelling here are that the absorber is to be connected together with pumps, pipework, control values, orifice plates, surge tanks and other equipment, and subject to external forcing functions including ambient temperature and solar irradiance. The system under consideration in the earlier stages of this work was to be isolated from the main plant by heat exchanger, with the result that the flow loop was to have a fixed mass and a constrained volume, which leads to some difficult-to-apply boundary conditions.

This section will review the numerical methods used for the solution of such systems and discuss some of the available tools.

2.6.1 Steady-state process modelling

The field of steady-state process modelling tackles the general problem of solving the behaviour of a system of interconnected units of equipment, with streams of energy, momentum and matter between them. The focus is on solving interacting systems of heterogeneous non-linear components, and is quite different from computational fluid dynamics, solid mechanics and many of the other numerical problems with which the mechanical engineer tends to be more familiar [170]. It is also in contrast
2.6. STEADY-STATE AND DYNAMIC SYSTEM MODELLING

to ‘spreadsheet modelling’ [129], because in that technique the engineer hard-wires a calculation sequence from start to finish; it requires the engineer to ‘unravel’ the defining equations manually, and that task is one that becomes increasingly unwieldy as the number of equations and feedback loops increase.

2.6.1.1 Sequential-modular modelling

There are two main approaches in steady-state process modelling. In the first approach, the sequential-modular approach, the models of each unit are created as a self-contained subroutine which takes inputs and parameters and calculates outputs. The inputs are the stream variables, such as pressure and enthalpy, for which values will have already been determined in ‘upstream’ calculations. Inside each unit model, code is embedded that solves for the outputs, possibly by direct calculation, or possibly with some iteration [129].

The unit models are then connected, or ‘wired up’, and a specialised solver program is called. The solver attempts to work out the precedence ordering for the units and streams so that they can be calculated in the right order and an overall solution found. The difficulty, as with spreadsheet modelling, comes when there are feedback loops, or recycles in the model. This is overcome by identifying a tearing for the system of equations, which is a set of streams that can be removed from the graph in order to make the remaining graph solvable without guesswork or iteration. A torn system must then be solved iteratively by adjusting the variables in the torn streams until their values match on either side of the tear [173, 157, 109, 166, 165]. In large systems, there can be a great many different possible tear sets, as each recycle can be cut at any point around the loop, and each tear set leads to quite different numerical behaviour in the iterative problem-solving step; much effort was put into algorithms for automatic selection of tear sets with the best numerical properties. Many of the early specialist process simulators used a sequential modular approach, including FLOWTRAN, PACES and CONCEPT [170], then Aspen and Pro/II, as well as the TRNSYS package used specifically in solar thermal engineering [154].

2.6.1.2 Equation-based modelling

The more recently favoured approach to process modelling has been the equation-based approach. In this case, models of individual units are broken apart into the equations that define their behaviour, and the solver tackles precedence ordering and tearing at the level of variables and equations, rather than streams and units. The primary reason for preferring this approach has been that the inscrutable black boxes of sequential modular modelling are replaced by transparent sets of equations; the method of solution is clearly separated from the declaration of the problem
Equation-based modelling has only become viable as computer memory has increased and sparse matrix solution methods have improved. Other reasons include that equation-based models are more reconfigurable: a parameter can be changed into a variable, or an input can be changed into an output, and this does not require fundamental reprogramming of the unit model. Also, equation-based models are immediately reusable as dynamic models: they can contain derivative variables that if set to zero reduce the dynamic model to a steady model.

In equation-based modelling, the same concepts of tearing can be applied [172, 173] but this approach is found at the equation-based level to offer little advantage over multi-variable Newton iteration [130, 7, 80], particularly if good ordering [30], block decomposition [157], scaling, and upper and lower bounds are first applied.

The earliest widely used equation-based process modelling program was Speed-Up [148, 122]. The dominant equation-based modelling environments today seem to be gPROMS and Dymola. The Engineering Equation Solver (EES) [80], is an equation-based modelling tool popular in the field of solar thermal engineering, and it offers block decomposition, but lacks object-oriented language constructs that allow model components to be bundled together to create larger system models. ASCEND is probably the most mature free, open-source equation-based modelling tool with that ability [171], although a new alternative, OpenModelica, has recently appeared and is evolving rapidly with the work of a group at Linköping University [42].

Equation-based modelling software can be designed so that it is equally applicable to both nonlinear algebraic (NLA) problems, which contain only algebraic variables, and differential-algebraic equation systems (DAEs), which are covered in the following section, and even sometimes partial-differential-algebraic equations (PDAEs).

In the case of NLA solvers, there are a number of components that make up the solver. At the highest level is the non-linear part of the algorithm, which is usually a Newton solver, or a variant [130, 7]. Inside the Newton solver is a linear solver, which may use either direct methods such as Gaussian elimination, or by iterative methods, primarily Krylov methods [76]. A current software package that provides NLA solver capabilities using both sparse and dense matrix storage, and both direct and indirect methods, is Kinsol [26] which is part of the SUNDIALS NLA/DAE solver library [60].

### 2.6.1.3 Hybrid approaches

As equation-based solvers have evolved, it has become apparent that not all problems can be easily solved using only equation-based methods. Sometimes it is necessary to ‘bury’ some intermediate equations and variables so that they are not visible in
the overall system. This approach leads to a hybrid system, in which ‘black box’ models are present along with other equation-based model parts [1]. Most large simulation packages provide for some form of ‘external’ or ‘plug-in’ system which allows this approach to be used, and this is often how thermodynamic properties are treated for high-accuracy simulations [153, 80].

In the present work, the open-source ASCEND program was enhanced with the ability to use ‘black box’ models and accurate steam property correlations were implemented using that technique. To the author’s knowledge this is the first time that steady-state equation-based simulations with accurate steam properties has been possible with open software.

2.6.2 Dynamic process modelling

We address here the problem of dynamic simulation using equation-based models. This is the case where a system of equations contains one or more time-derivatives; the problem consists of first calculating the initial conditions for all unknowns in the system, followed by stepping through a specified interval of time and recording the changing values of the variables of interest.

Two distinct approaches are possible for the dynamic modelling. In the first case, we divide the system into algebraic and differential parts. The variables for which derivatives are present, *states*, are given initial values, then the algebraic part of the system is solved using an NLA solver, as described above. The values of the algebraic variables are then substituted into the differential equations, which are solved using an ODE equation solver. A widely used ODE solver which uses dense direct matrix methods and provides both Adams-Moulton (AM) and Backwards Difference Formula (BDF) techniques [5] is *LSODE* [133]; a more recent solver is *CVODE* [25], which offers both AM and BDF integration, but allows dense and sparse, direct and iterative matrix methods. The important difference between AM and BDF methods is that AM is essentially a method for explicit integration, and is not suitable for *stiff* models that contain transients of widely varying decay rates. BDF is an implicit integration method; it involves solving a system of linear equations for each time-step, but is much better able to deal with stiff problems.

The other distinct approach for dynamic modelling is to treat the system holistically using a DAE solver. This type of solver does not require the algebraic equation to be solved separately from the differential ones, and it results in some efficiency gains as a result. A DAE solver is also able to solve *high-index* problems that are common in many ‘naturally expressed’ engineering problems. These problems are structured in such a way that the initial conditions of the system cannot be computed without first taking derivatives of some or all of the system equations [123]; this is called *index reduction* [23, 95, 96, 152]. Automated index reduction using
such methods is not always the best solution; it is often possible and preferable to formulate the problem in a different way so that there is no index problem [129]. A popular DAE solver is DASSL [5]; a more recent solver with support for sparse Krylov methods as well as dense direct matrix methods is IDA, another member of the SUNDIALS library [59].

In the present work, the IDA solver was connected with ASCEND, giving the ability to solve general non-linear DAE systems using implicit BDF integration on dense direct matrix methods.

2.6.3 PDAE modelling

PDAE modelling involves systems that include both partial differential equations and algebraic equations. The basic approach for computer solution of such systems is usually the method of lines [5]. In this approach, spatial derivatives are replaced by finite-different approximations by discretising the spatial variables. A number of different finite-difference approximations are possible. Once the spatial derivatives have been replaced, the system of equations is effectively a DAE system, and the methods of the above system can be applied. The finite-difference approximation introduces a number of numerical problems however, and these can cause unexpected results when the equations are integrated [21][178].

2.6.4 Tools for equation-based dynamic modelling

The TRNSYS modelling package [154] focuses on thermal and solar energy systems, but is primarily a sequential modular modelling program, although it possesses the hybrid ability to directly include equations in the model. TRNSYS comes with support for reading-in weather data in ‘TMY’ format. Being a sequential modular package, however, it is not particularly suited to systems with a large number of repeated small component having stiff-system behaviour. It is also rather difficult to use because it lacks an object-oriented modelling language and uses cryptic numerical sequences for the definition of model ‘decks’. TRNSYS supports ‘black box’ models for additional units that can be written in FORTRAN 90. TRNSYS does not include any models for two-phase fluid flow, and is generally considered to be appropriate for annual simulations based on simplified component models, rather than detailed short time-step models with difficult transients.

Wittwer et al, 2001 [176, 177], presented a simulation environment called ColSim, which performs many of the functions of TRNSYS but is released as open source and including a GUI environment based on other open source tools (namely xfig). It is written in pure ANSI C. Its focus is on short time-step simulations but is said to be fast enough for simulation of annual performance. Wittwer et al offer benchmark
results that compare well against industry BESTEST results. Internally, ColSim uses Euler integration for time stepping. Like TRNSYS, it appears to be focused on solving single-fluid thermal systems, rather than the more general classes of process engineering problems which much of the comparable software attempts to tackle. ColSim was used for transient simulation of the Solarmundo system.

### 2.6.4.1 Thermo-fluid modelling

Much of the established software for detailed process modelling of heterogeneous thermo-fluid systems still uses the sequential modular approach, although work from *Dymola* and *gPROMS* shows that progress on the solution of these systems with equation-based approaches has moved ahead recently. Elmqvist et al, 2003 [42], detail some techniques used in Dymola/Modelica for the incorporation of object-oriented units in an equation-based modelling environment. Communication with the Dymola developers revealed that although some early success is being reported in modelling thermo-fluid systems with the equation-based Dymola, these parts of their software are still considered quite new, and are unproven with large-scale simulations.

### 2.7 Solar radiation measurement and estimation

In the design of solar thermal energy systems, it is crucial that the ‘solar resource’ can be estimated, for the purposes of project finance as well as engineering design. Then, during operating, it is desirable that solar radiation be measured so that the efficiency of the collector can be monitored over time, so that operational decisions, such as when to clean the mirrors, can be made in an informed way.

Prediction of the solar resource at a particular site can be done using ground-based measurements, or using estimates derived from visible-wavelength satellite imagery. These methods are briefly described, along with a review of basic radiative heat transfer and the availability of ‘TMY’ radiation data sets, in Appendix A.

In the present work, the need for a prediction of the solar resource at the Liddell site was required for the purposes of predicting system output. Satellite-derived data for Australia offers only an estimation of global horizontal irradiance, with no estimate for direct beam irradiance. For that reason, correlations needed to be used to estimate direct beam radiation from global horizontal radiation.

Solar radiation database programs such as *Meteonorm* could not be used in the present case, because of the very sparse grid of measurement stations operated by the Australian Bureau of Meteorology. Other than Williamtown, the nearest solar radiation data sites are Sydney (200 km south), Brisbane (900 km north), and

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9Personal correspondence with Mats Gensberg, October 2006
Canberra (350 km south-west). The data grid interpolation method offered by the Meteonorm program is not reliable in this case, due to the significant gradient of solar radiation conditions from the eastern coast sites to inland sites. A more sophisticated method for estimating the solar resource at Liddell was pursued in the present work.

2.7.1 Diffuse fraction correlations

Diffuse fraction correlations can be used to give monthly, daily or hourly estimates of the diffuse fraction of solar radiation at a particular site, given information about the global horizontal radiation, as well as perhaps other weather or location data.

Daily correlations of Collares-Pereira and Rabl, 1979, of Orgill and Hollands, 1977, and of Reindl, Beckman and Duffie, 1990 [138, 32] gave correlations in terms of the clearness index, $K_T$, a ratio of measured GHI, $H$, to the extraterrestrial radiation on a horizontal surface, $H_o$ that would be incident on the same surface in the absence of Earth’s atmosphere. The correlation of Collares-Pereira and Rabl was recommended by Duffie and Beckman in their earlier 1980 edition [31] but they currently recommend their correlation found with Reindl. The Reindl correlation however is seen to be derived from only five locations, all of them in northern Europe and temperate United States regions [138], and is seen in the present work to correlate poorly with Australian data.

The monthly correlation of Erbs, Klein and Duffie is recommended by Duffie and Beckman, but another by Collares-Pereira and Rabl correlation is also cited and was previously recommended [32, 31].

More recently, Perez et al, 2002 [127] give a method for estimating diffuse fraction, which incorporates altitude, humidity, latitude, air mass and turbidity, but requires to be applied over shorter time periods than those for which the Australian satellite data are available. Perez et al particularly note some problems with earlier methods, including significant model bias seasonally and geographically, and bad correlation in ‘extreme’ climates such as the south-west United States. Perez et al found a much more elaborate correlation for direct normal irradiation that included factors for elevation, atmospheric turbidity and air mass. The Perez method however is really applicable only as part of the data processing that is applied to the raw satellite imagery data, and is difficult to apply to the data available from the Bureau sources.

Roderick, 1999 [146] performed an analysis of 25 sites from Australia (22 sites) and Antarctica to establish daily and monthly diffuse fraction correlations. He gave location-specific correlations for each site as a piecewise-linear ramp function in terms of two parameters $X_0$ and $X_1$: 
2.7. SOLAR RADIATION MEASUREMENT AND ESTIMATION

(a) Learmonth AMO
(b) Brisbane AMO
(c) Macquarie Island AMO

Figure 2.7: Daily diffuse fraction data sets, $H_d/H$ versus $K_T$, for three locations (a) Learmonth, (c) Brisbane and (c) Macquarie Island, from Roderick [146].

\[
\frac{H_d}{H} = \begin{cases} 
0.96 & \text{for } K_T \leq X_0 \\
0.96 - \left( \frac{0.96-0.20}{X_1-X_0} \right) k_T & \text{for } K_T \in [X_0, X_1] \\
0.05 & \text{for } K_T \geq X_1 
\end{cases} 
\]  

(2.1)

Roderick attempted to correlate $X_0$ with latitude, elevation and distance from coast but no apparent trends were found, so a constant value of $X_0 = 0.26$ was used. For $X_1$, Roderick found a correlation in terms of latitude, $\phi$,

\[X_1 = 0.80 - 0.0017|\phi| + 0.000044|\phi|^2\]

Roderick proceeds to attempt to find a universal correlation for monthly diffuse ratio. His key observation is that not just the mean $\bar{H}$ but also the variance $\sigma_H^2$ are necessary correlation variables for an accurate monthly diffuse fraction correlation, because of the site-to-site variation in the way daily points tend to be distributed (Figure 2.7).

In the present work, a new study is performed using updated Australian data for 28 sites, and a simple linear correlation is formed that confirms the problems, as found by Perez and Roderick, that are seen in the earlier fraction correlations. The results give a best-reasonable-estimate approach for the difficult case of calculating diffuse fraction at Australian locations where ground-station data points are too far apart for simpler interpolation methods, and where satellite data are only available on a daily basis.
2.8 Conclusions

A number of historical and current solar thermal energy projects were reviewed, the most important of which were the SEGS project, the DISS project and Solar One. For each of these projects a wide body of research was produced. The first of these was a full-scale benchmark system against which the CLFR must be judged. The latter two, particularly the DISS project, have provided a large amount of operational experience in direct steam generation, although only at prototype scale.

Important differences exist however between the historical systems and the CLFR system that is the subject of the present work. No systems have been studied that use flow-boiling in long uninterrupted pipes in the manner proposed for the CLFR.

Modelling of flow boiling can be done with the assistance of many techniques developed for application to the nuclear reactor design problem, however those models are of a level of detail not suitable for the present problem. Modelling performed specifically for the solar thermal direct steam generation problem was found, but systems of the scale of the CLFR do not yet exist and modelling has focused on problems with a higher degree of active control, more ancillary pipework and fittings, and shorter heated-pipe runs. It is not clear to what extent experiences in earlier simulation will be transferrable to the present problem.

Techniques of process modelling were reviewed including the methods that allow systems of heterogeneous components to be modelled using automated software. Some good system-level modelling of the DISS system has been performed by Hirsch, Eck and Steinmann, although this modelling uses simplified steam property correlations and lower-accuracy pressure drop correlations which may not provide the necessary accuracy for the long pipe runs in the present system, and is focused on the needs of superheated steam generation. Modelling of Hirsch et al does not include a consideration of the flow regimes that may be present in the collector flow. Although stiff-system behaviour may be expected in certain DSG configurations, to the author’s knowledge, implicit integration schemes have not yet been successfully applied to these problems.

Satisfactory control systems suitable for once-through saturated solar steam generation in long pipes have not yet been developed.

On the basis of other workers’ conclusions about DSG, the CLFR concept appears to hold promise as an exciting new solar steam collector design, and the possibilities of integration into existing power stations, such as Liddell, provide a viable way forward even in Australia’s economic climate of low incentives and very low energy costs.
Chapter 3

Site Evaluation

For concentrator solar thermal energy installations, the tasks of performance simulation and subsequent economic evaluation require accurate estimates of direct beam irradiation data. This data is available from some weather ground stations, but the quality of the data varies considerably depending on the type of equipment installed there. Solar irradiation data can also be derived from satellite imagery, where models of surface and cloud reflectivity allow the solar radiation at ground level to be estimated with reasonable accuracy.

For the CLFR prototype project at the Liddell power station, there is no local solar irradiation data. The nearest weather station is at Williamtown, 90 km distant, which can be expected to show quite different local climate effects.

The purpose of this chapter is therefore to use the available sources of solar irradiation data to arrive at a good estimate of the solar resource at the Liddell power station, in such a way that subsequent modelling can predict the CLFR system output with some confidence.

3.1 Climate data

3.1.1 Ground stations

Solar radiation data in Australia is primarily gathered by the Bureau of Meteorology, which maintains a large number of ground stations around the country. Unfortunately, only a small number of these ground stations are equipped with the necessary additional equipment required to measure solar radiation data. The five such stations closest to Liddell\(^1\) are show in Table 3.1.

These ‘reference’ weather stations are equipped with the means of measuring the beam and diffuse components of global irradiation at one-minute intervals\(^2\), as

\(^{1}\)32.373° S 150.979° E (Google Maps)  
^{2}Data is published by the Bureau in the form of half-hourly data, although the one-minute
Table 3.1: Bureau of Meteorology ‘reference’ ground stations closest to Liddell (those with radiation measurement facility)

<table>
<thead>
<tr>
<th>Location</th>
<th>Distance and direction</th>
<th>Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Williamtown*</td>
<td>90 km to the east-southeast</td>
<td>coastal</td>
</tr>
<tr>
<td>Sydney</td>
<td>160 km to the south</td>
<td>coastal</td>
</tr>
<tr>
<td>Wagga Wagga</td>
<td>440 km to the southwest</td>
<td>far inland</td>
</tr>
<tr>
<td>Canberra</td>
<td>350 km to the south-southwest</td>
<td>inland</td>
</tr>
<tr>
<td>Brisbane</td>
<td>570 km to the north-northeast</td>
<td>coastal</td>
</tr>
<tr>
<td>Dubbo**</td>
<td>220 km to the west</td>
<td>far inland</td>
</tr>
</tbody>
</table>

* 32.798° S 151.842° E
** Dubbo is not a ‘reference’ station but was one of the locations for a study of one-minute radiation data during 1993-1996. The data is considered to be of a much lower quality than that from the reference locations, however.

well as rainfall, temperatures, wind-speeds, and so on. The purpose of the reference stations is to provide additional statistics that are desirable but too expensive to record at all locations. They are also used to calibrate the satellite data observations described in the following section.

Although the closest weather station to Liddell capable of measuring radiation is Williamtown, there are some ‘synoptic’ ground stations that provide a reduced set of weather data, including long-term data for rainfall, temperature, wind-speed, and approximate cloud cover, amongst other things. The closest such stations are Jerrys Plains (approximately 15 km south-southwest) and Scone (approximately 37 km north). Data for these stations³, as well as the nearest reference weather station, Williamtown, are shown in Figure 3.1

From Figure 3.1 we see that there is some clear difference between Williamtown and the two inland stations. The maximum daily temperatures are up to 4 °C higher at Jerrys Plains compared to Williamtown. The number of clear days however is not much different between Jerrys Plains and Williamtown, although it is somewhat higher at Scone. Reasons for these differences would include the proximity to the sea (in the case of Williamtown), nearby hills, possibly effects due to the Wollemi National Park just beyond Jerrys Plains, and finally the presence of haze due to the coal fired power stations in the area.

3.1.2 Bureau of Meteorology satellite data

The Bureau has also received data feeds from a number of geostationary meteorological satellites from 1990 to present. These data are obtained by analysing visual-wavelength imagery of the earth from space. Correlations including seasonal data can be retrieved on special request. (private correspondence with Bruce Forgan, Bureau of Meteorology, 18 Aug 2006)

Figure 3.1: Local observations of maximum temperature (lines) and clear days per month (bars) for Williamtown and the two Bureau synoptic weather stations nearest the Liddell power station.
variation of surface reflectivity (surface ‘albedo’) as well as radiation transmission and reflection of air and water vapour (including clouds) are used to transform the satellite imagery into global solar radiation estimates at the earth’s surface. In order to identify the surface albedo, the effect of clouds is eliminated by filtering the image data for times when the pixels are at their darkest for a given level of extraterrestrial radiation. It is assumed that the clouds are always more reflective than the ground. Once the local surface albedo is estimated, models of upper atmosphere and cloud albedo can be used to calculate what fraction of the light received by the satellite is attributed to surface reflection, and hence, what the irradiance at ground level must be. These data are then integrated over the course of each day.

The estimates are calibrated against, and from time to time corrected using, data from the ‘reference’ ground stations, some of which were listed in the previous section.

The satellites that have been used to supply the Bureau data are shown in Table 3.2, including the years that they were used.

The GMS-4 and GMS-5 satellites are the only ones for which data was available at the time of the present work; it was used for the periods 1990-1994 and 1997-2003, with a period of operational problems in-between during which no data was collected. The Bureau published the data from 1990 to 2001 as NCCSOL 2.209, which was used for the present analysis. The spatial resolution of the NCCSOL data varies between 6 × 6 km and 24 × 24 km. In July 2001 the data from GMS-5 was collected with a reduced frequency.

The monitoring of the accuracy of the satellite-derived data has not been a continuous process; the GMS4 and GMS5A operation periods were analysed fairly thoroughly against ground station data (Oct 1990 to Jun 1994 and Jul 1997 to Apr 1998), but later periods of operation with the GMS and GOES satellites were not analysed as extensively.

Table 3.2 also shows error estimates from some of these data sources, including all of those that were used for the present analysis. These errors were obtained by comparing the satellite-derived data at locations where a ‘reference’ ground station exists with irradiation measurements. It is seen that the estimates have been improving significantly in accuracy, but that data from 1990 to 2001 has an average error of the order of 6 to 8 %, with higher errors occurring on cloudy days. It is not known if these errors are biased or not.

The full set of daily solar irradiation from the Bureau satellite data set (NCCSOL 2.209) was arranged by spreadsheet ‘pivot table’ to give the month-average daily global horizontal radiation values shown in Tables 3.3 and 3.4. These data, along

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3.1. CLIMATE DATA

Table 3.2: Mean magnitude of error between satellite estimate and surface observation of daily exposure, averaged over all surface pyranometer sites. Source cited in text (Note that the date for the end of the ‘GOES9A’ period was wrongly published as 29 May 2003; the correct date is assumed to be 29 May 2004).

<table>
<thead>
<tr>
<th>From date</th>
<th>To date</th>
<th>Satellite</th>
<th>H error</th>
<th>% error</th>
<th>Period</th>
</tr>
</thead>
<tbody>
<tr>
<td>Oct 1990</td>
<td>13 Jun 1995</td>
<td>GMS-4</td>
<td>4.5</td>
<td>17.5</td>
<td>7.7</td>
</tr>
<tr>
<td>Jun 1997</td>
<td>3 Jul 2001</td>
<td>GMS-5</td>
<td>0.8</td>
<td>1.6</td>
<td>1.4</td>
</tr>
<tr>
<td>4 Jul 2001</td>
<td>29 May 2003</td>
<td>GMS-5</td>
<td>3.8</td>
<td>3.5</td>
<td>-</td>
</tr>
<tr>
<td>10 Jul 2003</td>
<td>29 May 2004 (?)</td>
<td>GOES-9</td>
<td>1.7</td>
<td>4.2</td>
<td>-</td>
</tr>
<tr>
<td>21 May 2004</td>
<td>present</td>
<td>GOES-9</td>
<td>1.6</td>
<td>2.3</td>
<td>-</td>
</tr>
</tbody>
</table>

with standard deviations in the daily global horizontal radiation values, are shown in Figure 3.2. The Liddell location is shown to have significantly higher average daily radiation in the summer months, but the same or slightly less in the winter months. The standard deviation in daily radiation for both locations is of the order of ±30% throughout the year, although it should be noted that the distribution is not likely to be symmetric.

3.1.3 NASA satellite data

In addition to the Bureau satellite data, an independent set of data is available from an online database maintained by NASA\(^5\) [175]. This data is of a lower resolution (squares of one degree latitude by one degree of longitude – equivalent to a square approximately 95 km east-west \(\times\) 110 km north-south in New South Wales) and it covers a different time span (the 10 years to April 2007), but is useful as an additional source of data for comparison.

The data retrieved from the NASA website for the Liddell location is shown in Table 3.5. This is the ‘pure’ form of the NASA data; additional temporal or spatial resolution is not available. The website also provides beam and diffuse radiation, but this is derived information following similar techniques to those used here, but made without reference to the available ground station data that we will be using. Data for specific days is not available. The regions corresponding to the NASA data for the two locations are shown in Figure 3.3. It is clear from this figure that there is really not enough resolution in the NASA data to draw any detailed conclusions about the differences in radiation between the two locations.

\(^5\)http://eosweb.larc.nasa.gov/sse/ accessed April 2007
Table 3.3: Liddell month-averaged data for daily global horizontal radiation $H$, from Bureau satellite

<table>
<thead>
<tr>
<th>Year</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
<th>12</th>
<th>Year average</th>
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<tr>
<td>1990</td>
<td></td>
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<td></td>
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<td></td>
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<td>27.83</td>
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<td></td>
<td></td>
<td>28.55</td>
</tr>
<tr>
<td>1994</td>
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Table 3.4: Williamtown month-average data for daily total global horizontal radiation $H$, from Bureau satellite

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3.1. CLIMATE DATA

Figure 3.2: Daily global horizontal radiation from Bureau satellite data. Error bars show the standard deviation and the lines show minima and maxima for each month from the raw Bureau data.

Table 3.5: NASA average daily global irradiation data for Liddell and Williamtown

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3.1.4 Other data sources

The Australian and New Zealand Solar Energy Society published an *Australian Solar Radiation Data Handbook* (ASRDH), which gives a wide range of statistics derived from data from the Bureau reference stations [125]. Data from ASRDH for Williamtown is shown in Figure 3.4.

A publication of one-minute data for a limited set of 14 special ground stations (numbered 901-914) was made for approximately a two-year period from 1993 to 1996, published by the Bureau as SMNAWS 1.0 [46]. The data is not considered as accurate as that from the reference ground stations, but is in a convenient form for the study of short transients at the locations in question.

Although the Bureau only publishes its ground station radiation data as half-hourly irradiation, archived data is available on special request with a one-minute resolution, for weather stations numbered 101-899.

Daily global horizontal irradiation totals from the reference ground stations are also available in a convenient aggregate form from the World Radiation Data Center. At the time of access, this data was available for 1965 to 1988 in the case of Williamtown, 1968-1991 for Longreach and 1968-1990 for Wagga Wagga.

Typical meteorological year (TMY) data was compiled by Morrison and Litvak
for Williamtown (as well as the other Bureau reference stations), giving fine-scaled data to allow prediction of long-term behaviour of solar thermal systems in accordance with Australian Standards [106].

3.2 Daily global-to-beam correlations

Satellite radiation data for Liddell will only provide us with estimated global horizontal radiation data. In order to use that data for modelling the CLFR, we will need a way to estimate the beam component of that radiation. A number of correlations are available for this that take hourly (accumulated) irradiation and estimate the diffuse component. The beam component is then the remainder. Two correlations were evaluated here, that of Orgill and Hollands (as given by Duffie and Beckman [32]) and that of Reindl, Beckman and Duffie [138].

The Orgill and Hollands correlation allows estimation of hourly diffuse radiation $I_d$ as a ratio of the hourly global radiation $I$. The correlation is given in terms of the clearness index $k_T$, which is the ratio of hourly global horizontal irradiation to hourly extraterrestrial radiation, as follows:

$$\frac{I_d}{I} = \begin{cases} 
1.0 - 0.249k_T & \text{for } 0 \leq k_T \leq 0.35 \\
1.557 - 1.84k_T & \text{for } 0.35 \leq k_T \leq 0.75 \\
0.177 & \text{for } k_T \geq 0.75 
\end{cases}$$

The correlation of Reindl et al is provided in ‘full’ form, which gives the diffuse fraction in terms of clearness index plus solar altitude $\alpha$, ambient temperature $T_a$ and relative humidity fraction $\phi$:

$$\frac{I_d}{I} = \begin{cases} 
1.0 - 0.232k_T + 0.0239 \sin(\alpha) - 0.000682T_a + 0.0195\phi & \text{for } 0 \leq k_T \leq 0.3 \\
1.329 - 1.716k_T + 0.267 \sin(\alpha) - 0.00357T_a + 0.106\phi & \text{for } 0.3 \leq k_T \leq 0.78 \\
0.426k_T - 0.256 \sin(\alpha) + 0.00349T_a + 0.0734\phi & \text{for } k_T \geq 0.78 
\end{cases}$$

The above results are subject to the additional constraint that

$$\frac{I_d}{I} \in \begin{cases} 
(0.0, 1.0) & \text{for } 0 \leq k_T \leq 0.3 \\
[0.1, 0.97] & \text{for } 0.3 \leq k_T \leq 0.78 \\
[0.1, 1.0] & \text{for } k_T \geq 0.78 
\end{cases}$$

The ‘reduced Reindl correlation’ is also given; this is similar to the above, but only the clearness index and solar altitude are used as correlation variables[138]:

---

6This correlation is the ‘full Reindl correlation’ provided in the TRNSYS version 15 ‘radiation processor’ [154]
Figure 3.4: Solar irradiation data for Williamtown, for various receiving surface configurations [125].
\[ I_d/I = \begin{cases} 
1.020 - 0.254kT + 0.0123 \sin(\alpha) & \text{for } 0 \leq kT \leq 0.3 \\
1.400 - 1.749kT + 0.177 \sin(\alpha) & \text{for } 0.3 \leq kT \leq 0.78 \\
0.486kT - 0.182 \sin(\alpha) & \text{for } kT \geq 0.78
\end{cases} \]

again with the constraints that

\[ I_d/I \in \begin{cases} 
(0.0, 1.0) & \text{for } 0 \leq kT \leq 0.3 \\
[0.1, 0.97) & \text{for } 0.3 \leq kT \leq 0.78 \\
[0.1, 1.0] & \text{for } kT \geq 0.78
\end{cases} \]

To evaluate which of these correlations gave the best results for the Australian conditions, a simple TRNSYS simulation with five-minute resolution was used to add up the total irradiation for each month for each of the reference locations Williamtown, Wagga Wagga\(^7\) and Longreach\(^8\), and then to calculate the mean daily beam irradiation for each month. Two inland locations were chosen for contrast with the coastal location at Williamtown.

The results for this calculation are shown in Figure 3.5. It can be seen that for the three locations, the Orgill and Hollands correlation tends to give higher estimates, but that on average this correlation is the closest to the actual radiation data. Orgill and Hollands’ correlation also has significantly better correspondence to the values at Williamtown, which is the nearest location to Liddell. We therefore conclude that the best estimate of short time-frame beam irradiation from global irradiation is obtained from the Orgill and Hollands correlation.

It should be noted that the Reindl correlations [138] are based on only five locations: Albany (New York State), Cape Canaveral, Copenhagen, Hamburg and Valencia, with only one year of data for most locations (specifically, one year for all except Albany, for which there were four years of data). Clearly these locations have significantly different climates to those of the Australian locations in question. For example, the most equatorial location in the Reindl data is Cape Canaveral, at 28.4°N, which is located on a coastal spit and where the average maximum temperatures are 28-32 °C in summer. This is the best match in the Reindl data for Longreach, at 23.4°S, which is far inland and has average maximum temperatures of 36-37 °C in summer. European locations in the Reindl data are all at greater than 50°N. Humidity could also be assumed to be quite different at the Australian locations. It is interesting to note that the reduced Reindl correlation is better than the full correlation at the surveyed locations, which would suggest that ambient temperature and humidity ratio correlate somehow in an opposite way in Australia compared to the northern locations used by Reindl.

\(^7\)35.110°S 147.367°E
\(^8\)23.440°S 144.250°E
Figure 3.5: Predicted (points) and actual (lines) values of average daily direct beam radiation by month, for the Orgill and Hollands correlation and the two forms of the Reindl correlation, for (a) Williamtown, (b) Wagga Wagga and (c) Longreach.
3.3. Relocating ground station data using satellite data

Hourly radiation data, of any sort, does not exist for Liddell. In order to estimate hourly data, data could either be synthesised using some controlled random data, or else must be extrapolated from another location. It was decided that the Williamtown location was sufficiently close to Liddell that some extrapolation should be attempted.

In order to find an extrapolation, we must create a scaling ratio that, when applied to hourly ground-station radiation data for Williamtown will give an estimate of hourly radiation at Liddell. We have the choice of applying a flat scaling rate for every day of the year, or alternatively monthly, weekly, or daily scaling factors. Scaling ratios could even be non-linear, as some function of daily radiation or other data at the two locations.

Monthly scaling seems like a sensible time resolution for the scaling ratio. Seasonal differences between inland Liddell and coastal Williamtown can be captured in this way, without producing possibly exaggerated effects that would be likely to arise if scaling at short time scales. To see how this could happen, consider Figure 3.6. If attempting to scale hourly ground data from Williamtown on a day when it is cloudy half of the time (left side of Figure 3.6) but sunny in Liddell (right side of Figure 3.6) would result in an approximately 200% scaling ratio, and physically impossible radiation estimates for the Liddell location. It is therefore necessary to perform scaling using a longer sample interval. Monthly data is the obvious choice, as long term averages for a wide range of climate data are available in ready form.

It is important that when applying scaling between locations, it is applied so that the important parameter is as close as possible to that required. In this case, we are designing a concentrating solar thermal system, so beam radiation is what we

Figure 3.6: A hypothetical case to demonstrate that simple daily scaling of radiation from a cloudy location (A) to a less-cloudy location (B) can lead to instantaneous scaled radiation estimates that exceed the clear-sky limit.
need to estimate. This means that in attempting to determine the scaling between locations, we should first attempt to estimate the daily total beam radiation at Liddell. If we can estimate that on a month-average scale then we can ensure that our scaled Williamtown data will have month-by-month exactly that estimated value of daily total beam radiation over the course of a month.

### 3.3.1 Monthly diffuse ratio correlations

The next issue, therefore, is to find a way to estimate month-average daily total beam radiation for each month of the year at Liddell. Correlations for this exist, such as the monthly total-to-diffuse Collares-Pereira and Rabl correlation, and the correlation of Erbs, Klein and Duffie, both given by Duffie and Beckman [32]. The correlation of Collares-Pereira and Rabl gives the monthly diffuse-to-global fraction, $\tilde{H}_d/\bar{H}$, as

$$\frac{\tilde{H}_d}{\bar{H}} = 0.775 + 0.00606 (\omega_s - 90) - [0.505 + 0.00455 (\omega_s - 90)] \cos (115\bar{K}_T - 103)$$

where $\omega_s$ is the sunset angle and $\bar{K}_T$ is the month-averaged clearness index, calculated as the month-averaged ratio of daily global radiation to extraterrestrial radiation. The equation is plotted for the range of sunset angles and clearness index in Figure 3.7. Meanwhile, the correlation of Erbs et al is given by

$$\frac{\tilde{H}_d}{\bar{H}} = \begin{cases} 1.391 - 3.560\bar{K}_T + 4.189\bar{K}_T^2 - 2.137\bar{K}_T^3 & \text{for } \omega_s \leq 81.4 \circ \\ 1.311 - 3.022\bar{K}_T + 3.427\bar{K}_T^2 - 1.821\bar{K}_T^3 & \text{for } \omega_s > 81.4 \circ \end{cases}$$

with the additional requirement that $0.3 \leq \bar{K}_T \leq 0.8$.

### 3.3.2 Australian diffuse ratio data

The above correlations appear to have been created primarily to fit northern-hemisphere climates. The source data for these correlations is not known, however in their 1980 edition, Duffie and Beckman encouraged readers to seek new correlations that improved upon the Collares-Pereira and Rabl work. It was therefore decided to examine just how well these correlations fit Australian data, with a view to selecting the best one.

Monthly data for the 28 Bureau ‘reference’ ground stations were extracted from the Australian Solar Radiation Data Handbook, and plotted on the axes of diffuse fraction $H_d/\bar{H}$ versus clearness ratio $\bar{K}_T$. The clearness ratio data was calculated
3.3. RELOCATING GROUND STATION DATA USING SATELLITE DATA

Figure 3.7: The Collares-Pereira and Rabl correlation of daily diffuse fraction as a function of daily clearness fraction $K_T = \frac{H}{H_0}$ where $H$ is daily global radiation and $H_0$ is the daily extraterrestrial radiation.

using month-average daily horizontal extraterrestrial radiation $\bar{H}_o$ calculated according to the method of Duffie and Beckman (chapters 1 and 2 [32]); the diffuse fraction was derived directly from ASRDH data for $\bar{H}_d$ and $\bar{H}$. A plot of the diffuse fraction data for all 28 locations is shown in Figure 3.8.

First of all it must be noted with reference to Figure 3.7 that at high values of $K_T$, data points for Australia are showing diffuse ratios significantly lower than those predicted by the Collares-Pereira and Rabl correlation. There appears to be a significant problem with this correlation when applied to the Australian data.

It is also alarming to note that the Williamstown data appears in Figure 3.8 as an outlier, somewhat below the main cluster of data points. The Williamstown points are located along an apparent trend-line with Halls Creek (inland of Broome), Forrest (in the Australian Bight), Longreach (central Queensland) and Oodnadatta (inland South Australia). These locations are very widely dispersed, and have the full range of Australian sunset angles.

It seems that individual locations in the Australian data could all be correlated quite well against $K_T$ in isolation; the problem comes with variation between locations. As the observations about Williamstown show, sunset angle (which differs greatly between Broome and Williamstown, for example) is not a variable that (on its own) correlates the locational effects. Reindl et al [138] found that for their hourly diffuse fraction correlation the best correlation variables (after $k_T$) were the sine of the sun altitude angle, $\sin (\alpha)$, monthly average ambient temperature $\bar{T}_a$, and the instantaneous relative humidity as a ratio of the month’s average $\phi/\bar{\phi}$. It would be
valuable to investigate a set of analogous monthly variables in order to identify the ones that give the best correlation to the Australian data, but this was not done in the present work. Another variable that might be of use is the monthly mean number of cloudy days, which is recorded for a wide range of the Bureau synoptic weather stations using a Campbell-Stokes recorder, even though beam radiation is not recorded.

### 3.3.3 An improved monthly diffuse ratio correlation for Australia

Observing the spread of data points in Figure 3.8, a comparison of predicted and measured monthly diffuse ratios was made. The correlations of Collares-Pereira and Rabl and Erbs et al were calculated against the ASRDH data for the 28 Australian locations. Figure 3.9 shows the result. It can be see that the Collares-Pereira and Rabl correlation performs quite poorly, and the Erbs et al correlation is somewhat better, but still performs poorly in cases of high clearness ratio, which is very unfortunate, as these are exactly the conditions that we are concerned with in the field of solar thermal energy generation.

Also shown on Figure 3.9 is a new relationship for monthly diffuse ratio based on the above Australian data. The relationship is a linear regression produced by correlating only against $\bar{K}_T$ as the independent variable. The influence of sunset angle $\omega_s$ was shown to be insignificant, as shown in Figure 3.10. The reason for $\omega_s$ not being a significant factor in the Australian data for $\bar{H}_d/\bar{H}$ may be that there is lower seasonal variation (compared to the northern hemisphere) between summer and winter in the selected locations, or it might be that the tendency for more clear days in winter is an opposite trend to that of northern location; this would clearly need further investigation. The linear correlation was found to be

$$\frac{\bar{H}_d}{\bar{H}} = 1.073 - 1.339 \bar{K}_T$$

### 3.3.4 Month-average global horizontal and horizontal beam radiation

We now turn back to the task of obtaining the best possible estimate of beam radiation for Liddell. Our goal is to establish a scaling ratio between the Williamtown and Liddell location. Ideally we would use data that has been gathered in the same way at both Williamtown and Liddell, which means that we must make the scaling on the basis of either satellite derived global radiation, or else synoptic data from Jerrys Plains or some other ground station. We will pursue the option of scaling based on satellite data, but first, a check of these data against other sources is given.
Figure 3.8: Month-averaged diffuse fraction $\bar{H}_d/\bar{H}$ versus clearness index $\bar{K}_T$ based on all available Australian data, extracted from the Australian Solar Radiation Data Handbook, for Bureau reference ground stations (place names have been abbreviated).
Figure 3.9: Predicted versus measured monthly diffuse fraction for 28 Australian locations. Correlations of Collares-Pereira and Rabl, and Erbs et al, are shown, as well as a new linear fit (‘Pye’) which gives improved (but still not wonderful) correlation especially in clear conditions, where $H_d/H$ is low. Values of the correlation coefficients are shown in the legend.
3.3. RELOCATING GROUND STATION DATA USING SATELLITE DATA

![Graph: Ordinary linear regression for monthly diffuse ratio $\frac{H_d}{\bar{H}}$ against monthly clearness index $\bar{K}_T$ using data points for the 28 Australian locations from the Australian Solar Radiation Data Handbook.]

Figure 3.10: Ordinary linear regression for monthly diffuse ratio $\frac{H_d}{\bar{H}}$ against monthly clearness index $\bar{K}_T$ using data points for the 28 Australian locations from the Australian Solar Radiation Data Handbook.

Month-average daily global horizontal radiation is shown for satellite and ground stations from the NASA and Bureau data banks, plus the TMY data for Williamtown and ASRDH data, in Figure 3.11. Variation in data within a particular location (as opposed to between locations) is due to (a) different years of data used by different sources and (b) differences in methods for deriving radiation estimates from satellite imagery (c) large differences in the ‘pixel’ area for the satellite data.

We next take the above global horizontal radiation data and estimate the average horizontal beam radiation at the two locations using the linear regression results of Section 3.3.3. The results are shown, for NASA and Bureau satellite data only, as Figure 3.12. The difference between the two data sources is apparent, but fortunately the two data sources seem to agree quite well on the relativity between the two locations, considering the large difference in resolution in the NASA and Bureau data.

3.3.5 Scaling ratio

The monthly ratios of estimated beam radiation at Liddell and Williamtown are plotted in Figure 3.13. The results from the NASA satellite data predict annual-average horizontal beam radiation to be 7.1% higher at Liddell compared to Williamtown. Meanwhile, the results from the Bureau satellite data predict Liddell annual-average horizontal beam radiation to be 5.2% higher than Williamtown. The value of 5.2% is preferred, due to the higher resolution of the Bureau satellite, as discussed in Section 3.1.3.
Figure 3.11: Monthly average global horizontal radiation from Williamtown (Wil) and Liddell (Lidd) data sources including ground station data (G) and satellite data (S) as well as Typical Meteorological Year data (TMY).

Figure 3.12: Estimated daily-average beam radiation on a horizontal surface for each month, for Liddell and Williamtown, using both NASA and Bureau satellites.
Table 3.6: Scaling ratios for daily horizontal beam radiation at Liddell relative to Williamtown

<table>
<thead>
<tr>
<th>Month</th>
<th>$H_{b,Lidd}/H_{b,Wil}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.207</td>
</tr>
<tr>
<td>2</td>
<td>1.086</td>
</tr>
<tr>
<td>3</td>
<td>1.121</td>
</tr>
<tr>
<td>4</td>
<td>1.026</td>
</tr>
<tr>
<td>5</td>
<td>1.034</td>
</tr>
<tr>
<td>6</td>
<td>0.985</td>
</tr>
<tr>
<td>7</td>
<td>0.984</td>
</tr>
<tr>
<td>8</td>
<td>0.997</td>
</tr>
<tr>
<td>9</td>
<td>0.978</td>
</tr>
<tr>
<td>10</td>
<td>1.078</td>
</tr>
<tr>
<td>11</td>
<td>1.159</td>
</tr>
<tr>
<td>12</td>
<td>1.193</td>
</tr>
</tbody>
</table>

The month-by-month scaling values are shown in Table 3.6. These ratios can be applied to hourly (or instantaneous) horizontal beam radiation data for Williamtown and will give an estimate of horizontal beam radiation data at Liddell that, over the course of the year, will integrate to a reasonable value for the annual horizontal beam radiation.

Consider the clear-days-per-month data of Figure 3.1. The clear-days data shows that in winter, Williamtown is very slightly clearer than Scone and Jerrys Plains, but that in November and December, both Jerrys Plains and Scone are clearer. This is consistent with the radiation scaling data found above.

3.4 Conclusions

This chapter gives a review of available climate data at the location of the Liddell power station where the CLFR prototypes are being constructed. Satellite data and ground station data sources were described, as well as some other sources of aggregate data.

As satellite data only provides global radiation data at daily intervals, a study was presented that shows that the Orgill and Hollands daily global-to-beam radiation correlation gives the best estimate of beam radiation at the proposed location.

In order to provide a way of estimating hour-scale beam radiation at Liddell, a review of available monthly diffuse ratio correlations was performed using long-term data from 28 Australian locations. The Australian data were not shown to be well correlated by the established Collares-Pereira and Rabl correlation, nor by the Erbs et al correlation. A new correlation was proposed that improves the fit to Australian data, but more work is required to improve the correlation with the introduction of
Figure 3.13: Monthly ratios of average estimated daily beam radiation at Liddell compared to Williamstown, derived from both Bureau and NASA satellite data.

other correlation parameters.

Scaling ratios were presented that can be used to translate instantaneous Williamstown ground-station data for use at the Liddell location, in such a way that monthly beam radiation at Liddell is consistent with that estimated from daily satellite data. It must be emphasised that the data upon which this scaling is based all have significant uncertainties, and there is no substitute for ground-based radiation measurements when accurate data is required.

The method in this chapter was used in the published estimates of Mills et al, 2006 [103], of the radiation available to the CLFR system at the Liddell site.
Chapter 4

Absorber Cavity Modelling

4.1 Background

The CLFR design incorporates a trapezoidal cavity receiver, somewhat similar in shape to a large ceiling-mounted fluorescent light. It is broad at the base, in order to permit upwardly-reflected solar radiation to enter from a range of angles, and narrower at the top. On the inside upper surface are positioned a number of parallel pipes onto which the solar radiation is focussed. Water passes through the pipes to remove heat by forced convection boiling (covered later, in Chapter 5). In the cavity, sloping side-walls and vertical end-walls serve to contain the built-up hot air that accumulates near the hot pipes. A transparent cavity cover constitutes the lower surface, seeking to prevent outside wind from blowing away the accumulated hot air, which would be a significant form of convective heat loss. The cover also serves to block re-radiated long-wave radiation. Figure 4.1 shows the cavity’s general shape, as well as its radiative and convective heat loss modes. A photograph of the cavity during construction was earlier shown in Figure 1.6.

The design of the cavity has been chosen for simplicity of construction and installation. It uses sheet-metal cavity walls combined with a simple welded frame [98]. The overall scale of the cavity\(^1\) is of the order of 0.5 m wide at the top, and 300 m long in total, although constructed as modules, each of which is 60 m in length.

The task here was to choose an optimal configuration that would minimise total losses under the range of typical operating conditions, and to make an estimate of what the predicted losses would be. The next section gives details of steady-state modelling of the cavity heat loss modes, and is followed by a section which investigates possible transient heat loss effects which were observed. Next, there is an investigation into a slightly different cavity design that uses an inflated plastic-

\(^1\)These values are for the Stage 1 prototype; later designs have been modified slightly.
sheet cavity cover. This design was investigated as a cheap way to make the cover. A flat plastic-sheet cover would be easier, but it was anticipated that it would be in danger of flapping in wind conditions, which would disturb the stratified hot air in the cavity, leading to increased thermal losses. Finally, a more detailed radiation-only model of the cavity is presented, showing the effects of having tubes rather than a flat surface at the top of the cavity, and the effect of the cavity window transmittance is also investigated.

![Figure 4.1: The CLFR cavity cross-section, showing convective and radiative heat-loss modes.](image)

### 4.1.1 Initial prototype design

At the time of embarking upon this modelling work, the prototype design was a flat-bottomed cavity as shown in Figure 4.2, with a cavity depth of $D = 150 \text{ mm}$ and an absorber surface width of $W = 500 \text{ mm}$, and side-walls sloping at 30 degrees to the horizontal.

A TEFZEL plastic film membrane was tried on the prototype but was unsuccessful and low iron glass with an anti-reflective coating has now been adopted as the preferred design.

The top surfaces of the cavity will be insulated with at least 100 mm of rock-wool.

### 4.1.2 Approximate operating conditions

For the sake of modelling cavity heat loss, we will consider the range of absorber temperatures in the range 256 - 337 °C (530 - 610 K). At the upper end, these values are higher than planned, however they will provide useful data for the case where superheated steam is to be generated.

### 4.1.3 Intensity ratios at the receiver

Data from the optical modelling of Damien Buie is shown in Figure 4.3[18, 16].
4.2. ANALYTICAL MODELS

For the present work, we define the intensity ratio as the ratio of local radiative flux intensity to the direct normal irradiation flux intensity.\(^2\)

A peak intensity ratio of 30 was found to be possible across the width of the bank of absorber pipes. Interestingly, the modelling shows that the optical concentration of radiation at the plane of the cavity cover has several peaks. This is a result of the focussing strategy that aims to give as uniform concentration at the absorber plane as possible.

4.2 Analytical models

The cavity receiver heat loss processes involve radiation, conductive and convective heat transfer, and the interaction of these makes a purely analytical model impossible. We can however consider the magnitude of these processes acting in isolation, to gain an approximate idea of the relative magnitudes of each.

4.2.1 Radiation

We can estimate radiative heat transfer quite accurately with some fairly simple mathematics as outlined in Appendix A (section A.9)

**Exchange between parallel grey surfaces**  For the case of two perfect black-body surfaces ‘1’ and ‘2’, radiative heat transfer is given by

\[
Q_{12} = A_1 F_{12} \sigma \left( T_1^4 - T_2^4 \right)
\]  \hspace{1cm} (4.1)

where \(\sigma\) is the Stefan-Boltzmann constant, \(A_1\) is the area of surface 1, \(Q_{12}\) is the heat transferred from surface 1 (at temperature \(T_1\)) to surface 2 (at temperature

\(^2\)The term concentration ratio will apply here to the overall geometric concentration ratio, which can be calculated using an aperture area and a focal area, with no reference to the efficiency (scatter and absorptance) of the optical surfaces involved.
Figure 4.3: Intensity ratios on two horizontal planes through the cavity. The intensity ratio on the window is seen not to be as uniform as that on the absorber surface. [18]
4.2. ANALYTICAL MODELS

$T_2$, and $F_{12}$ is the corresponding view factor, which for infinite parallel surfaces is equal to one.

In the more general case of grey-body radiation transfer, which assumes uniform emissivity of surfaces in the irradiation and emission wavelength range of interest, the above equation becomes

$$Q = \frac{\sigma (T_1^4 - T_2^4)}{1 - \frac{1}{A_1} + \frac{1}{A_{12}} + \frac{1}{A_2}}$$

In the case where the surfaces are planar and parallel, $F_{12}$ will be one, and $A_1 = A_2 \to \infty$ and the equation becomes

$$Q = \frac{1}{1 - \frac{1}{A_1} + \frac{1}{A_2}} \sigma (T_1^4 - T_2^4)$$

The heat transfer will grow with area, so we divide through by $A_1$ and simplify

$$\frac{Q}{A_1} = \frac{1}{1 + \frac{1}{A_1}} - 1 \sigma (T_1^4 - T_2^4)$$

Comparing this result with that for perfect black bodies, we see that the fractional factor $F$ plays a role similar to that of the view factor $F_{12}$ in the black body expression:

$$F = \frac{1}{1 + \frac{1}{A_1}} - 1$$

The above equation applies only to the simple case of two infinite parallel grey surfaces with different emissivities. In the real case of the CLFR cavity absorber, we have multiple inter-radiating surfaces: the side walls, the top and bottom surfaces, and we also have the issue of the opacity/transparency of the cavity cover to consider, and the process of heat loss from the outside surface of the cavity cover.

We can assume that the side walls will play a minor role in the overall heat transfer. This can be justified by observing that the inclination of the side walls is such that the view factor to the upper surface is very low, but the view factor to the lower surface will be high. Furthermore, there is very little heat loss out through the side walls, since they are insulated (with an assumed external heat transfer coefficient of 0.5 W/m²). This means that there will be negligible heat transfer from the cavity cover to the side walls compared to that between the upper and lower surface, so the side walls have negligible effect on the temperatures and the overall heat transfer in the radiation-only case we are considering here. We will therefore approximate heat transfer using Eq.4.2.

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3 Incropera & DeWitt [75] page 738
4 See the following sub-section for an analysis that does not make this simplifying assumption.
CHAPTER 4. ABSORBER CAVITY MODELLING

Using an upper surface emissivity\textsuperscript{5} of 0.49 and a cavity cover emissivity of 0.9, and assuming equal areas, we obtain

\[\frac{Q}{A} = \frac{1}{1 + \frac{1}{\epsilon_1} - 1} \sigma \left( T_1^4 - T_2^4 \right) \quad (4.3)\]

\[= 0.46 \sigma \left( T_1^4 - T_2^4 \right) \quad (4.4)\]

The value of \( F \) for this case is 0.46, and the per-area heat transfer for the temperature range \( T_1 = 550 \text{ K}, \ T_2 = 332 \text{ K} \) is \( Q/A = 2090 \text{ W/m}^2 \). The effect of the grey surfaces at this temperature is to reduce radiative heat transfer by 54\% relative to the perfect black body case. If the absorber width is 0.5 m, the linear heat loss is 1045 W/m\(^2\).

**Surface-to-surface radiation estimate**  A more complete radiation-only cavity simulation is possible with a little more work. Using the network-flow formulation from Appendix section A.9, and calculating view factors using Appendix section A.11, we can model the surface-to-surface heat transfer within the cavity plus external losses by radiation and convection.

A basic model assuming uniform temperatures across each planar surface was constructed using the ASCEND modelling package \[171\], and is listed in Appendix section A.12. This model makes allowance for external heat loss by both radiation and convection; all heat transfer on the inside of the cavity in this base model is assumed to be by radiation alone.

In this calculation we use the definition

\[Q = F_{rad} \sigma A \frac{T_{absorber}^4 - T_{cover}^4}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} \quad (4.5)\]

In other words, \( F_{rad} \) is defined as the ratio of total radiative heat loss from the absorber surface to that which would occur between infinite parallel grey plates. The temperatures are that of the upper surface of the cavity (the absorber) and the lower surface of the cavity (the cover). It is important that we use these temperatures and not the ambient temperature, since we will assume that the cavity cover is effectively opaque at the re-radiation wavelengths.

The simple surface-to-surface model gives a value of \( F_{rad} = 1.035 \) for this case. The reason that this value exceeds the value for infinite grey plates is that the side walls, which are highly reflective with \( \epsilon = 0.1 \), are enhancing the heat transfer since

\textsuperscript{5}These are approximate emissivities for re-radiated heat at long wavelengths. Long wave (re-radiation losses) and short-wave effects (incident concentrated solar radiation) are being treated as independent for the purpose of this modelling.
4.2. ANALYTICAL MODELS

much of the re-radiated heat from the relatively ‘black’ lower surface is not being returned to the upper surface but instead to these reflective side-walls. This effect is not expected to be seen in the real case, since we are assuming here that temperature is uniform along the lower surface. In fact the temperature will be far from uniform, varying according to the local view factor between the upper and lower surfaces.

A quick sensitivity analysis here shows:

- Making the side-walls perfectly black doesn’t result in a significant change in the $F_{rad}$ value, since the view is mostly towards the lower surface, and the external convection coefficient is low so black side walls don’t absorb a significant fraction of overall heat.

- Reducing the emissivity of the lower surface to 0.4 has a much more significant effect: the result is $F_{rad} = 1.41$. We have enhanced the radiative loss to significantly above the parallel infinite plates case. This is because with side-wall emissivities of 0.1, and with the configuration of the cavity, the heat re-radiated from the lower surface will tend to be reflected back to the lower surface again anyway, thus increasing the total transfer to greater than that which would have occurred between parallel infinite plates with no side-walls. If the lower surface emissivity is reduced further to 0.1, the result is $F_{rad} = 2.49$.

- Increasing the emissivity of the upper surface to 1.0 has only a minor effect on $F_{rad}$: its value changes from 1.035 to 1.071. The overall radiative heat loss changes significantly though, as expected, it almost doubles from 1081 W/m to 2019 W/m. Decreasing the emissivity on the upper surface to 0.1 changes $F_{rad}$ to 1.007 and results in an overall radiative heat loss of only 235 W/m. This shows that variations in the emissivity of the upper surface are fairly well accommodated by the Eq 4.5.

- Changes in ambient and upper surface temperatures have little or no effect on $F_{rad}$.

- Changes in cavity aspect ratio and side-wall angle can be expected to change the value of $F_{rad}$ somewhat: however change to a depth of 0.1 m and a width of 1 m only caused the value of $F_{rad}$ to change to 1.013: a fairly small change towards the infinite plate case. Changing the side-wall angle from 30 deg to 15 deg (with the original absorber width of 0.5 m and cavity depth of 0.3 m) changed the value of $F_{rad}$ to 1.044 – also a small effect.

Overall, we can say that the expression given by Eq 4.5 will, for a fixed value of $F_{rad}$, give fairly accurate predictions of radiative heat loss from the upper surface. This is
provided the emissivity of the lower surface is correct: the sensitivity analysis shows that a correct value for this emissivity is the primary variable affecting the accuracy of the simple \( F_{\text{rad}} \) expression.

### 4.2.2 Losses by conduction and convection

Analytical results for the heat loss by convection are harder to achieve, since we expect to have a stable stratified layer of hot air near the top of the cavity. It is not clear what temperature differential will therefore be available to drive the convection, and how stable the stratified region will be.

However, if we do assume the stratified layer is stable then we can safely say that any convective heat loss must first pass through the stable stratified ‘barrier’ by conduction. We can therefore estimate an upper bound on the convective and conductive losses. For a steady 4 cm-deep stratified region, with an upper surface temperature of 550 K and a lower boundary temperature of 332 K (a value found for the lower surface in the radiation modelling), even for this extreme case (quite a thin layer with large temperature difference) we will only see conduction of 106 W/m through this layer\(^6\). So we can therefore say that so long as there is a small (4 cm or greater) stratified layer in the cavity, the convective and conductive losses shouldn’t exceed about 10% of the total.

### 4.2.3 Overall heat loss

The above simple analysis shows that radiation can be expected to dominate the cavity heat losses, and should make up approximately 90% of the heat loss from the top surface. We expect total heat loss from the cavity to be of the order of 1000 W/m (per metre of cavity length, assuming \( W = 0.5 \text{ m} \) and \( D = 0.3 \text{ m} \)).

The magnitude of the cavity heat loss was shown to be most sensitive to the value of the emissivity of the cavity cover. Coatings on either the inside or outside surface of the cavity cover could be considered as a way of reducing the cavity heat loss, since in the above, the emissivity on the inside and outside of the cavity cover were considered equal at \( \epsilon = 0.9 \).

### 4.3 Steady-state CFD modelling of trapezoidal cavity

In Section 4.2 we found some simple estimates for the radiative heat losses from the absorber surface, and assuming stratification we obtained approximate bounds on the heat loss by conduction and convection. The radiation model assumed uniform

\(^6\)This was assuming air in the cavity to be at 200 °C, which is a conservative value once again, giving \( k = 0.039 \text{ W/mK} \)[57].
temperatures, and we have not made any allowance for the interaction of radiation and convection effects. A steady-state computation fluid dynamics model was therefore developed to obtain a more detailed simulation of the heat loss processes, allowing for this interaction. A more detailed model also permitted examination of the degree of stratification that can be expected; this was not at all possible using the simple analytical approach.

4.3.1 Simplifying assumptions

It is anticipated that the CLFR cavity receiver will be operating under essentially steady-state conditions for the great majority of its productive part of the day, so we assume that a good knowledge of its predicted behaviour under these conditions will allow sufficiently accurate prediction of its role in the total system. We say this since the thermal mass of the cavity structure will be small compared with that of the pipes and the water passing through them, and also small compared with the rate of solar energy input. It is expected that the slower transients will arise from the changes in sunlight levels and the changes in average water temperature in the system.

Steady state behaviour of the cavity then is to be modelled assuming constant solar radiation input and constant water mass and state in the pipes.

We further simplify by assuming that the temperatures of all of the pipes are equal and that the surface of the pipes can be approximated by a flat surface (the upper surface is then flat and of uniform constant temperature).

Simplifying further, we therefore assume that a specified rate of heat is delivered as a uniform heat flux at the upper surface (some aspects of this assumption is examined in Section 4.5.2). Any losses incurred due to mirror reflectivity, mirror fouling, or losses in the cavity cover will be assumed to have already been allowed for in the specified rate of heat delivery. Because we have a steady state situation, we can then assume that the incoming heat flux supplied at the upper surface is exactly matched by the sum of heat removed from that surface, which is comprised of heat loss by conduction into the upper part of the structure, heat loss by conduction and convection to the air in the cavity, radiation to the air in the cavity, radiation to surfaces visible from the cavity’s upper surface and the useful heat taken away through forced convection by the water in the pipes. For the sake of modelling, we can therefore remove the component of the heat supplied which is absorbed by the water and by losses out through the upper surface\(^7\), and consider only that which is

\(^7\)To clarify the reason for removing upper surface losses from the model: we consider that these losses are not coupled to the convection/radiation behaviour in the cavity. The upper surface of the pipes will attain an equilibrium with the inner upper surface of the cavity, and once that equilibrium is established, the only effect of this loss on the lower portion of the cavity is that which could be equivalently represented by a variation in the total incident radiation. Therefore we can
lost, via the air in the cavity, by radiation and convection and conduction into the side-wall insulation. We will fix the upper surface temperature, and find out what losses result from a hot upper surface maintained at that temperature.

Finally, we can simplify the model further by assuming symmetry across the vertical mid-plane.

In order for the above heat loss model to be developed, the process by which heat is drawn by convection away from the internal upper surface must be modelled more broadly to include the properties of the cavity cover, and some form of external heat loss from the cover. We choose to assume that the cavity cover has negligible thermal mass, so we model it as having emissivity on both sides, a fixed external convection coefficient, and a surface temperature.

4.3.2 Earlier work

Reynolds performed a similar analysis for a range of modelling cases [140, 142] and also gathered some experimental data [141]. Experimental data were gathered by building a model cavity, shown in Figure 4.4, in which the upper surface was heated electrically and insulated above. The hot upper surface of the inside of the cavity radiated heat into the cavity and also caused convective cells to form in the trapped air in the cavity. This effectively reproduces the situation following the assumptions outlined above: a heat transfer model which cuts out the bulk supplied heat and investigates only the loss components.

Reynolds’ modelling work concluded with some numerical correlations matching the heat loss behaviour with fixed parameters including ambient temperature and absorber temperature.

The geometry modelled by Reynolds was different from that proposed in the current design: the slope of the cavity walls was changed, and later the lower surface was made to be curved or angled outwards (see Section 4.5). In addition, Reynolds’ correlations were considered to be rather unusual in their form. For the cavity heat transfer problem, it is desirable to form correlations only in terms of properties relating to the cavity and its boundary. Properties relating to the ambient air and the external heat transfer should be accommodated in a separate (but coupled) model. In the present work, an effort is made to observe the physical boundaries in the system when generating the necessary heat transfer correlations.

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model the upper surface losses using a simple thermal resistance $R$-value derived from experimental measurements, and perform the following cavity modelling without needing to worry about upper surface losses.
4.3. STEADY-STATE CFD MODELLING OF TRAPEZOIDAL CAVITY

Figure 4.4: Photograph of the experimental cavity used by Reynolds et al [141]. This cavity has a width of 1.2 m.
Table 4.1: Varied parameters for laminar steady-state cavity heat loss model

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cavity depth</td>
<td>100, 200, 300 mm</td>
</tr>
<tr>
<td>Cavity width</td>
<td>500, 1200 mm</td>
</tr>
<tr>
<td>Absorber temperature</td>
<td>530, 570, 610 K</td>
</tr>
<tr>
<td>Ambient temperature</td>
<td>290, 305 K</td>
</tr>
<tr>
<td>External convection coefficient</td>
<td>2.6, 10 W/m²</td>
</tr>
</tbody>
</table>

Table 4.2: Fixed parameters for laminar steady-state cavity heat loss model

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Absorber emissivity in cavity</td>
<td>0.49</td>
</tr>
<tr>
<td>Wall emissivity in cavity</td>
<td>0.1</td>
</tr>
<tr>
<td>Internal and external emissivity of the cover</td>
<td>0.9</td>
</tr>
<tr>
<td>Cavity temperature (Boussinesq approximation)</td>
<td>370 K</td>
</tr>
<tr>
<td>Side-walls external heat loss coefficient</td>
<td>0.5 W/m²K</td>
</tr>
<tr>
<td>Sky temperature</td>
<td>5 K above ambient</td>
</tr>
</tbody>
</table>

4.3.3 Laminar steady-state cavity heat-loss model

A series of 72 laminar steady-state model cases were run using FLUENT 6.0 software, with the new geometry as shown in Figure 4.2. The inclination of the side-walls, at 30° rather than 60°, is different from that used in the Reynolds work. The shallower side-wall angle was chosen to allow a greater acceptance angle in response to changes in the layout of the mirror array. The Boussinesq approximation was used to model convection effects, under the assumption that the temperature range inside the cavity would not be great enough for non-linear density variation to become significant.

Initially, models were run with combinations of parameters drawn from the following sets shown in Table 4.1. In all model cases, other parameters took the fixed values as shown in Table 4.2. The radiation ‘sky’ temperature was taken to be 5 K above ambient temperature to allow for the fact that the cavity will be facing mostly mirrors which will be opaque at the wavelengths emitted, and these mirrors will have heated somewhat in the sun.

Mesh The computational model uses a mesh size of 2mm, following the mesh sensitivity carried out by Reynolds et al. [140]. The structure of the grid used here was rectangular between the parallel surfaces, and triangular ‘paved’ under the sloping surfaces. The side walls are lined with a single row of 2mm rectangular elements for improved convection modelling at the boundary. By symmetry, only

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8See comments in following section
9This value is an estimate only; it was intended that this value be measured using the CLFR Stage 1 prototype, but data was not received in time for this modelling work. See also Section 6.1.
one side of the cavity needed to be simulated.

**Computation** The solution was found using the segregated steady-state two-dimensional solver, with the Boussinesq approximation used for convection effects. The Discrete Transfer Model (DTRM) was used to model radiation transfer within the cavity\textsuperscript{11}. Simple constant-value convection coefficients and constant-value emissivities were used to model external losses. The cavity cover was assumed to be opaque, because at this stage the cover was assumed to be made of glass whose transmittance at the radiated wavelengths is low.

Heat loss through the insulated side-walls was modelled using a very low overall heat transfer coefficient of (0.5 W/m\(^2\)).

For each of these models, 20,000 iterations were performed.

### 4.3.4 Laminar steady-state results

The results of the initial laminar modelling cases\textsuperscript{12} are shown in Table 4.3. An overall appreciation of the results can be gained from Figure 4.5, which shows the total absorber heat loss for all of the tabulated data points, but grouped visually by cavity width \(W\), depth \(D\) and heat transfer coefficient on the outside of the cover, \(h_g\). From the regular nature of these results we would expect to be able to attain a good correlation using a simple power-law relationship between variables.

A plot of the contours of stream function for a typical modelled case (Table 4.3 case #1590) is shown in Figure 4.6. Note the wavy flow structure near the lower half of the centreline: this flow structure was present in most of the modelled cases, and would probably be responsible for some of the difficulty involved in achieving convergence of these models. The wavy structure is essentially an unstable region in the flow pattern resulting from the localised maximum in buoyancy of air rising off the midpoint of the cavity cover. At this point, the surface of the cover is hottest because of the radiation view factors between the various surfaces.

A convective cell is established that takes the warm air up off the midpoint of the cover into the stratified layer of hot air above. Cooling on the sidewalls and cover drive the convective cell. It appears that the strength of the convection at the midpoint is great enough that the flow may be oscillatory or turbulent. This is investigated further in Section 4.4.

The stratified zone in the Figure 4.6 appears to take up a little over half of the depth of this 200 mm deep cavity. Also, the total heat transfer of 592 W/m from\textsuperscript{13}

---

\textsuperscript{11}The simpler S2S formulation could not be used since its view factor calculation algorithm cannot handle symmetry planes.

\textsuperscript{12}These results were presented at the 2003 ISES conference [132].
Table 4.3: Laminar half-cavity modelling results

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4.3. **STEADY-STATE CFD MODELLING OF TRAPEZOIDAL CAVITY**

Figure 4.5: Graph of average absorber heat-loss flux versus model case. This graph gives a general overview of the effect of the various parameters: absorber temperature is a strong effect as is external convection. Cavity geometry has a weaker effect, as does ambient temperature.

Figure 4.6: A plot of contours of stream function for a typical case in the steady-state laminar modelling. Fairly stable stratification is observed in the upper half of the cavity, with a convection cell occupying the lower half. Note the wavy flow structure near the lower half of the centreline. The cavity dimensions here are $D = 200 \, \text{mm}$, $W = 500 \, \text{mm}$ wide. The absorber temperature is 257 °C (530 K), and the ambient temperature is 17 °C (290 K). External heat transfer coefficient is 2.6 W/m²K.
the upper surface of the half-cavity was comprised of 555 W/m of radiation heat transfer, so the assumption of radiation heat transfer being dominant is confirmed.

A further point about the wavy flow structure: we have imposed symmetry in the flow in these cases. We would not expect this wavy flow structure to occur in reality: more likely is some kind of oscillating pattern such as a von Kármán vortex street, or possibly something more chaotic. Further analysis was performed to investigate this, see Section 4.9.

Finally, it should be noted that the Boussinesq approximation was used in this cavity modelling, without verification of its accuracy. A calculation using extreme air temperatures of 610 K and 370 K, and a mean air density of 0.75 kg/m$^3$ (460 K) gives a density variation ratio $\Delta \rho / \rho = 0.49$. Further modelling should be performed to quantify the variation in the flow patterns arising from the Boussinesq assumption. However, it is noted with reference to Reynolds [142] that flow patterns predicted here are consistent with his experimental results. Convective heat losses are seen to be small compared to radiative losses, so this further modelling has not been performed here.

4.3.5 Laminar steady-state correlation: Radiation

In order to create correlations that are physically-based, the radiation and convection heat loss components were treated separately, and dimensionless groups were sought from among the parameters in question. This approach will require that the radiation-and-convection interaction effects be negligible.

Following the approach taken in Section 4.2.1, we seek a correlation parameter that will allow prediction of the radiative heat loss for arbitrary un-simulated conditions. Since it is desirable to maintain physical significance in this correlation parameter, we will use the form of the equation given by

$$\frac{Q_{rad}}{A} = F_{rad} \sigma \epsilon_{absorber} \left( T_1^4 - T_2^4 \right)$$

(4.6)

This form is similar to, but not the same as, the earlier definition for $F_{rad}$, given in Eq 4.5. In Eq 4.6 we have simplified the expression with the assumption that emissivity of the cavity cover is close to one.

Heat loss by radiation. The above correlation was performed on the cases outlined in Section 4.3.3, and the resulting values of $F_{rad}$ are plotted in Figure 4.7. It is observed that the values are within a narrow range of 0.87 to 0.93.

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13Note that, although this is the value of the maximum temperature difference, much of the cavity is actually stratified, meaning that the density ratio for that portion of the fluid that is actually involved in convective heat transfer is rather lower.
4.3. STEADY-STATE CFD MODELLING OF TRAPEZOIDAL CAVITY

Figure 4.7: Calculated values of $F_{\text{rad}}$ for the laminar flow cases in Table 4.3.

The values of $F_{\text{rad}}$ from the laminar steady-state simulations are slightly lower than those predicted by the simple surface-to-surface radiation model. Mostly this will be because of the effects of non-uniform temperature on the lower and sidewall surfaces. Stratification and convection effects will also affect the temperature profiles, and this in turn will have had some effect on the values of $F_{\text{rad}}$. Another difference between the results of Section 4.2.1 is that the simulation used a ‘sky temperature’ of 5 deg C above ambient temperature, to account for heating up of the mirrors and ground underneath the absorber.

Geometric effects are seen in the results in Figure 4.7. These effects were predicted by the simple surface-to-surface model and as before are seen to be reasonably small. Likewise, the effects of changing external convection coefficient is also seen to have only a small effect.

4.3.6 Laminar steady-state correlation: Convection

Convection in the cavity is a free-convection process that appears to be primarily driven by the heat loss from the sidewalls, but arguably also by the temperature variation along the lower surface. We here identify non-dimensional parameters for the convective heat transfer and determine a correlation that will allow later prediction of cavity heat loss for arbitrary conditions.

Grashof number Free convective flow and heat transfer from a vertical plate is governed by physical equations that in normalised form contain a non-dimensional parameter named the Grashof number $\text{Gr}$ (sometimes written $\text{Gr}_L$, $\text{Gr}_D$, etc., to specify what is being used as the ‘characteristic length’ parameter), which takes the form of Eq 4.7:
CHAPTER 4. ABSORBER CAVITY MODELLING

\[ Gr = g \cdot \beta \cdot \left( T_s - T_\infty \right) \cdot D^3 \nu^2 \] (4.7)

where \( g \) is the gravitational acceleration, \( \beta \) is the coefficient of volumetric expansion for the fluid, \( T_s \) is the temperature from which the convective heat transfer is taking place and \( T_\infty \) is the bulk temperature of the fluid temperature, \( D \) is the ‘characteristic length’, or in other words the scale of the experiment (all other lengths are scaled by this length to make them dimensionless), and \( \nu \) is the kinematic viscosity.

The Grashof number represents a ratio of buoyancy forces to viscous (damping) forces, and hence will reflect the magnitude of convective flow liable to be taking place.

As with most free convection correlations, we will form a Grashof number from the available parameters in our model. In particular, the bulk temperature of the fluid is not well defined for a cavity with a convection cell, so we will use as our temperatures the absorber temperature and the mean cavity cover temperature. This is likely to be problematic, since the presence of the stratification in the upper half of the cavity will mean that the temperature difference driving the convection is likely to be something other than \( T_{\text{absorber}} - T_{\text{cover}} \). Perhaps it would be better to use something like \( T_{\text{membrane midpoint}} - T_{\text{base of sidewall}} \). However using these more physically significant temperatures gives a correlation that will not be useful in the prediction of heat-loss for arbitrary conditions, and that is what we really require here.

So, we will use the average cover surface temperature in place of the bulk cavity temperature. For the scale of the experiment we will use the depth of the cavity, since we presume this will have a more direct affect on the convective heat transfer taking place (we will eventually correlate \( Gr \) with an aspect ratio, however, to accommodate effects from the cavity width). For the fluid properties, we will evaluate these at the average of the two temperature values, \( T_c \). Our Grashof number is then best expressed as

\[ Gr = g \cdot \beta_c \cdot \frac{(T_{\text{absorber}} - T_{\text{cover}})}{\nu_c^2} \cdot D^3 \] (4.8)

\[ \beta_c = \beta(\text{water}, T_c, p_{\text{atm}}) \]
\[ \nu_c = \nu(\text{water}, T_c, p_{\text{atm}}) \]
\[ T_c = \frac{1}{2} (T_{\text{absorber}} + T_{\text{cover}}) \]

**Nusselt number** The Nusselt number is a dimensionless heat transfer coefficient conventionally used in convection situations (both free and forced) to describe the
4.3. STEADY-STATE CFD MODELLING OF TRAPEZOIDAL CAVITY

increase in heat transfer that occurs as a result of the motion of the fluid over
the bounding surface. That means that we are comparing the heat loss (usually
convective heat loss) with that which would have occurred with only conduction
taking place. Therefore, the standard Nusselt number expression has the form

\[
Nu = \frac{hL}{k}
\]  

(4.9)

Nusselt number correlations are then normally applied in practice by calculating
the theoretical heat loss by conduction alone, and multiplying by the Nusselt
number (usually calculated from an expression in terms of Grashof number or other
dimensionless numbers) to estimate the convective heat loss.

In order to calculate a Nusselt number for the cavity heat loss, we observe that
the pure-conductive heat loss can be estimated by Eq 4.10,

\[
\frac{Q_{\text{cond}}}{A} = \frac{k_c}{D} (T_{\text{absorber}} - T_{\text{cover}})
\]  

(4.10)

where \(k_c\) is the thermal conductivity of air evaluated at \(T_c\) and atmospheric pressure.
\(A\) is the area over which the heat \(Q_{\text{cond}}\) is convected. Then, taking

\[
Nu = \frac{Q_{\text{conv}}}{Q_{\text{cond}}} = \frac{Q_{\text{conv}}/A}{\frac{k_c}{D} (T_{\text{absorber}} - T_{\text{cover}})}
\]  

(4.11)

we arrive at our definition of Nusselt number for this case. For the value \(Q_{\text{conv}}\) we
will insert our simulation results, then we will correlate the Nusselt numbers with
the Grashof number and any other non-dimensional groups we require.

Note that the above approach assumes that conduction losses are purely one
dimensional. In fact the cavity is wider at the base, so we will be underestimating
the conductive heat loss using this approach; the deeper the cavity the greater the
error there will be.

Note also that the cavity temperature is being arbitrarily chosen as the mean of
the absorber and cover temperatures. Again, this is accurate in the 1-D conduction
case, but in the case of our wider-at-the-base cavity, it will introduce another error.

Also note that because of the stratification in the upper portion of the cavity,
the value of \(T_c\) will quite probably not really represent the bulk temperature of the
fluid undergoing convection.

The fact that stratification appears to be the dominant bottleneck on convective/conductive heat-loss suggests that perhaps an approach such as this one will
be only partly successful at best: the situation is not a pure convection case (due to
the stable stratified region), so we would not expect simple convection equations to
fit the data particularly well.
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Results From the simulation results, values for the above-defined Nusselt and Grashof numbers needed to be determined. We also added another correlation parameter, an aspect ratio, $D/W$. This will compensate to some extent for the 2-D conduction effects and possible changes in the behaviour of the convective cell in different cavity depths.

Using a least-squares fit to a power-law equation, the best fit for the data was found to be:

$$Nu = 1.19Gr^{0.104} \left( \frac{D}{W} \right)^{0.643} \quad (4.12)$$

An assessment of the correlation in Eq 4.12 can be made using Figure 4.8. Factors other than cavity dimensions are clearly systematically affecting the results however given the simplistic approach (ignoring the effect of the stratified layer) the correlation appears reasonable.

4.3.7 Applying the correlations

How to apply these correlations is not trivial: a full model incorporating the correlation equations requires that internal heat loss is calculated as well as external heat loss; this requires that we make estimates for the surface temperatures on the sidewalls and on the cavity cover. We must then iterate to find temperatures that balance the internal and external total heat loss rates.

The simplified external heat losses are modelled for this purpose as follows:
4.4. TRANSIENT MODELLING OF TRAPEZOIDAL CAVITY

\[ Q_{\text{sidewalls}} = N \, h_{\text{sidewalls}} \left( T_{\text{cavity}} - T_{\text{ambient}} \right) \]
\[ Q_{\text{cover,conv}} = B \, h_{\text{cover}} \left( T_{\text{cover}} - T_{\text{ambient}} \right) \]
\[ Q_{\text{cover,rad}} = B \, \epsilon_{\text{cover}} \left( T_{\text{cover}}^4 - T_{\text{ambient}}^4 \right) \]

Note that in the above, \( Q_{\text{sidewalls}} \) is calculated relative to the cavity temperature, \( T_{\text{cavity}} \) which is taken as the mean of the absorber and cover temperatures. This is a crude approximation, since the average temperature will be affected by the temperature variation within, and the thickness of, the stratified layers at the top of the cavity. This approximation also ignores the more dominant effects of cavity radiation on surface temperature: the internal heat loss model showed that temperatures on the side-walls could in some cases even be lower than those on the cavity cover. However, we have seen from the simple simulation in Section 4.2.1 that neither the (internal) emissivity of the sidewalls, nor the external heat transfer coefficient, were particularly significant in affecting the overall heat-loss, so we will assume that this approximation will not cause too much disruption in the final calculated thermal losses.

The overall process for calculating heat loss is as shown in Algorithm 1. This process gives a value of the heat losses from the absorber surface for a given value of absorber temperature. The absorber temperature, as discussed in Section 4.3.1, depends on the total incident radiation (the concentrated short-wavelength irradiation that was assumed independent of the heat loss effects modelled here) as well as the heat transfer on the other side of the absorber (to the operating fluid in the pipes). Therefore a complete system model will require the iteration given in the above steps, plus additional higher-level iteration for the absorber surface temperature \( T_{\text{absorber}} \). These further elements in the system model will be discussed in Chapter 5.

### 4.4 Transient modelling of trapezoidal cavity

A full-cavity model was generated in order to study possible flow instability indicated by the wavy pattern in the laminar half-cavity modelling (Section 4.3). The intention of this was to remove the forcing effect of the symmetry plane used in previous investigations.

Using a full cavity transient model, it was found that convergence was achieved more readily. The S2S radiation model, simple external boundary conditions, Boussinesq approximation and laminar flow model were used. The range of simulations was similar to those used previously, with the exception of the cavity depth. Instead
Algorithm 1 Solving for cavity heat-loss using both internal and external heat loss correlations

1. Take an assumed value of $T_{\text{absorber}}$
2. Guess the cavity cover temperature $T_{\text{cover}}$.
3. Calculate the cavity temperature $T_c$ as the mean of these
4. Calculate the external loss values $Q_{\text{sidewalls}}$, $Q_{\text{cover, conv}}$, and $Q_{\text{cover, rad}}$ and hence the overall external heat loss $Q_{\text{external}}$.
5. Calculate physical and transport properties for air at atmospheric pressure and the temperature $T_c$.
6. Calculate Grashof number $Gr$ for the cavity using Eq 4.8
7. Using the correlation in Eq 4.12, calculate $Nu$ and hence $Q_{\text{conv}}$
8. Using the value of $F_{\text{rad}}$ and Eq 4.6, calculate $Q_{\text{rad}}$.
9. Calculate the total internal heat loss $Q_{\text{internal}}$ at the assumed value of $T_{\text{cover}}$.
10. Use an iterative solver to adjust $T_{\text{cover}}$ in the range $(T_{\text{ambient}}, T_{\text{absorber}})$, repeating from step 2 to converge on a value where $Q_{\text{external}} = Q_{\text{internal}}$.

Table 4.4: Varied parameters for the transient modelling of the trapezoidal cavity.

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of using cavity depth, the aspect ratio $D/W$ was used, with the three values 1:5, 2:5 and 3:5. The range of simulations was therefore as shown in Table 4.4. Variation in ambient temperature was not considered\(^{14}\).

Model cases were run for one hour of simulated time using an adaptive time step of up to 0.5s. The grid mesh was 2mm, which was found to be satisfactory in a mesh dependency study by Reynolds[141]. All models were initiated with a uniform temperature in the cavity of 370 K.

The flow pattern that resulted (Figure 4.9) appears to have a basically stratified upper layer, with a pair of large convective cells in the lower half of the cavity. Near the centre-line, rising fluid from the hot spot in the middle of the cavity cover produces local instability in the flow, and vortex entrainment occurs between the

\(^{14}\)The effect of changing the external convection coefficient was considered to be enough to force any possible changes in internal flow structure, without varying the ambient temperature as well.
4.5. ‘BUBBLE’ CAVITY

Figure 4.9: Stream function contours after a one-hour transient laminar simulation with a \( D = 300 \text{mm}, W = 500 \text{mm} \) cavity.

Figure 4.10: ‘Bubble’ cavity geometry

left and right-hand sides, resulting in a flow pattern that alternately swings to the left and right. For shallower cavities, the pattern is stable and does not oscillate. Depending on the model parameters for the case in question, the oscillations in the pattern sometimes disappear as time progresses, and at other times appear to be stable, becoming steadily periodic.

4.5 ‘Bubble’ cavity

As indicated in Section 4.1, a new cavity geometry was proposed which included a gently inflated plastic film membrane to prevent disturbance of the cavity stratification due to flapping of the plastic film during windy conditions. In this section some modelling results for this geometry are presented to confirm that the addition of a curved cover does not adversely affect heat loss compared with earlier modelling.

4.5.1 Geometry

Geometry of the ‘bubble’ cavity is defined by two additional parameters (Figure 4.10). The first is the width \( L \) of the ‘lip’ used to retain the plastic film; the second is depth of the membrane bubble \( M \), or in other words, the distance by which the bubble bulges outwards.
4.5.2 Cavity film material

The cavity cover material proposed for use in the ‘bubble’ cavity design was a plastic film used in large-scale greenhouses, called TEFZEL, from DuPont. TEFZEL is an ethylene-tetrafluoroethylene film. Figure 4.11 shows the short and long-wave absorptance data for TEFZEL material. The film is relatively ultra-violet stable, possibly due its near-transparency in the UV range. The film is uni-axially oriented, which means that it will have a tendency to tear in one direction, as a result of the film drawing process. The film has low surface energy, which means that the surface should not attract dust and require cleaning. The film is stiff and has low tendency to creep. The heat-stabilised variety of the film shrinks by 1% at 180°C and melts at 270°C. It is rated for continuous use at 150°C and intermittent use at 200°C. [48, 33, 83]

In Section 4.3.1, we assumed that the cavity cover would be completely transparent to incoming radiation. We see that this assumption is not too far wrong for this plastic material: in the 300 to 800 nm wavelength band, the absorptivity ranges from 0.07 to 0.04. Numerically integrating the absorbed energy with respect to wavelength for the terrestrial radiation profile (Table 2, page 52, [107]) for air-mass 1.5, we obtain an average absorptivity of 0.044 for the wavelength range 15^200–800 nm. The irradiation in this wavelength band at air mass 1.5 is approximately 486 W/m^2, so at an intensity ratio of 20, the membrane will be absorbing about 420 W/m^2 from direct solar irradiation.

4.5.3 Model cases

The aim here was not to calculate a new set of correlation equations but instead simply to check that the modified design did not cause the heat loss to increase substantially.

A set of four simulations was run for the bubble cavity. Transient laminar models were used. The same 2mm grid construction technique was used as previously to construct a trapezoidal cavity with the addition of small side-lips and a cylindrically-curved lower surface. The cavity width in these cases was 605 mm, which is slightly wider than the 500 mm width used in earlier simulations.

In the four simulations, cavity cover external convection coefficients of 10 W/m^2 and 25 W/m^2 were used. Two absorber temperatures were also used, 530 and 570 K.

Parameter values that were held constant are shown in Table 4.5.

---

15 We would have used a broader 200–3000 nm range, however data for TEFZEL was not available for this full range.

16 Data from Damien Buie [18] shows a maximum intensity ratio of 20 on a cavity cover when the absorber intensity ratio is 30 (Figure 4.3)
4.5. ‘Bubble’ Cavity

(a) Wavelength range 200-800 Nm

(b) Wavelength range 650-1230 Nm

Figure 4.11: Absorptance TEFZEL 150 ZM film for ultraviolet, visible and short infrared [48]. TEFZEL 150 ZM was the plastic film proposed for use with CLFR for the ‘bubble’ cavity concept.
CHAPTER 4. ABSORBER CAVITY MODELLING

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cavity depth (upper section)</td>
<td>125 mm</td>
</tr>
<tr>
<td>Simulated time span</td>
<td>1 h</td>
</tr>
<tr>
<td>Absorber emissivity</td>
<td>0.5</td>
</tr>
<tr>
<td>Sidewall external convection coefficient</td>
<td>0.5 W/m²</td>
</tr>
<tr>
<td>Ambient temperature</td>
<td>290 K</td>
</tr>
<tr>
<td>Sidewall emissivity</td>
<td>0.1</td>
</tr>
<tr>
<td>Plastic film emissivity ( h_{\text{cover}} )</td>
<td>0.8</td>
</tr>
</tbody>
</table>

Table 4.6: Simulation results for the one-hour transient laminar model for the ‘bubble’ cavity

<table>
<thead>
<tr>
<th>case</th>
<th>( T_{\text{absorber}} )</th>
<th>( h_{\text{cover}} )</th>
<th>( Q_{\text{abs, tot}} )</th>
<th>( Q_{\text{abs, rad}} )</th>
<th>( T_{\text{window}} )</th>
<th>( T_{\text{wall}} )</th>
<th>( T_{\text{cavity}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>#2592</td>
<td>530</td>
<td>10</td>
<td>1090.7</td>
<td>1027.3</td>
<td>344</td>
<td>388</td>
<td>373</td>
</tr>
<tr>
<td>#2594</td>
<td>530</td>
<td>25</td>
<td>1156.3</td>
<td>1088.6</td>
<td>320</td>
<td>377</td>
<td>354</td>
</tr>
<tr>
<td>#2596</td>
<td>570</td>
<td>10</td>
<td>1477.0</td>
<td>1402.5</td>
<td>360</td>
<td>407</td>
<td>392</td>
</tr>
<tr>
<td>#2599</td>
<td>570</td>
<td>25</td>
<td>1624.0</td>
<td>1541.9</td>
<td>331</td>
<td>390</td>
<td>367</td>
</tr>
</tbody>
</table>

4.5.4 Results

Numerical data Table 4.6 shows the aggregate statistics at the end of the one-hour simulations for the ‘bubble’ cavity.

Flow pattern Figure 4.12 shows contours of stream function. We observe that the flow pattern still shows a thick stratified layer at the top of the cavity. For this case, the total heat loss is 1623 W/m. We also observe that radiative losses in this geometry are about 95% of the total losses: this appears to be a result of the thicker stratified region in this design.

Time-variation in heat loss Examination of one case (case #2592 of Table 4.6: absorber temperature 530 K, external convection 10 W/m²K) of the values of absorber heat loss over time show that the value decreases exponentially from a starting value of 2500 W/m to a slightly but erratically oscillating value in the range 1090.4 to 1091.4 W/m after about 3 minutes. Figure 4.13 shows the variation in absorber heat loss for the final one minute of the hour-long simulation.

All of the four cases show total heat flux varying over a range of no greater than 1 W/m after the initial establishment of the stratified layer.
4.5. ‘BUBBLE’ CAVITY

Figure 4.12: Contours of stream function for the ‘bubble’ cavity, after one hour of transient laminar simulation (case #2599 of Table 4.6). A thick stratified layer is still present, and relatively stable, in the upper part of the cavity.

Figure 4.13: Heat loss from the absorber surface for the last 60 s of the simulated time (case #2592 of Table 4.6). This transient modelling shows that even with an unsteady flow pattern such as seen in Figure 4.12, heat loss from the cavity will vary over only a very small range.
CHAPTER 4. ABSORBER CA VITY MODELLING

Table 4.7: Extract of result from Section 4.3.4, for comparison with transient modelling cases.

<table>
<thead>
<tr>
<th>case</th>
<th>D</th>
<th>$T_{absorber}$</th>
<th>$T_{∞}$</th>
<th>$h_{cover}$</th>
<th>$Q_{tot,absorber}$</th>
<th>$T_{window}$</th>
<th>$T_{absorber}$</th>
<th>$T_{cavity}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>#</td>
<td>(K)</td>
<td>(K)</td>
<td>(W/m²K)</td>
<td>(W/m)</td>
<td>(K)</td>
<td>(K)</td>
<td>(K)</td>
<td>(K)</td>
</tr>
<tr>
<td>1579</td>
<td>100</td>
<td>530</td>
<td>290</td>
<td>10</td>
<td>861.6</td>
<td>351.7</td>
<td>530.0</td>
<td>410.4</td>
</tr>
<tr>
<td>1583</td>
<td>100</td>
<td>570</td>
<td>290</td>
<td>10</td>
<td>1170</td>
<td>369.7</td>
<td>569.9</td>
<td>435.2</td>
</tr>
<tr>
<td>1591</td>
<td>200</td>
<td>530</td>
<td>290</td>
<td>10</td>
<td>900.6</td>
<td>337.7</td>
<td>529.9</td>
<td>382.5</td>
</tr>
<tr>
<td>1595</td>
<td>200</td>
<td>570</td>
<td>290</td>
<td>10</td>
<td>1220.8</td>
<td>352.3</td>
<td>569.9</td>
<td>403.3</td>
</tr>
<tr>
<td>1603</td>
<td>300</td>
<td>530</td>
<td>290</td>
<td>10</td>
<td>929.2</td>
<td>328.8</td>
<td>530.0</td>
<td>360.9</td>
</tr>
<tr>
<td>1607</td>
<td>300</td>
<td>570</td>
<td>290</td>
<td>10</td>
<td>1267</td>
<td>340.9</td>
<td>569.9</td>
<td>380.2</td>
</tr>
<tr>
<td>1611</td>
<td>300</td>
<td>610</td>
<td>290</td>
<td>10</td>
<td>1655.6</td>
<td>354.6</td>
<td>609.9</td>
<td>401.3</td>
</tr>
</tbody>
</table>

Table 4.8: Fixed parameters for the transient ‘bubble cavity’ model cases.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cavity depth</td>
<td>125 mm</td>
</tr>
<tr>
<td>Cavity width</td>
<td>605 mm</td>
</tr>
<tr>
<td>Absorber emissivity</td>
<td>0.5</td>
</tr>
<tr>
<td>Sidewall external convection coefficient</td>
<td>0.5 W/m²</td>
</tr>
<tr>
<td>Sidewall emissivity</td>
<td>0.1</td>
</tr>
<tr>
<td>Ambient temperature</td>
<td>290 K</td>
</tr>
</tbody>
</table>

Magnitude of losses  We can compare the heat losses in the cases here with those from Section 4.3.4. The results from that earlier modelling that have parameters consistent with those in the cases modelled here are shown in Table 4.7.

We observe that the magnitude of the losses in the ‘bubble’ cavity here are comparable with those from 200 or 300 mm depth cavity in the flat-cavity-cover results.

4.6 Further transient modelling of the bubble cavity

Some further transient modelling of the ‘bubble’ cavity was performed in order to generate more complete results and to assess the $F_{rad}$ value and the correlation parameters for this new geometry. Based on the findings in section 4.5, the simulation times were reduced to 20 minutes – ample time for the total heat transfer values to stabilise to their final values (final values of heat transfer were seen after 3 minutes).

In these simulations, the parameters for plastic film emissivity, external convection coefficient and absorber temperature were varied, as shown in the first columns of Table 4.9. The values of fixed parameters in these simulations are show in Table 4.8.
4.6. FURTHER TRANSIENT MODELLING OF THE BUBBLE CAVITY

Table 4.9: Results for the 20-minute-simulation-time ‘bubble’ cavity model cases described in Section 4.6.

<table>
<thead>
<tr>
<th>case</th>
<th>( T_{\text{absorber}} ) (K)</th>
<th>( h_{\text{cover}} ) (W/m(^2))</th>
<th>( \epsilon_{\text{cover}} )</th>
<th>( Q_{\text{abs,tot}} ) (W/m)</th>
<th>( Q_{\text{abs,rad}} ) (W/m)</th>
<th>( T_{\text{cover}} ) (K)</th>
<th>( T_{\text{wall,left}} ) (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2580</td>
<td>530</td>
<td>10</td>
<td>0.9</td>
<td>1129</td>
<td>1065</td>
<td>343.6</td>
<td>385.1</td>
</tr>
<tr>
<td>2581</td>
<td>530</td>
<td>10</td>
<td>0.8</td>
<td>1091</td>
<td>1028</td>
<td>343.9</td>
<td>387.9</td>
</tr>
<tr>
<td>2582</td>
<td>530</td>
<td>25</td>
<td>0.9</td>
<td>1196</td>
<td>1127</td>
<td>320.7</td>
<td>374.1</td>
</tr>
<tr>
<td>2583</td>
<td>530</td>
<td>25</td>
<td>0.8</td>
<td>1156</td>
<td>1089</td>
<td>320.2</td>
<td>376.9</td>
</tr>
<tr>
<td>2584</td>
<td>570</td>
<td>10</td>
<td>0.9</td>
<td>1530</td>
<td>1454</td>
<td>359.9</td>
<td>404.7</td>
</tr>
<tr>
<td>2585</td>
<td>570</td>
<td>10</td>
<td>0.8</td>
<td>1477</td>
<td>1403</td>
<td>360.5</td>
<td>408.0</td>
</tr>
<tr>
<td>2586</td>
<td>570</td>
<td>25</td>
<td>0.9</td>
<td>1624</td>
<td>1542</td>
<td>331.0</td>
<td>390.1</td>
</tr>
<tr>
<td>2587</td>
<td>570</td>
<td>25</td>
<td>0.8</td>
<td>1570</td>
<td>1490</td>
<td>330.4</td>
<td>393.5</td>
</tr>
<tr>
<td>2588</td>
<td>610</td>
<td>10</td>
<td>0.9</td>
<td>2017</td>
<td>1930</td>
<td>378.3</td>
<td>424.0</td>
</tr>
<tr>
<td>2589</td>
<td>610</td>
<td>10</td>
<td>0.8</td>
<td>1946</td>
<td>1861</td>
<td>379.2</td>
<td>428.3</td>
</tr>
<tr>
<td>2590</td>
<td>610</td>
<td>25</td>
<td>0.9</td>
<td>2147</td>
<td>2052</td>
<td>343.3</td>
<td>406.9</td>
</tr>
<tr>
<td>2591</td>
<td>610</td>
<td>25</td>
<td>0.8</td>
<td>2075</td>
<td>1983</td>
<td>342.7</td>
<td>411.3</td>
</tr>
</tbody>
</table>

4.6.1 Results

Results for the 20 minute transient simulations of the bubble cavity are shown in Table 4.9.

4.6.2 Analysis

The important conclusion from these ‘bubble cavity’ simulations is that the dominant heat loss mode remains the radiative losses. A concern was that the bubble shape have contributed to enhanced convection but this appears not to have been the case. The radiative losses comprise between 94 and 96% of the total thermal losses. It is noted that the cavity cover temperature \( T_{\text{cover}} \) is quite high in some of these cases, so this will be investigated further in the next section.

4.6.3 Membrane temperature

There were concerns over the temperature that may be reached on the TEFZEL plastic film membrane. The membrane temperature profile for one of the simulation cases (Case #2590) is shown in Figure 4.14. This plot is important in that it shows the variation in cavity cover temperature is very non-uniform. We attribute this primarily to the local variation in view factor between the cavity cover and the absorber. At operating temperatures in the range shown in Table 4.9, there appears to be a dangerous risk of entering the creep temperature range for the plastic film, especially in low-wind conditions, although with careful design, and a suitable control strategy, it might be possible.
CHAPTER 4. ABSORBER CA VITY MODELLING

4.7 Effect of tube bank geometry on heat transfer

The modelling presented in Section 4.2.1 on page 76 makes the assumption that the hot absorber surface may be approximated by a flat surface. In fact, the CLFR cavity contains absorber tubes inside the cavity, and it is from this bank of hot tubes that the lost heat originates. We seek here to create a more detailed model of the cavity geometry so that the effect of a non-planar absorber is understood. This work has been performed after that given in later chapters, so this result will simply be used to quantify possible errors that can be expected from the early plane-surface cavity modelling.

In general, when a smooth surface is replaced by a rough surface, we expect its effective emissivity to rise. This is because the surface has become more cavity-like (the concave portions of the surface become self-viewing, \( F_{11} > 1 \)), and so the radiation which emerges from it is more like that which emerges from a cavity ‘peephole’. The same logic applies to the CLFR absorber; the ‘rough’ surface of the tube bank will be more like a black body, and will have a consequently higher emissivity.

4.7.1 Model

In order to quantify the change in cavity heat loss due to correctly modelling the bank of tubes in the absorber cavity, we will use two 2D radiation-only cavity models. The model here considers only the flat-bottom cavity, not the ‘bubble’ design.

The first model contains four planar surfaces (the top surface, the window, and the left and right side-walls), with each surface subdivided into sub-surfaces for the sake of view factor calculation, so that accurate local temperatures may be
4.7. EFFECT OF TUBE BANK GEOMETRY ON HEAT TRANSFER

also calculated. The top surface is temperature to the a value that will be the absorber temperature. The window has external radiative and convective heat loss components. The top surface and side walls have an assigned heat transfer coefficient to correspond to the heat loss through the rockwool insulation.

The second model contains the same subdivided surfaces as the first model, with the addition of a series of octagonal ‘pipes’ placed near the top of the cavity. The surfaces of the second model are shown in Figure 4.15. In this model, the top surface segments have the same external heat loss coefficients, but the temperatures will not be constrained. Instead, we constrain the temperature of the pipe surface segments to the absorber temperature. The sidewalls and window are treated as for the first model.

Two-dimensional view factors for both models are then calculated using the 2D view factor calculation tool from the program View3D by George Walton. This program uses adaptive integration to accurately calculate numerical radiation view factors including for cases where surfaces are partially obstructed [169].

Surface emissivities are set according to the values also used for earlier modelling, as given Appendix C.

For each of the two models, the view factor output from View3D is converted into an ASCEND input file view3d.a4c that can be used to configure the surfaces, view factors, emissivities and boundary conditions for the general-purpose radiosity solver cavity.a4c given in Appendix A. This cavity model is then solved and average and local surface temperatures as well as overall heat flux can be calculated for varied absorber temperature.

4.7.2 Results

View factors For Model 1, results from View3D give the average view factor from the top of the cavity to the cavity window as 0.9387. This means that 93.87% of radiation that leaves the top surface strikes the cavity window.

For Model 2, the average view factor from the tubes to the cavity window is

\[ \text{View factors} \]

\[ 0.9387 \]

\[ \text{Model 1} \]

\[ 0.9387 \]

\[ \text{Model 2} \]

\[ \text{http://view3d.sourceforge.net} \]
Table 4.10: Radiation-only cavity heat loss model results for simple plane-surface cavity model (Model 1) and cavity with bank of absorber tubes (Model 2).

<table>
<thead>
<tr>
<th>Absorber temperature (K)</th>
<th>Model 1 (no tubes)</th>
<th>Model 2 (tubes)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$q_{\text{loss, total}}$ (W/m$^2$)</td>
<td>$T_{\text{window,avg}}$ (K)</td>
</tr>
<tr>
<td>450</td>
<td>443</td>
<td>310</td>
</tr>
<tr>
<td>500</td>
<td>707</td>
<td>322</td>
</tr>
<tr>
<td>523.15</td>
<td>856</td>
<td>329</td>
</tr>
<tr>
<td>570</td>
<td>1220</td>
<td>344</td>
</tr>
</tbody>
</table>

0.337. This value is skewed somewhat by the fact that we have allowed for radiation exchange between the tubes and the top surface of the cavity. If we count only the radiation leaving the tube that hits surfaces other than the top surface, the view factor from the tubes to the cavity window becomes 0.510. In any case, it is clear that our model shows a significant change in the view factors as a result of changed geometry.

**Net heat transfer** Running the two models with ASCEND yields the results shown in Table 4.10 for an ambient temperature of 290 K. We see that the cavity model with tubes gives an increase of 25% to 27% over the range of temperatures tested.

**Temperatures on the cavity window** Figure 4.16 shows the cavity window and cavity top surface temperatures for Model 2 (with tubes) for the case of an absorber surface of 523.15 K. Cavity window temperatures are seen to have risen significantly as a result of the changed geometry. Temperatures on the top surface are near-uniform at the top surface due to the limited field of view from top surface segments. These temperature profiles are consistent with Figure 4.14 which was calculated using FLUENT including both convection and radiation effects. These profiles tend to confirm the conclusion that radiation both dominates the cavity heat loss effects in the CLFR cavity design; an understanding of the convective cell in the lower portion of the cavity is not necessary in order to predict the temperature variation across the cavity window.

**4.7.3 Discussion**

The simple surface-to-surface radiative transfer model for the CLFR cavity was modified to include the effect of tubes being present at the top of the cavity, instead of just a plane surface. This was shown to increase heat loss from the cavity by 25% at typical operating temperatures. The new radiation transfer model also
4.8. Conclusions

A laminar computational fluid dynamics model of the CLFR absorber cavity was constructed and a large number of model cases computed, to allow prediction of heat loss from the cavity with variations in design and operating conditions. It was found that radiative heat loss dominates the convective losses, with radiative heat loss at about 90% of the total. Radiative heat loss is dominant because a stable stratified layer of hot air in the top portion of the cavity prevents excessive convection at the absorber surface. No operating conditions were found that caused serious disturbance of this stratified layer.
CHAPTER 4. ABSORBER CAVITY MODELLING

The total heat loss per length of absorber cavity at operating conditions is of the order of 800 to 1600 W/m and the temperature of the cavity cover will be of the order of 350 to 400 K.

A Nusselt-Grashof correlation for the convective component of heat loss is found, and a calculation procedure is presented that allows prediction of the total cavity heat loss for given conditions and surface properties. The procedure uses iterative approach to find the equilibrium between heat transfer inside and outside the cavity, in both cases with radiative and convective heat transfer. The radiative component of heat transfer is also checked against a simple surface-to-surface heat loss model excluding convection.

Transient laminar modelling shows that some unsteadiness (oscillation and/or turbulence) can be expected near the centre of the cavity cover, although it appears that it is not sufficient to destroy the stratified layer that effectively insulates the absorber.

The ‘bubble cavity’ concept was examined and it was observed that temperatures at the centre of the plastic film cavity surface were sufficiently high to cause concern in this case.

A radiation-only cavity heat loss model was implemented to investigate the effect of altered absorber geometry on cavity heat loss. The flat-plat configuration is used throughout this thesis, however a model of a tube-bank absorber shows 25% higher heat loss due to altered radiation view factors. Other absorber configurations are also possible, and might include single-pipe absorbers and/or pipes with secondary concentrators as used in the SolarMundo system, however these are not investigated here. It is reiterated that a cavity with an absorber tube bank would exhibit heat loss 25% higher than the flat plat cavity model assumed in the remaining chapters.
Chapter 5

Two-phase absorber pipe flow model

In the modelling of the cavity heat losses, in Chapter 4, we found a way to estimate the losses from the cavity receiver for assumed values of ambient temperature, absorber surface temperature, external convection coefficient, as well as cavity geometry and material properties.

In this chapter, the focus is on the steam generation in the pipes. Heat will be absorbed from the hot absorber surface according the the internal convection coefficient inside the pipes. Depending on the water state, this will cause an increase in the temperature, or a change in steam quality. Furthermore, we will calculate the effect of pipe friction on the flow. This will cause a pressure drop that will also depend on the pipe size and steam state.

Steam properties are non-linear over the subcooled, saturated and superheated regions, so we divide the flow path into small elements and calculate using localised linear approximations. The desired steady-state solution is then achieved by stepwise integration down the length of the pipe, with the absorbed heat, pressure drop, exit fluid state and pipe surface temperature being solved at each step in turn.

This chapter will detail the methods used to create this model for the flow of water in the absorber, and the work that has been done to validate it.

5.1 Modelled phenomena

5.1.1 Conservation equations

We consider mass, momentum and energy conservation for one-dimensional pipe flow. In the case of steady-state flow, the total mass flow rate must be constant at each point in the flow, although the mass-proportion of the flow present in the gas
phase, $x$, may vary:

$$\frac{d\dot{m}}{dz} = \frac{d}{dz} (x\dot{m}_g + (1 - x)\dot{m}_f) = 0$$

For conservation of energy, we use the first law of thermodynamics applied to pipe flow to obtain that

$$\dot{m} \frac{dh}{dz} = \dot{q}_t$$

where $\frac{dh}{dz}$ is the gradient in specific enthalpy, and $\dot{q}_t$ is the rate of heat transfer to the fluid per pipe length. The enthalpy $h$ can be computed from $xh_g + (1 - x)h_f$ in the two-phase region. No work is done by the fluid other than flow work, which is already included in enthalpy.

Finally, for conservation of momentum, a detailed transient-flow derivation (given later) gives

$$\frac{1}{A} \frac{\partial \dot{m}}{\partial t} = -\frac{\partial p}{\partial z} - \frac{f}{D} \frac{1}{2} \rho v^2 - \frac{\partial (\rho v^2)}{\partial z}$$

from which the steady flow case is

$$\frac{\partial p}{\partial z} = -\frac{f}{D} \frac{1}{2} \rho v^2 - \frac{\partial (\rho v^2)}{\partial z}$$

where the average flow velocity\(^1\) is $v$ and the average density is $\rho$, and the friction factor $f$ is not yet known. This can be simplified by using that $\rho A v = \dot{m}$, and by neglecting compressibility of the fluid phase $\frac{\partial v}{\partial z}$, to give

$$\frac{\partial p}{\partial z} = -\frac{f}{D} \frac{1}{2} \rho v^2 - \frac{m_x^2 v}{A^2} \frac{\partial v}{\partial z}$$

This reduces to slightly simpler expressions in the case of single-phase flow.

\subsection*{5.1.2 Pressure drops}

Pressure drop correlations use the dimensionless friction factor $f$, defined as the pressure drop over a pipe length (direction $z$) of one internal pipe diameter $D$, divided by the kinetic power of the flow, $\frac{1}{2} \rho v^2$ (calculated from the fluid density $\rho$ and the bulk fluid velocity $v$):

$$f = \frac{-\frac{\partial p}{\partial z} D}{\frac{1}{2} \rho v^2}$$

\footnote{In this thesis we use $v$ to represent a velocity and $v$ to represent a specific volume.}

Once the friction factor is known, it is simple to determine the pressure drop $\Delta p$ for any given length of pipe $L$ by rearranging Eq 5.1 in terms of the pressure
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5.1.2.1 Single-phase pressure drops

For single-phase flow, pipe flow can be laminar, transitional, or fully turbulent. For the laminar (potential, irrotational) flow case, we use the purely analytical expression

\[ f = \frac{64}{\text{Re}} \]

where \( \text{Re} \) is the Reynolds number for the flow,

\[ \text{Re} = \frac{\rho v D}{\mu} = \frac{v D}{\nu} = \frac{\dot{m} D}{A \mu} = \frac{G D}{\mu} \]

This laminar flow relationship is seen to be valid up to around \( \text{Re} < 3000 \). Once Reynolds number rises above this, pressure drop becomes dependent also on pipe roughness. For the case where the pipe roughness is negligible, the smooth pipe correlation of Petukhov is used [75]:

\[ f = [0.790 \cdot \log(\text{Re}) - 1.64]^{-2} \]

For rough pipes, we use the relative roughness ratio \( \epsilon/D \), and the Colebrook equation [159]:

\[ \frac{1}{\sqrt{f}} = -0.86 \cdot \log \left( \frac{\epsilon/D}{3.7} + \frac{2.51}{\text{Re} \sqrt{f}} \right) \quad (5.2) \]

In this last case, \( f \) must be solved iteratively. A good technique is to repeatedly evaluate \( 1/\sqrt{f} \) using the value of the right hand side of Eq 5.2 until its value stops changing by more than a chosen tolerance. A good starting value for \( 1/\sqrt{f} \) seems to be 1.0.

In the present work, transition effects (\( \text{Re} \approx 3000 \)) are ignored. In fact, flow will almost always be turbulent for water and the pipe sizes in question here.

5.1.2.2 Two-phase pressure drops

Two phase frictional pressure drops\(^2\) are typically defined in terms of a multiplier \( \phi^2 \) that gives the pressure gradient as a ratio of a ‘fluid only’ pressure gradient:

\(^2\)The CLFR collector is comprised almost solely of horizontal pipes, so gravitational pressure drops are ignored in the present work.
\[ \frac{dp}{dz}_{TP} = \phi^2 \frac{dp}{dz}_{FO} \]

where \( \frac{dp}{dz}_{TP} \) is the two-phase pressure gradient, and \( \frac{dp}{dz}_{FO} \) is the pressure gradient were the same total flow rate to be present but entirely as saturated liquid. The latter can be easily calculated from single-phase pressure drop correlations. The value of \( \phi^2 \) is correlated based on experimental observations.

**Friedel correlation.** The Friedel correlation\[14\] for horizontal flow\(^3\) gives \( \phi^2 \) in the form

\[ \phi^2 = A + 3.43x^{0.695} (1 - x)^{0.24} \left( \frac{\rho_f}{\rho_g} \right)^{0.8} \left( \frac{\mu_g}{\mu_f} \right)^{0.22} \left[ 1 - \frac{\mu_g}{\mu_f} \right]^{0.89} \text{Fr}_f^{-0.047} \text{We}_f^{-0.0334} \]

where \( x \) is the steam quality, \( \rho_f \) and \( \rho_g \) and liquid and gas densities, and \( \mu_f \) and \( \mu_g \) and liquid and gas dynamic viscosities. The first term, \( A \) is defined as

\[ A = (1 - x)^2 + x^2 \left[ \frac{\rho_f f_{fo}}{\rho_g f_{go}} \right] \]

in which \( f_{fo} \) and \( f_{go} \) are liquid-only and gas-only friction factors.

The Froude and Weber numbers are calculated in terms of pipe diameter \( D \) and mass flux \( G \) as follows:

\[ \text{Fr}_f = \frac{G^2}{gD\rho_f^2} \]

\[ \text{We}_f = \frac{G^2 D}{\rho_f \sigma} \]

In the calculation of \( \text{We}_f \), the surface tension \( \sigma \) for water can be calculated using the IAPWS Release on Surface Tension [69].

**Martinelli-Nelson correlation.** The Martinelli-Nelson correlation [94] gives the multiplier \( \phi^2 \) in terms of inlet pressure \( p \) and quality \( x \) as shown in Figure 5.1. For the present work, a sixth order polynomial least-squared fit was made to each of the pressure levels tabulated in the original paper, with weighting applied to ensure the curves pass through the origin. This was changed from the third- and fourth-order curve-fits used by Odeh and Reynolds in their modelling after some errors in their curve-fit were found.\(^3\) This form applies to horizontal and upwards vertical flow. A different form is used for vertical downward flow.
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A comparison of the pressure drops computed for Friedel and Martinelli-Nelson correlations for a few simple cases is given in Section 5.3.2.

5.1.2.3 Minor losses

For compatibility with the approach used by Rheinländer et al [144], we will use the method for computing minor losses in two-phase flow by Chisholm [22]. Chisholm defines the pressure drop attributable to a pipe bend $\Delta p_b$ during two-phase flow via the ratio equation

$$\frac{\Delta p_b}{\Delta p_{bLO}} = 1 + \left( \frac{\rho_f}{\rho_g} - 1 \right) \left\{ B x (1 - x) + x^2 \right\}$$

where

$$B = 1 + \frac{2.2}{k_{LO} \left( 2 + \frac{R}{D} \right)}$$

In the above, $R$ is the bend radius (tested by Chisholm for the range 0 to 5.02), $k_{LO}$ is the K-factor for liquid-only flow and $\Delta p_{bLO}$ is the pressure drop for liquid only flow, given as

$$\Delta p_{bLO} = k_{LO} \frac{G^2}{2\rho_f}$$

For the 90° bends in the present work, we will use the K-factor $k_{LO} = 0.234$ quoted by Chisholm (p 365) for long-radius $R/D = 5.0$. 

Figure 5.1: Martinelli-Nelson friction factor multiplier for two phase flow [94].
For minor losses other than bends, Rheinländer cites a paper by Paliwoda [121], which gives a system for calculating minor losses in refrigeration systems. For ball valves, a pressure drop equal to a single Chisholm pipe bend is used.

5.1.3 Heat transfer correlations

In the pipe flow in the CLFR absorber, we are dealing with a situation of internal forced convection heat transfer. Behaviour of forced convection heat transfer is typically predicted using dimensionless correlations in terms of the Nusselt number, which is defined as

\[ \text{Nu} = \frac{h D}{k} \]

where \( h \) is the internal convection coefficient, defined as \( Q = h A (T_{\text{fluid}} - T_{\text{pipe wall}}) \). The pipe diameter is \( D \) and the fluid conductivity is \( k \). It is noted that defining \( k \) in the context of two-phase flow is not simple, so Nusselt number correlations are not always used in this case.

5.1.3.1 Single-phase heat transfer in pipe flow

For laminar flow with \( \text{Re} < 3000 \), where the surface heat flux is uniform and the flow is fully developed, the Nusselt number for fully developed flow is seen to be constant, with a value \( \text{Nu} = 4.36 \). This is for the case of constant surface heat flux. Laminar flow will not occur in the present modelling except for very briefly during start-up.

For turbulent flow with \( \text{Re} \gtrsim 10000 \), the Dittus-Boelter equation applies (as given by Incropera and DeWitt [75]), which correlates Nusselt number as a function of Reynolds number and Prandtl number \( \text{Pr} = \nu/\alpha \). The Prandtl number is the ratio of momentum diffusivity to thermal diffusivity, and hence the relative sizes of the momentum and thermal boundary layer effects.

The Dittus-Boelter equation is

\[ \text{Nu}_D = 0.023 \text{Re}_D^{4/5} \text{Pr}^n \]  \hspace{1cm} (5.3)

where \( n = 0.3 \) for heating, and \( n = 0.4 \) for cooling.

For the transition region between laminar and turbulent, we use the the Gnielinski correlation (as given by Incropera and DeWitt [75]), which in addition to dependence on Reynolds number and Prandtl number is dependent on the friction factor. The friction factor must therefore be first calculated using the correlations in Section 5.1.2.1.
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\[ \text{Nu}_D = \frac{(f/8)(\text{Re}_D - 1000) \text{Pr}}{1 + 12.7(f/8)^0.5(\text{Pr}^{2/3} - 1)} \] (5.4)

It is observed that Nusselt numbers from the Dittus-Boelter equation and the Gnielinski correlation do not agree at the point \( \text{Re} = 10000 \). For that reason, a linearly-interpolated smoothing region will be used here between the values of \( \text{Re} = 10000 \) and \( \text{Re} = 11000 \).

In all of the above correlations, Nusselt number is independent of the pipe wall temperature\(^4\), so the heat transfer coefficient can be calculated independently of the pipe wall temperature. The final value of internal heat transfer is then calculated iteratively by adjusting the pipe wall temperature until internal heat transfer at the selected pipe wall temperature matches the external heat transfer at the selected pipe wall temperature, giving energy conservation. This approach is suitable for steady-state modelling and where the pipe thermal mass is small.

5.1.3.2 Flow boiling

For two-phase saturated flow boiling, we use the Kandlikar correlation (as supplied by Lienhard & Lienhard [84]). The Kandlikar correlation is somewhat simpler than the Gungor and Winterton correlation (as given by Stephan [156]) used by earlier workers, and is claimed to give greater accuracy.

The Kandlikar correlation gives the forced convection boiling coefficient \( h_{TP} \) as a ratio

\[ \frac{h_{TP}}{h_L} = \max \left( \frac{h_{TP}}{h_L}_{nbd}, \frac{h_{TP}}{h_L}_{cbd} \right) \] (5.5)

where

\[ \frac{h_{TP}}{h_L}_{nbd} = (1 - x)^{0.8} \left[ 0.6683 \text{Co}^{-0.2} f_o + 1058 \text{Bo}^{0.7} F \right] \]

and

\[ \frac{h_{TP}}{h_L}_{cbd} = (1 - x)^{0.8} \left[ 1.136 \text{Co}^{-0.9} f_o + 667.2 \text{Bo}^{0.7} F \right] \]

These latter two expressions relate to ‘nucleate boiling dominant’ and ‘convective boiling dominant’ heat transfer. The dimensionless group Co is the convection number, given by

\(^4\)Strictly speaking, the Dittus-Boelter correlation is a function of pipe wall temperature. However for steady-state modelling we can normally determine whether heating or cooling occurs without needing to know the exact value of pipe wall temperature.
\[ C_0 = \left( \frac{1 - x}{x} \right)^{0.8} \sqrt{\frac{\rho_g}{\rho_f}} \]

The boiling number, Bo, is also present, given by

\[ Bo = \frac{q_w}{Gh_{fg}} \]

The value of \( F \) is fluid-dependent. For water, a value of 1.0 is used. The factor \( f_o \) is influenced by whether or not stratification occurs, and hence is dependent on the orientation of the pipe. For the case of horizontal pipe\(^5\), we can use

\[
 f_o = \begin{cases} 
 1 & \text{for } Fr_{LO} \geq 0.04; \\
 (25 Fr_{LO})^{0.3} & \text{for } Fr_{LO} < 0.04 
\end{cases}
\]

where the Froude number for liquid-only flow is

\[ Fr_{LO} = \frac{G^2}{\rho_f g D} \]

For Eq 5.5, the value of the liquid-only convection coefficient \( h_L \) is computed using the Gnielinski correlation (Eq 5.4), with

\[ Re_L = \frac{GD}{\mu_f} \]

and liquid properties at the saturation temperature. Here we note that the Kandlikar correlation sidesteps the need to calculate the average conductivity \( k \), as the correlation has not been made in terms of Nusselt number.

### 5.1.3.3 Flow condensation

For condensation heat transfer, which will occur in connecting and reticulating pipework, we will use the Rohsenow correlation (as given by Behnia [14], Eq. 7.50):

\[ Nu = \frac{h_f D}{k_f} = Pr \cdot \frac{Re^{0.9}}{F_2} \left( 0.15 \frac{X_H}{X_H} + 0.4275 X_H^{0.476} \right) \]

where

\(^5\)For vertical pipe, stratification does not occur, and \( f_o = 1 \).
\[
\begin{align*}
F_2 &= \begin{cases} 
5 \cdot \Pr_f + 5 \cdot \log (1 + 5 \cdot \Pr_f) + 2.5 \cdot \log \left(0.0031 \cdot \text{Re}_f^{0.812}\right) & \text{for } \text{Re}_f > 1125 \\
5 \cdot \Pr_f + 5 \cdot \log \left(1 + \Pr_f \left(0.0964 \cdot \text{Re}_f^{0.585}\right)\right) & \text{for } 60 < \text{Re}_f < 1125 \\
0.707 \cdot \Pr_f \cdot \text{Re}_f^{0.5} & \text{for } \text{Re}_f < 60
\end{cases}
\end{align*}
\]

and

\[
X_{\text{HT}} = \sqrt{\frac{\rho_g}{\rho_f}} \left(\frac{\mu_f}{\mu_g}\right)^{0.1} \left(\frac{1 - x}{x}\right)^{0.9}
\]

and Prandtl number \( \Pr_f = \mu_f c_{p,f} / k_f \) and Reynolds number \( \text{Re}_f = G(1 - x)D/\mu_f \).

### 5.1.4 Flow regimes

It is important to be able to estimate what flow regime is occurring in the two phase flow in the pipes, since two-phase flow is frequently unsteady and if allowed to enter into equipment items such as pumps and heat exchangers will possibly cause damage to equipment and will therefore be a safety risk. For the smooth running of the DSG system, we need to study the flow regimes that occur in the collector under different conditions, and determine a strategy that allows steady two phase flow structures, such as stratified and annular flow, to be the only ones that are seen at the collector exit.

The overall method used for flow regime modelling here is the “unified model for predicting flow-pattern transitions for the whole range of pipe inclinations” given by Barnea[11], simplified so as to only cater for horizontal pipes\(^6\). This in turn utilises the horizontal and near-horizontal regime transitions of Taitel and Dukler[160], the annular-to-dispersed-bubble transition of Barnea[10] and the upward vertical flow transitions of Taitel and Barnea[161]. The code used here was based on the FORTRAN code written as part of Odeh’s thesis[115], but re-written in C++ and checked against the original publications.

The simplifying assumption of horizontal flow has the effect of making the gravitational acceleration parameter \( Y \) of Taitel and Dukler[160] become identically zero, such that the stratified liquid level (Figure 5.4) becomes a simple function of just the Lockhart-Martinelli parameter \( X \). This means that, as opposed to inclined pipes, where one must know the pressure drop gradient before the flow regime can be cal-

---

\(^6\)It should be noted that the Taitel-Dukler-Barnea flow regime transition models were developed for the case of adiabatic two-phase flow. It has been shown by Frankum, Wadekar and Azzopardi [49] that adiabatic flow regime maps are adequate for modelling flow regime transitions in the flow boiling, as is the case here.
CHAPTER 5. TWO-PHASE ABSORBER PIPE FLOW MODEL

Figure 5.2: Flow regime map computed using the present computer code, according to the method of Barnea et al [11], for a pressure of 50 bar in a pipe with a 21 mm inside diameter.

Figure 5.3: Diagram of the flow structures in different horizontal flow regimes of two-phase flow. From Taitel et al [162] (p 921).
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Figure 5.4: Stratified liquid level curves. $Y$ is the relative gravitational acceleration, which in the present case with horizontal pipes is zero. $X$ is the Lockhart-Martinelli parameter, which gives the relative magnitude of liquid and gas frictional pressure gradients. For inclined pipes, computing the value of $Y$ requires iteration to solve for pressure drop equilibrium between phases. The figure is from Taitel and Dukler [160].

culated, we are here able to calculate the flow regime at any point knowing only the fluid state and flow rate.

A map of flow regimes, as generated by the new model, for a typical operating condition of 50 bar and 21 mm inside diameter is shown in Figure 5.2. These are the flow regimes as described\footnote{Here, the ‘slug’ and ‘plug’ (or ‘elongated bubble’) flow regimes have been lumped together under the banner of ‘intermittent’, since the structures are quite similar, and distinction between them is not necessary here: it only matters that they are unsteady.} by Taitel et al [162]. The flow regimes are illustrated in Figure 5.3. The process used to generate Figure 5.2 is to perform a flow regime calculation for each pixel in the flow map; the boundaries emerge in the resulting image and don’t need to be calculated explicitly.

The main flow regime of interest here are the intermittent regimes shown in red: ‘slug’ and ‘plug’. These flow regimes are undesirable because the large variations in
fluid momentum between the gas and the liquid ‘slugs’ results in large forces when the flow encounters resistance in the form of pumps, heat exchangers, pipe bends and other equipment. In order to ensure the safe operation of the CLFR, we desire steady flow regimes at the outlet, including stratified, annular or dispersed bubble flow.

5.1.5 Integrating the system of equations

The above system of differential-algebraic equations allows us to consider the steady state flow as an initial value problem, with the independent variable being pipe flow distance. We specify inlet flow conditions (flow rate plus two thermodynamic properties) then integrate the differential equations down the length of the pipe. Hence, as with any initial value problem, we have the choice of a range of integration techniques, or, alternatively, the problem could be viewed as a boundary value problem, with some combination of inlet and outlet properties specified [5].

For this chapter, the steady-state model being described is an ad-hoc semi-explicit Euler integration scheme. Upstream conditions are used to calculate friction factor and hence pressure drop, and to determine whether the flow segment is treated as subcooled, superheated or saturated. Then pipe wall temperature is solved iteratively using internal and external heat transfer. Finally, a two-variable Newton iteration inside the freesteam library code is used to find the remaining fluid properties from the calculated values of pressure and specific internal energy. The dependencies that necessitate this approach are shown in Figure 5.5.

In cases where there is a need to specify some downstream flow properties, the shooting method has proven satisfactory, so there has been no need to use any special boundary-value problem techniques.

One side-effect of this fully explicit integration technique is that there can be unexpectedly high heat transfer in the flow segment where superheated flow is first seen. This is due to the heat transfer coefficient being calculated based on two-phase conditions. This is likely to be a minor problem as the number of nodes increases, but deserves some attention when looking at mesh sensitivity.

5.2 Implementation

Using the above, it is possible, given the steam state at the inlet to a section of pipe, to calculate the outlet state. First, pressure drop can be calculated, since there is no need to know the temperature of the pipe wall. Next, the heat transfer coefficient can be calculated by iteratively solving the internal and external heat balance, by adjusting the pipe wall temperature. Finally, knowing the heat transfer coefficient
Figure 5.5: Calculation dependencies for the steady-state DSG model. Nodes requiring iteration are shown in blue.
and the pressure drop, we can use these along with the energy and momentum conservation equations to determine the steam state at the outlet of the flow segment.

5.2.1 Steam properties correlations

The present work uses the complete IAPWS-IF97 correlation published by the International Association for the Properties of Water and Steam, which is an improvement on the simple linearly-interpolated and bespoke correlations used by other workers. C++ code for this was written as a part of the present work, and these steam property routines have been made available as the free open-source software project freesteam. This software implements the simpler IAPWS-95 correlation as well as the IAPWS-IF97 correlation, and a fairly complete test suite ensures the the computed values agree with the test-case values published in the IAPWS releases, and also ensures that values from the two correlations agree to within the tolerances as published. The freesteam code includes two-variable Newton solvers to allow the IAPWS-IF97 correlation to be reliably back-solved in terms of non-correlated property pairs including the pairs \((u, v)\) and \((p, h)\) that are required in the present application.

The freesteam library also implements a method for checking dimensionality and units-of-measurement in C++ code using the method of overloaded C++ arithmetic operators combined with C++ template parameters (compile-time) to express the dimensionality of a value [139]. This eliminates the risk of errors arising from incorrect units of measurement and also works to reduce the risk of other coding errors by ensuring that values of incompatible dimensions are not assigned, added or subtracted.

In addition, the library provides a C++ implementation of the single-variable Brent solver as described in Press et al [130].

5.2.2 Software design

5.2.2.1 Data structures

Figure 5.6 shows a collaboration diagram for classes used to represent DSG pipe flow in the present work. This architecture has evolved from the original C++ architecture developed by Reynolds, but generalised to allow interconnection with other types of equipment, and to allow flexible assignment of different types of heat loss models for the pipe exterior.

Pipe length, size, roughness, density, conductivity, etc are kept as properties of the Pipe object, which also contains a vector of PipeNode objects. A PipeCalculator object is used to iterate through the pipe nodes, and depending on local flow conditions one of two types of PipeCalculatorState are used: PipeCalculatorSat for
saturated flow, or PipeCalculatorSubSup for superheated and subcooled flow.

A PipeNode object inherits from the SteamPort class, which stores just a steam state\footnote{in the form of a SteamCalculator object from the freesteam http://freesteam.sourceforge.net codebase} and mass flow rate. PipeNode objects additionally store data on the pipe wall temperature.

A PipeExterior object (Figure 5.7) is also referenced from the Pipe object. This contains the details of the external heat transfer correlation and external conditions such as ambient temperature; only one PipeExterior is present for each pipe, meaning that if varying external heat transfer correlations must be used along the pipe, then separate (connected) Pipe objects must be used. As shown in the figure, subclasses of PipeExterior developed here include classes for evacuated tubes (implemented according to Odeh [115]), the heat loss model of the present work (see Chapter 4) and a simple external loss (which uses the relation $Q = hA\Delta T$). For the case of the AbsorberPye and EvacuatedTube pipe exteriors, incident radiation and concentration ratio values are also stored.

Pipe objects inherit from the SteamEquipment class, and PipeNode inherits from SteamPort, which allows Pipe objects to be connected in series with other items (pumps, throttles, heat exchangers) inheriting from SteamEquipment, with common SteamPort objects shared by the inlet and outlet of adjacent equipment items. These SteamEquipment objects together are assigned a parent SteamSystem object, which coordinates the solving process for all the SteamEquipment objects it contains.

The DesignByContract class is a base class providing some error checking func-

Figure 5.6: Collaboration diagram for classes in the DSG software written for the present work. A purple dotted line indicates a class contained by another class. A solid blue line indicates a class from which properties and/or methods are inherited.
A private class `FlowRegimeCalculator` class used to compute flow regime by a call in the appropriate `PipeCalculatorState` object.

### 5.2.2.2 Pipe flow calculation procedures

The architecture outlined above allows a connection of pipes and other components to be specified programmatically and a sequential modular approach (Section 2.6.1.1) written to permit their steady-state solution. In the case of the pipe flow, this reduces to an explicit Euler integration. This section gives a high-level overview of the procedures used calculate steady state pipe flow. For more detail, the code documentation (source code or Doxygen output) can be consulted.

A `Pipe` is first defined (pipe diameter, density, thickness, length, segment length) and its `PipeExterior` defined (external loss model, ambient temperature) and assigned to it using `Pipe::setPipeExterior`. The inlet flow conditions are specified using by accessing methods of the `SteamPort` object at `Pipe::inletState`, and the pipe is connected to downstream equipment by specifying its `Pipe::outletState`. The `Pipe::createSegments` method is called, which fills the `Pipe` nodes vector with copies of the `Pipe::inletState`. With the pipe properties, internal nodes, and external loss properties specified, the `Pipe::solveSteady` method can be called. For each node along the pipe in sequence, excluding the inlet node, this performs a call to `PipeCalculator::calculateSteady`.

In the `PipeCalculator::calculateSteady` method, pipe flow-rate is set equal to the upstream value, and depending on the upstream node’s steam state, either a saturated or subcooled/superheated `PipeCalculatorState` is activated. A call to `prepareCalculationData` pre-calculate any values that will not change during iterative solving of the pipe node. Then, since our pressure drop models don’t require knowing the heat transfer coefficient, a call to `getPressureDrop` is used to compute $dp$. With this value, the heat transfer can then be calculated\(^9\) via a call

---

\(^9\)Note that internal heat transfer using the Gnielinski correlation requires a value for the friction
to \texttt{getHeatRate} to give a value for \texttt{du}. With \texttt{dp} and \texttt{du} these differences can be added to the \texttt{p} and \texttt{u} values for the upstream node, and a method from the steam properties library used to compute the new steam state.

In the case of \texttt{PipeCalculatorSat::getHeatRate}, which implements the heat transfer calculation described in Section 5.1.3.2 and Section 5.1.3.3, the pipe wall temperature is calculated by solving with the Brent algorithm for the pipe inner wall temperature $T_1$. At each iteration both the external and then the internal heat transfer are solved, and the error used as the Newton residual. The value of the external heat transfer is passed to the internal heat transfer method (\texttt{getInternalHeatRate}), since a value for the wall heat flux is required for calculation of the boiling number, Bo.

In the case of \texttt{PipeCalculatorSubSup::getHeatRate}, for which the Gnielinski correlation is used, the internal heat transfer coefficient can be calculated without any knowledge of the pipe wall temperature, so this is first calculated, and iterative solution for the pipe wall temperature thus simplified a little.

Once the state of the current pipe node has been computed and stored in the \texttt{PipeNode} object, the flow regime there is also calculated via a call to \texttt{getFlowRegime}.

5.2.2.3 Scripting language wrapper

The above C++ classes and methods are wrapped using SWIG (Simple Wrapper Interface Generator) [13] to facilitate higher-level modelling using the Python scripting language [89]. Both \texttt{freesteam} and the DSG classes were wrapped in this way, with the result of a simple MATLAB-like interactive interface for the calculation of two phase pipe flow problems. Through the wrapper interface, a set of test routines was also established (Section 5.3.1) to ensure that the various classes were functioning as expected.

5.2.3 Important differences from previous work

Pressure drops. The Friedel pressure drop correlation (as given by Behnia [14]) has been preferred to the Martinelli-Nelson correlation [94]. The Friedel correlation was used by Rheinländer in his DISS modelling and agreed well with experiment. Friedel’s correlation was also based on a larger bank of experimental data than that used by Martinelli and Nelson.

An error in curve-fits to the Martinelli-Nelson correlation was found in the code used by Odeh [115], indicating that his pressure drops may actually be underestimated for pressures between 20 and 100 bar. Reynolds’ work [142] for validating the pressure drop model also contained these errors, which casts some doubt on the as-
sertion that the pressure drop model used there was a better match to experimental results than the Rheinländer model. Further details of the error found are given in Appendix B.

Reynolds work [142] for validating the Rheinländer pressure drop [144] also did not include K-factor losses.

**Heat transfer** The new Kandlikar correlation (as given by Lienhard & Lienhard [84]) is being used here in place of the approach based on Stephan [156] and Gungor and Winterton [55] used by Odeh and continued by Reynolds. The reason for the change was primarily that this approach was quite complicated, and it was very difficult to identify a coherent approach from these sources. Newer correlations have appeared and seem to have much reduced errors when validated by experiment, for example by Liu and Winterton [86]. In any case, it is clear that the resistance to heat transfer will be much greater outside the pipes than inside, so it makes sense to choose a simple correlation as is the case with the Kandlikar correlation.

**Condensation correlations.** The code written for the above flow boiling case has been tied together with heat transfer correlations for condensation, and these correlations are switched automatically when saturated heating is replaced by cooling. This allows estimation of heat losses in connecting and reticulating pipework, which was beyond the scope of the validation performed by Reynolds.

### 5.3 Validation

This section details the efforts made to ensure the accuracy of the absorber pipe flow model, including low-level ‘unit tests’ as well as checks of intermediate and high-level results against published and experimental data.

#### 5.3.1 Code checking

A range of test-cases has been implemented to ensure that the code routines behave as expected. These test-cases are presented as an integral part of the codebase and can quickly be run *en masse*\(^\text{10}\) before any ‘real’ analysis proceeds. These tests include:

- ensuring that steam properties match known values from published tables,
- checking that units of measurement conversions given the expected values,

\(^{10}\text{Using the command:}\)

```
cd dsg-transient/python && scons -U && python test.py
```
• testing that data is passed into and out of data structures as expected,
• testing that points on the Taitel, Barnea and Dukler flow map give the expected flow regime values,
• testing single-phase friction factor values against values in the laminar, turbulent-rough and turbulent-smooth parts of the Moody diagram,
• testing that Martinelli-Nelson pressure drop multipliers agree with several values from published table of values
• testing the Friedel value for $\phi^2$ agrees with a sample calculation by Behnia ([14], p. 82 ff)
• testing that Friedel values for $\phi^2$ are within 50% of Martinelli-Nelson values, for ranges of pressures, diameters, qualities and mass flow rates in the region of interest.

5.3.2 Comparison of Martinelli-Nelson and Friedel pressure drops

In the course of the code checking, as noted above, it was seen that in some cases there was a significant discrepancy between pressure drops calculated using the models of Friedel and that calculated using the model of Martinelli-Nelson. The errors can be seen to be greatest at lower pressures and at high values for steam quality. In this section the discrepancies are examined with a view to understanding how the use of the Friedel correlation may affect computational results compared with results obtained with the Martinelli-Nelson correlation as used by earlier workers. It is found that Friedel friction factors vary by up to $\pm 40\%$ from Martinelli-Nelson values, so this work investigates where the differences occur.

5.3.2.1 Comparison map at $D = 50$ mm

We will compare the pressure drop given by both the Friedel correlation and the Martinelli-Nelson [94] correlations for a range of pressures, steam qualities, pipe diameter and mass flow rates.

By plotting graphs showing the magnitude of the Friedel pressure drop gradient relative to the Martinelli-Nelson pressure drop gradient for a range of inlet pressures and steam qualities, we will be able to approximately see what overall discrepancy can be expected in a real situation in which the quantities are varying along the length of the pipe.

Figure 5.8 shows such a plot for the case of a 50mm pipe with 0.5 kg/s flow of saturated water and steam\textsuperscript{11}. A discrepancy in pressure drop gradient of -30 Pa/m

\textsuperscript{11}The script friedelvsmartnels.py was used to generate these plots.
Figure 5.8: Contours of discrepancy in pressure drop gradient (in Pa/m) for a pipe with 50mm internal diameter and 0.5 kg/s flow rate. The contour values are for the Friedel correlation relative to the Martinelli-Nelson correlation, so a value of -500 Pa/m means that the Friedel correlation returns a pressure drop gradient that is 500 Pa/m smaller than the value given by the Martinelli-Nelson correlation.

would correspond to a pressure drop of 18000 Pa or 0.18 bar over a length of 600m.

We see that quite high discrepancies occur at low pressures, in which case the Friedel correlation gives smaller values of the pressure drop gradient, and high qualities, in which case the Friedel correlation gives larger values of the pressure drop gradient.

Figure 5.9 shows a similar graph for the greater mass flow rate of 1.0 kg/s. Here we see that the discrepancy in pressure drop gradient has increased significantly in magnitude.

Consider the cases of a 300 m length of pipe, in which force convection boiling at 50 bar takes place from $x = 0.3$ to $x = 0.7$. We would expect the Friedel correlation to give a pressure drop at least 0.6 bar smaller than the value given from the Martinelli-Nelson correlation. This lower pressure drop estimate would be balanced somewhat by the higher estimate in the $x > 0.9$ range.

5.3.2.2 Comparison map at $D = 25$ mm

Figure 5.10 shows the above graph generated for a small pipe and smaller flow rate. Although the mass flux is equal, the magnitude of the discrepancy in pressure gradient is higher here (note the contour scale on the graph). The majority of pressure and quality positions again given lower estimates of the pressure drop from the Friedel correlation compared with the Martinelli-Nelson correlation.
5.3. VALIDATION

Figure 5.9: Errors in pressure drop gradient (in Pa/m) for a 50mm pipe (as per Figure 5.8) but with a higher mass flow rate of 1 kg/s.

Figure 5.10: Discrepancies in pressure drop gradient (in Pa/m) for a 25mm pipe (as per Figure 5.8), with flow rate of 0.25 kg/s. This flow rate corresponds to the same value of mass flux as was used in 5.9.
5.3.3 Flow regime maps

Some studies were performed to assess the way that the flow regime maps change with pressure and pipe diameter, to aid in an understanding of the effects of these variables on the flow regime transitions through the CLFR absorber. Since the flow regime map computation is dependent only on fluid state\textsuperscript{12}, mass flow rate and pipe diameter, being able to reproduce a complete flow regime map constitutes a good check on the code implementation.

The flow regime map shown in Section 5.2 (Figure 5.2) compares well with the map by Reynolds [142] (page 136).

Effect of pressure on flow map: The position of the flow regime boundaries in the Taitel et al flow maps is determined by two variables: the pressure and the pipe diameter. We can observe (Figure 5.11) that the effect of varying the pressure from 20 bar to 80 bar for a 21mm inside diameter pipe is to move the flow regime boundaries towards the left (transitions occur at a lower superficial gas velocity).

Effect of pipe diameter: Pipe diameter effects the flow regime boundaries in a different way (Figure 5.12). The boundary of the dispersed bubble and stratified regimes move up the liquid axis as the diameter increase.

Flow map ‘grid’: In order to ensure that the flow regime code functions correctly over the full range of pressures, a ‘grid’ of flow maps was created showing the variation in flow map behaviour over a wide range (Figure 5.13). Diameters from 5mm to 100mm were used (down the grid), and pressures from 2 bar to 100 bar (across the grid). Other than the irregularity in the stratified-annular boundary, the flow map appears to behave consistently with data from Odeh [115] and Reynolds [142]. We assume that pipe diameters of 5mm diameter do not fall within the scope of the correlations used.

5.3.4 Validation of pressure drops using PSA DISS experimental data

In order to validate the pressure drop model, we can use the pressure drop results from the DISS project at the Plataforma Solar de Almería. A technical report given to the German DLR by Rheinländer et al [144] details some limited experimental data, and full details of 12 different experimental tests are available in the spreadsheet [143] that was developed with that work. This appears to be the only available

\textsuperscript{12}we are assuming homogeneous flow, so the two-phase fluid state can be represented by two properties in all cases, for example \((p, x)\) or \((u, v)\), etc.
Figure 5.11: Effect of inlet pressure on the location of flow regime boundaries, for horizontal flow in a 21-mm-internal-diameter pipe, with inlet pressure of 20 bar, 50 bar and 80 bar, as labelled.
Figure 5.12: Effect of pipe diameter of flow regime boundaries, for horizontal flow in a pipe with inlet pressure of 50 bar. Positions of flow regime boundaries for DN20, DN25, DN40 and DN65 pipes (SCH40S) are shown. As the pipe diameter increases, the most notable difference is that boundary of the stratified flow regime moves upward and to the right; this implies that stratified flow continues at higher flow rates in large pipes.
Figure 5.13: A matrix of flow regime maps for horizontal pipes, evaluated according to the method of Barnea[11]. Pressures in the range 2 - 100 bar are shown across the matrix, and diameters in the range 5 - 100 mm are shown down the matrix. For each flow map within the figure, $U_{LS}$ values from 0.001 to 10 m/s are shown on the vertical (logarithmic) scale, and $U_{GS}$ values from 0.001 to 100 m/s are shown on the horizontal (logarithmic) scale. Some irregularities are noted in the case of 5 mm pipe.
data from a large scale study that provides data on long-pipe pressure drops for two-
phase boiling. The data has been reproduced for reference purposes as Appendix E.

The spreadsheet gives steady-state experimental pressure-drop data for 12 different test runs with the Direct Solar Steam (DISS) prototype system at the Plataforma Solar de Almería. It accompanies this experimental data with modelled pressure drop data (calculated using a program called IPSEpro) for each case. The recorded data includes inlet and outlet enthalpy (in the spreadsheet), system geometry (given in the report), and pressures at the inlet and outlet of each absorber section (given in the spreadsheet).

Figure 5.14 shows the results of a modelling case (‘experiment 7’) which aims to reproduce the results of the 60 bar once-through case in the Rheinländer report. Although the results are qualitatively similar, the overall pressure drop shown is significantly greater.

The data from Rheinländer does not provide sufficient breakdown of the absorber heat loss characteristics to allow this aspect of the present model to be validated. In order to make a same-with-same comparison, the heat supplied to the present model was adjusted in each case such that the overall enthalpy rises matched with the enthalpy data calculated by Rheinländer. This enables the pressure drops to be compared for equivalent overall heating.

The computed results here seem to track those of Rheinländer quite closely. This validation is not a substitute for experimental measurement but it confirms the validity of the computational model to the extent possible until data from the CLFR is available.

5.3.4.1 Effect of Friedel versus Martinelli-Nelson pressure drop correlation

For comparison, the results plotted in Figure 5.14 were re-computed with the Martinelli-Nelson correlation used in place of the Friedel correlation\(^\text{13}\). The new results are show in Figure 5.15. As expected from Section 5.3.2, we see smaller overall pressure drops predicted by the Friedel results.

5.3.4.2 Full validation using all of Rheinländer’s data

Figure 5.16 shows the results of the above simulation reproduced for all 12 of the sets of experimental data from the Rheinländer report. The pressure drop values are shown to track both the experimental data and the Rheinländer model results closely.

\(^{13}\text{See notes in Section 5.2.3}\)
Figure 5.14: Graph showing the Rheinländer once-through 60 bar results compared to a comparison case computed here. Pipes in the present work have been assumed horizontal (in practice they are inclined at 4 degrees). Approximate values of K-factor losses have been used. The optical characteristics were not fully specified, so the optical efficiency was adjusted to give the same overall enthalpy change as stated in Rheinländer’s spreadsheet.
Figure 5.15: A repeat of Figure 5.14 but with the ‘present work’ curve calculated using the Martinelli-Nelson correlation for two-phase pressure drops.
5.3. VALIDATION

From these results, we conclude that the Friedel correlation is suitable for the modelling of the long-run pipes in question here. This data only applies for a single value of pipe diameter, however from the large data bank that Friedel used to create his correlation we can have some confidence that the agreement with experimental values here should extrapolate to the smaller diameters likely to be used in CLFR systems.

5.3.5 Validation of internal heat transfer

Due to the much greater magnitude of resistance to heat transfer outside the pipe, it is impossible to validate the internal heat transfer from the results of Rheinländer.

5.3.6 Validation of flow regime transitions

It is not possible to validate the flow regime transitions of the current model other than to note that the flow regime map above agrees well with other published flow regime maps based on the Barnea et al. The flow maps are also consistent with those shown by Reynolds. The experimental results published by Rheinländer do not include information relating to flow regimes, and to the author’s knowledge, this has not been a feature of the analysis performed by Eck and his colleagues.

5.3.7 External heat transfer

The external heat transfer correlation can not be validated until experimental data have been gathered. The cavity geometry that is scheduled for construction is now the angled glass cavity cover. This cavity cover was chosen after the earlier heat loss correlations had been completed. It is proposed that this work be revisited when experimental data becomes available.

5.3.8 Comparison with results of Reynolds

The above validation was also performed by Reynolds (using the Martinelli-Nelson correlation), but the results showed significantly overestimated pressure drops. This is thought to be due primarily to the fact that the Martinelli-Nelson correlation predicts larger pressure drops, as was shown in Section 5.3.2. The faulty correlation equations inherited by Reynolds from Odeh appears likely to have reduced the over-estimation (Appendix B). Because of this deviation caused by the incorrect pressure drop correlation, it is not possible to accurately verify the effect of simplifications made by Reynolds such as neglecting the effects of minor losses and heat loss in connecting pipe sections.
Figure 5.16: Results of DISS pressure drop validation modelling for all of the experimental data provided by Rheinländer (legend as for Figure 5.15).
5.3. VALIDATION

Figure 5.17: Results of a simulation using the present model to demonstrate the presence of the Ledinegg instability. Inlet pressure is 100 bar.

5.3.9 Ledinegg Instability

The Ledinegg instability (see Tong [164] p 460) refers to an adverse pressure gradient effect where increasing the flow rate has the effect of decreasing the pressure drop across the collector. As part of model validation, it is important that we are able to reproduce this effect in calculations involving two-phase flow in the absorber, as it is a major source of likely instability and uncontrollability in the DSG process.

The Ledinegg instability is illustrated in the present case by plotting pressure drop versus inlet mass flow rate, for a range of inlet temperatures. The important factor is the degree to which the inlet fluid is subcooled.

A simulation was performed using a single-run 500m pipe (no bends or valves) with a pipe with 50 mm ID, 70 mm OD, and LS-2 heat loss parameters, as specified in the report by Rheinländer [144]. An optical efficiency of 0.5 was used to compute the radiation received at the absorber surface. The overall curves for inlet subcooling of 50, 100 and 150°C are shown in Figure 5.17.

These results agree well in a qualitative sense with the results of Rheinländer, and with the partial information available on the simulation case, this appears to validate our ability to simulate the Ledinegg phenomenon.
The Ledinegg instability arises due to the movement of the subcooled, two-phase, and superheated regions moving through the pipe. When the outlet is saturated, and the flow rate is increasing, the specific enthalpy change is reducing, and so the part of the pipe subject to enhanced two-phase pressure drops is also shrinking, with the result of an overall decrease in pressure drop. The consequence of this instability is that conventional feedback control cannot be used in this mode to regulate the outlet pressure. If a pump is to be used to regulate the flow such that a desired outlet pressure is achieved, then the controller needs additional signals other than outlet pressure in order for it to operate deterministically.

Quantitative differences between the present results and those of Rheinländer appear most significant in the high flow rate part of the graphs. They can be explained in the following ways:

- an estimate of optical efficiency for the collector was required, and this value was not given in the Rheinländer, but had to be estimated using the overall efficiency (including losses).
- no information on the pipe roughness was available. Differences in pipe roughness could have a significant effect in the high flow rate cases.

5.3.9.1 Effect of bends and valves on the Ledinegg instability

Tong notes that the Ledinegg instability may be mitigated by adding throttling at the inlet to the two-phase flow section of pipe, as has been used in heat exchangers. This effect will be discussed in Chapter 6.

5.3.10 Mesh sensitivity

The segment length in the DISS validation model was varied between 0.05 m and 1.0 m without any appreciable change in the results. It is anticipated that mesh effects will become more significant in the case of transient modelling, where moving dry-out boundaries will cause rapid fluctuations in local heat transfer coefficient.

5.4 Summary

A new computer model for steady two-phase pipe flow was developed that includes the Friedel pressure drop correlation and the Kandlikar internal heat transfer coefficient correlation. The new model was validated against results from Rheinländer, 2002 [144], and was found to be in good agreement. The component parts of the computer code were also tested individually with a range of further validity checks and appear correct against available data.
The model also corrects some errors in estimated pressure drops found in the earlier modelling work of both Odeh and Reynolds. It was demonstrated that the Martinelli-Nelson correlation over-predicts pressure drops in this case, especially if minor losses are included.

The computer model was shown to predict the occurrence of the Ledinegg instability in long pipe flow, as anticipated. Importantly, this instability means that conventional feedback control can not be applied to the CLFR absorber, and steps are required to either modify the physical system to eliminate the instability, or else to find an advanced control system for the pressure/flow-rate response that can make the instability in some way controllable.

The flow regime map of Taitel, Dukler and Barnea was implemented and the results validated against those from Reynolds. It is noted that the model of flow regimes does not affect pressure drop or heat transfer in the present computer model; flow regimes can therefore be computed in post-processing if desired.

Although the present model requires experimental validation that will be possible at the completion of the Stage 2 prototype, the model now appears to be the most rigorous two-phase flow model that has been applied to long-pipe flow-boiling problems. It includes industry-standard steam properties, a more accurate pressure drop correlation, and prediction of flow regimes present in the flow. To the author’s knowledge, no other modelling of direct steam generation has combined all of these elements.

The present model is suitable for steady-state modelling but as discussed it requires some structural adaption before it can be applied to transient modelling. In later chapters, the code from the present chapter is used as a foundation for a modified approach that deals with the transient case.
Chapter 6

Thermohydraulic analysis of the CLFR absorber

The previous chapter gave an approach for computing steady-state flow in the direct steam generation pipe flow, including friction factor correlation, internal heat transfer correlation, conservation equations and an integration scheme, and provided some validation of that approach. In this chapter, we apply that approach specifically to the CLFR design problem, with a view to understanding how it might be optimised and controlled.

6.1 CLFR base-case configuration

We will concentrate on the design configuration of the Stage 2 CLFR prototype (Figure 6.2), which is modified from the earlier Stage 1 prototype in a number of ways. Full details of the Stage 2 CLFR configuration are given in Appendix C. Some of the key parameters are reproduced here:

- Absorber is 310 m long and contains 12 parallel pipes of size DN 32 and schedule 10S in stainless steel of grade 304. The upper surface of the cavity is 575 mm wide and the cavity is 200 mm deep.

- The cavity is covered with low-iron anti-reflection-coated glass.

- Concentration ratio is 27 at absorber plane. Optical losses, such as mirror reflectivity, spillage and cavity cover reflectivity, are assumed to have already been allowed for in this value.

- A single DN 150 schedule 10S pipe in 304 stainless is used to reticulate water to the far end of the array, where it enters the absorber and returns to the near end.
A steam drum is mounted up in the air, above the level of the absorber.

For ambient conditions, a temperature of 300 K is assumed. For a base case, external convection coefficient on the cavity cover is taken as 20 W/m²K. Base-case solar irradiation is taken as 1000 W/m².

For the radiative losses, the ‘sky’ temperature is taken as equal to the ambient temperature, because effective temperature of the surfaces in view of the absorber are likely to be close to ambient, rather than cold like the sky. The radiative losses for the cavity are calculated from the flat-plate absorber design, which as discussed in Chapter 4, introduces an error in the cavity heat loss of the order of 25% compared with a tube-bank absorber design.

For the present analysis, we are concerned with a single path through the absorber. In order to perform this in a way that will scale correctly when all pipes in the absorber are considered, we divide the radiation received at the absorber plane by the number of pipes in the absorber, ignoring the fact that the pipes do not actually completely cover the width of the absorber plane (Figure 6.1). This fact will need to be included in any optical efficiency value that precedes this analysis. With this approach, we assume that both incident radiation and thermal losses are uniformly distributed on the absorber plane.

### 6.1.1 Alternative configuration

Subsequent designs are planned to dispense with the reticulating pipe and instead to make two passes of the absorber (‘there and back’) for a total of 620 m of absorber pipe. This alternative scenario will be included in the analysis by simple extending the flow path considered. No consideration is given to minor losses in the bend or manifold at the far end of the absorber.

### 6.2 Flow behaviour at the expected operating point

At the expected operating point, it is intended that the absorber should provide steam at an outlet quality of 0.8. We should estimate what flow rates might be

\[\text{In the absence of experimental measurements to confirm the assumption of Section 4.3.3, this more conservative assumption was instead used.}\]
Figure 6.2: Configuration of the Stage 2 CLFR prototype.
required to achieve this, and estimate what the pressure drops through the collector will be in this case. We should also observe what the expected flow regimes throughout the length of the pipe will be in this case. The variable that we do not have any control over is the solar irradiation, so we will plot against that as our independent variable.

Figure 6.3 shows the results of simulations for the CLFR Stage 2 absorber in the single-pass and dual-pass configuration, with fixed inlet conditions and flow rate varied to give steam quality of 0.8 for each case. The curves are all much as might be expected, with the exception of the average absorber temperature, which is seen to vary non-linearly with solar radiation.

Mass flow rate here is proportional to radiation level, due to the fact that the required specific enthalpy is constant so only the flow-rate must be varied. The changes in two-phase flow heat transfer behaviour appear to be second order as flow rate varies with exit steam quality held constant.

Heat loss is linear with radiation level, but not proportional. The important factor here is that the heat loss mechanisms are determined only by the absorber temperature, not by the solar irradiation level. The absorber temperature is in turn determined by the internal heat transfer, which in the saturation region, where temperatures are constant, is mostly only being affected by changing mass flow rate. So small linear changes in the absorber temperature are causing the relatively small linear changes in heat loss. In fact the heat loss is radiation dominated and varies with the fourth power of temperature; however here the changes in absorber temperature are small, so we see only a linear effect.

6.2.1 Variation in average absorber temperature

In order to examine the variation in average absorber temperature seen in the previous section, we will study the case of the 310 m absorber in more detail, and extract the location of the phase transitions and flow regime boundaries. Using the model, it is seen that flow regimes are changing, and specifically, in the $I=100$ W/m$^2$ case, laminar flow is occurring in the subcooled region, owing to the very low flow rate of 11.3 g/s.

Full details are show in Table 6.1. It is seen that the annular flow regime occurs at the exit for higher solar radiation levels, but that at lower radiation, stratified flow regimes can be expected at the exit. Also, at $I=100$ W/m$^2$, the transition to two phase flow occurs earlier in the pipe, which will result in the higher predicted average absorber temperature.
Figure 6.3: Steady-state absorber simulation results for the CLFR Stage 2 prototype, showing single-pass (310 m) and dual-pass (620 m) configurations, with fixed inlet conditions of 42 bar, 150 °C, and flow rate adjusted to provide constant outlet steam quality of 0.8 at varying solar input.
Table 6.1: Locations of flow regime transitions for the 310 m CLFR Stage 2 prototype with flow rate varied to keep exit steam quality at 0.8. ‘turbulent’ and ‘laminar’ refer to subcooled single-phase flow in this table.

<table>
<thead>
<tr>
<th>Solar irradiation (W/m²)</th>
<th>Inlet turbulent</th>
<th>Slug stratified smooth wavy annular</th>
<th>Exit</th>
</tr>
</thead>
<tbody>
<tr>
<td>1000</td>
<td>turbulent</td>
<td>86</td>
<td>112  annular</td>
</tr>
<tr>
<td>800</td>
<td>turbulent</td>
<td>86</td>
<td>112  annular</td>
</tr>
<tr>
<td>600</td>
<td>turbulent</td>
<td>85</td>
<td>91   162 annular</td>
</tr>
<tr>
<td>400</td>
<td>turbulent</td>
<td>84</td>
<td>85   95 stratified wavy</td>
</tr>
<tr>
<td>200</td>
<td>turbulent</td>
<td>81</td>
<td>140  stratified wavy</td>
</tr>
<tr>
<td>100</td>
<td>laminar</td>
<td>34</td>
<td>72   stratified smooth</td>
</tr>
</tbody>
</table>

6.3 Fixed outlet pressure

The above analysis was performed by using a fixed inlet pressure, and allowing the outlet pressure to float. However the configuration of the CLFR Stage 2 prototype is likely to be such that the outlet pressure would be fixed due to way the collector is connected to the power station. In this section, we reproduce the above analysis for the case where the exit quality is maintained at 0.8 and the exit pressure is maintained at 42 bar, and the inlet temperature is 150 °C.

Results of analysis of the single-pass collector using a fixed outlet pressure of 42 bar is shown in Figure 6.4. As expected the behaviour is much the same as for the fixed inlet pressure case.

6.3.1 Inlet temperature

The assumption that the inlet temperature to the collector is 150 °C (= 423 K) needs to be examined. In fact this temperature is the temperature of the feedwater supplied to the steam drum. During operation, saturated steam will be returned to the steam drum at approximately the saturation temperature at 42 bar, which is 526 K (= 253 °C). Depending on the ratio of absorber and feedwater flow-rates, the temperature of the drum will therefore be in the range of 423 K (= 150°C) and 526 K (= 253 °C). The result of this will be a smaller region of subcooled flow in the absorber, which will affect some of the relationships. We will investigate the case of fixed outlet pressure, fixed outlet quality, varying inlet temperature, varying solar radiation, for the single-pass configuration.

Figure 6.5(a) shows the effect of varying inlet temperature for the fixed outlet pressure and fixed outlet quality scenario. The primary cause for this is that higher inlet temperature means lower specific energy change through the collector, which
6.3. **FIXED OUTLET PRESSURE**

Figure 6.4: Pressure drop and total heat loss for the CLFR Stage 2 prototype of length (single-pass configuration), with the exit pressure held at 42 bar and the flow rate adjusted to give an exit steam quality of 0.8
means a higher flow rate to achieve the same outlet quality. A secondary effect would be that as inlet temperature rises, the two phase flow is present in more of the flow path, and two phase flow has higher pressure drops than liquid-only flow, causing pressure drop to be further increased slightly.

Figure 6.5(b) shows the the absorber efficiency for these cases, which is defined here\(^2\) as

\[
\eta_{\text{absorber}} = \frac{\dot{m}(h_{\text{outlet}} - h_{\text{inlet}})}{Q_{\text{incident}}}
\]

The first comment with regard to the absorber efficiency is that it rises significantly as the solar radiation increases. The reason for this is that, with the outlet pressure at 42 bar and the outlet quality at 0.8, most of the flow is at saturation temperate for all cases, but the heat flux is much lower for low solar radiation. Thermal losses meanwhile are roughly constant; they depend on the temperature only. Hence, higher heat fluxes give higher efficiencies, so long as we stay in the saturation region at a fixed operating pressure.

We see that the absorber efficiency is highest when the inlet temperature is 526 K, which is the saturation temperature at 42 bar. This can be explained by the fact that in two phase flow, the pipe internal heat transfer coefficient is significantly higher than for two phase flow. As the inlet temperature rises, the part of the pipe containing single phase flow becomes smaller, and it is in that single-phase region that losses are a higher fraction of the incident radiation.

A conclusion here is that we would like to run the entire CLFR collector in the two-phase region if possible, as it will ensure that the absorber temperature, the main driver of thermal losses, will be as low as possible for a given desired fluid temperature.

### 6.3.2 Outlet quality

Up to now, we have assumed an outlet quality of 0.8 for all cases. The motivation for this is that we want to ensure that we can avoid entering the superheated region anywhere in the CLFR absorber, as has been earlier discussed. The value of 0.8 allows a margin of error such that the automated controller should be able to keep the steam safely out of the superheated region. Too low a value of steam quality would mean higher flow rates would be required, and this would increase the required pumping power. There is a significant mass holdup in the collector, which means that quite elaborate control strategies may be necessary in order to correctly anticipate

\(^2\)Reynolds [142] and Odeh [115] defined an absorber efficiency that included ‘losses’ for pumping power as well. For the moment, we will ignore those effects and instead give a fuller treatment of the whole-system efficiency in Chapter 8.
Figure 6.5: (a) Pressure drop and (b) absorber efficiency for the CLFR Stage 2 prototype with fixed outlet pressure 42 bar and fixed outlet steam quality 0.8, plotted against varying solar irradiation (x-axis) and inlet temperature (legend).
CHAPTER 6. THERMOHYDRAULIC ANALYSIS OF THE CLFR ABSORBER

Figure 6.6: Mass holdup versus solar irradiation for three values of exit quality, with outlet pressure fixed at 42 bar, and inlet temperature set at 150 °C.

‘lag’ effects and to avoid controller overshoot (as noted for example by Valenzuela et al, 2003 and 2003 [168, 167]).

Here we will consider the effect of increasing the outlet quality set-point to 0.90 and 0.95. The results of the analysis are given in terms of absorber efficiency (Table 6.2) and pressure drop (Table 6.3).

The changes as expected are small; there is a slight increase in the collector efficiency and a slight reduction in the pressure drop. The reduction in pressure drop is a little unintuitive so is investigated in 6.6.

With both of these factors being favourable, it would appear that a good plant design will involve taking the exit steam quality as high as possible without risking entry into the superheated region. Additional controls, such as injection of water near the pipe end could be used to improve controllability, as was done for the DISS project at Plataforma Solar de Almería [38]. These considerations will be dealt with in Chapter 8.

It interesting to note (Figure 6.6) that the increase in exit quality has a fairly significant effect on the mass holdup in the collector. We see that for full radiation, the mass of water in the collector during operation decreases from 100.1 kg to 89.1 kg. The effect would be even more observed if the inlet temperature were raised closer to saturation, due to the length of pipe in which flow is subcooled.
6.4. V ARIED OUTLET PRESSURE

Table 6.2: Absorber efficiency for the CLFR Stage 2 prototype, for varying outlet quality, at fixed outlet pressure 42 bar and fixed inlet temperature 150 °C

<table>
<thead>
<tr>
<th>Solar irradiation</th>
<th>x=0.8</th>
<th>x=0.9</th>
<th>x=0.95</th>
</tr>
</thead>
<tbody>
<tr>
<td>W/m²</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>100</td>
<td>0.5165</td>
<td>0.5144</td>
<td>0.5135</td>
</tr>
<tr>
<td>300</td>
<td>0.8162</td>
<td>0.8172</td>
<td>0.8176</td>
</tr>
<tr>
<td>600</td>
<td>0.8890</td>
<td>0.8907</td>
<td>0.8915</td>
</tr>
<tr>
<td>1000</td>
<td>0.9179</td>
<td>0.9200</td>
<td>0.9210</td>
</tr>
</tbody>
</table>

Table 6.3: Pressure drop for the CLFR Stage 2 prototype, for varying outlet quality, at fixed outlet pressure 42 bar and fixed inlet temperature 150 °C

<table>
<thead>
<tr>
<th>Solar irradiation</th>
<th>x=0.8</th>
<th>x=0.9</th>
<th>x=0.95</th>
</tr>
</thead>
<tbody>
<tr>
<td>W/m²</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>100</td>
<td>-0.0058</td>
<td>-0.0055</td>
<td>-0.0053</td>
</tr>
<tr>
<td>300</td>
<td>-0.0742</td>
<td>-0.0718</td>
<td>-0.0703</td>
</tr>
<tr>
<td>600</td>
<td>-0.2880</td>
<td>-0.2799</td>
<td>-0.2750</td>
</tr>
<tr>
<td>1000</td>
<td>-0.7586</td>
<td>-0.7401</td>
<td>-0.7285</td>
</tr>
</tbody>
</table>

6.3.3 Mass holdup, exit quality and inlet temperature

We will briefly examine the mass holdup in different regions of the flow for base-case operation.

In the previous section it was found that with an exit quality of 0.95, the total mass holdup is 89.1 kg. Of this, 75.5 kg is seen to be located in only the first 78 m of pipe, as subcooled liquid. If the exit quality is very high, then there will be only very little saturated water being added to the feed water, and the result will be that the inlet temperature is not much higher than the feedwater. If on the other hand, the exit steam quality is lower this will result in a higher temperature at the absorber inlet, and hence slightly higher absorber efficiency. It would appear that there may be an optimisation that can be made here, but this will need to be investigated as a closed-loop effect, in Chapter 8.

6.4 Varied outlet pressure

We saw in Section 6.3.1 that an important factor in absorber efficiency was the operating pressure. Here we will investigate how that efficiency varies with pressure. In the case of the CLFR Stage 2 prototype, the operating pressure is determined by the power station. However for other CLFR applications, this may be a free choice. We will use a range of outlet pressure, fixed outlet quality, varying solar radiation.

Figure 6.7 shows the result of this analysis. As could be expected, the increasing outlet pressure results in a reduction in absorber efficiency for fixed exit quality. The reason is that at higher pressures, the saturation temperature is higher and
6.5 Locations of flow regime boundaries

A study of the location of flow regimes in one of the earlier CLFR design concepts was performed [103], shown here as Figure 6.8. This behaviour is in qualitative agreement with Table 6.1, although this work used the Barnea et al [11] pressure drop model and the Stephan/Gungor and Winterton [156] heat transfer model, and a significantly different collector geometry.

6.6 Performance maps for the CLFR Stage 2 prototype

The above analysis has given an understanding of the sensitivity of the CLFR Stage 2 design to changes to a number of variables near its chosen operating point. In this section we will look more broadly at the full range of possible operating conditions for the single-pass CLFR collector. This will inform us on possible start-up and shut-down effects, as well as the possible benefits of changing the pressure or inlet temperature, and also ‘map out’ the extents of the saturated flow region in terms of operating parameters.
6.6. PERFORMANCE MAPS FOR THE CLFR STAGE 2 PROTOTYPE

Figure 6.8: Flow regime transitions in a 20mm direct-steam generation pipe as presented in the paper with Mills et al [103].

6.6.1 Pressure drop

Figure 6.9 shows a contour plot of absorber pressure drop for varying inlet pressure and mass flow rate. The inlet temperature is set at 150 °C in these cases, and the solar irradiation is fixed at 1000 W/m². The important observation is that the pressure drop decreases with increasing flow rate for much of the two-phase region. This is the Ledinegg instability that was observed in Section 5.3.9. We can expect that as the inlet temperature is raised, this instability will be reduced.

Another observation from Figure 6.9 is that there is a danger of intermittent flow regimes if the flow rate is too high, but that the annular flow region is adjacent to the superheated region at all pressures, so provided the steam quality is kept high, we should be able to avoid intermittent flow at the absorber outlet.

Figure 6.10 shows the effect of raising the inlet to 175 °C, 200 °C and 225 °C. The Ledinegg instability has been reduced and from 200 °C appears to have been eliminated for pressures beneath 40 bar. As a result of the higher inlet enthalpy, we see that higher flow rates are possible while still remaining in the two phase region, and we see that pressure drops have increased slightly at any chosen inlet pressure.

6.6.2 Efficiency

The absorber efficiency data from the above simulations has also been compiled as shown in Figure 6.11. These curves will be useful for predicting the steady-state
Figure 6.9: Contours of pressure drop at $T_{inlet}=150$ °C. Exit flow regime is shown by background colour. The uneven bottom part of the coloured region corresponds to the area where the pressure drop exceeded the inlet pressure.
Figure 6.10: Contours of pressure drop at higher inlet temperatures (a) 175 °C, (b) 200 °C and (c) 225 °C. Note that the glitch in the 225 °C case is due to that temperature being about the saturation temperature at the lower pressures.
thermal losses for the CLFR under full sun. Higher efficiencies at a given inlet pressure are possible as the inlet temperature is raised.

6.7 Performance maps for other collectors

6.7.1 DISS prototype

As a follow-on to the earlier validation of the present computational model, the performance maps shown in Section 6.6 have been reproduced for the DISS collector as described in Section 5.3.4. The inlet enthalpy is kept constant in all cases here (in order to avoid the discontinuity seen in Figure 6.10(c)) at a value of 1030 kJ/kg. The pipe diameter is larger in this model (50 mm) and there are a number of pipe bends and valves in the model, as well as lengths of unheated pipe connecting the different DISS collector modules.

The results of this analysis are shown in Figure 6.12.

An important difference is that there is no apparent Ledinegg instability – at all flow rates and pressures, the contours of outlet pressure are monotonically increasing as flow rate increases. There are additional plots showing the contours of outlet quality and outlet temperature, which show that in the subcooled region, outlet temperature does not vary with increasing pressure.

It is also interesting to note that there are minima in the curves of outlet quality at higher flow rates. For most of the performance map, the usual effect dominates: increased flow rate causes a decreased outlet steam quality because the specific enthalpy is smaller for the same total amount of heat absorber. For the small region where this is not the case, note that the steam quality is low. Therefore most of the flow is subcooled and only the last little portion of the pipe is saturated. In this high flow-rate, predominantly subcooled, flow, incremental increases in pressure drop with flow rate are larger than elsewhere on the map. The result is that the effect of moving downwards on the p-h diagram at constant pressure causes an increase in steam exit quality without any change in specific enthalpy.

6.7.2 SEGS collector

Another simulation was performed using a simple DN 32 pipe with length 309 m, almost the same as for the CLFR Stage 2 prototype, but attached to a SEGS-style concentrator (500 W/m² of solar radiation concentrated to 13045 W/m²) and using the SEGS thermal loss model (as given by Odeh [115]). The results of this simulation are shown in Figure 6.13. The interesting observations here are that no Ledinegg instability is seen at 40 bar, and that the dispersed bubble flow regime has been seen at the collector exit for certain higher pressure, higher flow rate cases.
6.7. PERFORMANCE MAPS FOR OTHER COLLECTORS

Figure 6.11: Contours of efficiency for the CLFR Stage 2 collector at different inlet temperatures (a) 150 °C. Note that the glitch in the 225 °C case is due to that temperature being about the saturation temperature at the lower pressures. Contours in the superheated region were removed for clarity. The solar irradiation was constant in all of the above cases.
CHAPTER 6. THERMOHYDRAULIC ANALYSIS OF THE CLFR ABSORBER

Figure 6.12: Performance maps for the DISS prototype as described by Rheinländer [144]. Variation in pressure drop, outlet pressure, outlet temperature and outlet steam quality are shown. The inlet specific enthalpy in all cases was 1030 kJ/kg. Contours in the superheated region were suppressed for clarity.
Figure 6.13: Contours of pressure drop at lower radiation of 500 W/m², for a DN32 309m absorber with SEGS trough concentrator and SEGS thermal loss model.
6.8 Conclusions

The model of the preceding chapter was applied to the specific case of the CLFR Stage 2 prototype, and the CLFR absorber steady-state behaviour was predicted. A number of sensitivity analyses around the planned operating point were made. These showed that

- at lower solar radiation levels, intermittent flow regimes are likely to occur at the absorber outlet, and
- steady-state Ledinegg instability can be expected to occur at the 40 bar operating pressure, but only if the inlet temperature is below 200 °C, and
- choosing a lower exit steam quality may act to increase the system efficiency by causing preheating of feedwater before it enters the collector; this was identified as an area needing system-level modelling.

This chapter also presented a new form of ‘performance map’, which is a tool for predicting the steady-state response of a DSG collector. Specific performance maps were given for the CLFR Stage 2 prototype, the DISS prototype, and one other simple configuration. These maps give some insights into desirable ranges of operating parameters for DSG collectors. Each map has an assumed inlet temperature, pipe configuration and solar irradiation level, which must be known before the performance map can be generated.

As well as predicting the output state, the steady-state model was used to calculate the expected locations within the absorber of flow regime boundaries at varying levels of solar irradiance.
Chapter 7

Unsteady effects in absorber two-phase flow

In this chapter, transient modelling of two-phase flow in the CLFR absorber is presented using a homogeneous flow model for two phase flow and using a stationary moment equation. The initial work is based on a continuation of that performed by Odeh [115] and later Reynolds [142], but later the form of the equations is altered for improved numerical convergence.

An important distinction in the present work is that it is being performed with a view to integrating the absorber model into a larger system model. This imposes some very different requirements on the modelling technique. Specifically, the state variables of the system (such as pressure and flow rate at the node-points along the absorber pipe) need to be exposed to the high-level solver algorithm, rather than buried in subroutines, as all states need to be included in the time-stepping algorithm for the larger system.

The present work uses an equation-based solver to integrate the differential equations, as discussed by Piela and Westerberg [129]. The software being used is the ASCEND modelling environment [171], to which a number of enhancements were made in the present work, including the connection of a implicit BDF DAE solver code called IDA [61, 59].

The transient model is used to study the effects of sudden increases in solar irradiance in the CLFR system, and some basic results are found that agree with those seen by other workers. Directions for further work that will allow more flexible simulation are identified.
7.1 Theoretical background

7.1.1 Conservation equations

The following is a derivation of the transient flow equations for homogeneous two phase flow, similar to that used in earlier work [115, 142, 14]. For this model, the properties of the two-phase fluid are presumed to be satisfactorily approximated by a fluid with aggregate properties $\rho$ and $u$. How the aggregate properties are connected with the properties of the two phases is not yet being specified. The homogeneous flow equations that result are equivalent to inviscid compressible gas flow.

Take a short segment $\delta z$ of a pipe and its contained flow (Figure 7.1). At the inlet we have inlet mass flow rate $\dot{m}$, pressure $p$ and specific internal energy $u$. At the outlet of the segment, we have mass flow rate $\dot{m} + \frac{\partial \dot{m}}{\partial z} \delta z$, pressure $p + \frac{\partial p}{\partial z} \delta z$, and specific internal energy $u + \frac{\partial u}{\partial z} \delta z$. The pipe has inside radius $r_1$ and outside radius $r_2$. The pipe has thermal conductivity $k$ and specific heat capacity $C$. For the pipe, the temperature on the inside surface is $T_1$ and on the outside surface is $T_2$. Heat absorption occurs at the outside surface of the pipe at a rate $q_s$. Heat loss processes also occur at the outside surface of the pipe, removing heat at the rate $q_l$. Heat is also conveyed to the fluid inside the pipe at the raw $q_l$. The bulk temperature of the fluid is $T$ (with a spatial gradient of $\frac{\partial T}{\partial z}$). Flow moving down the pipe is subject to an inner-wall shear force $\tau_w$ acting against the flow. The mass of fluid in the segment is $\delta m$ and the volume of fluid in the segment is $V$. 

![Figure 7.1: Homogeneous two phase flow model and parameters](image-url)
Mass balance Using the above definitions, we examine the total mass flow rate through the boundary of the segment, and equate that with the partial time derivative of the mass contained within the segment:

\[
\dot{m} - \left( \dot{m} + \frac{\partial \dot{m}}{\partial z} \delta z \right) = \frac{\partial \delta m}{\partial t}
\]  
(7.1)

Using the fact that \( \delta m = \rho \delta V = \rho A \delta z \), we can simplify this to

\[
- \frac{\partial \dot{m}}{\partial z} \delta z = \frac{\partial \rho A \delta z}{\partial t} = \frac{\partial \rho}{\partial t} A \delta z
\]

hence

\[
\frac{\partial \rho}{\partial t} = - \frac{1}{A} \frac{\partial \dot{m}}{\partial z}
\]  
(7.2)

Energy balance Next we consider the energy balance equation for the flow element. By the first law of thermodynamics, the total of the heat rate supplied \( \delta Q \), minus power expended on the environment \( \delta W \), must equal the total energy change, being the net rate of energy moving into the control volume, \( \dot{m} (h + e_k + e_p) \), plus the time rate of change of the total energy inside the control volume \( \delta E \).

\[
\delta Q + \sum_{\text{inlet}} \dot{m} (h + e_k + e_p) = \delta W + \sum_{\text{outlet}} \dot{m} (h + e_k + e_p) + \frac{\partial \delta E}{\partial t}
\]

If we ignore changes in translational potential energy \( e_p \) (gravitational potential) and kinetic energy \( e_k \), and only consider changes in the specific enthalpy of the flow, \( h \), then

\[
\delta Q + \sum_{\text{inlet}} \dot{m} h = \delta W + \sum_{\text{outlet}} \dot{m} h + \frac{\partial \delta E}{\partial t}
\]

\[
\delta Q - \delta W = - \dot{m} h + \left( \dot{m} + \frac{\partial \dot{m}}{\partial z} \delta z \right) \left( h + \frac{\partial h}{\partial z} \delta z \right) + \frac{\partial \delta E}{\partial t}
\]

\[
= h \frac{\partial \dot{m}}{\partial z} \delta z + \dot{m} \frac{\partial h}{\partial z} \delta z + \frac{\partial \delta E}{\partial t} + O \left( \delta z^2 \right)
\]

If we ignore terms \( O (\delta z^2) \) and then substitute \( \delta Q = \frac{\partial \dot{W}}{\partial z} \delta z \), \( \delta W = \frac{\partial \dot{W}}{\partial z} \delta z \) and \( \delta E = u \delta m \), then

\[
\frac{\partial \dot{Q}}{\partial z} \delta z - \frac{\partial \dot{W}}{\partial z} \delta z = \delta z \frac{\partial (\dot{m} h)}{\partial z} + \frac{\partial (u \delta m)}{\partial t}
\]

Divide by \( \delta z \) to give:

\[
\frac{\partial \dot{Q}}{\partial z} - \frac{\partial \dot{W}}{\partial z} = \frac{\partial (\dot{m} h)}{\partial z} + \frac{\partial (u \delta m)}{\partial t} \frac{1}{\delta z}
\]
The final term can be expanded to $\frac{\partial u}{\partial t} \delta m \frac{\delta z}{\delta z} + \frac{\partial \delta m}{\partial t} u \delta z$. Then, using $\delta m = \rho A_1 \delta z$, this is found to equal $\frac{\partial u}{\partial t} \rho A_1 + \frac{\partial \rho A_1}{\partial t} u = \frac{\partial \rho u}{\partial t} A_1$ (with $A_1 = \pi r_1^2$ being the flow area), so

$$\frac{\partial \dot{Q}}{\partial z} - \frac{\partial \dot{W}}{\partial z} = \frac{\partial (\dot{m} h)}{\partial z} + \frac{\partial (\rho u)}{\partial t} A_1$$

Now, allowing for the fact that no work is being done on the environment, we rearrange in terms of the time derivative, to get a final energy balance equation for the flow:

$$\frac{\partial (\rho u)}{\partial t} = \frac{1}{A_1} \left[ \frac{\partial \dot{Q}}{\partial z} - \frac{\partial (\dot{m} h)}{\partial z} \right]$$

We rewrite $\frac{\partial \dot{Q}}{\partial z} = \dot{q}_t$ as the heat transmitted to the fluid per unit length, to give

$$\frac{\partial (\rho u)}{\partial t} = \frac{1}{A_1} \left[ \dot{q}_t - \frac{\partial (\dot{m} h)}{\partial z} \right] \quad (7.3)$$

Note that the above treatment does not make any assumptions about the nature of the fluid which is flowing, and applies equally to single-phase and two phase fluids. One must ensure that suitable expressions for $u$, $h$, and $\rho$ can be calculated. It should also be necessary that only one mass flow rate is included in this relation. In cases where it is important to understand the ways in two-phase flow in which gas and liquid move independently of each other, this equation is replaced by a separate equation for each phase. The experience of other modellers has been that this detail is not required for large systems with the relatively slow transients considered for solar thermal applications [34, 63, 136, 142]. The case for separate fluid flow comes when greater accuracy is needed in order to make highly optimised designs in the field of nuclear reactor engineering [9, 82].

**Energy balance in the tube wall** We must also apply the first law of thermodynamics to the pipe wall in order to model the effect of the thermal mass in the pipe. There is no work being done by the pipe, so the first law is simply a relationship between the rate of heat absorbed by the segment of pipe wall, $\delta \dot{Q}_w$, and the internal energy of the segment of the pipe wall, $\delta U_w$:

$$\delta \dot{Q}_w = \frac{\partial \delta U_w}{\partial t}$$

The heat absorbed by the pipe segment is the sum of the heat fluxes shown in Figure 7.1:
\[ \delta \dot{Q}_w = \delta \dot{Q}_s - \delta \dot{Q}_l - \delta \dot{Q}_t \]

Consider for a moment the components of this heat balance. The heat absorbed on the outer surface, \( \delta \dot{Q}_s \), is an independent variable in the present work, and will be a constant value \( \dot{q}_s \delta z \), where \( \dot{q}_s \) is the rate of heat absorbed at the outer surface per unit of pipe length.

The heat loss from the outer surface, \( \delta \dot{Q}_l \), can likewise be represented in terms of a per-unit-length heat loss, hence \( \delta \dot{Q}_l = \dot{q}_l \delta z \). The use of a correlation for the heat loss \( \dot{q}_l \) was introduced in Section 4.3.7 and its application to the steady-state case was discussed in Section 5.2. This correlation is, in the present work, a function of the absorber temperature, ambient temperature, sky temperature and external heat transfer coefficients on the cavity. Of primary interest is the dependency of the heat loss \( \dot{q}_l \) on the temperature \( T_2 \).

The heat loss from the inner surface, \( \delta \dot{Q}_t \), is discussed in Section 5.1.3. As before we set \( \delta \dot{Q}_t = \dot{q}_t \delta z \). The heat transfer is a function of the bulk temperature of the fluid \( T \) and the temperature of the inner wall of the pipe \( T_1 \). The total heat absorbed can now be written

\[ \delta \dot{Q}_w = \delta z (\dot{q}_s - \dot{q}_l - \dot{q}_t) \]

Accurate modelling of these heat fluxes requires a model\(^1\) that includes the variation in temperature through the pipe wall from \( T_1 \) to \( T_2 \). As resistance to heat transfer by conduction in the pipe wall will be much lower than the resistance to heat transfer by external convection, it will be sufficient to model the pipe with a ‘lumped capacity’ approach [84]. We assume the pipe wall will be at an almost-uniform temperature \( T_w \), and that this temperature can be used to approximate both \( T_1 \) and \( T_2 \). We next express the internal energy of the pipe segment \( \delta U_w \) in terms of this lumped temperature

\[ \delta U_w = \delta U_w (T_{w,ref}) + c_w [T_w - T_{w,ref}] \delta m_w \]

where \( T_{w,ref} \) is some reference temperature, \( \delta U_w (T_{w,ref}) \) is the internal energy of the pipe segment at that reference temperature, \( \delta m_w \) is the mass of the pipe wall segment, and \( c_w \) is the specific heat capacity (assumed constant) of the pipe wall material at the temperature \( T_w \). We only need to be able to calculate the time derivative of \( \delta U_w \), which is simply

\(^1\)For a two-dimensional model of two-phase pipe flow that includes non-lumped wall conduction, see the work of Mousseau [110].
\[ \frac{\partial \delta U_w}{\partial t} = \frac{\partial \delta U_w}{\partial T_w} \frac{\partial T_w}{\partial t} = c_w \delta m \frac{\partial T_w}{\partial t} \]

The mass of the wall element can be written in terms of the pipe density \( \rho_w \) and pipe cross-sectional area \( A_w \) and segment length \( \delta z \), so that \( \delta m_w = \rho_w A_w \delta z \). Hence,

\[ \frac{\partial \delta U_w}{\partial t} = c_w \rho_w A_w \delta z \frac{\partial T_w}{\partial t} \]

The overall expression for energy conservation in the pipe wall is now

\[ \frac{\partial T_w}{\partial t} = \frac{1}{\rho_w A_w c_w} (q_s - q_l - q_t). \quad (7.4) \]

Note that this modelling assumes that circumferential aspects of the pipe wall heat transfer can be ignored. In earlier work by Eck et al \[41\] on direct steam generation in parabolic trough systems, a key concern was that a significant fraction of solar irradiation would be incident on the top of the absorber pipe, while in the stratified flow case, the water would only be present in the lower portion of the pipe. This situation was predicted to lead to significant circumferential variation in pipe temperature. The changed geometry of the CLFR avoids this problem, as stratified flow will, if it occurs, occur in the same part of the pipe as that upon which solar irradiation is focussed. Other work by Dey \[29\] showed that temperature variations are reduced with closer packing of pipes, although the specific configuration studies by Dey were different from that which has been finally adopted for the CLFR.

**Momentum balance** For the momentum balance in the fluid, we set the time rate of change of momentum within the control volume to be be equal to the sum of the forces applied to the control volume, plus the net influx of momentum into the control volume. We only consider the \( z \) direction. Firstly, the momentum within the control volume is equal to the mass times the velocity. The mass is \( \delta m = \rho \delta V = \rho A \delta z \). The mean fluid velocity \(^2\) is \( v = \frac{\dot{m}}{\rho A} = \dot{m}/\rho A \). Hence the momentum in the control volume is \( \rho A \delta z \dot{m}/\rho A = \dot{m} \delta z \), and the momentum balance in the \( z \) direction becomes:

\[ \frac{\partial (\dot{m} \delta z)}{\partial t} = pA - (p + \frac{\partial p}{\partial z} \delta z)A - \pi D \delta z \tau_w + \dot{m}v - \left( \dot{m}v + \frac{\partial (\dot{m}v)}{\partial z} \delta z \right) \]

\[ = \frac{\partial p}{\partial z} A \delta z - \pi D \delta z \tau_w - \frac{\partial (\dot{m}v)}{\partial z} \delta z \]

\(^2\)In this thesis, the symbol \( v \) is used for for velocity and \( v \) for specific volume.
Dividing by $\delta z$ (which is independent of $t$, so this is OK), and noting that $\dot{m} = \rho \dot{V} = \rho Av$

$$\frac{\partial \dot{m}}{\partial t} = -\frac{\partial p}{\partial z} A - \pi D \tau_w - \frac{\partial (\rho A v^2)}{\partial z}$$

Finally, dividing by $A$, and using $A = \frac{1}{4} \pi D^2$ so $\pi D/A = \pi D/\frac{1}{4} \pi D^2 = 4/D$,

$$\frac{1}{A} \frac{\partial \dot{m}}{\partial t} = -\frac{\partial p}{\partial z} - \frac{4}{D} \tau_w - \frac{\partial (\rho v^2)}{\partial z}$$

Note here that the wall shear stress $\tau_w = C_f \frac{1}{2} \rho v^2$ uses the Fanning friction factor (Incropera & DeWitt [75], Eq 8.17, p. 424), and in order to use the more standard Moody friction factor, $f = 4C_f$, the expression needs to be as shown below:\footnote{Note that Taitel [160] uses the symbol $f$ for the Fanning friction factor, and doesn’t use the Moody factor at all.}

$$\tau_w = \frac{f}{4} \frac{1}{2} \rho v^2$$

(7.5)

We can rewrite the momentum balance therefore as

$$\frac{1}{A} \frac{\partial \dot{m}}{\partial t} = -\frac{\partial p}{\partial z} - f \frac{\rho v^2}{D} - \frac{\partial (\rho v^2)}{\partial z}$$

(7.6)

An important observation from Stewart and Wendroff [158] is that pressure propagation as modelled by the momentum equation above can lead to transients of about two orders of magnitude faster than convection propagation:

In a typical 2-phase flow problem the characteristic times might be
of the order of $10^{-5}$ s for inter-phase exchanges, $10^{-3}$ s for pressure
propagation and $10^{-1}$ s for convection.

This is typical ‘stiff system’ behaviour: accurate simulation requires either explicit
integration with very short time-steps able to follow the fastest transients, or else
an implicit integration scheme that ‘rides over’ the short transients.  The alternative
is to modify the problem to use a stationary momentum equation, which removes
the fast transients completely.  For this approach, we imagine for the purpose of
momentum conservation that the fluid is in steady state: no variation of momentum
in the control volume, and no rate of change of mass flow rate along the length of
the pipe.  From the same starting point as the full transient formulation, this yields

$$0 = pA - (p + \frac{\partial p}{\partial z}) A - \pi D \delta z \tau_w + \dot{m} V - \dot{m} v$$

With the friction factor expression substituted for $\tau_w$, this simplifies to
CHAPTER 7. UNSTEADY EFFECTS IN ABSORBER TWO-PHASE FLOW

Figure 7.2: Propagation using the simple advection equation [21] of a square step signal using the backwards difference formula integrator IDA [59] in ASCEND, with three different spatial discretisation schemes. The inherent instability of the central difference formula for this case is clear.

\[ \frac{\partial p}{\partial z} = -f \frac{\frac{1}{2} \rho v^2}{D} \]

It is noted that this simplification transforms one of the differential equations of the system into an algebraic equation: there is no temporal derivative in this equation. The stationary momentum assumption has a very significant effect on the numerical behaviour of the model system, and reduces by one the number of state variables per flow ‘node’.

7.2 Model development

The system of equations developed in the previous section can not in general be solved analytically\(^4\), so we must turn to numerical methods to solve the system of differential equations. We will use the numerical ‘method of lines’ [5]. This requires spatial discretisation, that is, to divide the continuous spatial domain into discrete points, with a system of estimating the spatial derivatives at those discrete points based on the finite difference approximations. Carver and Hinds show that for advection problems such as this one, the choice of the finite difference approximation to the spatial derivatives can have a huge bearing on the success of the efforts at modelling [21], and in particular show that careless use of the central difference method will cause numerical instability (Figure 7.2).

7.2.1 Finite difference formulae

We will assume spatial discretisation of the continuous domain \(z\) into nodes numbered \(i \in \{1, ..n\}\). The locations of the nodes will be \(z_1\) to \(z_n\), with \(z_i = (i - 1)\Delta z\)

\(^4\)In some cases the semi-analytical method of characteristics, or ‘wave tracing’ method, can be applied. This approach gives very accurate results for the wave propagation behaviour of the flow, and without numerical diffusion effects [9], but is not of relevance to the longer time-scale convective phenomena of interest here.
and $\Delta z = L/(n - 1)$. All the continuous variables on the domain are also replaced with values at these points, so a continuous variable $X(z)$ becomes discretised as \(\{X_1, ..., X_n\}\) with $X_i = X(z_i)$. Finite difference formulae are now chosen that give an approximation to the derivative of the continuous variable using only the discretised values at the locations \(\{z_1, ..., z_n\}\). The simplest approximations are backwards difference formula

$$\frac{\partial X_i}{\partial z} \approx \frac{X_i - X_{i-1}}{\Delta z}$$

or the central difference formula

$$\frac{\partial X_i}{\partial z} \approx \frac{X_{i+1} - X_{i-1}}{2\Delta z}$$

or the forwards difference formula

$$\frac{\partial X_i}{\partial z} \approx \frac{X_{i+1} - X}{\Delta z}$$

Also possible are the upwind three-point formula and the upwind-biased four-point formula, respectively [21]:

$$\frac{\partial X}{\partial z} \approx \frac{3X_i - 4X_{i-1} - X_{i-2}}{2\Delta z}$$

and

$$\frac{\partial X}{\partial z} \approx \frac{-X_{i+1} + 6X_i - 3X_{i-1} - 2X_{i-1}}{6\Delta z}$$

These formulae are used in various combinations in the work that follows.

### 7.2.2 Numerical integration

The ‘numerical method of lines’ allows the integration of a system of partial differential equations having one temporal and one spatial dimension. The spatial dimension is discretised as above; what is done with the temporal dimension is not specified. The point is that by discretising in space, and replacing spatial derivatives with finite dimensional approximations, the problem is transformed to an ODE system, or in full generality, a differential-algebraic system of equations. There is a broad range of techniques available for the solution of such problems including single step and multi-step methods and explicit, semi-implicit and fully implicit methods [5].

Established systems for modelling two phase flows have predominantly used either explicit integration or ad hoc semi-implicit methods [82, 158], or sometimes the method of characteristics [9], and with only occasional efforts to use fully implicit methods [110]. The semi-implicit schemes follow the same general approach as used
for compressible single-fluid flow [126], but require several additional relations to allow for independent motion of the two phases, and for mass and heat transfer between them. The details of these models is quite incredible: the fundamental models are ill-posed, which implies that numerical diffusion is required even to make the model convergeable [158]. Then, there are complex models for interfacial heat and mass transfer [82, 15] that go far beyond what, to the author’s knowledge, has been applied in modelling by any workers in the field of solar thermal energy.

Previous work at the University of New South Wales by Odeh and later Reynolds used an ad-hoc fully explicit ODE method to simulate the flow [115, 142]. In the context of the present system-level modelling, it was recognised that transient modelling would require a very different architecture if the collector model was to be connected with other system components. In particular, the fixed-mass fixed-volume nature of the Stage 1 system concept (Section 8.1) involves stiff system behaviour that prevents simply encapsulating the explicit-integration absorber model of Odeh and Reynolds. The present work tackles the absorber model using an implicit integration approach for that reason.

### 7.2.2.1 Stiffness

A key concern in deciding on an approach to this problem was the observation that, especially in the case of a fixed-volume, fixed-mass system, there were likely to be transient modes with very different time constants. Pressure transients would be rapidly passed throughout the system, but convection and pipe wall temperature changes would occur much more slowly. This is the feature of so called ‘stiff’ systems that leads to difficulty in their efficient solution [5]. When solving stiff systems with explicit integration techniques, one is constrained to use very short time steps dictated by the fastest transient mode in the system. However if an implicit method is chosen (or, as in the case of the some of the big two-phase flow solvers, a carefully-crafted semi-implicit technique is used) then these shortest transients can effectively be ‘skipped over’, permitting much longer time-steps to still be solved accurately. This benefit comes at the cost of needing to perform a matrix solution $Ax = b$ at each time-step, but this is often found to be worthwhile.
7.2.2.2 DAEs and ODEs

Another key decision to be made was whether or not to use a DAE solver\(^5\) for the system.

There was the concern that the fixed-mass, fixed-volume system might be a high-index problem\(^6\) (see Ascher and Petzold [5]). Another aspect was that the solution of fluid properties, being solved in an inner iteration loop, would be more efficiently solved if they were present as equations in the (DAE) model. Finally, iteration of the cavity heat loss behaviour could also possibly be eliminated by adding those equations into a DAE representation of the system.

7.2.3 Thermodynamic properties

It becomes clear that accurate simulation of fluid systems depends very much on how the thermodynamic properties of the fluid are calculated, especially in the context of equation-based modelling. The reason for this is that equation-based modelling requires calculation of Jacobian matrices in order to apply Newton iteration at each time-step (and equally when calculating steady-state solutions). Thermodynamic properties must therefore ideally be implemented in a form that provides continuously differentiable functions, but then it must be observed that although the properties will be continuous, the property derivatives certainly will not be. Discontinuous derivatives, especially in the context of index-1 (or higher) differential-algebraic systems of equations cause great numerical difficulties, as noted by Petzold [128]. A range of ways of resolving these problems is given in the following paragraphs.

Restrict modelling to a single fluid phase The first of these approaches is probably the simplest, but least useful. By modelling behaviour of only a single flow region, such as only the saturated region, or only the subcooled region, phase-boundary discontinuities can be avoided. We would like, however, to model the behaviour of the CLFR system from start-up, at which time the fluid is subcooled

\(^5\)DAEs are differential/algebraic equations systems. In an ODE system, all variables are present with both differential (\(\frac{dx}{dt}\)) and derivative (\(\frac{d^2x}{dt^2}\)) forms. DAE systems, by contrast, contain some additional algebraic (\(z\)) variables along for the ride, for which no derivatives are present. Sometimes these variables could be eliminated by substituting their values into the other equations. This isn’t always desired, however, and even in cases where it is, it can lead to systems that take longer to solve.

\(^6\)A high-index problem is one where there are ‘missing equations’ and it is not possible to calculate derivatives of all of the variables in the system of equations without first differentiating some of the equations in the system. Engineering systems in natural form are commonly ‘index 1’ but this is sidestepped by eliminating variables or by solving them in an inner loop, so that the integrating solver does not ‘see’ them. Many problems in certain fields, such as robotics, are of higher index (2 and above). In these cases ‘index reduction’ is an automated process that can allow such systems to be solved when they would otherwise be impossible to solve using standard ODE techniques. Ascher and Petzold describe these problems and how they can be solved using DAE techniques[5].
liquid. We would also like to be able to model all the way into the superheated region: although this is outside the design scope, it will be part of risk evaluation scenarios.

**Ignore the discontinuity** When using a sophisticated predictor-corrector integration algorithm, there is a reasonable chance that a discontinuous derivative can be ‘ridden over’, because the variable step-size will shrink in response to the increased disagreement between the predictor and the corrector. This is a dangerous approach, however, and one that is likely to lead to models that are not robust.

**Use lumped capacity methods to smooth the discontinuity** This approach was used in the system modelling in Chapter 8, where the whole CLFR absorber was encapsulated in a ‘black box’ calculation. Whereas the absorber contains thermodynamic property discontinuities at each node in the flow path, the black box model does not expose these to the solver, with the result that its Jacobian matrix is smoother. This worked in the case of the steady-state model but is not easily applicable to the transient model, due to the fact that the state of the absorber must be known by the solver in order to be able to integrate for its behaviour over time; a lumped capacity model of the absorber would eliminate the key aspects of the physics that we need to model here.

**Detect discontinuities as boundaries, and carefully traverse them** This approach is probably the most elegant, but requires sophisticated software. The idea is that one can always detect when a boundary has been crossed, and from there, one can backpedal to the exact point in time where the discontinuity emerged\(^7\). Once that spot has been found, the model can be re-examined and the model formulation adjusted\(^8\). For example, the saturation steam property equations can be replaced by the superheated steam property equations. Then the model may need reanalysis (if new intermediate variables have been added, for example). The ASCEND modelling environment, used for the modelling in Chapter 8, provides the conditional modelling solver CMSlv [145] that can manage the solution of such models in the steady-state, but at the time of writing provides no support for dynamic modelling involving conditions\(^9\). As part of the present work, the ASCEND modelling environment was extended to provide support for the differential-algebraic

\(^7\)This is a root-finding problem with \(t\) as the independent variable

\(^8\)Equations that apply at time \(t^b\) before the boundary \(t_b\) are replaced with equations that apply after the boundary, at \(t^a\)

\(^9\)Note that in many ways, the steady-state conditional modelling problem is more complex, as one doesn’t know which region the solution will lie in. In the dynamic case, if was can assume that initial conditions are known, any changes in the conditional model are then made as incremental steps from the starting state. It should therefore be much easier to identify the correct behaviour at boundaries in this case.
equation solver, up to the point of boundary detection but it was not possible to complete the implementation of boundary traversal. The active research area of Mixed Integer/NonLinear Programming (MINLP) applies to this problem and only a few software packages are available that tackle this problem with generality. The approach to boundary traversal which was to be implemented was that provided by the IDA integrator, part of the SUNDIALS suite by Hindmarsh et al [60]. One dilemma with the boundary-detection-and-traversal approach is that it requires an extensive re-analysis of the problem each time a boundary is crossed: the intention is, for example, that the equations that represent saturated fluid properties would be removed from the model and replaced with equations that represent superheated properties. In the case here of dynamic two-phase flow in a solar absorber, the saturated/superheated boundary is progressively moving through the modelled pipe, so this approach would result in many model re-analysis steps, to the point where it might be impractical for method-of-lines models such as this.

**Smooth out the discontinuity** If a way can be found to remove the discontinuity from the model without it changing the results too much, then this might be a good approach. For example, in some $\delta$-neighbourhood of the saturation boundary, steam properties could be returned as a ‘blurred’ average of values on either side of the boundary. This option was not investigated, as attempts with this approach had not been found in the literature. It seems likely that the degree of smoothing required to resolve the numerical problems would be greater than could be tolerated for accuracy of results.

**Move the nodes as the discontinuity moves** One way to avoid moving over the discontinuity is to move the nodes as the discontinuity moves. A simplified form of this approach was used by Ray [136] in modelling for the Solar One system. The ColSim simulation also implements this strategy by considering pipe flow to occur as coherent ‘slugs’ passing down the pipe in strict order [177]. This result requires a significantly different solver framework, in which the slug positions must be tracked through time. The relatively new ‘adaptive’ method of lines also uses this approach, although using a more automated approach to following the boundaries, ensuring that nodes move closer together at locations where fast transients are occurring [178].

### 7.2.3.1 Incorporating IAPWS steam tables

With the desire to present modelling of the CLFR system to large power-generating companies it was recognised that international-standard steam tables should be adopted if at all possible.
It is noted that many of the standard ‘equations of state’ used in the chemical and process engineering field do not include compressibility of the fluid phase (for example, see Reid [137]). At the pressures employed in steam engineering, compressibility of the fluid is a significant feature that affects the mass flow rates and fluid properties. It was considered that the simplified equations of states used in that field would not be sufficient for the present work.

The IAPWS-IF97 correlations were used in Chapters 5, 6 and 8. However, some difficulty was starting to become apparent with their application in Chapter 8 when solving certain kinds of constrained system configurations. This is attributed to the fact that the steam property derivatives were computed by ASCEND using finite difference approximations. This fact exacerbates the problems discussed in the previous section, as the values of derivatives resulting from finite different approximations that span across discontinuities can clearly be incorrect, and lead to glitches in the Newton iteration process.

For the present work three approaches were used:

**Implement \((p,h)\) formulation for all regions** The IAPWS-IF97 correlations were implemented as a black-box unit in ASCEND (Section 2.6.1.3) that allows precise evaluation of properties in terms of pressure and enthalpy, using only the ‘forward’ equations from the IAPWS-IF97 release. This minimises the discontinuities across the boundaries but does not eliminate the discontinuities in the derivatives of the properties. Implementing using the black-box feature of ASCEND is necessary if conditional modelling is not to be used. The black-box approach has the additional benefit that superfluous intermediate variables are not present in the equation system. Additional variables such as those used to compute the property partial derivatives in the IAPWS-IF97 formulation, if present in the ‘main’ system of equations, can easily lead to a model much larger than it needs to be.

**Symbolic differentiation** Using the symbolic manipulation toolbox GiNaC [12], an attempt was made to implement a version of the IAPWS-95 steam tables including exact derivatives based on symbolic differentiation of the correlation equations. This approach showed some promise but the work of connecting to a modelling environment satisfactory for all the other parts of the problem was considered too great and it was not pursued further. Note that, in the work of Hirsch et al [63], a bespoke set of polynomial correlations for thermodynamic properties was created in order that they could easily be differentiated using the commercial solver Dymola; this allowed them to achieve numerical convergence. In general, the development of bespoke correlations should, however, be avoided, due to the difficulty in quantifying the deviations that arise compared to analysis using the standard correlations such
7.3. THREE-REGION FORMULATION

as IAPWS-IF97 and IAPWS-95

**Single-region formulation**  A single region formulation for two-phase flow was produced based on the IAPWS supplementary release for saturation properties [68]. While not in full conformance with the IAPWS-IF97 release [71], this enabled a fully equation-based steam property formulation to be implemented in ASCEND.

### 7.3 Three-region formulation

In this formulation, the full ‘freesteam’ IAPWS-IF97 steam properties routines were connected to ASCEND as a black box function in terms of pressure and specific enthalpy \((p, h)\). The homogeneous two-phase flow model, with the stationary momentum expression, was constructed using equation objects in ASCEND. A flow path length of 100 m was simulated with an internal diameter of 60 mm and an outside diameter of 70 mm\(^{10}\). A fixed internal friction factor of 0.03, fixed internal heat transfer coefficient of 1000 W/m\(^2\)K and fixed external heat transfer coefficient of 5 W/m\(^2\)K were used. The inlet conditions were set to 0.26 kg/s, 150 °C, 42 bar. The model contained 40 nodes\(^{11}\) for a spatial resolution of 2.5 m. Steady state conditions were first computed with a solar irradiation level of \(\dot{q}_s=335\) W/m. After steady state was achieved (using the NLA solver) the solar irradiation was set to \(\dot{q}_s=6000\) W/m.

Figure 7.3 shows the result of running this model until equilibrium was again achieved. The heat flux selected was such that the exit flow passed from subcooled all the way up to superheated, representing the system start-up process. The results appear plausible, and are in keeping with the steady-state solution at the specified irradiation level. The initial wall temperature is almost the same as the fluid temperature but then rises as the thermal mass of the pipe responds to the greater rate of heating. Eventually its temperature reaches a new equilibrium at about 30 K above the fluid temperature. The thermal inertia of the pipe is also seen at about \(t=250\) s, where it takes a minute or so to stabilise after the fluid temperature enters the saturation region.

The most interesting part of Figure 7.3 is the plot of mass flow rate against time. The broad effect is a peak in float rate as the mass holdup in the pipe is expelled before again returning to a steady flow situation. The puzzling part of the plot is the sudden increase in flow rate in the first \(\sim30\) s. We would have expected the

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\(^{10}\)These dimensions are modelled on dimensions of the DISS collector.

\(^{11}\)This limitation is a result of the dense direct matrix methods available in the ASCEND integration solver at this stage. With sparse matrix methods, much larger numbers of nodes should become possible.
thermal mass to have buffered the system from this effect, so we will investigate further with a detailed plot of the first few seconds.

Figure 7.4 shows the results again in detail for the first 30 seconds of the simulation. The increase in the flow rate does appear to start off with a flat gradient. But by the end of 30 s, the flow rate has increased to 0.35 kg/s, which seems high, but when we consider that there is 100 m of water that has increased in temperature by 5 degrees, we see that this requires a volume of 1.5 L water to have been expelled in that time, so the order of magnitude appears right, allowing for the fact that nothing much happens for the first few seconds.

The other interesting part of Figure 7.3 is the second sharp rise in flow rate at about 250 s, which is the time when the exit steam enters the saturation region. At this time, flow upstream of the exit is also entering the saturation region, which is causing bubbles to form which displace water from the end of the pipe at an even greater rate. After 400 s, the effect is equalised, and the flow rate starts to reduce back to the steady state flow.

This simulation shows that this simple model is able to reproduce the expected transient behaviour for a sample case. The model however does not include an accurate pressure loss correlation, nor does it include correct internal or external heat transfer correlations. These things must be attended to before a realistic transient model of the CLFR is possible. Another limitation is that the model convergence is hard to attain for longer pipes and for more nodes. The main cause for that appears to be the fast transients seen each time a phase boundary is crossed, as discussed in Section 7.2.3.

7.4 Single-region formulation

In order to assess the degree to which boundary-crossing discontinuities were responsible for the numerical difficulties with the three-region steam properties formulation in Section 7.3 above, the model was rebuilt using the smooth equation-based steam property correlations, with only the saturation region included (Section 7.2.3).

This was done using another IAPWS release, the Revised Supplementary Release on Saturation Properties of Ordinary Water Substance [68]. This formulation provides steam properties in the form of $p_{\text{sat}}(T), \rho_f(T), \rho_g(T), \alpha(T)$ and $\phi(T)$. The property $\alpha$ is used to calculate saturation enthalpies using the additional relations

$$h_f = \alpha + \frac{T}{\rho_f} \left. \frac{dp}{dT} \right|_{\text{sat}}$$

$$h_g = \alpha + \frac{T}{\rho_g} \left. \frac{dp}{dT} \right|_{\text{sat}}$$
Figure 7.3: A simple formulation of transient flow in a 100 m pipe with 60 mm internal diameter, subject to a step increase in solar irradiation.
Figure 7.4: Detail of the first 30 s of simulated time from Figure 7.3. Note the logarithmic time-scale.
while the property $\phi$ is used to calculate the specific entropy, which is not required for the present work. In order to model fluid within the saturation region, it was then necessary to add interpolation relations

$$h - h_f = x (h_g - h_f)$$

and

$$u_f = h_f - \frac{p}{\rho_f}$$

$$u_g = h_g - \frac{p}{\rho_g}$$

$$u - u_f = x (u_g - u_f)$$

and, correctly performing the linear interpolation on specific volume instead of density,

$$\rho_f \rho_g = [\rho_g + (\rho_f - \rho_g) x] \rho$$

These steam property correlations were carefully tested and were seen to converge reliably (in stand-alone form) for a range of different known/unknown variable configurations.

The model from Section 7.3 was then updated to use these single-region steam properties. The steam quality, $x$, was limited to the range [0,1] so that the solver would fail if the solution migrated outside those bounds.

It was observed that with this steam properties formulation, the model appeared to be less stable, and harder to integrate. The pipe length had to be reduced to 16 m, and the number of nodes had to be reduced to 7 nodes, and the finite difference schemes had to be adjusted to central difference for the mass conservation equation. This was surprising, as central difference schemes were observed to be destabilising for many advection problems [21].

The limited results of a simulation using this model are shown in Figure 7.5. For this case, the inlet conditions were initially $\dot{m}=0.26 \text{ kg/s}$, $x=0.23$, $p =10 \text{ bar}$, $\dot{q}_s=100 \text{ W/m}$. After steady-state conditions had been computed, the solar irradiation was increased to $\dot{q}_s=6000 \text{ W/m}$.

As before, the peak and decline in the exit mass flow rate is observed, again in apparent anticipation of the rise in exit quality; again we attribute that to the fluid expansion in the pipe, as in Section 7.3.

Some slight instability in the mass flow rate result can be observed. When the length of the pipe is increased or more nodes are added, the instability grows
until eventually causing the model to become fully divergent. The instability also appeared in the three-region model, but was less pronounced, perhaps due to the fact that there were not as many variables in the model. (The implementation of the single-fluid model has introduced new variables such as $h_f$, $h_g$, $\alpha$ and so on, that were previously hidden inside the ‘black box’ used for thermodynamic property evaluation in the three-region model). It is also possible that the method used here to calculate the saturated fluid density by interpolation between the saturation boundaries could be introducing some numerical problems.

On the other parts of the graph, the expected results are seen. The increased heat flux causes an increase in the pipe temperature owing to the greater temperature difference needed to drive the increased heat flux into the flowing fluid. The temperature of the fluid however has not changed as it remains in the saturation region. Enthalpy and quality increase smoothly as expected.

### 7.5 Further work

Some efforts have been made to implement conditional modelling support for ASCEND in order that the problems of boundary traversal could be overcome when integrating the three-region model. Boundary detection was successfully (and generically) implemented as part of the new IDA integrator support in ASCEND that was added for the purpose of the present work; boundary traversal would require additional work, and was thought to go beyond the scope of the thesis, but it is hoped that it may be pursued as a later project.

The above models used fixed constant values of friction factor and internal heat transfer coefficient. In fact these are variable quantities that furthermore depend on the phase of the local fluid flow. In the above modelling, numerical difficulties have prevented adoption of the more accurate models used in the steady-state case. It is thought that with boundary crossing support in ASCEND, the detailed friction and heat transfer correlations could be added to the transient model without problems.

### 7.6 Conclusions

A transient homogeneous two-phase model of flow in the CLFR absorber was derived from conservation equations. The model is solved using the method of lines and modern implicit DAE integration methods, and maintains a clean separation of model definition and solution method. To the author’s knowledge, no other researchers have applied implicit DAE methods to the direct steam generation problem. It is anticipated that these methods will facilitate more robust modelling of the CLFR collector in a full-system model context.
7.6. CONCLUSIONS

Figure 7.5: Simulation results for a 16 m long pipe of 60 mm internal diameter, using the equation-based single-region thermodynamic properties correlation.
The model was used to demonstrate the transient response of the collector to a sudden large increase in solar irradiance. The response reproduces behaviour noted by other workers, including the short term spike in exit flow rate followed by the gradual return to steady state conditions.

The successful model uses complete IAPWS-IF97 steam properties, which are industry-standard but were not used in work by Reynolds, Odeh, or Hirsch et al. The steam property routines were developed as part of the present work, and have been released into the public domain.

Difficulties noted in transient modelling of the direct steam generation problem were discussed and a number of opportunities for further work were identified. It is the author’s understanding that earlier workers experienced the same difficulties in achieving stable solutions for the sharpest transients, and that further work in this field is still required.
Chapter 8

Overall system modelling

In Chapters 5 and 6, a model of the two-phase flow in the CLFR absorber was constructed and validated against experimental data. In this chapter, the absorber model is connected with the other components necessary for a complete CLFR system to function, with a view to understanding likely operating modes at full and partial sun, evaluating pressure drops in pipework, designing the circulating pumps, and attempting to determine the most efficient operating point.

The work for this chapter was performed in two stages corresponding to two quite different system designs that emerged as the project developed. The first design was a fixed-mass, fixed-volume, closed-loop system; the second system was open in the sense that mass could be displaced to the power station loop. We will describe the models and show results for each of these systems in turn.

8.1 Stage 1 design

At the time of the CLFR Stage 1 Prototype, work was underway to determine how the later-stage collector would ultimately be connected to the power station. The process risk involved in ‘plugging in’ directly to the power station fluid loop was considered to be quite high, so the following approach was pursued, in which the CLFR loop was isolated from the main power station by a heat exchanger. The configuration is shown in Figure 8.1.

The closed loop in this case results in a fixed-mass, fixed-volume two-phase flow circuit. Clearly, as the absorbed heat varied, there would be a wide range of pressures resulting from water being boiled in a confined space. In order for the fluid to expand, a surge tank was made a part of this design.

The aim here is to predict steady-state operating conditions, required pumping power and pressure, mass transfer around the system and likely start-up, shutdown and cloud transient effects. Flow regimes within the absorber at different irradiance
Figure 8.1: Stage 1 model for the CLFR steam circuit
level are found. The interaction of the absorber pressure, irradiance and pump speed is discussed. A simple model for steady-state system pressure is presented, built around an assumed controller that maintains the absorber steam exit quality as 80%. Finally, a more complex steady-state model, in which the total mass of fluid in the system is used to drive fluctuations in system pressure, is presented. The initial design for the CLFR power-station tie-in included a large heat exchanger due to difficulties in achieving agreement with Macquarie Generation regarding direct tie-in to the boiler feedwater heating line.

The first effort made here to produce a steady-state system model was using an ad-hoc shooting method (as described by Ascher and Petzold, 1998 [5]) with the earlier steady-state absorber model combined with various other system components. The shooting method was found to be simple and usable, but relatively expensive computationally, and not flexible for changes to system configurations and parameters.

The focus of the present work is to determine the operating points of the final system when connected to the power station. In particular, the required pumping power and surge tank capacity need to be determined. Also, the behaviour of the system during start-up and shutdown, as well as during cloud transients, must be simulated, in order that a suitable control strategy can be devised.

A steady-state system model has been built with C++ code, and is depicted in Figure 8.1. An overview of each of the components being modelled follows.

### 8.1.1 Heat exchanger

A heat exchanger is required for the transfer of energy from the solar array to the power station. This is required to protect the integrity of the power station, and ensure that contaminated water does not enter the turbines. It is hoped that a stand-alone CLFR system would not need a heat exchanger. The heat exchanger is effectively a sub-cooling steam condenser. The component is modelled in two parts: firstly the latent heat transfer that occurs as the steam on the hot-side is reduced to saturated water, and secondly the sensible heat transfer that occurs as the saturated hot-side water is sub-cooled (8.2). This calculation requires an iterative procedure, since the hot-side outlet temperature must be guessed and an error based on the heat transfer rate in the sensible stage minimised.

### 8.1.2 Absorber

The absorber in the earlier modelling consists of 16 parallel DN 25 pipes, each 60m long, made of 304 stainless steel, and mounted side-by-side, for a total absorber width of 575 mm. Absorber pipes in the prototype are connected via a set of manifolds into
A sizing calculation is performed based on expected hot-side inlet conditions, design hot-side outlet subcooling, and required total heat transfer. For the prototype system, taking overall heat transfer coefficients to be $U_f = 4000 \text{ W/m}^2\text{K}$ and $U_l = 1200 \text{ W/m}^2\text{K}$ (Incropera & DeWitt [75], p. 586), the approximate required heat transfer area is 12 m$^2$. For the full-scale system, the heat exchanger will be large, and a major cost item.

![Figure 8.2: Heat transfer in the heat exchanger](image)

A four-pass configuration, so that the total length of a flow path is approximately 240m. About 300mm beneath the absorber surface, a transparent cavity cover forms the lower surface of a sealed air cavity, acting to limit convective losses from the hot absorber. Pye et al [131], Reynolds et al [140] and Reynolds & Jance et al[141] show modelling results for the air cavity. The cavity model from Pye et al [131] has been used here. Inside the absorber pipes, wet steam at approximately 60 bar at 270°C with a final quality of 0.8 to 1.0 will be produced, with a pressure drop of 2 to 10 bar as it passes through the absorber. Modelling of the forced-convection boiling process has been performed using the method of Odeh [115] and Reynolds [140]. This entails dividing the flow path into a series of one-dimensional elements, and then solving the pressure drop and enthalpy rise for each element in sequence, using local flow properties based on the outlet of the previous segment. For the pressure drops, the Colebrook equation is used in single-phase elements, and the Martinelli-Nelson correlation [94] is used in two-phase elements. Likewise, for the heat transfer coefficients, the Dittus-Boelter and Gnielinski correlations (Incropera & DeWitt [75], p. 444 ff.), and the Gungor-Winterton method (1987) are used. Solution of the absorber model in isolation shows that under normal steady radiation, unsteady two-phase flow will not be expected at the absorber outlet (Figure 8.3, and [103]). However, this does not prove that during transient conditions unsteady flow patterns will not occur, so further analysis is required, necessitating a full transient system model.
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Figure 8.3: Flow regimes expected along the absorber pipes in the full-scale CLFR. For each irradiation level, the flow rate has been adjusted to give outlet quality of 0.8

8.1.3 Pump

The pump is modelled by an ideal quadratic performance curve. A reference pump curve, defined at the speed $N = N_{ref}$, is taken to be a quadratic $H = f(Q)$, with a maximum head at $Q = 0$ of $H = H_{max}$, and a specified flow rate $Q_{ref}$ at $H = 0.9H_{max}$.

The isentropic efficiency of the pump $\eta_{is}$ is also specified, allowing all of the pump outlet conditions to be calculated when the pump speed is specified along with the inlet conditions and flow rate.

The above constraints allow the pump to be solved for any chosen speed using the equation

$$H = C_0 \frac{N}{N_{ref}} + C_1 \frac{N}{N_{ref}} Q + C_2 Q^2$$  \hspace{1cm} (8.1)

8.1.4 Other components

A simple model for the steam separator is used: if the inlet is saturated, the liquid component goes to the throttle, and the gas component goes to the heat exchanger; if the inlet is superheated, then all flow goes to the heat exchanger; if the inlet is sub-cooled, then all flow goes to the throttle. Obviously in a real separator, these
assumptions for sub-cooled and superheated flow are not correct, and will need to be corrected for an accurate transient model.

The surge tank can be ignored for the purpose of this steady-state system model. However, when the mass transfer from component to component is included in the system, this component will be required for an accurate representation of the system. The purpose of the surge tank is to contain during normal operation the water that will be present in the absorber and associated pipe work when the system is starting up. It will also act as a time-lag component, which may aid controllability when trying to achieve a constant temperature at the heat exchanger cold-side outlet.

The throttle is required to drop the pressure from the steam separator outlet down to that of the pump inlet. In reality, a control valve would likely be used here. For the modelling, the throttle is isenthalpic. There will be a pressure drop here since a small pressure drop has been allowed for in the heat exchanger model, and this will be the pressure drop seen across the throttle.

Where the throttle outlet joins the pump inlet, a mixer is required in the model. The mixed outlet flow is chosen to have the mass-weighted average of the specific internal energy of the two inlet streams, and inlet pressure of the surge tank outlet.

### 8.1.5 Model results

The overall model has been solved using the approach shown in Figure 8.4. There are a number of model parameters which can be used as inputs into the calculation. These are shown in Table 8.1. Other system parameters which were taken as constants in this study are shown in Table 8.2. The method of solution was to take an assumed operating point at the absorber inlet, then to move around the cycle, solving each component in turn, based in each component’s inlet conditions. Where components had variable parameters as yet unspecified, such as pump pressure rise, these values were guessed. Then, when the whole cycle had been solved in this way, the Brent solver algorithm [130] was used to adjust the parameters such as pump speed so that the steam pressure and enthalpy at the end of the cycle match those at the start of the cycle.

The parameters shown in Table 8.2 were initially chosen since it is relatively simple to solve the system with these values fixed.

An absorber inlet temperature of 240 °C is used as the first value for the iterations.

Figure 8.5 shows a series of results for operating conditions where the inlet pressure of the absorber is 60 bar. As the pump flow rate is increased, the required pumping pressure increases. Higher radiation also requires a higher pumping pressure. Note that these are artificial conditions since the absorber inlet pressure would
Figure 8.4: Calculation procedure for the first steady-state system model. Bold boxes show input/outputs that can be specified. The dotted lines show predictor/corrector value-pairs that can be used along with the circuit calculation to determine the remaining unknowns. Therefore, two operating parameters are required in order to find a valid operating point using the above method.
### Table 8.1: System operating parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$G$</td>
<td>Irradiation (W/m²)</td>
<td>500 - 1100 W/m²</td>
</tr>
<tr>
<td>$p_1$</td>
<td>Pressure at absorber inlet (bar)</td>
<td>50 - 90 bar</td>
</tr>
<tr>
<td>$\dot{m}_1$</td>
<td>Pump mass flow rate (kg/s)</td>
<td>0.4 - 1.0 kg/s</td>
</tr>
<tr>
<td>$x$</td>
<td>Absorber outlet steam quality</td>
<td>(aim for 80%)</td>
</tr>
</tbody>
</table>

### Table 8.2: System design variables

<table>
<thead>
<tr>
<th>CLFR design</th>
<th>Values used are those for the prototype system. Absorber length is 60 m, 16 x DN 25 SS304 pipes in four-pass configuration. Optical concentration is 40 and optical efficiency is 0.8.</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>Heat exchanger contact area (m²)</td>
</tr>
<tr>
<td>$U_f$</td>
<td>Overall heat transfer coefficient in latent heat transfer in the heat exchanger (W/m²K)</td>
</tr>
<tr>
<td>$U_l$</td>
<td>Overall heat transfer coefficient in sensible heat transfer in the heat exchanger (W/m²K)</td>
</tr>
<tr>
<td>$T_8$</td>
<td>Heat exchanger cold-side inlet temperature (°C)</td>
</tr>
<tr>
<td>$T_9$</td>
<td>Heat exchanger cold-side outlet temperature (°C)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Pump design</th>
<th>At a reference speed of 2000 RPM, the maximum pump pressure is 10 bar at zero flow and at this same speed but a flow rate of 2 kg/s, the pump generates 9 bar pressure. The density at reference conditions was taken as 704 kg/m³</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta_s$</td>
<td>Pump isentropic efficiency</td>
</tr>
<tr>
<td>$T_{amb}$</td>
<td>Ambient temperature (°C)</td>
</tr>
</tbody>
</table>
naturally ‘float’ to a different pressure if the pump flow rate were altered. In effect, by fixing the absorber inlet pressure, a variation in the mass of fluid in the system is being permitted.

The upper half of Figure 8.5 shows the steam quality at absorber outlet for the same range of conditions. As the pump flow rate increases, the outlet quality at the absorber decreases. At high radiation, the outlet is superheated at low flow rates. For low radiation and higher flow rates, the available energy in the hot-side flow was not sufficient to achieve the desired cold-side outlet temperature in the heat exchanger, so no solution is shown.

Figure 8.6, shows a series of results for cases where the pump flow-rate has been held constant. The absorber inlet pressure has been varied between cases, which is equivalent to the overall system pressure varying, and different levels of solar radiation are shown. As the pump flow rate is constant, the absorber outlet quality (in the upper half of Figure 8.6) shows a dependency only on the irradiance. The pump pressure rise is seen to decrease for increasing absorber inlet pressure, however. This reflects the Ledinegg instability, which was discussed in 5.3.9. This is a direct consequence of the two-phase flow that is present in the absorber. At higher pressure, the steam is more highly compressed and this is resulting in a lower average friction factor for the flow in the absorber.
Figure 8.6: Closed-circuit behaviour for fixed pump mass flow rate. Pressure rise across the pump is shown for the fixed pump flow rate of 0.6 kg/s, with varying pump outlet pressure and radiation.

8.1.6 Effect of fixed total mass in the system

A study of the effect of a fixed total mass of water in the system has been performed by considering only the volume of the absorber, a pipe between the absorber and the heat exchanger, and a pipe from the heat exchanger to the surge tank. This corresponds to points (3) and (6) on Figure 8.1. The approach used was to take an operating point as before, then to adjust the (previously arbitrarily fixed) absorber inlet pressure until the mass of water in the system is the desired (fixed) amount. This is an effective ‘shooting’ method as described by Ascher and Petzold [5].

Using 50 m-long, DN 50, schedule 10S, 304 stainless steel pipe in each of these locations, with an external heat transfer coefficient of 0.5 W/m², and then setting a fixed mass of 185 kg of water in the system, the operating pressure for fixed fluid mass and for fixed absorber outlet quality \( x = 0.80 \) are found as shown in Figure 8.7. This shows the expected increase in the pressure in the system following from a greater amount of heat being transferred by steam of the same quality but in the same sized space; the steam must be compressed in order for the required steady-state heat transfer to occur.

More detail will be added to the modeling to allow for the volume of the heat exchanger and surge tank as well as other pipe-work. Under the fixed-total-mass-of-water constraint, the system pressure was found to be quite sensitive to pump flow
Figure 8.7: Variation of operating pressure and system flow rate with irradiance
8.1.7 Conclusions and further work

A basic steady-state steam circuit model for the CLFR was presented, and operating conditions for the fixed-volume, fixed-mass case were investigated. The modelling results are plausible. The system-level effect of the Ledinegg flow instability was seen in the fact that significantly reduced pumping pressures occur as the absorber inlet pressure rises. The effect of irradiance on the system operating pressure was found for the fixed-volume, fixed-flow-rate case.

Solutions were obtained by an ad-hoc shooting technique [5], with Euler integration along the length of the absorber. This approach was found to be relatively straightforward to implement, although there are clear problems with generalising code from this approach to the intended transient model that will be required if start-up, shut-down and cloud-transient scenarios are to be investigated.

At this stage, the system model does not conveniently output information about two-phase flow regimes. This will need to be added before a transient model is developed since two-phase flow regimes at the absorber outlet during low-radiation periods are a concern, and the modelling will need to reflect this.

Line losses were not included in this initial model, but should be included, as there will be significant lengths of reticulating pipework in the final system.

8.2 Stage 2 design

The Stage 2 CLFR prototype design implements direct tie-in to the power station direct feedwater heating line. It eliminates the need for surge tank capacity, as mass can be displaced from the CLFR loop into the main power station loop, and it removes the possibility of, and need for, floating-pressure operation. The system to be modelled is shown in Figure 8.8.

8.2.1 Methodology

In order to progress slightly towards the ‘correct’ methodology for use in a closed-loop transient system, an effort was made to revisit the above results using a more flexible ‘systems’ approach, and by using a general-purpose solver for the simulation. The ASCEND modelling environment [171] was selected after number of other (free) alternatives were evaluated, as it provided block-decomposition of a system Jacobian matrix, which increases the ease of solving many such systems [129].

In taking the modelling into ASCEND, the earlier absorber model was used as-is, as a nested ‘black box relation’ in ASCEND, with the inputs being the design
variables and inlet flow properties, and the outputs being the outlet flow properties as well as mass holdup, exit flow regime and average pipe wall temperature. This allowed the existing tested validated code to be used, although there were certain arguments for reimplementing the model natively in equation form under ASCEND. The resolution of the nested model is controlled by parameters supplied in the ASCEND model file (for example, the number of numerical finite-difference flow segments along the length of the absorber).

Use of ASCEND here allowed the ‘wiring up’ to be done in a more flexible manner, and also permitted various system configurations to be tried out more easily. On the downside, solving the system this way sometimes requires a few steps; for a closed-loop model to converge with the Newton iterations, it must often first be solved in an open-loop configuration first. This is referred to as ‘creeping up on the solution’ [129].

The systems approach adopted in this Stage 2 work is much more adaptable for later-stage modelling of the CLFR; it provides an object-oriented, testable approach that will be familiar to engineers in the process modelling field. It also separates the task of specifying the problem from the task of solving it, which means that comparisons between solution methods and problem definitions can be made more reliably.

The model file resulting from the following work is given in Appendix G. Efforts have been made to design the simulation in such a way that all relevant parameters for the CLFR design are specified in this ASCEND model file, rather than hidden...
away as numerical constants buried in subroutines.

8.2.2 Component models

The following sections describe the components used in modelling the CLFR Stage 2 prototype system. These models were developed as ASCEND model objects and with the exception of the steam node, pipe and the absorber-pipe components, they were implemented directly as equations. The others were implemented as linked-in C and C++ code.

8.2.2.1 Steam node

For the calculation of steam properties, a single unified approach was adopted throughout the model. A *steam_node* model was defined, containing a flow rate $\dot{m}$, plus the usual thermodynamic properties $p$, $h$, $u$, $v$, $T$, $x$, and $s$ as well as the transport property $\mu$. These properties were then related using the ASCEND ‘black box’ functionality to link to the ‘freesteam’ IAPWS-IF97 steam properties library that was developed as part of the present work. All properties are evaluated in terms of $(p, h)$ which gives good convergence properties and reasonable decoupling of mass, energy and momentum effects, as pointed out by Hirsch et al [63]. For each *steam_node* model, there are 6 equations, 8 unknown fluid properties and 1 unknown flow rate, giving a total of three degrees of freedom that in each case are constrained by the equations of the component models in the sections that follow.

8.2.2.2 Pipe component

A general pipe flow component was developed which includes treatment of single-phase as well as two-phase pressure drops and heat transfer, using the same code as developed for Chapter 5. The pipe component also includes external heat loss, but, for the purpose of modelling reticulating pipework, the heat loss is modelled using a fixed linear heat transfer coefficient $h_{c,\text{ext}}$. Table 8.3 shows the set of fixed constants in the pipe model. The other parameters, treated by ASCEND as ‘variables’\(^1\), are shown in Table 8.4.

In modelling the CLFR Stage 2 prototype, the pipe component can be used for any short length of connecting pipework. The value of $h_{c,\text{ext}}$ is important for accurately modelling the amount of heat loss. The pipe component is also used for a longer length of pipe which is the reticulating pipe running from the bottom of the steam drum (see Figure 6.2) to the far end of the absorber array. The CLFR

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\(^1\)These can be either held constant, or can be ‘freed’ meaning that the ASCEND solver adds them to the list of unknown values for which to solve. For normal solving, one must fix one variable and free another, in order that the problem remains ‘square’.
Table 8.3: Fixed constants for the Pipe component used in the Stage 2 model

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>$h_{e,ext}$</td>
<td>external heat transfer coefficient</td>
</tr>
<tr>
<td>$c_{p,pipe}$</td>
<td>pipe heat capacity</td>
</tr>
<tr>
<td>$\epsilon_{pipe}$</td>
<td>absorber pipe absolute roughness (for friction in internal flow)</td>
</tr>
<tr>
<td>$k_{pipe}$</td>
<td>thermal conductivity of the absorber pipe material</td>
</tr>
<tr>
<td>$n_{segments}$</td>
<td>the number of finite-difference segments used within</td>
</tr>
<tr>
<td>$\rho_{pipe}$</td>
<td>mass density of the absorber pipe material</td>
</tr>
</tbody>
</table>

Table 8.4: Other parameters for the Pipe component used in the Stage 2 model (can be varied during a solution run, for example for optimisation or sensitivity studies)

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_{i}$</td>
<td>inside diameter of the absorber pipe</td>
</tr>
<tr>
<td>$D_{o}$</td>
<td>outside diameter of the absorber pipe</td>
</tr>
<tr>
<td>$I$</td>
<td>normal direct-beam solar irradiation (energy per area)</td>
</tr>
<tr>
<td>$L$</td>
<td>length of each absorber (and equivalently, the length of each absorber pipe)</td>
</tr>
<tr>
<td>$T_{amb}$</td>
<td>ambient temperature</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>the inlet flow rate</td>
</tr>
<tr>
<td>$x$</td>
<td>the exit steam quality</td>
</tr>
<tr>
<td>$p_{inlet}$</td>
<td>inlet pressure</td>
</tr>
<tr>
<td>$h_{inlet}$</td>
<td>inlet specific enthalpy</td>
</tr>
<tr>
<td>$p_{outlet}$</td>
<td>outlet pressure</td>
</tr>
<tr>
<td>$h_{outlet}$</td>
<td>outlet specific enthalpy</td>
</tr>
<tr>
<td>$m_{holdup}$</td>
<td>mass holdup in the absorber</td>
</tr>
</tbody>
</table>
Table 8.5: Constant-valued parameters in the Absorber Pipe component of the Stage 2 steady system model.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>CR</td>
<td>net concentration ratio at absorber plane (after optical losses)</td>
</tr>
<tr>
<td>(D_{cavity})</td>
<td>depth of the cavity (see Chapter 4)</td>
</tr>
<tr>
<td>(F_{rad})</td>
<td>equivalent view factor for internal heat transfer in the cavity</td>
</tr>
<tr>
<td>(K_{cavity})</td>
<td>magnitude factor in the Nusselt number correlation</td>
</tr>
<tr>
<td>(pow_{D/W})</td>
<td>power of the (D/W) factor in the Nusselt number correlation</td>
</tr>
<tr>
<td>(pow_{Gr})</td>
<td>power of the Grashof number in the Nusselt number correlation</td>
</tr>
<tr>
<td>(W_{cavity})</td>
<td>width of the cavity</td>
</tr>
<tr>
<td>(\epsilon_{absorber})</td>
<td>emissivity of the absorber pipes (after coating)</td>
</tr>
<tr>
<td>(\epsilon_{cover})</td>
<td>emissivity of the cover material at re-radiated wavelengths</td>
</tr>
<tr>
<td>(h_{c,cover})</td>
<td>convection coefficient on the outside of the cavity cover</td>
</tr>
<tr>
<td>(h_{c,sidewall})</td>
<td>convection coefficient of the cavity side-walls</td>
</tr>
<tr>
<td>(n_{absorbers})</td>
<td>the number of absorbers present in the array</td>
</tr>
<tr>
<td>(n_{pipes\ per\ absorber})</td>
<td>the number of ganged pipes inside each absorber cavity</td>
</tr>
</tbody>
</table>

Stage 2 prototype uses a once-through configuration with the heating being on the ‘return’ part of the out-and-back loop. By dividing the pipe model into segments, we attain an accurate calculation of pressure drops due to friction that accounts for changes in fluid properties with pressure.

With regard to thermal losses, it is considered that radiative losses will be second order for heat loss here, so the linear dependence on temperature difference is appropriate. The thermal losses are applied using

\[
\dot{q}_l \delta z = h_{c,ext} \pi D_o \delta z (T_1 - T_{amb})
\]

where \(T_1\) is the pipe wall temperature.

8.2.2.3 Absorber pipe component

The absorber pipe component is a specialisation of the above pipe component. Following the approaches from Chapter 4 and Chapter 5, the pipe external heat loss parameter \(h_{c,ext}\) from the pipe model in Section 8.2.2.2 is replaced by the set of fixed constants shown in Table 8.5. It is reiterated that these constants were calculated using the flat plate absorber model, and that a more accurate tube-bank absorber model would result in absorber pipe losses approximately 25% higher than calculated here. An additional output variable is also provided, \(T_{absorber}\), which is the average temperature of the absorber surface, along its length.

The standard configuration is to fix the inlet state \((p_{inlet}, h_{inlet})\) and flow-rate \(\dot{m}\) as well as the length and pipe size and ambient conditions, and then to solve for the exit state including \(x\), \(p_{inlet}\) and \(h_{inlet}\). Alternatively, one can fix the desired outlet
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8.2.2.4 Absorber component

A single CLFR absorber contains 12 absorber pipes. Rather than modelling this as separate parallel pipes (as was done for the case of two parallel pipes by Natan et al [113]), we here choose to model a single pipe then apply a multiplication factor to the resulting flow rate and absorbed energy values. This allows the realistic absorber model to be used without introducing instabilities of the type found by Natan et al. Additionally, an orifice plate is added to the inlet of the the absorber. The resulting component model is shown in Figure 8.9.

8.2.2.5 Absorber group component

In the Stage 2 design, three absorbers are joined together via a manifold at each end, and thereby share a single pump and steam drum (Figure 8.8). The entire CLFR is approximated using a model of a single absorber to which is assigned an appropriate slice of the total radiation and mass flow rate. The parameters for the

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These instabilities are however investigated in Section 8.2.3.1
resulting \textit{clfr} model are as for the absorber component, but the mass flow rates on
the inlet and outlet are many times greater, depending on the number of absorbers
and the number of pipes per absorber.

In order to model the effect of varying sunlight levels and varying optical per-
formance through the day, only the concentration ratio is provided as a parameter.
Optical modelling has not been incorporated into the present model, but could eas-
ily be done by connecting further models with the value of this concentration ratio
parameter (which would then become a variable).

\subsection{Pump component}

A pump model was written for ASCEND that allows generalised pump curves to
be fed into it. Pump curves were digitised and then curve-fit from the original
data shown in Appendix F. Pump curves are translated from the plotted reference
conditions (values with subscript ‘1’) to the actual operating conditions (sans sub-
script) using the speed ratio $N_r = \omega_{\text{operating}}/\omega_{\text{reference}}$, and the following similarity
relations:

\begin{align*}
\dot{V} &= N_r \dot{V}_1 \\
\dot{W} &= N_r^2 \dot{W}_1 \\
H &= N_r^2 H_1 \\
\eta &= \eta_1
\end{align*}

where $\dot{V}$ is the volumetric flow-rate, $\dot{W}$ is the shaft power, $H$ is the pump head
and $\eta$ is the first law efficiency, hence

$$\eta \dot{W} = \dot{m}(h_{\text{out}} - h_{\text{in}})$$

For the current pump, the curves used are

\begin{align*}
H_1/[1 \text{ m}] &= 171.91 - 0.0349789 \dot{V}_1 - 0.00193442 \dot{V}_1^2 \\
\dot{W}_1/[1 \text{ kW}] &= 28.2514 + 0.120906 \dot{V}_1 + 0.00360997 \dot{V}_1^2 \\
&\quad - 2.80389 \times 10^{-5} \dot{V}_1^3 + 6.91525 \times 10^{-8} \dot{V}_1^4 \\
\eta \times 100 &= 1.35898 \dot{V}_1 - 0.0124358 \dot{V}_1^2 \\
&\quad + 5.85841 \times 10^{-5} \dot{V}_1^3 - 1.26982 \times 10^{-7} \dot{V}_1^4
\end{align*}

where the pump speed in the above has been normalised to
8.2. STAGE 2 DESIGN

\[ \dot{V}_{IN} = \dot{V}/[1 \text{ m}^3/\text{h}] \]

8.2.2.7 Separator and mixer components

The remaining item in the CLFR Stage 2 prototype that needs modelling is the steam drum which is planned to have the configuration shown Figure 8.10. An accurate model of the steam drum requires allowing for mass transfer between phases: the water in the bottom of the drum is being replenished at 150 °C from feedwater from the plant whereas the steam is at the saturation temperature of 253 °C at 42 bar. However, the extent to which such mass transfer occurs would depend on the design of the steam drum. One could configure it as a kettle re-boiler, which would ensure that the water was heated right up to 253 °C, or one could configure it so that the steam separation occurs in a completely separate chamber.

For the present analysis, we will approximate the steam drum by two simplified components, the separator and the mixer.

Separator The separator is configured with a single inlet stream and two outlet streams. Any gas component of the inlet flow is sent to the first outlet stream; any liquid goes to the second outlet stream. The inlet stream corresponds to the steam supply from the absorber array; the first outlet corresponds to the steam line to the power plant and the second outlet corresponds to the flow of water into the bottom of the steam drum.

In the context of a nonlinear equation-based solver, the separation calculation requires some ‘switching’ logic, since the outlet enthalpies can be either the phase saturation enthalpy \((h_f \text{ or } h_g)\) or the overall fluid enthalpy \(h\) depending on whether the inlet flow is saturated, subcooled or superheated. This fact makes modelling the separator in simple equation-based form a little difficult, although a work-around was found possible using absolute-value expressions:

\[ h_{out,1} = h_{in} - \frac{1}{2} ((h_{in} - h_g) - |h_{in} - h_g|) \]

which has the desired result that \(h_{out,1} = h_g\) for cases where \(h_{in} < h_g\) and \(h_{out,1} = h\) for cases where \(h_{in} > h_h\). A similar expression is used for \(h_{out,2}\) using \(h_f\).

The mass flow rates assigned to each stream are calculated directly using the value of the inlet steam quality. The pressures at the inlet and the two outlets are set to be equal.

Mixer The mixer has two inlet streams and a single outlet stream. The first inlet corresponds to the liquid stream from the separator above. The second inlet corresponds to the feedwater line from the power plant. As with the separator, various combinations of phases are possible on the inlets, but the calculation is
Figure 8.10: Steam drum configuration for the CLFR stage two prototype. The steam drum is mounted up in the air, above the level of the absorber, so that the absorber will be filled by gravity from water in the steam drum. Each steam drum is shared by a set of three linear absorbers.

much simpler, as we can just use mass and energy balance to determine the outlet flow rate and specific enthalpy. Again, it is assumed that pressures at all ports are equal.

### 8.2.2.8 Control valve

Downstream of the pump a control valve is being used to regulate flow in the CLFR Stage 2 prototype. This component is modelled with a variable K-factor loss. No special treatment of two-phase behaviour is required as it is not considered possible that the valve could be exposed to two-phase flow. The equations determining the behaviour of the control valve are therefore:

\[
\begin{align*}
    h_{in} &= h_{out} \\
    \dot{V} &= K \sqrt{\Delta p} \\
    \dot{m}_{in} &= \dot{m}_{out}
\end{align*}
\]
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8.2.2.9 Orifice plate

An orifice plate is present in the CLFR Stage 2 Prototype design, although at the time of writing its optimal size was not known. The equations for a corner-tapped orifice plate from ISO 5167 are used (as given by Mohitpour et al [104]), with the resulting equations for the orifice plate being

\[ \dot{m}_{in} = \dot{m}_{out} \]
\[ h_{in} = h_{out} \]
\[ \left( \frac{\dot{V}}{C A_{orif}} \right)^2 = -2(D\Delta p)\nu_{in} \]
\[ C = \frac{C_d}{\sqrt{1 - \beta^4}} \]
\[ \beta = \frac{D_{orif}}{D_{pipe}} \]
\[ C_d = 0.5959 + 0.312\beta^{2.1} - 0.1840\beta^8 + 0.0029\beta^{2.5} \left( \frac{10^6}{Re_D} \right)^{0.75} \]
\[ Re_D = \frac{\dot{m}_{in}}{\frac{1}{2}D_{pipe}\mu_{in}} \]

In the CLFR Stage 2 prototype, the orifice is present in a DN 100 Schedule 10S pipe just upstream of the absorber inlet manifold. In order to have a pressure drop across the orifice of the order of 1 bar at 36 kg/s, 150 °C, 42 bar, the required diameter is 0.2177. It is not clear yet what orifice size is necessary to stabilise the macro-scale Ledinegg instability in the absorber flow.

8.2.3 Integration of system components

The system configuration was shown in Figure 8.8. The above component models are used to create an equivalent model system as shown in Figure 8.11.

8.2.3.1 Removing the Ledinegg instability

In the previous chapter, it was seen that the Ledinegg instability was likely to occur at expected operating pressures of ~40 bar, provided the inlet temperatures were below 200 °C. Here we will use the absorber component described above, which includes an orifice plate, to determine the size of the orifice plate that is required to eliminate that instability. We will initially assume an inlet flow rate of 44 bar in all cases, in order simplify the calculation. In the earlier work it was seen that the Ledinegg instability increased slightly with increasing pressure, so this should be conservative.
CHAPTER 8. OVERALL SYSTEM MODELLING

Figure 8.11: Model system configuration. The multiple absorbers of Figure 8.8 are replaced by a single absorber ‘bracketed’ by a mass flow-rate divider and multiplier, and the steam drum is approximated by a separator and a mixer. Note that the absorber model, in turn, contains the \texttt{absorber\_pipe} component, as each absorber is composed of 12 pipes.

Figure 8.12 shows some pressure drop versus mass flow rate curves with the orifice plate added. Each plot is for a given inlet pressure and temperature, with a number of curves shown for different values of the orifice ratio $\beta$ (Section 8.2.2.9). The effect of the orifice plate is to add a parabolic pressure drop curve onto the two-phase pressure drop curve with its inflexion point. When the orifice is made sufficiently small, the inflexion is removed.

In Figure 8.12 (a), a 44 bar, 175 °C inlet (roughly standard operating conditions with full sun) is seen to require an orifice plate with $\beta \lesssim 0.40$.

In Figure 8.12 (b), a 44 bar, 120 °C inlet (which corresponds approximately to start-up conditions of the CLFR), a smaller orifice must be used, with $\beta \lesssim 0.325$.

In Figure 8.12 (c) and (d), the same temperatures are shown but with a lower pressure of 34 bar. In this case, the required orifice diameters are about the same: $\beta \lesssim 0.45$ for 175 °C and $\beta \lesssim 0.325$ for 120 °C.

Finally, consider the case where the inlet temperature rises to 200 °C and 225 °C (Figure 8.13), and keep the inlet pressure at 44 bar. We see that this allows the orifice to become much larger: $\beta \lesssim 0.6$ at 200 °C, and none required at 225 °C. The reason for this is that we are getting close to the saturation temperature (256 °C at 44 bar), and a significant fraction of the pipe needs to contain subcooled flow before the Ledinegg instability is seen.

From the above analysis, a sensible choice of orifice ratio appears to be $\beta = 0.325$, 

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Figure 8.12: Study of the effect of orifice ratio $\beta$ on the pressure-drop-vs-flow-rate curve. This shows that the Ledinegg stability seen in Section 5.3.9 is eliminated using an orifice plate with $\beta \approx 0.30$ for the conditions described in Section 8.2.3.1. The solar irradiation is 1000 W/m$^2$ in all cases.
Figure 8.13: Further study of the effect of orifice ratio \( \beta \) on the pressure-drop-vs-flow-rate curve, with inlet temperatures at 200 °C and 225 °C. At these higher inlet temperatures, much less constrictive orifice plates can be used.
as this will prevent instabilities over the full range of operating conditions including the ‘warm up’ state, where the steam drum is heated to 120 °C by plant steam. By removing the Ledinegg instability, we make the task of controlling the CLFR system much more straightforward: firstly, because there will be resistance to greater rates of flow in all pipe branches, flow will equalise between the branches; secondly, an monotonic $\Delta p$ vs $\dot{m}$ curve allows the mass flow rate to be calculated from the pressure drop, which reduces the necessary instrumentation and permits simpler feedback-based control mechanisms.

There is a remaining issue, which is that the current configuration of absorber and orifice places the orifice plates upstream of the absorber manifold. There is still the distinct possibility of instability occurring between the individual absorber pipes in the way described by Natan et al [113]. At the CLFR Stage 2 prototype, it has not been possible to place orifice plates upstream of each absorber pipe, however instrumentation should be added so that fluctuations in individual absorber pipe pressures and flow rates can be monitored. This is a central concern for the design of the CLFR control system, and if any problems are discovered, it will be important that they can be monitored and corrected before a full-scale system system design is finalised.

Finally, it is noted that with the orifice plate at $\beta = 0.325$, and at the expected operating conditions, with the absorber outlet adjusted to $x = 0.8$ and $p = 42$ bar, with inlet at 175 °C, the required inlet pressure is 43.53 bar, with a pressure drop of 0.145 bar over the orifice plate and 1.38 bar over the absorber. This calculation did not include minor losses.

### 8.2.3.2 Pump duty calculation and addition of minor losses

In above section, an orifice plate has been selected for use in the CLFR Stage 2 system. Next the remaining pressure drops must be investigated so that the pump duty can be estimated. We first model the absorber with associated pipework as shown in the upper portion of Figure 8.14 (reticulating pipe PI1 and absorber group AG).

An approximate set of minor losses (abbreviated ML) was added to the system model as shown in Table 8.6. No consideration of vertical distances was made\(^3\). Minor losses are lumped at three locations: upstream of the orifice (ML1), upstream of the steam drum (ML2) and upstream of the pump (ML3). Thermal losses from the pipe sections are considered with a constant heat transfer coefficient of 0.5 W/m\(^2\)K. Thermal losses from the absorber are as modelled in Chapter 6. The orifice with

\(^3\)The down-comer from the steam drum would have a pressure drop almost opposite to that in the riser at the far end of the array, so the effect of the vertical distance would be second- or third-order only in its effect on pipe friction in the reticulating pipe and the pump behaviour.
Figure 8.14: Pipe configuration for the pump duty design. Minor losses are lumped at three locations; Pipe PI2 is the pipe from the absorber outlet to the steam drum; its length depends which absorber it comes from.

Table 8.6: Minor losses for the CLFR Stage 2 prototype loop

<table>
<thead>
<tr>
<th>Pipe branch</th>
<th>Label</th>
<th>DN</th>
<th>Elbows</th>
<th>Pipe entry</th>
<th>Pipe exit</th>
<th>K factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Upstream of orifice</td>
<td>ML1</td>
<td>100</td>
<td>4</td>
<td>1</td>
<td>1</td>
<td>1.64</td>
</tr>
<tr>
<td>Inlet to steam drum (long)</td>
<td>ML2</td>
<td>150</td>
<td>4</td>
<td>1</td>
<td>1</td>
<td>1.57</td>
</tr>
<tr>
<td>Inlet to steam drum (short)</td>
<td>ML2s</td>
<td>150</td>
<td>2</td>
<td>1</td>
<td>1</td>
<td>1.07</td>
</tr>
<tr>
<td>Inlet to pump</td>
<td>ML3</td>
<td>150</td>
<td>3</td>
<td>1</td>
<td></td>
<td>1.27</td>
</tr>
</tbody>
</table>

\( \beta = 0.325 \) is as calculated above.

This model can be solved for the desired operating conditions with an outlet steam quality of 0.8 and outlet pressure of 42 bar, given an inlet temperature of 175 °C. The solution for a range of solar irradiation levels is shown in Figure 8.15. The pressure drops across the different components at the 1000 W/m² operating point are shown in 8.7.

From these results we can see that the pump and control valve assembly must be able to supply 7.735 kg/s at an outlet pressure of 43.107 bar. Assuming a 42 bar

Table 8.7: Component pressure drops at standard operating point, \( \dot{m} = 7.735 \text{ kg/s}. \)

<table>
<thead>
<tr>
<th>Component</th>
<th>Label</th>
<th>Pressure drop</th>
<th>Inlet pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pipe</td>
<td>PI1</td>
<td>-0.036</td>
<td>43.107</td>
</tr>
<tr>
<td>Minor loss</td>
<td>ML1</td>
<td>-0.001</td>
<td>43.071</td>
</tr>
<tr>
<td>Orifice</td>
<td>OR</td>
<td>-0.099</td>
<td>43.070</td>
</tr>
<tr>
<td>Absorber pipe</td>
<td>AB</td>
<td>-0.906</td>
<td>42.971</td>
</tr>
<tr>
<td>Pipe</td>
<td>PI2</td>
<td>-0.041</td>
<td>42.064</td>
</tr>
<tr>
<td>Minor loss</td>
<td>ML2</td>
<td>-0.024</td>
<td>42.026</td>
</tr>
</tbody>
</table>
8.2. STAGE 2 DESIGN

Figure 8.15: Inlet pressure and total flow rate for the open-loop system with inlet temperature of 175 °C, outlet pressure of 42 bar and outlet quality of 0.8.

inlet pressure, the pump and control valve together will have a pressure rise across them of 1.107 bar. Clearly the pipework is performing well; by far the greatest pressure drop is in the absorber pipe, which by necessity is small; other pipework has been sized to keep pressure drops low. At the flow rate shown here, the orifice is experiencing only quite a small pressure drop, although we know from Section 8.2.3.1 that the pressure drop on this item rises significantly if the flow rate increases.

8.2.3.3 An estimate size for the control valve pressure drop

A control valve was needed in the Stage 2 design as part of the conservative design process required by the power station operations company. The control valve gives some scope for some active control in the case that the orifice is not sufficient to eliminate the Ledinegg instability. It would be hoped that a later-stage design could remove the control valve and use only a pump with a variable speed drive.

The rule of thumb for control valve sizing is that the pressure drop across them should be between 20% and 50% of the flow around the whole circuit [43]. As we are being conservative here, and this is a prototype with many uncertainties, we choose 50%, giving a pressure drop of 0.553 bar across the control valve. If we approximate the control valve by an orifice plate as described in Section 8.2.2.9, and assume a pipe size of DN 150 SCH 10S, then calculation shows that the desired pressure drop
and outlet pressure of 43.107 bar, inlet temperature 175 °C, and flow-rate 7.735 kg/s, gives a required orifice ratio $\beta = 0.254$.

With this pressure drop added to the circuit, the pump must now provide 7.735 kg/s with a pressure rise of 1.615 bar, and outlet pressure of 43.614 bar. Using the pump proposed for installation in the Stage 2 CLFR prototype, this corresponds to running the pump at a 34.7% speed ratio (1031 RPM).

We note here that the pump selected for installation in the Stage 2 prototype is overpowered for the expected load. This is a result of the very conservative design process required by the power station operations company. There is uncertainty regarding the accuracy of the two-phase flow correlations, and questions of system controllability that need to be resolved. A programme of experiments and measurement is planned for the Stage 2 prototype, but has not been achievable within the scope of this thesis. Once Stage 2 is operational, there will be experimental data to permit a more optimal and hence more efficient design to be selected with confidence.

### 8.2.3.4 Effect of differences in flow path length

The outlets of the absorbers are connected by pipe P12 to the steam drum; there are three absorbers per steam drum. The absorbers further from the steam drum have a flow path length of 62 m, as was modelled in the preceding sections; the one next to the steam drum has a flow path of only 5 m, and reduced minor losses also ($ML_2 / ML_{2s}$). It is a concern that these differing flow paths are present in the system, so we here evaluate the difference in flow rate and exit conditions if an equal pressure drop is assumed across the two branches.

Table 8.8 shows the result of changing the pipe configuration and then solving for specified outlet conditions of 0.8 quality at 42 bar, given an inlet of 175 °C, by varying the inlet pressure and flow rate. The change in flow rates and inlet pressure drop is seen to be small.

Table 8.9 shows a different case. Here the first row is as for Table 8.8, but the second row gives the case where the flow-rate is varied such that the inlet and outlet pressures for the short pipe configuration match the pressures for the long pipe configuration. We see in this case that a surprisingly large 7% change in flow-rates is required to match the pressure drops. The reason for this is seen in the second table: shortening the pipe between the absorber and the steam drum causes a 0.04 bar pressure drop deficit which can only be made up by a change in flow rate. However, the Ledinegg instability ensures that increasing the flow rate decreases the pressure drop across the absorber. So quite a large flow rate change is required before the pressure drop over the orifice plate is able to make up the deficit.

The difference seen here in exit quality between the two flow paths (0.80 from
8.2. STAGE 2 DESIGN

Table 8.8: Flow-path comparison with long and short pipes between absorber and steam drum. Inlet pressure and flow rate were adjusted to give equal outlet conditions of 0.8 quality and 42 bar, given inlet of 175 °C.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Flow rate</th>
<th>Inlet pressure</th>
<th>Mass holdup</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>kg/s</td>
<td>bar</td>
<td>kg</td>
</tr>
<tr>
<td>62m pipe, K=1.57</td>
<td>2.5806</td>
<td>43.073</td>
<td>1076.0</td>
</tr>
<tr>
<td>5m pipe, K=1.07</td>
<td>2.5828</td>
<td>43.030</td>
<td>1049.1</td>
</tr>
</tbody>
</table>

Table 8.9: Flow-path comparison with long and short pipes between absorber and steam drum. For the first case, inlet and outlet pressures were determined as in the above Table 8.8. For the second case, with the shorter exit pipe, the mass flow rate was adjusted until the same overall pressure drop was seen.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Flow rate</th>
<th>Exit pressure</th>
<th>Pressure drops</th>
<th>Mass</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>kg/s</td>
<td>bar</td>
<td>bar</td>
<td>kg</td>
</tr>
<tr>
<td>62m pipe, K=1.57</td>
<td>2.5806</td>
<td>0.800</td>
<td>-0.001</td>
<td>-1.008</td>
</tr>
<tr>
<td>5m pipe, K=1.07</td>
<td>2.7880</td>
<td>0.724</td>
<td>-0.001</td>
<td>-0.936</td>
</tr>
</tbody>
</table>

Analogous for the long path, but 0.72 from the short path) is enough to warrant some redesign. A more constrictive orifice would be advisable on the short flow path. This area deserves some experimental investigation when the Stage 2 prototype is completed.

8.2.3.5 Closed loop operating conditions

Up until this point, only open-loop solutions have been sought. Here we add the missing steam drum component, approximated by two much-simplified components: a separator and a mixer, as described in 8.2.2.7.\(^4\)

Solving the closed loop model was not possible with only the Newton nonlinear algebraic solver. Three outer-loop Brent solvers were added, effectively ‘tearing’ the system of equations on \(p\), \(h\) and \(\dot{m}\) at the pump inlet\(^5\). Attempting to converge the whole system solution with the Newton solver was not successful, but solving the torn system in this way using an ‘outer loop’ approach worked reliably.\(^6\)

\(^4\) Because this is a steady state model, the mass holdup of the steam drum can not be predicted; this will require dynamic modelling.
\(^5\) ‘tearing’ is described by Piela and Westerberg [129], Brent solvers are described by Press et al [130].
\(^6\) One possible reason for the difficulty is that many elements of the models involved ‘black box’ models for which the ASCEND modelling does not have access to exact derivatives. ASCEND must instead use finite-difference estimates of the derivatives, which carries with it some destabilising numerical error. Attempts were made to implement equation-based replacements for the black-box components, but this was overly limiting in the case of thermodynamic properties: either one had to restrict oneself to a single phase region, or else use a more complex ‘conditional solver’, such as CMSlv in ASCEND [145]. This looks like an appealing option for further work. Other workers have used closed-source commercial software packages to tackle these problems, but these were not available for the present work (Hirsch et al [63], Rheinländer et al [144]).
Firstly the system was solved for the expected operating point with a plausible set of conditions at the pump inlet. Then the Brent solvers were applied as follows:

1. Adjust pump inlet enthalpy to achieve equal enthalpies across the tear
2. Adjust pump flow rate to achieve desired exit steam quality of 0.8
3. Adjust pump speed to achieve equal pressures across the tear

This approach gave the results shown in Table 8.10. The small change in some values from the earlier open-loop model can be explained by the fact that the pump inlet conditions changed slightly in order to balance properties around the loop. The collector efficiency here is as defined in Chapter 6. The system efficiency is defined as

$$\eta_{\text{sys}} = \frac{Q_{\text{absorbed}} - \frac{W_{\text{pump}}}{\eta_{\text{tm}}}}{Q_{\text{incident}}}$$

where $\eta_{\text{tm}}$ is the power-station-and-pump thermal-to-mechanical efficiency, taken to be 0.3. This approximate value makes an allowance for the fact that pump work is provided here by an electric motor, and that motor uses power ultimately derived from the thermodynamic cycle to which the CLFR output heat is being provided.

It can be seen that, at the current design point, the pumping work for the CLFR Stage 2 prototype is a very small energy flow. By contrast, in the modelling by Reynolds [142] and Odeh [115], it was seen that pump work was significant and led to a clearly defined optimum operating point. Essentially, the additional energy gathered by extending the absorber length will *not* be offset by the increased energy required to pump the fluid, so it appears that a longer absorber should be used, flow stability issues notwithstanding. A multi-pass configuration, an option already under consideration for the next stage of the project, appears to be a good solution, as this eliminates the need for the reticulating pipe at the same time, hence increasing the efficiency *and* reducing the system cost.

### 8.2.3.6 Effect of varying the outlet quality set-point

A key control parameter available in the CLFR design is the exit steam quality. In Chapter 6 it was seen that for the open-loop cases modelled there,

- a raised absorber inlet temperature gave better collector efficiency
- a higher outlet steam quality gives better collector efficiency

Both of these effects appear to be due to the fact that a smaller region subcooled flow leads to higher efficiency. The reason for this is that in subcooled flow, the
8.2. *STAGE 2 DESIGN*

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flow rate in the loop</td>
<td>7.66 kg/s</td>
</tr>
<tr>
<td>Collector efficiency</td>
<td>91.94%</td>
</tr>
<tr>
<td>System efficiency*</td>
<td>91.89%</td>
</tr>
<tr>
<td>Pump pressure difference</td>
<td>1.576 bar</td>
</tr>
<tr>
<td>Pump efficiency**</td>
<td>55.9%</td>
</tr>
<tr>
<td>Volumetric flow-rate at pump</td>
<td>8.53 L/s</td>
</tr>
<tr>
<td>Pump head</td>
<td>17.9 m</td>
</tr>
<tr>
<td>Control valve pressure drop</td>
<td>-0.495 bar</td>
</tr>
<tr>
<td>Net absorbed heat</td>
<td>13,285 kW</td>
</tr>
<tr>
<td>Heat loss from each absorber</td>
<td>1153 kW</td>
</tr>
<tr>
<td>Pressure drop across orifice</td>
<td>-0.097 bar</td>
</tr>
<tr>
<td>Pressure drop across absorber pipe</td>
<td>-0.884 bar</td>
</tr>
<tr>
<td>Temperature at pump inlet</td>
<td>445 K</td>
</tr>
<tr>
<td>Feedwater flow-rate</td>
<td>6.13 kg/s</td>
</tr>
<tr>
<td>Feedwater inlet temperature</td>
<td>423 K</td>
</tr>
<tr>
<td>Enthalpy at feedwater outlet</td>
<td>2800 K</td>
</tr>
<tr>
<td>Temperature at feedwater outlet</td>
<td>526 K</td>
</tr>
<tr>
<td>Pump work</td>
<td>2.1 kW</td>
</tr>
<tr>
<td>Heat loss from reticulating pipe</td>
<td>11.2 kW</td>
</tr>
</tbody>
</table>

* Allowing for pumping energy, assuming a power station thermal-to-mechanical efficiency of 0.3

** From the pump curve in F
internal heat transfer coefficient is much lower than for two-phase flow, and this has the result that the absorber pipe temperature rises and gives higher heat loss.

In the closed-loop configuration, the above trends are in opposite directions, because higher outlet steam quality means dryer outlet steam, so there is less vapour in the steam acting to preheat the feedwater, which means lowered absorber inlet temperature.

To determine which of these competing tendencies resulted in the better collector efficiency, several different exit steam quality set-points were solved using the technique of the previous section. The results, shown Table 8.11, show that the heat loss from the absorber increases slightly as the exit quality decreases, and the pumping energy also increases owing to the higher flow rates for lower exit quality. The result is that there is just a slight (+0.3%) efficiency improvement gained by choosing a higher exit steam-quality set-point (such as 0.95), compared to a low one (0.5), or a 0.1% improvement if moving the steam quality from 0.80 to 0.95.

<table>
<thead>
<tr>
<th>$x$</th>
<th>$\dot{m}_{PU}$</th>
<th>$W_{PU}$</th>
<th>$Q_{loss}$</th>
<th>$p_{AB,inlet}$</th>
<th>$T_{AB,inlet}$</th>
<th>$\eta_{sys}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>kg/s</td>
<td>kW</td>
<td>kW/absorber</td>
<td>bar</td>
<td>K</td>
<td></td>
</tr>
<tr>
<td>0.50</td>
<td>12.25</td>
<td>7.00</td>
<td>1169.82</td>
<td>43.73</td>
<td>476.22</td>
<td>0.917</td>
</tr>
<tr>
<td>0.65</td>
<td>9.43</td>
<td>3.54</td>
<td>1160.42</td>
<td>43.30</td>
<td>460.42</td>
<td>0.918</td>
</tr>
<tr>
<td>0.80</td>
<td>7.66</td>
<td>2.10</td>
<td>1152.84</td>
<td>43.04</td>
<td>444.42</td>
<td>0.919</td>
</tr>
<tr>
<td>0.95</td>
<td>6.46</td>
<td>1.37</td>
<td>1144.81</td>
<td>42.87</td>
<td>428.23</td>
<td>0.920</td>
</tr>
</tbody>
</table>

It should be noted that these results depend closely on the simple approximation to the steam drum that has been used here. The model does not allow for any heat transfer from the dry steam to the feed water. If this heat transfer were allowed, the absorber inlet temperature would rise higher and might possibly alter the efficiency trend.

### 8.3 Conclusions

Steady-state system models were constructed for two different CLFR system concepts, incorporating the two-phase flow absorber model of the previous chapter as well as pumps, pipework and other components. The first of the two system concepts is a fixed-mass, fixed-volume flow circuit. It was shown that the operating pressure can vary significantly with solar radiation in this design. Concerns about the controllability of this fixed-mass flow loop, as well as the need for an expensive heat exchanger unit, ultimately led to investigation of the second concept.

In the second CLFR system concept, the boiler feedwater loop passes directly through the array; the system is no longer fixed-mass, and the heat exchanger is
eliminated. In recognition of the modelling difficulties found with the fixed-mass flow-loop, this second model was implemented using an object-orient modelling language that separated the model description from the solution method. The new model was then used to address a range of design questions for the Stage 2 design.

A process was described for selecting an orifice plate to eliminate the Ledinegg instability from the two-phase absorber flow.

The pump duty was calculated and it was found that the pump intended for installation in the Stage 2 prototype is likely to be significantly overpowered, for reasons of conservative design.

It was shown that the difference in flow path lengths between the absorbers and their shared steam drum could be a cause for concern and will require some experimental investigation.

Closed-loop modelling showed that the CLFR Stage 2 collector is expected to have a 91.9% efficiency under peak solar conditions, not including optical losses. It was found that, subject to some assumptions about the behaviour of the steam drum, only slightly higher (+0.1%) system efficiencies were predicted at higher exit steam quality.
Chapter 9

Conclusions and recommendations

9.1 Conclusions

The CLFR solar collector has been shown to be a promising new concept for large-scale solar thermal energy generation. A second stage prototype nearing completion at Liddell power station will provide some proof of this concept and will provide important experimental data required for further modelling and design work.

In this thesis, a detailed thermodynamic model of the CLFR collector was created that includes thermal losses from the cavity receiver, and the two-phase flow-boiling process in the pipes running along the length of the absorber.

Thermal losses from the cavity receiver heat loss were modelled using a series of computational fluid dynamics simulations. A Nusselt-Grashof correlation for the convective heat loss was found, and together with a gray-body radiative model for the cavity interior and simple external heat loss relationships, permits estimation of the steady-state cavity heat loss knowing only the absorber temperature, ambient temperature, and surface properties.

The absorber pipe-flow model models the way in which heat is transferred to water in the pipes, and allows prediction of pressure drops through the CLFR. It uses the Friedel two-phase flow pressure drop correlation, Kandlikar flow-boiling heat transfer correlation, and uses industry-standard IAPWS-IF97 steam property correlations. The model was validated using experimental data from the DISS project, and showed good agreement for a range of cases.

The absorber pipe-flow model and cavity heat loss model were combined with ancillary equipment models to create two separate full-system models for the CLFR up to the point of tie-in to the existing power station cycle. The current CLFR project is a ‘coal saver’ add-on to the Liddell power station, and this was the focus
of modelling work performed here. Parametric studies were performed to investigate closed-loop behaviour of the CLFR collector in the full-system configuration. A study of the conditions leading to Ledinegg instability in absorber pipes was made, and this was used to determine suitable sizes for orifices to be placed upstream of the absorber. Pump and control-valve sizing was also investigated.

In the present work, the migration from ad-hoc sequential-modular modelling to the use of an equation-based simulation environment was made, which will facilitate re-use of the models developed. The software library for steam property calculation using the industrial IAPWS-IF97 steam tables was also released into the public domain as part of the present work.

For the purpose of predicting site conditions, some analysis of available solar irradiation data was performed. A method for predicting the hourly beam component of solar radiation for Liddell power station was determined, with some weakness in existing methods identified.

A transient model of the CLFR was developed using homogeneous two-phase flow and a stationary momentum equation. This transient model gives the expected dynamic response to sudden increases in solar irradiation received by the collector and will serve as a basis for further dynamic modelling.

### 9.2 Recommendations

The study of solar radiation data showed that there is no substitute for ground-based instrumentation, and this should certainly be installed at the prototype location before active testing commences. Experience with the Solar One system indicates that calculations based on inaccurate solar data can cause errors in energy output estimates, which can jeopardise ongoing financing.

The CFD modelling of the CLFR cavity receiver is another area that will benefit from experimental validation. It will be important for the purpose of future economic modelling to ensure that thermal losses from the CLFR are well quantified for the range of operating conditions. The cavity cover design selected for construction is not one of those that was modelling using CFD. Some further CFD modelling would allow an assessment of the accuracy of the modelling methods used here, and might also allow some optimisation of the current cavity cover design before the next construction stage. Radiation-only cavity modelling of a tube-bank absorber should be investigated in more detail, having predicted 25% higher thermal losses, and the results from this analysis should be worked through into the subsequent modelling work.

The transient modelling performed here gave a model that exhibited the expected response to step-change solar input, but in development of a complete model
9.2. RECOMMENDATIONS

with accurate thermodynamic properties some challenges were found. To progress further with the transient modelling, firstly, the thermodynamic properties should be implemented in a form that furnishes the integration algorithm with exact partial derivatives of each property with respect to each other property. Secondly, the integration algorithm needs to be given a boundary-crossing re-initialisation capability. Thirdly, the sparse matrix solution methods of IDA need to be connected to ASCEND to allow larger simulation models to be integrated efficiently. These steps will give more stable transient models and will eliminate problems relating to the numerical integration of non-smooth thermodynamic property relationships.

System level simulation performed here should be updated once detailed engineering has been completed, with a full treatment of all intermediate pipework and minor losses. With an accurate full-system model, it should be possible to perform annual simulations leading to high-quality economic analysis. With accurate design data, the Ledinegg instability should be further addressed. In particular, some of the concerns of Natan et al [113] regarding instability arising from multiple parallel tubes deserve investigation.
Bibliography


tional Association for the Properties of Water and Steam, Erlangen, Germany, September 1997.


Appendix A

Radiative heat transfer

The following is a brief review of radiative heat transfer serving as a point of reference for the chapter on cavity heat loss modelling.

A.1 Radiation emission

Surfaces emit energy in the form of ‘radiation’ as a result of energised atoms in their surface layers forming photon ‘energy packets’ that leave the surface before they can be reabsorbed. Photons have both wave-like and particle-like behaviour; in the case of mechanical engineering, the usual way to consider radiation is as a wave.

Radiation, viewed then as a wave, has a source and a destination, and has a wavelength $\lambda$. It also has a spectral intensity, which is a measure of the ability of the radiation to transfer energy from the source to the target:

**Spectral intensity** $I_{\lambda,e}(\lambda, \theta, \phi)$ Defined as the rate at which radiant energy $dq_\lambda$ is emitted at wavelength $\lambda$ in the $(\theta, \phi)$ direction, per unit area of the emitting surface normal to this direction $dA_n$, per unit solid angle $d\omega$ about this direction$^1$, and per unit wavelength about $\lambda$.

The area of the emitting surface normal to $(\theta, \phi)$ can be calculated as $dA_n = dA_1 \cos \theta$. Here, $\theta$ is the angle between the emitting surface normal and the radiation direction.

We can write the definition of spectral intensity therefore in equation form:

$$I_{\lambda,e}(\lambda, \theta, \phi) = \frac{dq_\lambda}{d\omega \cdot dA_n \cdot d\lambda}$$

or, substituting $A_n$,

---

$^1$A solid angle $\omega$ of a surface $S$ with respect to a point $P$ is the area projected by all rays from $P$ to $S$ onto a unit sphere centred at $P$. A hemisphere about $P$ therefore has a solid angle of $2\pi$ and a sphere about $P$ has a solid angle of $4\pi$. 

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If we then define the rate of heat transfer per wavelength and per source area \(q''_\lambda\) as

\[
q''_\lambda = \frac{dq_\lambda}{d\lambda dA_1}
\]

then we have

\[
dq''_\lambda = I_{\lambda,e} \cdot d\omega \cdot \cos \theta
\]

which, choosing our definition of the solid angle \(d\omega\) as a square patch with angular ranges \(d\theta\) and \(d\phi\) at radius \(r\), can be written as

\[
dq''_\lambda = I_{\lambda,e} \cdot d\theta \sin \theta d\phi \cdot \cos \theta
\]

In order to have a finite value of heat transfer, we must integrate with respect to a finite solid angle \(\Omega\) before we can have finite values for heat transfer:

\[
q''_\lambda (\lambda) = \int_\Omega \int I_{\lambda,e} (\lambda, \theta, \phi) \sin \theta \cos \theta d\theta d\phi
\]

**Emissive power** \(E_\lambda(\lambda)\) Defined as the rate at which energy is emitted as radiation at wavelength \(\lambda\) in *all directions* from a surface, per unit surface area. It is therefore the integral above with \(\Omega = 2\pi\), a hemisphere:

\[
E_\lambda (\lambda) = \int_0^{2\pi} \int_0^{\pi/2} I_{\lambda,e} (\lambda, \theta, \phi) \sin \theta \cos \theta d\theta d\phi
\]

If we assume that the spectral intensity is uniform in all directions, then this integral becomes the

\[
E_\lambda (\lambda) = I_{\lambda,e} (\lambda) \int_0^{2\pi} d\phi \int_0^{\pi/2} \cos \theta \sin \theta d\theta = I_{\lambda,e} (\lambda) \cdot 2\pi \cdot \frac{1}{2} = \pi I_{\lambda,e} (\lambda)
\]

The **total hemispherical emissive power** is then

\[
E = \int_0^\infty E_\lambda (\lambda) \, d\lambda = \pi I_e
\]

where \(I_e\) is the **total intensity**, found by integrating \(I_{\lambda,e} (\lambda, \theta, \phi)\) with respect to wavelength, and over the hemisphere.
Irradiation $G_\lambda(\lambda)$ In the above, we were integrating over the solid angles surrounding the source of the radiation. If we integrate instead the incident radiation $I_{\lambda,i}(\lambda, \theta, \phi)$ around solid angles at the destination of the radiation, we obtain similar results:

$$G_\lambda(\lambda) = \int_{\phi=0}^{2\pi} \int_{\theta=0}^{\pi/2} I_{\lambda,i}(\lambda, \theta, \phi) \sin \theta \cos \theta \ d\theta \ d\phi$$

$$G_\lambda(\lambda) = \pi I_{\lambda,i}(\lambda)$$

$$G = \int_0^{\infty} G_\lambda(\lambda) \ d\lambda = \pi I_i$$

Radiosity $J_\lambda(\lambda)$ In the above, irradiation is the total incident radiation (from whatever sources), and emissive power is the total radiation leaving a surface due to surface emission. What is missing is the total energy leaving a surface due to both emission and radiation. This is defined as radiosity and has equations of the same form as above, as follows:

$$J_\lambda(\lambda) = \int_{\phi=0}^{2\pi} \int_{\theta=0}^{\pi/2} I_{\lambda,e+r}(\lambda, \theta, \phi) \sin \theta \cos \theta \ d\theta \ d\phi$$

$$J_\lambda(\lambda) = \pi I_{\lambda,e+r}(\lambda)$$

$$J = \int_0^{\infty} J_\lambda(\lambda) \ d\lambda = \pi I_{e+r}$$

This distinction becomes important when considering that reflected radiation will have the spectral distribution of its original source, whereas emitted radiation will have a spectral distribution particular to the local surface.

A.2 Blackbody emission

A blackbody is an idealised surface that (a) absorbs all radiation, without reflection, (b) emits ideally – no surface can emit more energy than a blackbody – and (c) is a diffuse emitter, uniform in all directions. Emission from a black body have a well defined mathematical spectral intensity and emissive power called the Planck distribution. The peak spectral intensity of the Planck distribution varies with temperature $T$ such that
\[ \lambda_{\text{max}} = \frac{C_3}{T} \]

where \( C_3 = 2897.8 \mu\text{m.K} \). The Planck distribution also has the property that the total emissive power follows the form

\[ E_b = \sigma T^4 \]

where \( \sigma \) is the Stefan-Boltzmann constant, \( \sigma = 5.670 \times 10^{-8} \text{W/m}^2\).

### A.3 Emission from real objects

**Emittance** is the ratio of real to blackbody spectral intensity. We can define spectral directional emissivity \( \epsilon_{\lambda,\theta} (\lambda, \theta, T) \), spectral emissivity \( \epsilon_{\lambda} (\lambda, T) \), and total hemispherical emissivity \( \epsilon (T) \) in this way.

Directionality of emissivity is often ignored, although different types of surfaces have different properties that affect the emissivity and larger angles \( \theta \gtrsim 45^\circ \). Nevertheless, hemispherical emissivity is usually within 5% of the normal emissivity for most non-conducting surfaces [75].

### A.4 Reflection, absorption, transmission

When irradiation reaches a surface, there is an energy balance between the incoming radiation and its ongoing forms, which can be divided into reflected, absorbed and transmitted components:

\[ G_\lambda = G_{\lambda,\text{reflected}} + G_{\lambda,\text{absorbed}} + G_{\lambda,\text{transmitted}} \]

**Absorptivity** We define absorptivity similarly to emissivity. It is the ratio of absorbed to total incident spectral intensity:

\[ \alpha_{\lambda,\theta} (\lambda, \theta, \phi) = \frac{I_{\lambda,\text{absorbed}} (\lambda, \theta, \phi)}{I_{\lambda,\text{i}} (\lambda, \theta, \phi)} \]

Similarly we define hemispherical spectral absorptivity \( \alpha_{\lambda} (\lambda) \) and total hemispherical absorptivity \( \alpha \). Integration of the above proves that

\[ \alpha = \frac{G_{\text{absorbed}}}{G} \]

Note that the value for total absorptivity \( \alpha \) is dependent on the spectral distribution: if the irradiation is in a narrow band that does not happen to absorb well, the value of \( \alpha \) for that situation will vary, even though the surface is a constant.
therefore define a 'reference' absorptivity as the absorptivity of the surface under blackbody radiation at 5800 K as the 'solar absorptivity' $\alpha_s$, because the sun's spectral distribution (at least outside the atmosphere) is similar to that of a blackbody at 5800 K.

**Reflectivity** Reflectivity is defined in the same way, such that

$$\rho_{\lambda}(\lambda) = \frac{G_{\lambda,\text{reflected}}(\lambda)}{G_{\lambda}(\lambda)}$$

and

$$\rho = \frac{G_{\text{reflected}}}{G}$$

Reflectivity is special in that it is quite common for reflected radiation to be reflected mirror-like so that the reflected radiation distribution $\rho_{\lambda,\theta}(\lambda, \theta, \phi)$ has a peak at the point $(-\theta, \phi)$. Rough surfaces on the other hand will tend to have near-uniform reflectivity: incoming radiation is then scattered and the surface reflection is diffuse.

**Transmissivity** Transmitted radiation continues through the interior of the material after passing through the surface. It is not absorbed at the surface and will, if the surface on the far side permits, pass right through the body. Transmissivity is defined as

$$\tau_{\lambda}(\lambda) = \frac{G_{\lambda,\text{transmitted}}(\lambda)}{G_{\lambda}(\lambda)}$$

and

$$\tau = \frac{G_{\text{transmitted}}}{G}$$

**Total of reflected, absorbed and transmitted irradiance** We can take the equation for $G_{\lambda}$ above and divide it by $G_{\lambda}$ to give

$$1 = \rho_{\lambda}(\lambda) + \alpha_{\lambda}(\lambda) + \tau_{\lambda}(\lambda)$$

and for the total hemispherical case

$$1 = \rho + \alpha + \tau$$
A.5 Kirchoff’s Law

Cavities  Although ’real’ surfaces never have exact blackbody properties, a cavity is a close approximation. In a cavity, we see repeated reflections, absorptions and re-emission from the internal surface. Repeated reflection, with a small absorption at each reflection, will ultimately result in all the radiation being absorbed. Once under thermal equilibrium, all parts of the cavity surface will have the same temperature, and so temperature is the only properties of the cavity: the internal surface properties don’t matter.

Real bodies inside an ideal cavity  If we imagine a group of small bodies inside a blackbody cavity, as above, and we assume that they are small enough not to ’break’ the blackbody properties of the cavity, then we can assume that once a state of thermal equilibrium has been acheived, the small bodies will all have temperatures $T$equal to that of the blackbody cavity wall, and they will all have total hemispherical irradiation following that of the cavity, so $G = E_b(T)$. For equilibrium, there must also then be zero net radiation transfer between bodies. For any given body $i$ then,

$$\alpha_i G A_i = E_i(T) A_i$$

Since $\alpha \leq 1$, hence $E_i(T) \leq E_b(T)$. We have put a real body in the cavity and concluded that it must have a lower emissive power than the blackbody. This helps to prove that a cavity behaves in a way similar to a blackbody.

Kirchoff’s law  Following on from above, Kirchoff’s law states that for all the real bodies in the cavity,

$$\frac{E_1(T)}{\alpha_1} = \ldots \frac{E_n(T)}{\alpha_n} = E_b(T)$$

This can also be written in terms of emissivities, by dividing through by $E_b(T)$:

$$\frac{\epsilon_1}{\alpha_1} = \ldots \frac{\epsilon_n}{\alpha_n} = 1$$

In other words for the real bodies inside the blackbody cavity,

$$\epsilon_i = \alpha_i$$

This is a direct result of the assumption that the bodies are subject to and in equilibrium with blackbody radiation. We can repeat the analysis to find that for all the surfaces $\epsilon_\lambda(\lambda) = \alpha_\lambda(\lambda)$ and that $\epsilon_{\lambda,\theta}(\lambda, \theta, \phi) = \alpha_{\lambda,\theta}(\lambda, \theta, \phi)$. These last results are less restrictive than the the original result. We can say that all real
surfaces reflect and absorb equally when talking about directional spectral radiation at a specific direction and a specific wavelength.

A.6 Grey surfaces

A grey surface has the property that the spectral absorptivity $\alpha_\lambda$ and spectral emissivity $\epsilon_\lambda$ are equal and not a function of $\lambda$. If we take this assumption and apply it to the equations defining total absorptivity and total emissivity,

$$\alpha = \frac{G_{\text{absorbed}}}{G} = \int_0^\infty \alpha_\lambda G_\lambda(\lambda) \, d\lambda = \frac{\alpha_\lambda \int_0^\infty G_\lambda(\lambda) \, d\lambda}{G} = \alpha_\lambda$$

and

$$\epsilon = \frac{E}{E_b(T)} = \int_0^\infty \frac{E_\lambda(\lambda) \, d\lambda}{E_b(T)} = \frac{\int_0^\infty \epsilon_\lambda E_{\lambda,b}(\lambda,T) \, d\lambda}{E_b(T)} = \epsilon_\lambda = \alpha_\lambda$$

Then we can conclude that $\epsilon = \alpha$.

In fact if we know that there is no significant part of the above integrals outside a wavelength interval $(\lambda_1, \lambda_2)$, for example if the emissive power and irradiation are both zero outside this interval, then that is sufficient for the above behaviour to hold, even if the emissivity and absorptivity are not constant and equal outside this interval.

Once we know that the grey surface assumption is valid, we will see that treatment of radiation transfer between bodies becomes simpler to model.

A.7 View factors

In order to know what part $\int G_{ji} dA_j$ of the overall radiation $\int J_i dA_i$ leaving an emitting surface $i$ reaches a target surface $j$, we need to calculate the irradiation at each point on the target surface (integrating over the solid angle occupied by the remote emitting surface) and then integrate the irradiation over all points on the target surface. In the case of diffuse emitters and absorbers, this integration turns out to be independent of the magnitude of $J_i$, and is purely a function of the geometric configuration of the two surfaces, and is called the view factor $F_{ij}$, defined as

$$F_{ij} = \frac{1}{A_i} \int_{A_i} \int_{A_j} \frac{\cos \theta_i \cos \theta_j}{\pi R^2} dA_i dA_j = \frac{\int G_{ji} dA_j}{\int J_i dA_i}$$

From the symmetry of the above relation we can observe that
\[ F_{ij} A_i = F_{ji} A_j \quad (A.1) \]

We can express the view factor definition more simply in terms of average irradiation and radiosity:

\[ F_{ij} = \frac{A_j \bar{G}_{ji}}{A_i J_i} \]

or equivalently,

\[ F_{ij} A_i J_i = A_j \bar{G}_{ji} \]

In an enclosure, the total radiation \( J_i \) emitted from surface \( i \) must be equal to the sum of the radiation incident on the surfaces \( j \) from \( i \). Hence

\[ \sum_{j=1}^{N} F_{ij} = 1 \]

Similarly, the total radiation received at a surface \( j \)

\[ \sum_{i=1}^{N} F_{ij} = 1 \]

We can divide up any geometry we wish into small, infinitesimal surface fragments and then use the above relations to compute the fractions of emitted radiation that travel between the various surface fragments. Analytic expressions exist for many simple surface configurations and are given in [75] and [31].

For a large set of interactive surfaces, we place the view factors in a matrix as shown:

\[
\begin{bmatrix}
F_{11} & F_{12} & \cdots & F_{1N} \\
F_{21} & F_{22} & \cdots & F_{2N} \\
\vdots & \vdots & \ddots & \vdots \\
F_{N1} & F_{N2} & \cdots & F_{NN}
\end{bmatrix}
\begin{bmatrix}
A_1 J_1 \\
A_2 J_2 \\
\vdots \\
A_N J_N
\end{bmatrix}
= 
\begin{bmatrix}
A_1 G_1 \\
A_2 G_2 \\
\vdots \\
A_N G_N
\end{bmatrix}
\quad (A.2)
\]

Using the above symmetry relation, the calculation of the upper triangle of the matrix can be avoided, since

\[ F_{ji} = F_{ij} \frac{A_i}{A_j} \]

The above techniques are used in the FLUENT Surface-to-surface (S2S) radiative heat transfer modelling code [45], with these same assumptions of non-absorbing media between surfaces, and grey-diffuse surfaces. Furthermore, we will see that
there is a simplification of the relationship between $J_i$ and $G_i$.

**A.8 Net heat transfer between black-body surfaces**

If we consider the case where two black bodies $i$ and $j$ are near each other, the net radiative heat transfer $q_{ij}$ from $i$ to $j$ is the difference between the radiosity leaving $i$ and arriving at $j$ and that leaving $j$ and arriving at $i$. We can use the view factors to give

$$q_{ij} = F_{ij} A_i J_{bi} - F_{ji} A_j J_{bj}$$

For a black-body, all radiation is absorbed and so $J_b = E_b = \sigma T^4$. Hence,

$$q_{ij} = F_{ij} A_i \sigma T_i^4 - F_{ji} A_j \sigma T_j^4$$

Applying the view factor symmetry relation,

$$q_{ij} = F_{ij} A_i \sigma (T_i^4 - T_j^4)$$

**(A.3)**

**A.9 Net heat transfer between grey surfaces**

The matrix method in equation A.2 can be made more useful assuming

- The surfaces are grey-diffuse, so $\alpha = \epsilon$, and these are constant with direction and wavelength for the interface in which emission and absorption are occurring.
- The surfaces have uniform $J_i$ and $G_{ji}$.
- Surfaces are opaque, there is no transmissivity.

We will try to eliminate the $G_j$ terms such that we have everything in terms of the surface radiosities and black body radiation functions.

The net heat transferred away from surface $i$ is

$$q_i = A_i (J_i - G_i)$$

and we know that $J_i = E_i + \rho_i G_i$ so $J_i - E_i = \rho_i G_i$, so

$$q_i = A_i \left( J_i - \frac{J_i - E_i}{\rho_i} \right)$$

Now, using $E_i = \epsilon_i E_{bi}$ and $1 = \rho_i + \alpha_i = \rho_i + \epsilon_i$,
\[ q_i = A_i \left( J_i - \frac{J_i - \epsilon_i E_{bi}}{1 - \epsilon_i} \right) = \frac{A_i J_i}{1 - \epsilon_i} \left( 1 - \epsilon_i \right) - \left( 1 - \epsilon_i \right) \frac{E_{bi}}{J_i} \]

Hence

\[ q_i = \frac{A_i \epsilon_i}{1 - \epsilon_i} (E_{bi} - J_i) \quad (A.5) \]

Equation A.5 is a powerful way to calculate the heat leaving surface \( i \) as a simple function of the radiosity of surface, its temperature, emissivity and area. The difficult part is calculating the radiosity, since this depends on the incident radiation from all the other surfaces.

Equation A.2 gives an expression for the irradiation at each surface:

\[
\begin{bmatrix}
A_1 G_1 \\
A_2 G_2 \\
\vdots \\
A_N G_N
\end{bmatrix}
= 
\begin{bmatrix}
F_{11} & F_{12} & \ldots & F_{1N} \\
F_{21} & F_{22} & \ldots & F_{2N} \\
\vdots & \vdots & \ddots & \vdots \\
F_{N1} & F_{N2} & \ldots & F_{NN}
\end{bmatrix}
\begin{bmatrix}
A_1 J_1 \\
A_2 J_2 \\
\vdots \\
A_N J_N
\end{bmatrix}
\]

We can substitute these equations in to A.4 to give

\[
\begin{bmatrix}
q_1 \\
q_2 \\
\vdots \\
q_N
\end{bmatrix}
= 
\begin{bmatrix}
A_1 J_1 \\
A_2 J_2 \\
\vdots \\
A_N J_N
\end{bmatrix}
- 
\begin{bmatrix}
A_1 G_1 \\
A_2 G_2 \\
\vdots \\
A_N G_N
\end{bmatrix}
\]

\[
= 
\begin{bmatrix}
A_1 J_1 \\
A_2 J_2 \\
\vdots \\
A_N J_N
\end{bmatrix}
- 
\begin{bmatrix}
F_{11} & F_{12} & \ldots & F_{1N} \\
F_{21} & F_{22} & \ldots & F_{2N} \\
\vdots & \vdots & \ddots & \vdots \\
F_{N1} & F_{N2} & \ldots & F_{NN}
\end{bmatrix}
\begin{bmatrix}
A_1 J_1 \\
A_2 J_2 \\
\vdots \\
A_N J_N
\end{bmatrix}
\]

Now take equation A.5, substitute it in each row of the left hand side of the above:

\[
\begin{bmatrix}
\frac{A_1 \epsilon_i}{1 - \epsilon_i} (E_{b1} - J_1) \\
\frac{A_2 \epsilon_i}{1 - \epsilon_i} (E_{b2} - J_2) \\
\vdots \\
\frac{A_N \epsilon_i}{1 - \epsilon_i} (E_{bN} - J_N)
\end{bmatrix}
= 
\begin{bmatrix}
A_1 J_1 \\
A_2 J_2 \\
\vdots \\
A_N J_N
\end{bmatrix}
- 
\begin{bmatrix}
F_{11} & F_{12} & \ldots & F_{1N} \\
F_{21} & F_{22} & \ldots & F_{2N} \\
\vdots & \vdots & \ddots & \vdots \\
F_{N1} & F_{N2} & \ldots & F_{NN}
\end{bmatrix}
\begin{bmatrix}
A_1 J_1 \\
A_2 J_2 \\
\vdots \\
A_N J_N
\end{bmatrix}
\]

The above system gives a linear system of equations that can be solved for each surface radiosity given the black-body emissive power of each surface. So we simply
calculate our surface temperatures, calculated $E_{bi}$ for each surface, then solve the system for $J_i$.

There is a strong analogy here between an electrical circuit. $q_i$ is like an electrical current, $(E_{bi} - J_i)$ is like a voltage potential, and $\frac{A_i\epsilon_i}{1 - \epsilon_i}$ is like a resistance (we call it the surface radiative resistance).

In summary, the above technique can be used to calculate heat transfer between any number of uniform-temperature bodies that satisfy the grey-body assumptions. So long as emissivity can be assumed constant and equal to the absorptivity, and so long as surfaces are all opaque, we can divide any large object up into smaller finite elements. Assuming constant temperature only within the elements, we can use this technique to estimate the temperature variation on a body subject to radiation from other bodies. This is one of the techniques used in the calculations in Chapter 4.

A.10 Parallel grey surfaces

The simple case of a cavity consisting of only two grey surfaces can be examined as a special case of the above:

$$\begin{bmatrix} q_1 \\ q_2 \end{bmatrix} = \begin{bmatrix} \frac{A_1\epsilon_1}{1 - \epsilon_1} (E_{b1} - J_1) \\ \frac{A_2\epsilon_2}{1 - \epsilon_2} (E_{b2} - J_2) \end{bmatrix} = \begin{bmatrix} A_1J_1 \\ A_2J_2 \end{bmatrix} - \begin{bmatrix} F_{11} & F_{12} \\ F_{21} & F_{22} \end{bmatrix} \begin{bmatrix} A_1J_1 \\ A_2J_2 \end{bmatrix}$$

Assuming that the surfaces are both infinite parallel planes, the areas cancel, and all the view factors will become either one or zero, as follows:

$$\begin{bmatrix} q_1 \\ q_2 \end{bmatrix} = \begin{bmatrix} \frac{\epsilon_1}{1 - \epsilon_1} (E_{b1} - J_1) \\ \frac{\epsilon_2}{1 - \epsilon_2} (E_{b2} - J_2) \end{bmatrix} = \begin{bmatrix} J_1 \\ J_2 \end{bmatrix} - \begin{bmatrix} 0 & 1 \\ 1 & 0 \end{bmatrix} \begin{bmatrix} J_1 \\ J_2 \end{bmatrix}$$

$$= \begin{bmatrix} \frac{\epsilon_1}{1 - \epsilon_1} (E_{b1} - J_1) \\ \frac{\epsilon_2}{1 - \epsilon_2} (E_{b2} - J_2) \end{bmatrix} = \begin{bmatrix} J_1 - J_2 \\ J_2 - J_1 \end{bmatrix}$$

Solving simultaneously gives

$$J_1 - J_2 = \frac{E_{b1} - E_{b2}}{\frac{1 - \epsilon_1}{\epsilon_1} + \frac{1 - \epsilon_2}{\epsilon_2} + 1}$$

Which is equivalent to the following expression for the net heat leaving surface $1$:

$$q_1 = \sigma \left( T_1^4 - T_2^4 \right) \frac{1 - \epsilon_1}{\epsilon_1} + \frac{1 - \epsilon_2}{\epsilon_2} + 1$$

This simplifies slightly, to:

$$q_1 = \sigma \left( T_1^4 - T_2^4 \right) \frac{1 - \epsilon_1}{\epsilon_1} + \frac{1 - \epsilon_2}{\epsilon_2} + 1$$
Figure A.1: Intersecting straight edges, for the case when calculating the view factor from $a$ to $b$.

$$q_1 = \frac{1}{\varepsilon_1 + \varepsilon_2 - 1}$$

A.11 View factors in two-dimensions

View factors can be easily calculated for some simple cases:

A.11.1 Between intersecting straight edges

For the view factor from $a$ to $b$ between intersecting straight edges, as in Figure A.1, we use the expression

$$F_{ab} = \frac{a + b - c}{2a}$$  \hspace{1cm} (A.6)

A.11.2 Between non-intersecting straight edges

The view factor from $a$ to $b$ for non-intersecting straight edges, as in Figure A.2, providing there are no obstructions between the two surfaces (they are in clear view of each other), can be calculated using an equation called the 'crossed string rule', easily derived from the above Eq A.6 along with A.1:

$$F_{ab} = \frac{(e + f) - (c + d)}{2a}$$  \hspace{1cm} (A.7)

A.12 Modelling a grey cavity using ASCEND

The following is a simple ASCEND model file that was developed here for a simple cavity heat transfer problem. We assume all surfaces are grey, and we allow that either the temperature or the heat flux can be fixed for each surface. Specifying upper (‘W’) and lower (‘B’) surface temperatures we get an estimate of the radiative heat transfer through the cavity.
Figure A.2: Crossed-string rule, used for calculating the view factor $F_{ab}$ from edge $a$ to edge $b$.

```ascend
REQUIRE "atoms.a4l";
REQUIRE "johnpye/thermo_types.a4c";

MODEL cavity_base;
(*
This model implements the grey body equilibrium equations
but no specific geometry or boundary conditions
*)
  n "surfaces in the cavity" IS_A set OF symbol_constant;
  A[n] "length of the surfaces (in 2D)" IS_A distance;
  F[n] IS_A factor;

  (* Radiation equations *)
  q[n] "net heat transfer from each surface" IS_A energy_rate_per_length;
  E_b[n] IS_A energy_flux;
  J[n] IS_A energy_flux;
  T[n] IS_A temperature;
  eps[n] IS_A factor; (* emissivity *)

  FOR i IN n CREATE
    z_q[i]: q[i] = SUM[(J[i]-J[j])*(A[i]*F[i][j]) | j IN n];
  END FOR;

  FOR i IN n CREATE
    z_E_b[i]: E_b[i] = 1*SIGMA_C * T[i]^4;
  END FOR;

  FOR i IN n CREATE
    z_J[i]: q[i] * (1-eps[i]) = (E_b[i] - J[i]) * (eps[i]*A[i]);
  END FOR;
END cavity_base;

MODEL cavity REFINES cavity_base;
(* specific test case for the CLFR cavity receiver *)
  W,B,S,N,C,E,D IS_A distance;
  theta, phi, psi IS_A angle;
  F_WN, F_WW, F_WB IS_A factor;
  F_NB, F_NW, F_NN, F_BW, F_BN IS_A factor;
```
APPENDIX A. RADIATIVE HEAT TRANSFER

\[ F_{WS} \text{ IS A factor; } \]
\[
z_N: N \cdot \sin(\theta) = D; \]
\[
z_S: S \cdot \tan(\theta) = D; \]
\[
z_B: B = W + 2 \cdot S; \]
\[
z_E: E^2 = W^2 + D^2; \]
\[
z_{phi}: \phi = \arctan(D/W); \]
\[
z_{psi}: \psi = 1\pi - \phi; \]
\[
z_C: C^2 = E^2 + S^2 - 2 \cdot E \cdot S \cdot \cos(\psi); \]
\[
z_F_{WN}: F_{WN} = (W + N - C)/2/W; \]
\[
z_F_{WW}: F_{WW} = (2E - 2D)/2/W; \text{ (* from top to directly opp part of bottom *)} \]
\[
z_F_{WB}: F_{WB} = 1 - 2 \cdot F_{WN}; \]
\[
z_F_{WS}: F_{WS} = (1 - 2 \cdot F_{WN} - F_{WB})/2; \]
\[
z_F_{NB}: F_{NB} = (N + B - C)/2/N; \]
\[
z_F_{NW}: F_{NW} = (N + W - C)/2/N; \]
\[
z_F_{BN}: F_{BN} = F_{NB} \cdot N/B; \]
\[
z_F_{NN}: F_{NN} = 1 - F_{NW} - F_{NB}; \]
\[
z_F_{BW}: F_{BW} = F_{WB} \cdot W/B; \]
\[
n := ['W', 'B', 'L', 'R']; \]

\text{(* Put lengths into a vector *)}
\[
A['W'], W \text{ ARE_THE_SAME}; \]
\[
A['B'], B \text{ ARE_THE_SAME}; \]
\[
A['L'], N \text{ ARE_THE_SAME}; \]
\[
A['R'], N \text{ ARE_THE_SAME}; \]

\text{(* View factor matrix *)}
\[
F['W']['L'], F['W']['R], F_{WN} \text{ ARE_THE_SAME}; \]
\[
F['W']['B'], F_{WB} \text{ ARE_THE_SAME}; \]
\[
z_{F_{WW1}}: F['W']['W'] = 0; \]
\[
F['B']['L'], F['B']['R], F_{BN} \text{ ARE_THE_SAME}; \]
\[
F['B']['W'], F_{BW} \text{ ARE_THE_SAME}; \]
\[
z_{F_{BB1}}: F['B']['B'] = 0; \]
\[
F['L']['R'], F['R']['L'], F_{NN} \text{ ARE_THE_SAME}; \]
\[
F['L']['B'], F['B']['L'], F_{NB} \text{ ARE_THE_SAME}; \]
\[
F['L']['W'], F['W']['L'], F_{NW} \text{ ARE_THE_SAME}; \]
\[
z_{F_{LL}}: F['L']['L'] = 0; \]
\[
z_{F_{RR}}: F['R']['R'] = 0; \]

METHODS

METHOD on_load;
\quad RUN default_self;
\quad RUN reset;
\quad RUN values;
\quad RUN bound_self;
END on_load;

METHOD default_self;
\quad \psi := 120 \text{ (deg)};
A.12. MODELLING A GREY CAVITY USING ASCEND

END default_self;

METHOD bound_self;
  phi.lower_bound := 0 {deg};
  phi.upper_bound := 90 {deg};
  psi.lower_bound := 90 {deg};
  psi.upper_bound := 180 {deg};
END bound_self;

METHOD specify;
  FIX T['W', 'B'];
  FIX q['L', 'R'];
  FIX W,D,theta;
  FIX eps[n];
END specify;

METHOD values;
  T['W'] := 550 {K};
  T['B'] := 373.15 {K};
  q['L', 'R'] := 0 {W/m};
  W := 575 {mm};
  D := 200 {mm};
  theta := 32 {deg};
  eps['W'] := 0.49;
  eps['B'] := 0.9;
  eps['L', 'R'] := 0.1;
END values;

END cavity;

(*
The following model adds external convection coefficients to
the model, and an ambient temperature.

We also calculate the 'F_rad' correlation parameter (see
http://pye.dyndns.org for more information).
*)

MODEL cavity_losses REFINES cavity;
  h_B, h_N IS_A heat_transfer_coefficient;
  T_amb IS_A temperature;
  h_B, h_N IS_A heat_transfer_coefficient;
  T_amb IS_A temperature;

  (* external heat loss *)

  q_ext_rad IS_A energy_rate_per_length;
  q_ext_conv IS_A energy_rate_per_length;
  q_ext_conv = h_B * B * (T['B'] - T_amb);
  q_ext_rad = B * eps['B'] * 1(SIGMA_C) * (T['B']^4 - T_amb^4);
  z_q_B: - q['B'] = q_ext_conv + q_ext_rad;

The following model extends the previous model. In this case, the external heat
loss from the base of the cavity is given as the sum of a convective and radiative
heatloss to an ambient-temperature heat sink. Heat loss through the side-walls is
also allowed for, using a heat transfer coefficient to approximate the overall heat-loss process from those surfaces. The correlation parameters $F_{\text{rad}}$ and $F_{\text{rad}_1}$ are calculated, which are used in Chapter 4 to assess the results of the CFD modelling in which both convective and radiative losses are modelled.

\[
\begin{align*}
  z_{q,L} &= -q[L] = h_N N (T[L] - T_{\text{amb}}); \\
  z_{q,R} &= -q[R] = h_N N (T[L] - T_{\text{amb}});
\end{align*}
\]

(* Determine 'F_rad' from overall heatloss... *)

$F_{\text{rad}}$ IS A factor;
$F_{\text{rad}_1}$ IS A factor;

\[
\begin{align*}
  z_{F_{\text{rad}}} &= q[W] = F_{\text{rad}} W \varepsilon[W] 1\{\sigma_c\} (T[W]^4 - T[B]^4); \\
  z_{F_{\text{rad}_1}} &= q[W] = F_{\text{rad}_1} W \varepsilon[B] 1\{\sigma_c\} (T[W]^4 - T[B]^4) / \left(1/\varepsilon[B] + 1/\varepsilon[W] - 1\right);
\end{align*}
\]

METHODS

METHOD specify;
  FIX $T[W], T_{\text{amb}}$;
  FIX $h_B, h_N$;
  FIX $W, D, \theta$;
  FIX $\varepsilon[n]$;
END specify;

METHOD values;
  $T[W] := 550 \text{ (K)}$;
  $T_{\text{amb}} := 290 \text{ (K)}$;
  $W := 575 \text{ (mm)}$;
  $D := 200 \text{ (mm)}$;
  $\theta := 32 \text{ (deg)}$;
  $\varepsilon[W] := 0.49$;
  $\varepsilon[B] := 0.9$;
  $\varepsilon[L,R] := 0.1$;
  $h_B := 10 \text{ (W/m}^2\text{/K)}$;
  $h_N := 0.5 \text{ (W/m}^2\text{/K)}$;
  (* free values *)
  $T[L,R] := 500 \text{ (K)}$;
  $T[B] := 400 \text{ (K)}$;
END values;

END cavity_losses;

(*
This final model will examine the possible convection behaviour inside the stratified cavity
*)

MODEL cavity_convection;
  $W, D_{\text{strat}}$ IS A distance;
  $k$ IS A thermal_conductivity;
  $Q$ IS A energy_rate_per_length;
  $T[1,2]$ IS A temperature;

  $Q \ast D_{\text{strat}} = W \ast k \ast (T[1] - T[2])$; (* conduction through stratified zone*)
A.13Solar radiation

Light arriving at the surface of the Earth originates from the surface of the Sun, where the temperature is of the order of 6000 K. The radiated light has a spectrum that reflects the chemical makeup of the sun and some local variation in surface temperature. The most intense wavelengths in solar radiation are in the visible range of 300 to 700 nm, although there is a significant ‘tail’ component of solar radiation at both longer and shorter wavelengths. The intensity of solar radiation that reaches the outside of Earth’s atmosphere (extraterrestrial radiation) varies slightly throughout the year (and also longer-term, with sunspot activity) but averages 1367 W/m². The extraterrestrial solar spectrum can be approximated by that of a black body at 5762 K. As the light passes down through the atmosphere, it is attenuated. A fraction of the radiation is absorbed, at specific wavelengths corresponding to the chemical makeup of the atmosphere. This fraction is a function of the air mass, which is the ‘amount’ of atmosphere that direct beams pass through: a function of sun angle and latitude.

By the time radiation reaches the Earth’s surface, its intensity has been reduced

---

2The terms ‘light’ and ‘solar radiation’ are used interchangably here.
to a maximum of about 1000 W/m². A portion of the light received at ground level will have been scattered by the atmosphere, clouds or dust. This component of the radiation is called diffuse and is not useful in concentrating solar collectors. The unscattered component of the radiation is called direct beam radiation and is the part that is of interest here. It should be noted also that the intensity of measured incident light (irradiance) is dependent on the orientation of the receiving surface. We talk of direct normal irradiance (DNI) which refers to just the direct beam radiation (excluding the diffuse component) on a surface normal to the beam direction, and global horizontal irradiance (GHI) which refers to the total beam plus diffuse irradiance on a horizontal surface.

To be able to predict the output of, and to measure the efficiency of, a concentrating solar collector, we require a way of measuring or predicting the direct normal irradiance. The following sections examines how this is achieved at present using available data sources.

A.13.1 Ground-based measurement of solar radiation

Direct measurement of irradiance relies on a number of specialised devices (Figure A.3). The pyranometer measures global horizontal irradiance, whereas a pyrheliometer measures direct normal irradiance, and in so doing must follow the sun using a two-axis tracking mechanism. A simple solution to the DNI measurement problem is the use of a shade-ring pyranometer, although there are some measurement errors introduced with the use of that device. Finally, a device that does not measure irradiance directly, but which can be of assistance in estimating it when better instruments are not available, is the Campbell-Stokes recorder. This device uses a glass sphere to burn a trail on a piece of paper whenever there is direct sunlight. It is typically used for reporting of the number of hours of sunlight each day.

The above range of instruments is not necessary for the recording of basic weather trends such as rainfall and temperature, and so not all weather stations are equipped with them. The Australian Bureau of Meteorology maintains a network of 28 radiation-measuring weather stations [47], and provides a long history of half-hourly data for most of those stations. The Australian and New Zealand Solar Energy Society has prepared aggrerate data based on the Bureau ground-station measurements suitable for use in solar energy design work [125]. The Bureau also publishes the more basic ‘synoptic’ weather for many more stations around Australia; many of these lesser weather stations have Campbell-Stokes recorders.

The sparsity of accurate radiation measurement devices in Australia, particularly in outback locations typically under consideration for solar energy installations, has led to a need for other approaches to the gathering of solar radiation data in the present study.
Figure A.3: Devices for measuring solar radiation: the pyranometer (a) measures global radiation; the pyrheliometer (c) measures beam radiation. The shade-ring pyranometer (b) is a low-cost solution for measuring direct solar radiation through the course of the day. The Campbell-Stokes recorder (d) estimates hours of direct sunlight. [31, 52]
APPENDIX A. RADIATIVE HEAT TRANSFER

A.13.2 Satellite-derived estimation of solar radiation

Visible-wavelength satellite imagery of the Earth has been used to derive estimates of solar radiation. A series of models are used to derive these estimates, including sun position correlations, clear-sky radiation scattering, local atmospheric humidity and turbidity, terrain effects, cloud reflectivity, surface albedo and relative angles between the sun, the satellite, and the location being photographed. Perez et al, 2002 [127] give a complete overview of a method from image pixel analysis through to direct beam and global horizontal radiation estimation.

Australian satellite data at the time of this research were provided in the form of a data CD called NCCSOL from the Bureau of Meteorology [47]. Only daily global horizontal radiation data is available from this data set (no diffuse/beam breakdown), which means that the more recent methods of Perez et al cannot be used to estimate the direct beam radiation component, and instead correlations for daily and monthly diffuse fractions must be used to compute estimated beam radiation from the satellite-derived global horizontal radiation data, in combination with other available data such as latitude and sunset angle.

The method used for the NCCSOL data is given by Weymouth and Marshall, 1994 [174], although subsequent improvements have been made and applied to the more recent data.

A.13.3 TMY data sets

Solar radiation is subject to wide variations from year to year, but that for the purpose of solar thermal engineering design, there is a need for a representative sequence of data, short enough that it can be used manageably for simulation. Efforts have been made to compile ‘typical meteorological year’ (TMY) data sets which aim to match overall solar radiation to long-term averages, give a reasonable spread of unusual storm, wind, and rain events, and give realistic daily temperature ranges. These data sets are composed of selected ‘slices’ of data from real ground-station measurements; there is no synthesised data. TMY data has been compiled for North America [93], Australia and elsewhere. In Australia, Morrison and Litvak, 1999 [106], compiled the most current such data sets.

TMY data has been used in the present work for the some preliminary evaluation of daily beam-to-diffuse radiation correlations at several sites around Australia.

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Appendix B

Martinelli-Nelson correlation by Odeh

As noted in section 5.2.3. In the preceding work, an error was found in the curve fits used for the Martinelli-Nelson correlation in earlier work by Odeh [115]. Code from the Odeh direct steam generation computer code (in FORTRAN) shows the following polynomial curve fit in use at the 68.9 bar pressure level:

\[
c_{68.9\text{ bar}} = (-32.536* x**3) - (6.0142 * x**2) + (41.881 * x) + 0.8298
\]

This curve-fit was also present in the Reynolds C++ code [142].

When checked against the present implementation of the Martinelli Nelson correlation this curves appears as shown in Figure B.1. It would appear that a typographical error or other error has caused the incorrect polynomial to be used. This polynomial is used in linear interpolation in the pressure range 20-100 bar, and has the potential to affect calculated pressure drops throughout that entire range. The error in the friction factor multiplier is more than 100% for high levels of steam quality.
Figure B.1: Polynomial curve fits for the Martinelli Nelson used in the present work (solid lines) and the Odeh polynomial curve fit for 68.9 bar.
Appendix C

CLFR Configuration

The following appendix contains all the details of the CLFR configuration used in the present work. The data is for the CLFR stage 2 prototype at the Liddell power station (except where noted), also described in Chapters 1 and 6.

C.1 Overall system

- Estimated electrical output equivalent: 12 MW
- Number of absorbers: 10
- Steam drum at one end of the array, water pumped to far end of the array (in pipe lagged with 75mm rockwool), passes through the array (310 m) and returns to the steam drum. Flow rate is adjusted to ensure exit quality of 0.8.

C.2 Optics

- Width of mirrors: 2.25 m
- Total mirror area: 27 000 m$^2$
- Array orientation: NW axis (Liddell prototype)
- Mirror spacing (between edges of the glass): 500mm (near an absorber) to 750 mm (far from absorbers)$^1$
- Height of mirror pivot from foundation: 2.1 m
- Centre of mirrors is very close to the pivot axis.

$^1$Spacing between mirrors was uniform in the stage 1 prototype, approximately 700mm (estimated).
• Height of absorber pipes from foundation: 12 m

• At Liddell the length and alignment of the mirrors is the same as the absorber. For higher latitudes the mirrors may be displaced towards the equator relative to the absorber.

• With sun directly overhead the average concentration on the absorber surface is 27

• Cavity cover: flat glass with anti-reflective coating. Reflectivity see below

• Number of mirror lines per absorber: 10 (but 12 for the stage 1 prototype)

For details of the method used to optimise the optics, see [149].

C.3 Absorber

• Length: 310 m

• Width of absorber ‘focus area’: 575 mm

• Module length: 25.7 m

• Connection of pipes between absorber modules: welded

• Emissivity see below

• Number of pipes mounted in each absorber: 12

C.4 Pipework

In the absorber it is DN32 Sched 10S. 12 pipes per absorber. In the pipework from the steam drum to the far end of the absorber (stage 2) the pipe is DN 150 Sch 10S. In the pipework from the three-way manifolds to the actual absorbers, it is DN 100 Sch 10S. There is an orifice plate upstream of each absorber pipe. The size was not yet finalised, although there was an intentioned total of ~3 bar pressure drop from pump outlet to steam drum, with the pipe and absorber losses only accounting for ~1 bar. The steam drum configuration is as shown in Figure C.1.

Estimated pipework expansion at the far end of the absorber is 1.5 m

The manifold in use in the Stage 1 prototype is shown in Figure C.2. Because of the multi-pass configuration needed for the small prototype, a simpler manifold can be expected in the final design.
Figure C.1: Three CLFR modules showing each module with a steam drum connected to a pump and control valve and three absorbers; each absorber has an orifice plate upstream.
Figure C.2: Absorber manifolds from the Stage 1 prototype. The prototype uses a four-pass configuration down its 60 meter length to achieve a total flow path length of 240 m.

Figure C.3: CLFR cavity geometry
C.5 Cavity

- Emissivity of absorber pipes = 0.4 (in stage 1, it was 0.15)
- \( W = 575 \text{ mm}, \quad D = 200 \text{ mm}, \quad \theta = 32^\circ \) (see Figure C.3). Other parameters defined by trigonometry.
- Optical properties of cover material: low iron glass with anti-reflection coating transmission 0.92
- Insulation: Rockwool, 29 cm thickness on top, 15 cm thickness on sides
- Absorber pipe mounting arrangements: supported from roof

C.6 Steam drum

- Steam drum is kept at 42 bar and 120 °C at night. During the day, solar heating takes it up to 253 °C (saturation temperature at 42 bar)
- Three absorbers feed into each steam drum, each via their own flange. The nearest absorber one is connected via about 5m of DN 150 SCH 10S. The other two are connected via 62m of DN 150 SCH 10S.

C.7 Power plant tie-in

- Flow from CLFR to power plant: 42 bar at 253 °C (approx at saturation)
- From power plant to CLFR is water at 42 bar and 150 °C (100 °C subcooled)
- Other operational requirements at the power station side: nil
- Flow rate and/or control mechanisms planned for the power station side: None the flow will be continuous 24hr/day.
- Emergency stop triggers/conditions, if any :
- Pressure relief conditions :

Earlier designs have included an expansion tank, since water flowing into the CLFR was to have been contained within its own fixed-mass, fix-volume flow loop. Recent changes (August 2006) in agreements with the power station have permitted direct connection of steam from the CLFR collector into the boiler feedwater. This eliminates the need for an expansion tank.
C.8 Pumping

Before the direct-connection had been allowed, a pump had been envisaged for circulated the flow through the system. This pump is not required in the new design, as the power stations own pumps will provide the necessary pressure difference to drive the fluid through the collector.
Appendix D

Heat Loss from Evacuated Tubes

This appendix summarises the heat loss correlation developed by Odeh [115] based on thermal test data from Sandia for the SEGS LS-2 collector.

The correlation gives the heat loss flux at the absorber surface $q_l$ as a function of ambient temperature $T_{amb}$, absorber temperature $T$, sky temperature $T_{sky}$, wind velocity $V$ and absorber emissivity $\epsilon_{ab}$:

$$q_l = (a + cV)(T - T_{amb}) + \epsilon_{ab}b\left(T^4 - T_{sky}^4\right)$$

The correlation parameters found by Odeh were as follows:

$$a = 1.9182 \times 10^{-2} \text{ W/m}^2\text{K}$$
$$b = 2.02 \times 10^{-9} \text{ W/m}^2\text{K}^4$$
$$c = 6.612 \times 10^{-3} \text{ W/m}^2\text{K(m/s)}$$

The region of applicability was as follows:

$$150^\circ C < T < 430^\circ C$$
$$0 < V < 5 \text{ ms}^{-1}$$

For the LS-2 collector, the absorber emissivity was $\epsilon_{ab} = 0.14$. 

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Appendix E

Rheinländer Data

This appendix gives details of the experimental and modelling data that was published by Rheinländer [144, 143]. First some formulae from the spreadsheet are shown, followed by the experimental data and simulation results.

There were 12 experimental data sets given, each of which gave the system inlet pressure $p_{in}$, pressure readings $p_1...p_{11}$ at the inlet of each of the 11 collectors, collector, plus the pressure $p_{9out}$ at the outlet of collector 9 and the exit pressure $p_{out}$. The ambient temperature $T_{amb}$ and the values of the direct normal irradiation multiplied by the incident angle multiplier $DNI * IAM$ are given, as well as collector inlet temperature $T_{in}$ and flow rate $\dot{m}_{in}$. In every case the injection mass flow rate at the inlet of collector 10 was zero, but in the 11th data set, the injection flow rate $\dot{m}_{11in}$ was non-zero.

The formula use to give the efficiency of the field $\eta_{field}$ is seen in the spreadsheet to be

$$\eta_{field} = \frac{Q}{\max\left\{ G_n K_a W \sum_{i=1}^{11} L_i, 1 \right\}} \times 100\%$$

where $W = 5.76 \text{m}$ is the mirror aperture width and $L_i$ is the collector length ($L = 48 \text{m}$ for collectors 1 to 8 and collector 11; $L = 24 \text{m}$ for collectors 9 and 10).

The formula for absorbed heat $Q$ is also given, as follows. This formula allows for injection of water (with properties at the inlet state) at points 10$\text{in}$ and 11$\text{in}$, and (lumped) heating sections along the sections (1$\text{in}$,9$\text{out}$), (10$\text{in}$,10$\text{out}$) and (11$\text{in}$,11$\text{out}$). This facilitates two different injection points for the collector when operating in that mode.

$$Q = \dot{m}_{1} (h_{9out} - h_{in}) + \dot{m}_{1} (h_{10out} - h_{9out}) + \dot{m}_{10in} (h_{10out} - h_{in})$$
$$+ (\dot{m}_{1} + \dot{m}_{10in}) (h_{out} - h_{10out}) + \dot{m}_{11in} (h_{out} - h_{in})$$
Table E.1: Summarised minor loss data for the DISS system (from [144])

<table>
<thead>
<tr>
<th>section</th>
<th>Δp [number]</th>
<th>absorber [m]</th>
<th>connections [m]</th>
<th>bends 90°</th>
<th>ball joints</th>
<th>gate valves</th>
<th>T-junctions</th>
<th>Δh [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>coll. 1</td>
<td>1</td>
<td>48</td>
<td>11</td>
<td>8</td>
<td>3</td>
<td>0</td>
<td>0</td>
<td>0.6</td>
</tr>
<tr>
<td>coll. 2</td>
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<td>48</td>
<td>11</td>
<td>8</td>
<td>3</td>
<td>0</td>
<td>0</td>
<td>0.6</td>
</tr>
<tr>
<td>coll. 3</td>
<td>3</td>
<td>48</td>
<td>11</td>
<td>8</td>
<td>3</td>
<td>0</td>
<td>0</td>
<td>0.6</td>
</tr>
<tr>
<td>coll. 4</td>
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<td>48</td>
<td>11</td>
<td>8</td>
<td>3</td>
<td>0</td>
<td>0</td>
<td>0.6</td>
</tr>
<tr>
<td>coll. 5</td>
<td>5</td>
<td>48</td>
<td>11</td>
<td>8</td>
<td>3</td>
<td>0</td>
<td>0</td>
<td>0.6</td>
</tr>
<tr>
<td>coll. 6</td>
<td>6</td>
<td>48</td>
<td>11</td>
<td>8</td>
<td>3</td>
<td>0</td>
<td>0</td>
<td>0.6</td>
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<tr>
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<td>8</td>
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<td>0</td>
<td>0.6</td>
</tr>
<tr>
<td>coll. 8</td>
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<td>48</td>
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<td>8</td>
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<td>0</td>
<td>0.6</td>
</tr>
<tr>
<td>coll. 9</td>
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<td>8</td>
<td>3</td>
<td>1</td>
<td>2</td>
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<td>3</td>
<td>1</td>
<td>2</td>
<td>0.6</td>
</tr>
<tr>
<td>coll. 11</td>
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<td>48</td>
<td>6</td>
<td>4</td>
<td>2</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

The method used to calculate the collector efficiency $\eta_{\text{coll}}$, which would no doubt be central to the calculation of the enthalpies $h_{9\text{out}}$, $h_{10\text{out}}$ and $h_{\text{out}}$ isn’t clear from the report; from personal communication with Jürgen Rheinländer, it was learnt that this data was obtained from the FLAGSOL company, who performed other modelling of the DISS collector.

Data on the flow path length and minor losses is shown in Table E.1. The report gives more detailed data on the interconnecting pipework and bends, which is not shown here.

Table E.3 gives the operating conditions for the 12 experimental data sets given by Rheinländer, as well as his computed values of inlet and outlet enthalpy. Table E.2 gives the experimental pressure drops for this same experimental data set, and Table E.4 gives the corresponding simulation results found by Rheinländer.

Rheinländer’s modelled pressure drops are compared with the experimental values for all 12 experimental data sets in Figure E.1.
Table E.2: Pressure readings for the experiments reported by Rheinländer [143]. The flow distances are shown in the second row (in metres). The first row gives the names of the points where the various pressure readings are taken. Pressure readings are shown in bar.

<table>
<thead>
<tr>
<th>Experiment</th>
<th>Date</th>
<th>0</th>
<th>2m</th>
<th>3m</th>
<th>4m</th>
<th>5m</th>
<th>6m</th>
<th>7m</th>
<th>8m</th>
<th>9m</th>
<th>9out</th>
<th>10m</th>
<th>11m</th>
<th>12m</th>
</tr>
</thead>
<tbody>
<tr>
<td>flow distance / m</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
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<td>8-Jun-00</td>
<td>34.05</td>
<td>33.88</td>
<td>33.79</td>
<td>33.69</td>
<td>33.54</td>
<td>33.31</td>
<td>32.97</td>
<td>32.55</td>
<td>32</td>
<td>31.74</td>
<td>31.58</td>
<td>31.4</td>
<td>30.92</td>
</tr>
<tr>
<td>experiment 2</td>
<td>26-Jul-00</td>
<td>63.28</td>
<td>63.12</td>
<td>62.99</td>
<td>62.89</td>
<td>62.78</td>
<td>62.63</td>
<td>62.42</td>
<td>62.14</td>
<td>61.74</td>
<td>61.49</td>
<td>61.41</td>
<td>61.23</td>
<td>60.87</td>
</tr>
<tr>
<td>experiment 3</td>
<td>26-Jul-00</td>
<td>64.16</td>
<td>64.02</td>
<td>63.91</td>
<td>63.8</td>
<td>63.63</td>
<td>63.4</td>
<td>63.1</td>
<td>62.71</td>
<td>62.2</td>
<td>61.94</td>
<td>61.74</td>
<td>61.55</td>
<td>61.09</td>
</tr>
<tr>
<td>experiment 4</td>
<td>8-Aug-00</td>
<td>34.52</td>
<td>34.36</td>
<td>34.26</td>
<td>34.16</td>
<td>34</td>
<td>33.75</td>
<td>33.39</td>
<td>32.94</td>
<td>32.36</td>
<td>32.11</td>
<td>31.84</td>
<td>31.64</td>
<td>31.14</td>
</tr>
<tr>
<td>experiment 5</td>
<td>22-Sep-00</td>
<td>32.94</td>
<td>32.74</td>
<td>32.65</td>
<td>32.57</td>
<td>32.48</td>
<td>32.34</td>
<td>32.15</td>
<td>31.91</td>
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<tr>
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<td>10-Aug-00</td>
<td>37.56</td>
<td>37.39</td>
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<td>61.85</td>
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<td>61.31</td>
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<td>31.63</td>
<td>31.54</td>
<td>31.39</td>
<td>30.99</td>
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</table>
### Table E.3: Test case specification data from Rheinlander [143].

<table>
<thead>
<tr>
<th>Experiment</th>
<th>t_in (°C)</th>
<th>h_in (kJ/kg)</th>
<th>mass (kg/s)</th>
<th>h_out (kJ/kg)</th>
<th>T_out (°C)</th>
<th>p_in (bar)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>28.50</td>
<td>0.8664</td>
<td>198.10</td>
<td>844.6</td>
<td>0.494</td>
<td>2783.1</td>
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<tr>
<td>2</td>
<td>28.00</td>
<td>0.9560</td>
<td>238.50</td>
<td>1030.7</td>
<td>0.653</td>
<td>2670.8</td>
</tr>
<tr>
<td>3</td>
<td>32.00</td>
<td>0.9275</td>
<td>239.44</td>
<td>2856.8</td>
<td>335.3</td>
<td>3079.5</td>
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<tr>
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<td>212.40</td>
<td>909.3</td>
<td>0.732</td>
<td>2629.7</td>
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<tr>
<td>5</td>
<td>25.50</td>
<td>0.7631</td>
<td>193.44</td>
<td>823.8</td>
<td>0.401</td>
<td>2744.3</td>
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<tr>
<td>6</td>
<td>30.10</td>
<td>0.9560</td>
<td>238.50</td>
<td>1030.7</td>
<td>0.653</td>
<td>2670.8</td>
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<td>7</td>
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### Table E.4: Simulation results from Rheinlander [143].

<table>
<thead>
<tr>
<th>Experiment</th>
<th>p1_in (bar)</th>
<th>p1_out (bar)</th>
<th>p2_in (bar)</th>
<th>p2_out (bar)</th>
<th>p3_in (bar)</th>
<th>p3_out (bar)</th>
<th>p4_in (bar)</th>
<th>p4_out (bar)</th>
<th>p5_in (bar)</th>
<th>p5_out (bar)</th>
<th>p6_in (bar)</th>
<th>p6_out (bar)</th>
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<tbody>
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<td>33.93</td>
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<td>63.16</td>
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<td>32.88</td>
<td>32.71</td>
<td>32.65</td>
<td>32.42</td>
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</table>
Figure E.1: Experimental (blue line) and modelled (red crosses) pressure drops (bar) plotted against flow path length (m), for the 12 cases given in Rheinländer’s work. Results are very close, with the worst deviations seen in experiments 3 and 10.
APPENDIX E. RHEINLÄNDER DATA
Appendix F

Pump curves

Figure F.1 shows the pump curve for the pump currently selected for use in the CLFR Stage 2 prototype, as used for the steady-state system model in Chapter 8.
Figure F.1: Pump curves for the CLFR Stage 2 prototype circulation pump.
Appendix G

Steady-state ASCEND model

This appendix contains the listing of the ASCEND model file clfr.a4c, as referred to in Chapter 8.

```ascend
REQUIRE "atoms.a4l";
REQUIRE "johnpye/thermo_types.a4c";

IMPORT "freesteam";
IMPORT "johnpye/extpy/extpy";
IMPORT "johnpye/solvernotes";
IMPORT "sensitivity/solve";
IMPORT "clfrplot";
IMPORT "johnpye/brent/brent";
IMPORT "dag";

(*) Thermo properties -- IAPWS-IF97 (*)
MODEL steam_state;
   p IS_A pressure;
   h IS_A specific_enthalpy;
   u IS_A specific_energy;
   v IS_A specific_volume;
   T IS_A temperature;
   x IS_A fraction;
   s IS_A specific_entropy;
   mu IS_A viscosity;

   thermo: iapws97_uvTxsmu_ph(
      p,h : INPUT;
      u,v,T,x,s,mu : OUTPUT
   );
METHODS
   METHOD default;
      p := 10 (bar);
      p.nominal := 42 (bar);
      v.nominal := 10 (L/kg);
      u := 2000 (kJ/kg);
      T := 400 (K);
```

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APPENDIX G. STEADY-STATE ASCEND MODEL

```ascend
x := 0.8;
END default;
METHOD solve;
   EXTERNAL do_solve(SELF);
END solve;
METHOD on_load;
   RUN default_all;
   FIX p, h;
END on_load;
END steam_state;

(* a simple connector that includes calculation of steam properties *)
MODEL steam_node;
   state IS_A steam_state;
   p ALIASES state.p;
   h ALIASES state.h;
   u ALIASES state.u;
   v ALIASES state.v;
   T ALIASES state.T;
   x ALIASES state.x;
   s ALIASES state.s;
   mu ALIASES state.mu;
   mdot IS_A mass_rate;

METHODS
   METHOD default;
      mdot. nominal := 2 \{kg/s\};
   END default;
   METHOD solve;
      EXTERNAL do_solve(SELF);
   END solve;
   METHOD on_load;
      RUN default_all; RUN reset; RUN values;
      FIX p, h;
   END on_load;
END steam_node;

(*------------------------------------------------------------------------------
CONSTANTS
*)
MODEL pipe_data;
(* nothing here that might need to be optimised; only material properties and suchlike *)
epsilon_pipe IS_A length_constant;
epsilon_pipe ::= 0.05 \{mm\};
   (* Conservative value: 0.05 mm (Graham).
   A good value might be 0.015 mm --
   http://www.thermexcel.com/english/ressourc/pdcline.htm
   *)

(* properties of 304 stainless *)
k_pipe IS_A constant;
k_pipe ::= 16.2 \{W/m/K\}; (* from matweb *)

Cp_pipe IS_A constant;
Cp_pipe ::= 0.5 \{J/g/K\}; (* from matweb *)
```
rho_pipe IS_A constant;
rho_pipe := 0.285 \text{ (lbm/in}^3\text{)};
END pipe_data;

(* model of a CLFR stage-2 prototype absorber *)
MODEL clfr_data REFINES pipe_data;
(* ideally, nothing here that might need to be optimised; only material properties and such *)

W_cavity, D_cavity IS_A length_constant;
W_cavity := 0.575 \text{ (m)}; (* stage2 *)
D_cavity := 0.200 \text{ (m)}; (* stage2 *)

CR IS_A constant;
CR := 27; (* includes losses *)

emissivity_absorber IS_A constant;
emissivity_absorber := 0.4; (* stage2 = 0.40. *)

emissivity_cover IS_A constant; (* LONG WAVE *)
emissivity_cover := 1.0;

hc_cover, hc_sidewall IS_A constant;
hc_cover := 0 (W/m\text{^2/K}); (* 20 W/m2K *)
hc_sidewall := 0 (W/m\text{^2/K}); (* 0.5 W/m2K *)

(* this is the number of segments in each black-box *)
n_segments IS_A constant;
n_segments := 155;

n_pipes_per_absorber IS_A constant;
n_pipes_per_absorber := 12;

n_absorbers_per_group IS_A integer_constant;
n_absorbers_per_group := 3;

(* these are the parameters that define how the heat transfer in the cavity goes about *)
K_cavity, POW_DW_cavity, POW_GR_cavity, F_RAD_cavity IS_A constant;
K_cavity := 0.163;
POW_DW_cavity := 0.316;
POW_GR_cavity := 0.196;
F_RAD_cavity := 0.944;

(* SEE ALSO... the FIXED variables further down *)
END clfr_data;

MODEL lagged_pipe_data REFINES pipe_data;
(* nothing here that might need to be optimised; only material properties and suchlike *)
hc_ext IS_A constant;
hc_ext := 0.5 \text{ (W/m}^2\text{/K)};

n_segments IS_A constant; (* cast to integer by the blackbox *)
n_segments := 10;
END lagged_pipe_data;
APPENDIX G. STEADY-STATE ASCEND MODEL

(*------------------------------------------------------------------------------
PIPE AND ABSORBER MODELS
*)

(* Note that there are plenty of possible numerical pitfalls in this... the
Ledinegg instability means that solving for a given pressure drop can have
multiple solutions!
*)

MODEL absorber_pipe;

(* this is the number of blackbox nodes *)

n IS_A integer_constant;
n := 2;
N[1..n] IS_A steam_node;

inlet ALIASES N[1];
outlet ALIASES N[n];

mdot ALIASES N[1].mdot;
m[2..n] IS_A mass;

reg_out[2..n] IS_A factor;
region_name[2..n] IS_A symbol;
T_absorber[2..n] IS_A temperature;

D_i, D_o, L IS_A distance;

L_blackbox IS_A distance;
L_blackbox = L/(n-1);

I IS_A irradiance;

T_amb IS_A temperature;

data IS_A clfr_data;

FOR i IN [2..n] CREATE
    clfr_expr[i]: dsg_clfr_absorber(
        N[i-1].p, N[i-1].h, N[i-1].mdot, D_i, D_o, L_blackbox, I, T_amb : INPUT;
        N[i].p, N[i].h, reg_out[i], m[i], T_absorber[i] : OUTPUT;
        data : DATA
    );
    N[i].mdot, N[i-1].mdot ARE_THESAME;
END FOR;

Q_incident, Q_absorbed IS_A energy_rate;

Q_incident = data.CR * I * L * data.W_cavity / data.n_pipes_per_absorber;


eta_absorber IS_A factor;
eta_absorber = Q_absorbed / Q_incident;

Q_loss IS_A energy_rate;
\[ Q_{\text{loss}} = Q_{\text{incident}} - Q_{\text{absorbed}}; \]

\[ h_{\text{eff}} \text{ IS A heat_transfer_coefficient}; \]
\[ Q_{\text{loss}} = h_{\text{eff}} \times (\text{data.W_cavity} / \text{data.n_pipes_per_absorber}) \times L \times (T_{\text{absorber[2]}} - T_{\text{amb}}); \]

\[ dp \text{ IS A delta_pressure}; \]
\[ dp = \text{outlet.p} - \text{inlet.p}; \]

**METHODS**

**METHOD** specify;

\[ \text{FIX} D_i, D_o, L; \]
\[ \text{FIX} I, T_{\text{amb}}; \]

**END** specify;

**METHOD** specify_all;

\[ \text{FIX} N[1].p, N[1].T, N[1].mdot; \]

**END** specify_all;

**METHOD** values;

\[ D_i := 36.62 \text{ (mm)}; \]
\[ D_o := 36.62 \text{ (mm)} + 2 \times 2.77 \text{ (mm)}; \]
\[ L := 310 \text{ (m)}; \]
\[ I := 1000 \text{ (W/m}^2\text{)}; \]
\[ T_{\text{amb}} := 300 \text{ (K)}; \]
\[ N[1].p := 42 \text{ (bar)}; \]
\[ N[1].T := 273.15 \text{ (K)} + 200 \text{ (K)}; \]
\[ N[1].mdot := 0.23 \text{ (kg/s)}; \]
\[ T_{\text{absorber[2..n]}} := 400 \text{ (K)}; \]
\[ m[2..n].nominal := 100 \text{ (kg)}; \]

**END** values;

**METHOD** default;

\[ L_{\text{blackbox.nomal}} := 100 \text{ (m)}; \]

**END** default;

**METHOD** on_load;

\[ \text{EXTERNAL} \text{defaultself_visit_childatoms(SELF)}; \]
\[ \text{EXTERNAL} \text{defaultself_visit_submodels(SELF)}; \]
\[ \text{RUN} \text{reset}; \text{RUN} \text{specify_all}; \text{RUN} \text{values}; \]

**END** on_load;

**METHOD** lookup_region_name;

\[ \text{FOR} \ i \ \text{IN} \ [2..n] \ \text{DO} \]
\[ \text{EXTERNAL} \text{dsg_lookup_region_name(SELF,reg_out[i],region_name[i]);} \]
\[ \text{END FOR}; \]

**END** lookup_region_name;

**END** absorber_pipe;

**MODEL** pipe;

\[ D_i, D_o, L \text{ IS A distance}; \]

\[ T_{\text{amb}} \text{ IS A temperature}; \]

\[ m_{\text{holdup}} \text{ IS A mass}; (* \text{mass holdup in the pipe *}) \]
APPENDIX G. STEADY-STATE ASCEND MODEL

\[ n \text{ IS\_A integer\_constant}; \]
\[ n := 2; \]
\[ N[1,2] \text{ IS\_A steam\_node}; \]

\text{inlet ALIASES N[1];}
\text{outlet ALIASES N[2];}

\text{data IS\_A lagged\_pipe\_data;}

\text{reg\_out IS\_A factor;}

\text{pipe\_expr: dsg\_lagged\_pipe(}
\text{  N[1].p, N[1].h, N[1].mdot, D_i, D_o, L, T_amb : INPUT;}
\text{  N[2].p, N[2].h, reg\_out, m\_holdup : OUTPUT;}
\text{  data : DATA}
\text{);}

\text{N[2].mdot, N[1].mdot ARE\_THE\_SAME;}

\text{Q\_loss IS\_A energy\_rate;}

\text{dp IS\_A delta\_pressure;}
\text{dp = outlet.p - inlet.p;}

\text{(*)}
\text{A IS\_A area;}
\text{A = 0.25(PI) * D_i^2;}

\text{V IS\_A volume;}
\text{V = A * L;}

\text{rho\_avg IS\_A mass\_density;}
\text{rho\_avg = m\_holdup / V;}
\text{v\_avg IS\_A specific\_volume;}
\text{v\_avg = V / m\_holdup;}

\text{*)}
\text{METHODS}
\text{METHOD specify;}
\text{  FIX D_i, D_o, L, epsilon\_pipe;}
\text{  FIX T\_amb;}
\text{END specify;}

\text{METHOD values;}
\text{  D_i := 151.47 \{mm\};}
\text{  D_o := D_i + 2 * 3.40 \{mm\};}
\text{  L := 310 \{m\};}
\text{  T\_amb := 300 \{K\};}
\text{  m\_holdup\_nominal := 1000 \{kg\};}
\text{END values;}

\text{METHOD specify\_all;}
\text{  FIX N[1].p, N[1].h;}
\text{  FIX N[1].mdot;}
\text{END specify\_all;}

\[N_r \text{ IS A positive_factor;}
\]
\[\text{(* similarity laws (only given change of speed, not size) *)}
V_{dot} = V_{dot1} (m^3/h) \times N_r;
W_{dot} = N_r^3 \times W_{dot1};
H = N_r^2 \times H_1;
\]
\[\text{(* 1st law, with pump thermodynamic efficiency *)}
\eta \times W_{dot} = \text{inlet.mdot} \times (\text{outlet.h} - \text{inlet.h});
\]
\[dp \text{ IS A delta_pressure;}
\]
\[dp \times \text{inlet.v} = 1 \text{(EARTH_G)} \times H;
\]
\[\text{inlet.p} + dp = \text{outlet.p};
\]
\[\eta \text{ IS A factor;}
\]
\[\text{METHODS}
\]
\[\text{METHOD specify;}
\]
\[\text{FIX } N_r;
\]
\[\text{END specify;}
\]
\[\text{METHOD specify_all;}
\]
\[\text{FIX inlet.p, inlet.h, inlet.mdot;}
\]
\[\text{END specify_all;}
\]
\[\text{METHOD values;}
\]
\[N_r := 1;
\]
\[\text{inlet.p := 40 (bar);}
\]
\[\text{inlet.mdot := 0.3 (kg/s);}
\]
\[\text{inlet.h := 1000 (kJ/kg);}
\]
\[N_r.\text{lower_bound} := 0.2;
\]
\[N_r.\text{upper_bound} := 1.0;
\]
APPENDIX G. STEADY-STATE ASCEND MODEL

END values;
METHOD on_load;
  EXTERNAL defaultself_visit_childatoms(SELF);
  EXTERNAL defaultself_visit_submodels(SEL);
RUN reset;
RUN values;
  FIX outlet.p, inlet.T, inlet.mdot;
  outlet.p := 44.345 \text{bar};
  inlet.mdot := 7.73 \text{kg/s};
  inlet.T := 175 \text{K} + 273.15 \text{K};
  FIX inlet.p;
  inlet.p := 42 \text{bar};
END on_load;
METHOD solve;
  EXTERNAL do_solve(SEL);
END solve;
END pump_base;

(*
 This pump is the ZF 80-3400, SULZER reference 'AUS.0294.BSL.07.4534-F1
 Graham sent the pump curve to me in Mar 2007.
*)
MODEL pump REFINES pump_base;
  (* note the assumption that Vdot1 is scaled by 1(m^3/h) *)
  (* pump heat curve at reference speed, using fityk *)
  head1: H1 = (171.91 - 0.0349789 \times Vdot1 -0.00193442 \times Vdot1^2) \times 1 \text{m};
  (* power curve at reference speed, using fityk *)
  Wdot1 = (28.2514 + 0.120906 \times Vdot1 + 0.00360997 \times Vdot1^2
          - 2.80389e-05 \times Vdot1^3 + 6.91525e-08 \times Vdot1^4) \times 1 \text{kW};
  (* efficiency at reference speed, using fityk *)
  eta = (1.35898\times Vdot1 -0.0124358\times Vdot1^2+5.8584 1e-05\times Vdot1^3-1.26982e-07\times Vdot1^4) / 100;
METHODS
  METHOD values;
    RUN pump_base::values;
    N_r := 0.6;
END values;
END pump;

(*
 Model of a separator using conditional modelling
*)
MODEL separator_conditional;
  inlet IS_A steam_node;
  outlet[1,2] IS_A steam_node; (* outlet[1] is for gas *)
  (* the enthalpy of the liquid and gas phases at the steam drum pressure *)
  h_f, h_g IS_A specific_enthalpy;
  satprops: iapws97_hfg_p(
    inlet.p : INPUT;
    h_f, h_g : OUTPUT
  );
outlet[1,2].p, inlet.p ARE_THESAME;

CONDITIONAL
   subcooled: h_f >= inlet.h;
   superheated: inlet.h >= h_g;
END CONDITIONAL;

issub, issup IS_A boolean_var;
issub == SATISFIED(subcooled,1e-8(kJ/kg));
issup == SATISFIED(superheated,1e-8(kJ/kg));

sub1: outlet[1].mdot = 0 (kg/s);
sub2: outlet[1].h = h_g;
sub3: outlet[2].mdot = inlet.mdot;
sub4: outlet[2].h = inlet.h;

sup1: outlet[1].mdot = inlet.mdot;
sup2: outlet[1].h = inlet.h;
sup3: outlet[2].mdot = 0 (kg/s);
sup4: outlet[2].h = h_f;

sat1: inlet.mdot*inlet.x = outlet[1].mdot;
sat2: inlet.mdot*(1-inlet.x) = outlet[2].mdot;
sat3: outlet[1].h = h_g;
sat4: outlet[2].h = h_f;

WHEN (issub,issup)
   CASE TRUE, FALSE:
      USE sub1; USE sub2; USE sub3; USE sub4;
   CASE FALSE, TRUE:
      USE sup1; USE sup2; USE sup3; USE sup4;
   OTHERWISE :
      USE sat1; USE sat2; USE sat3; USE sat4;
END WHEN;

METHODS
   METHOD specify_all;
      FIX inlet.p, inlet.mdot, inlet.h;
   END specify_all;
   METHOD values;
      inlet.p := 42 (bar);
      inlet.h := 3500 (kJ/kg);
      inlet.mdot := 1 (kg/s);
   END values;
   METHOD on_load;
      RUN default_all;
      RUN reset; RUN specify_all; RUN values;
   END on_load;
END separator_conditional;

(*
   Same things as the above separator model, but using absolute value functions
   instead of conditional modelling techniques.
*)
MODEL separator;
inlet IS_A steam_node;
outlet[1,2] IS_A steam_node; (* [1] is for gas, [2] is for liquid *)

(* the enthalpy of the liquid and gas phases at the steam drum pressure *)
h_f, h_g IS_A specific_enthalpy;
satprops: iapws97_hfg_p(
    inlet.p : INPUT;
    h_f, h_g : OUTPUT
);

inlet.p, outlet[1,2].p ARE_THE_SAME;

outlet[1].mdot = inlet.mdot * inlet.x;
outlet[1].h = inlet.h - (0.5*(inlet.h - h_g) - 0.5 * abs(inlet.h - h_g));

outlet[2].mdot = inlet.mdot * (1 - inlet.x);
outlet[2].h = inlet.h + 0.5 * ( h_f - inlet.h ) - 0.5 * abs(h_f - inlet.h);

METHODS
    METHOD specify;
    END specify;
    METHOD values;
    END values;
    METHOD on_load;
        RUN default_all; RUN reset; RUN values;
        FIX inlet.p; inlet.p := 42 {bar};
        FIX inlet.h; inlet.h := 500 {kJ/kg};
        FIX inlet.mdot; inlet.mdot := 1 {kg/s};
    END on_load;
    END separator;

(*
This separator removes all of the steam plus a variable fraction of the
condensate.
*)

MODEL separator_fractional;
    inlet IS_A steam_node;
    outlet[1,2] IS_A steam_node; (* [1] is for gas, [2] is for liquid *)

(* the enthalpy of the liquid and gas phases at the steam drum pressure *)
h_f, h_g IS_A specific_enthalpy;
satprops: iapws97_hfg_p(
    inlet.p : INPUT;
    h_f, h_g : OUTPUT
);

inlet.p, outlet[1,2].p ARE_THE_SAME;

frac IS_A factor;

outlet[1].mdot = inlet.mdot - outlet[2].mdot;

outlet[2].mdot = frac * inlet.mdot * (1 - inlet.x);
outlet[2].h = inlet.h + 0.5 * ( h_f - inlet.h ) - 0.5 * abs(h_f - inlet.h);
MODEL orifice;
  inlet IS_A steam_node;
  outlet IS_A steam_node;
  inlet.h, outlet.h ARE_THE_SAME;
  inlet.mdot, outlet.mdot ARE_THE_SAME;
  dp IS_A delta_pressure;
  inlet.p + dp = outlet.p;

  mdot ALIASES inlet.mdot;

  Vdot IS_A volume_rate;
  Vdot = inlet.mdot * inlet.v;

  D_pipe IS_A distance;
  D_orifice IS_A distance;

  beta IS_A fraction;
  beta = D_orifice / D_pipe;

  A_orifice IS_A area;
  A_orifice = 0.25(PI)*D_orifice^2;

(* this is the pressure drop across a corner-tapped orifice. it's not quite the right equation but it will do given we're guessing the orifice size in the first place *)

C_d IS_A factor;
C_d = 0.5959 + 0.312*beta^2.1 - 0.1840*beta^8 + 0.0029* beta^2.5 * (1e6/Re_D)^0.75;

Re_D IS_A factor;
Re_D = inlet.mdot / (0.25(PI)*D_pipe) / inlet.mu;

C IS_A factor;
C * sqrt(1 - beta^4) = C_d ;

(Vdot / (C * A_orifice))^-2 = -2 * dp * inlet.v;
METHODS

METHOD specify;
  FIX D_pipe, beta;
END specify;

METHOD values;
  Re_D.n, := 100000;
  D_pipe := 108.20 {mm};
  beta := 0.325;
END values;

METHOD on_load;
  EXTERNAL default_self_visit_childatoms(SELF);
  EXTERNAL default_self_visit_submodels(SEL); run reset; run values;
  FIX inlet.mdot, inlet.p, inlet.T;
  inlet.mdot := 12. * 0.3 {kg/s};
  inlet.p := 42 {bar};
  inlet.T := 150 {K} + 273.15 (K);
END on_load;
END orifice;

(*
Model of a control value. We will just reuse the orifice calculation,
as the orifice ratio gives some physical understanding (compared with
a K-factor).
*)

MODEL control_valve REFINES orifice;

(* nothing added *)

METHODS

METHOD values;
  run orifice::-values;
  D_pipe := 161.47 {mm};
  beta := 0.254;
END values;

METHOD on_load;
  EXTERNAL default_self_visit_childatoms(SELF);
  EXTERNAL default_self_visit_submodels(SEL); run reset; run values;
  FREE beta;
  FIX dp;
  dp := -0.553 {bar};
  FIX outlet.p, inlet.h, inlet.mdot;
  outlet.p := 43.107 {bar};
  inlet.T := 175 {K} + 273.15 (K);
  inlet.mdot := 7.735 {kg/s};
  EXTERNAL solver_notes(SEL);
END on_load;
END control_valve;
ADD NOTES IN control_valve;
'solver' name {QRSlv}
'QRSlv' convopt {RELNUM_SCALE}
END NOTES;

(*
General K-factor loss for bends, inlet/outlets etc. We will lump them
together, an approximations.
*)
MODEL minor_loss;
  K IS_A factor;
  inlet, outlet IS_A steam_node;
  A IS_A area;
  D_i IS_A distance;
  A = 0.25*(PI)*D_i^2;
  dp IS_A delta_pressure;
  Vdot IS_A volume_rate;
  Vdot = inlet.mdot * inlet.v;
  vel IS_A speed;
  vel = Vdot / A;
  inlet.p + dp = outlet.p;
  -dp = 0.5 * K * inlet.mdot * inlet.v * inlet.mdot / A^2;
  inlet.mdot, outlet.mdot ARE_THE_SAME;
  inlet.h, outlet.h ARE_THE_SAME;

METHODS
  METHOD specify;
    FIX K, D_i;
  END specify;
  METHOD values;
    K := 0.3; (* flanged regular elbow *)
    D_i := 161.47 (mm); (* DN 150 SCH 10S *)
  END values;
  METHOD on_load;
    EXTERNAL defaultself_visit_childatoms(SELF);
    EXTERNAL defaultself_visit_submodels(SELF);
    RUN specify; RUN values;
    FIX inlet.mdot, inlet.p, inlet.T;
    inlet.mdot := 40 (kg/s);
    inlet.p := 42 (bar);
    inlet.T := 150 (K) + 273.15 (K);
  END on_load;
  METHOD test_example;
    FREE inlet.mdot, K;
    FIX Vdot, dp;
    D_i := 25 (mm);
    dp := -67 (kPa);
    Vdot := 0.0015 (m^3/s);
    inlet.p := 1 (bar);
    inlet.T := 300 (K);
    inlet.h := 100 (kJ/kg); (* nec. for convergence only *)
  END test_example;
  METHOD solve;
    EXTERNAL do_solve(SELF);
  END solve;
  END minor_loss;

MODEL mixer;
  n IS_A integer_constant;
  n := 2;
  inlet[1..n] IS_A steam_node;
  outlet IS_A steam_node;

  inlet[1..n].p, outlet.p ARE_THE_SAME;
  SUM[inlet[i].mdot | i IN [1..n]] = outlet.mdot;
SUM[inlet[i].mdot\times inlet[i].h | i \text{ IN } [1..n]] = outlet.mdot \times outlet.h;

METHODS

METHOD specify;
END specify;
METHOD values;
END values;
METHOD on_load;
RUN default_all; RUN reset; RUN values;
FIX inlet[1].T; inlet[1].T := 300 \text{ (K)};
FIX inlet[1].mdot; inlet[1].mdot := 1 \text{ (kg/s)};
FIX inlet[2].h; inlet[2].h := 4000 \text{ (kJ/kg)};
FIX inlet[2].mdot; inlet[2].mdot := 0.3 \text{ (kg/s)};
END on_load;
METHOD solve;
EXTERNAL do_solve(SELF);
END solve;
END mixer;

(*
Model of the steam drum. Steady-state separator plus fluid level and
mass holdup.

Inlet[1] is from the absorber array, outlet[1] is to the absorber array.
Inlet[2] is the supply feedwater from the plant; outlet[2] is the heated
steam going to the power station.
*)

MODEL steam_drum;
inlet[1,2], outlet[1,2] IS_A steam_node;
sep IS_A separator;

inlet[1], sep.inlet ARE_THE_SAME;
outlet[2], sep.outlet[1] ARE_THE_SAME;
mix IS_A mixer;

(* sep.outlet[2], mix.inlet[1] ARE_THE_SAME; *)
sep.outlet[2].h, mix.inlet[1].h ARE_THE_SAME;
dp IS_A pressure;
mix.inlet[1].p = sep.outlet[2].p = dp;
outlet[1], mix.outlet ARE_THE_SAME;

(* mixing ratio *)
MR IS_A factor;
MR = inlet[2].mdot / outlet[1].mdot;

METHODS

METHOD specify;
FIX inlet[2].T;
FIX inlet[2].p;
FIX inlet[2].mdot;
END specify;
METHOD specify_all;
FIX inlet[1].h;
FIX inlet[1].mdot;
END specify_all;
METHOD values;
  inlet[2].T := 150 {K} + 273.15 {K};
  inlet[2].p := 42 {bar};
  inlet[2].mdot := 1 {kg/s};
  inlet[1].h := 2600 {kJ/kg};
  inlet[1].mdot := 0.3 {kg/s};
END values;
METHOD on_load;
  RUN default_all;
  RUN reset;
  RUN specify_all;
  RUN values;
END on_load;
END steam_drum;

(*--------------------------------------------------------------------------
SYSTEM COMPONENTS:
  ABSORBER=multiple pipes
  ABSORBER_BRANCH=ABSORBER & pipework between pipe-branches
  ABSORBER_GROUP=several ABSORBER_BRANCHES
*)

(* Make allowance for the fact that our single absorber pipe is multiplied
  many times in the whole system
*)
MODEL absorber;
  data IS_A clfr_data;

  inlet IS_A steam_node;

  OP IS_A orifice;
  OP.inlet, inlet ARE_THE_SAME;

  AB IS_A absorber_pipe;
  AB.inlet.state, OP.outlet.state ARE_THE_SAME;
  AB.inlet.mdot * data.n_pipes_per_absorber = OP.outlet.mdot;

  outlet IS_A steam_node;
  AB.outlet.state, outlet.state ARE_THE_SAME;
  AB.outlet.mdot * data.n_pipes_per_absorber = outlet.mdot;

I ALIASES AB.I;

mdot ALIASES inlet.mdot;

m_holdup IS_A mass;

m_holdup = AB.m[2]* data.n_pipes_per_absorber;

Q_incident IS_A energy_rate;

Q_incident = AB.Q_incident * data.n_pipes_per_absorber;

dp IS_A delta_pressure;

inlet.p + dp = outlet.p;
x ALIASES outlet.x;

METHOD values;
RUN OP.values;
RUN AB.values;
END values;
METHOD specify;
RUN OP.specify;
RUN AB.specify;
END specify;
METHOD on_load;
EXTERNAL default_self_visit_childatoms(SELF);
EXTERNAL default_self_visit_submodels(SELF);
RUN reset; RUN values;
FIX inlet.mdot, inlet.p, inlet.T;
inlet.mdot := data.n_pipes_per_absorber * 0.27 {kg/s};
inlet.p := 44 {bar};
inlet.T := 225 (K) + 273.15 (K);
EXTERNAL solvenotes(SELF);
END on_load;
METHOD plot;
EXTERNAL clrfrplot(SELF);
END plot;
END absorber;
ADD NOTES IN absorber;
'solver' name (QRSlv)
'QRSlv' convopt (RELNOM_SCALE)
END NOTES;
(*
All the pipework and losses associated with a single absorber. From the
far-end manifold back to (but not including) the steam drum.
*)
MODEL absorber_branch;
ML1 IS_A minor_loss;
AB IS_A absorber;
P12 IS_A pipe;
ML2 IS_A minor_loss;

inlet ALIASES ML1.inlet;
ML1.outlet, AB.inlet ARE_THE_SAME;
AB.outlet, P12.inlet ARE_THE_SAME;
P12.outlet, ML2.inlet ARE_THE_SAME;
ML2.outlet.state, outlet.state ARE_THE_SAME;
outlet ALIASES ML2.outlet;

m_holdup IS_A mass;
m_holdup = AB.m_holdup + P12.m_holdup;

Q_incident ALIASES AB.Q_incident;

Q_absorbed IS_A energy_rate;
Q_absorbed = outlet.mdot*outlet.h - inlet.mdot*inlet.h;

Q_loss IS_A energy_rate;
Q_loss = Q_incident - Q_absorbed;
x ALIASES outlet.x; mdot ALIASES inlet.mdot; dp IS_A delta_pressure; inlet.p + dp = outlet.p;

METHODS

METHOD values;
  RUN AB.values;

  RUN PI2.values;
  PI2.D_i := 151.47 {mm};
  PI2.D_o := PI2.D_i + 2 * 3.40 {mm};
  PI2.L := 62 {m};

  RUN ML1.values; (* from reticulation pipe to absorber *)
  ML1.K := 0.07 + 0.57 + 0.25 * 4; (* a manifold (inlet and outlet) and four elbows *)
  ML1.D_i := 108.20 {mm}; (* DN 100 SCH 10S *)

  RUN ML2.values; (* PI2 bends plus steam drum inlet *)
  ML2.K := 0.25 * 4 + 0.57; (* four long-radius elbows, exit to drum *)
  ML2.D_i := 108.20 {mm}; (* DN 100 SCH 10S *)

m_holdup.nominal := 3000 {kg};
END values;

METHOD specify;
  RUN AB.specify;
  RUN PI2.specify;
  RUN ML1.specify;
  RUN ML2.specify;
END specify;

METHOD on_load;
  EXTERNAL defaultself_visit_childatoms(SELF);
  EXTERNAL defaultself_visit_submodels(SELF);
  RUN values; RUN specify;
  FIX mdot, inlet.p, inlet.T;
  mdot := 3.0 {kg/s};
  inlet.p := 43 {bar};
  inlet.T := 175 {K} + 273.15 {K};
END on_load;

END absorber_branch;

MODEL absorber_branch_vary REFINES absorber_branch;
  x_err IS_A factor;
  x_err = x - 0.8;
  p_err IS_A delta_pressure;
  p_err = outlet.p - 42 {bar};

METHODS

METHOD default;
  mdot.lower_bound := 2 {kg/s};
  mdot.upper_bound := 4 {kg/s};
  inlet.p.lower_bound := 42 {bar};
  inlet.p.upper_bound := 50 {bar};
END default;

METHOD solve_outlet_quality;
  EXTERNAL brent(SELF,x_err,inlet.mdot);
APPENDIX G. STEADY-STATE ASCEND MODEL

METHOD solve_outlet_quality;
METHOD solve_outlet_pressure;
  inlet.p.lower_bound := outlet.p;
  inlet.p.upper_bound := outlet.p + 10 \{bar\};
  EXTERNAL brent(SELF.p_err,inlet.p);
END solve_outlet_pressure;
METHOD solve_flowrate_for_outlet_pressure;
  inlet.p.lower_bound := 40 \{bar\};
  inlet.p.upper_bound := 50 \{bar\};
  EXTERNAL brent(SELF.p_err,mdot);
END solve_flowrate_for_outlet_pressure;
METHOD solve_outlet_px;
  RUN solve_outlet_quality;
  RUN solve_outlet_pressure;
END solve_outlet_px;
METHOD solve;
  EXTERNAL do_solve(SELF);
END solve;
END absorber_branch_var;

MODEL absorber_group;
  AX IS_A absorber_branch;
  data ALIASES AX.AB.data;
  I ALIASES AX.AB.I;
  mdot ALIASES inlet.mdot;
  m_holdup IS_A mass;
  m_holdup = data.n_absorbers_per_group * AX.m_holdup;
  Q_incident IS_A energy_rate;
  Q_incident = data.n_absorbers_per_group * AX.Q_incident;
  inlet, outlet IS_A steam_node;
  AX.inlet.state, inlet.state ARE_THE_SAME;
  AX.outlet.state, outlet.state ARE_THE_SAME;
  AX.inlet.mdot * data.n_absorbers_per_group = inlet.mdot;
  AX.outlet.mdot * data.n_absorbers_per_group = outlet.mdot;
  dp ALIASES AX.dp;
  x ALIASES outlet.x;
  Q_absorbed IS_A energy_rate;
  Q_absorbed = outlet.mdot*outlet.h - inlet.mdot*inlet.h;
  Q_loss IS_A energy_rate;
  Q_loss = Q_incident - Q_absorbed;

METHODS
  METHOD on_load;
    EXTERNAL defaultself_visit_childatoms(SELF);
    EXTERNAL defaultself_visit_submodels(SELF);
    RUN values; RUN specify;
    FIX mdot, inlet.p, inlet.T;
    mdot := data.n_absorbers_per_group * 3.0 \{kg/s\};
    inlet.p := 43 \{bar\};
    inlet.T := 175 \{K\} + 273.15 \{K\};
  END on_load;
  METHOD values;
    RUN AX.values;
    m_holdup.nominal := 3000 \{kg\};
  END values;
  METHOD specify;
    RUN AX.specify;
(*---------------------------------------------------------------

SYSTEM MODEL
*)

MODEL clf_openloop_pipework;
PI1 IS_A pipe;
AG IS_A absorber_group;

inlet ALIASES PI1.inlet;
PI1.outlet, AG.inlet ARE_THE_SAME;
outlet ALIASES AG.outlet;

dp IS_A delta_pressure;
inlet.p + dp = outlet.p;

x ALIASES outlet.x;
mdot ALIASES inlet.mdot;

p_out ALIASES outlet.p;
p_err IS_A delta_pressure;
p_err = outlet.p - 42 {bar};

x_err IS_A factor;
x_err = outlet.x - 0.8;

m_holdup IS_A mass;
m_holdup = PI1.m_holdup + AG.m_holdup;

Q_absorbed IS_A energy_rate;
Q_absorbed = outlet.mdot*outlet.h - inlet.mdot*inlet.h;

Q_loss IS_A energy_rate;
Q_loss = AG.Q_loss + PI1.Q_loss;

METHODS
METHOD specify;
    RUN PI1.specify;
    RUN AG.specify;
END specify;

METHOD values;
    RUN PI1.values;
    RUN AG.values;

    PI1.D_i := 151.47 {mm};
    PI1.D_o := PI1.D_i + 2 * 3.40 {mm};
    PI1.L := 310 {m};
END values;

METHOD on_load;
    EXTERNAL defaultself_visit_childatoms(SELF);
    EXTERNAL defaultself_visit_submodels(SELF);
    RUN reset; RUN values;

    FIX inlet.mdot, inlet.T, inlet.p;
inlet.mdot := 7.5 {kg/s};
inlet.T := 175 {K} + 273.15 {K};

END values;

END specify;
END absorber_group;
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\[ \text{inlet.p} := 43.5 \text{ (bar)}; \]

\[ \text{EXTERNAL solvernotes(SELF);} \]

\[ \text{inlet.mdot.lower_bound} := 1 \text{ (kg/s)}; \]
\[ \text{inlet.mdot.upper_bound} := 10 \text{ (kg/s)}; \]

\text{END on_load;}

\text{METHOD solve_outlet_quality;}
\[ \text{EXTERNAL brent(SELF,x_err,inlet.mdot);} \]
\text{END solve_outlet_quality;}

\text{METHOD solve_outlet_pressure;}
\[ \text{inlet.p.lower_bound} := \text{outlet.p}; \]
\[ \text{inlet.p.upper_bound} := \text{outlet.p + 10 (bar)}; \]
\[ \text{EXTERNAL brent(SELF,p_err,inlet.p);} \]
\text{END solve_outlet_pressure;}

\text{METHOD solve;}
\[ \text{EXTERNAL do_solve(SELF);} \]
\text{END solve;}

\text{END clfr_openloop_pipework;}

\text{ADD NOTES IN absorber_and_pipes;}
\[ \text{’solver’ name \{QRSlv\}} \]
\[ \text{’QRSlv’ convopt \{RELNOM_SCALE\}} \]
\text{END NOTES;}

\text{(*}
\text{Another open-loop model. Includes from the flange upstream of the pump}
\text{to the flange upstream of the steam drum.}
\text{*})

\text{MODEL pump_to_drum;}
\[ \text{ML3 IS_A minor_loss; PU IS_A pump; CV IS_A control_valve; PW IS_A clfr_openloop_pipework; }\]
\text{inlet ALIASES ML3.inlet; ML3.outlet, PU.inlet ARE_THE_SAME; PU.outlet, CV.inlet ARE_THE_SAME;}
\text{CV.outlet, PW.inlet ARE_THE_SAME; outlet ALIASES PW.outlet; }
\text{mdot ALIASES inlet.mdot; x ALIASES PW.AG.x; m_holdup ALIASES PW.m_holdup; }
\text{dp IS_A delta_pressure; inlet.p + dp = outlet.p; }

\text{METHODS}
\[ \text{METHOD specify; RUN ML3.specify; RUN PU.specify; RUN CV.specify; }
\text{RUN PW.specify; }\]
END specify;
METHOD values;
  RUN PU.values;  RUN CV.values;
  RUN PW.values;
  RUN ML3.values; (* from drum to pump *)
  ML3.K := 0.25 * 3 + 0.5; (* pipe entry loss, three elbows *)
  ML3.D_i := 161.47 {mm}; (* DN 150 SCH 10S *)
END values;
METHOD on_load;
  EXTERNAL defaultself_visit_childatoms(SELF);
  EXTERNAL defaultself_visit_submodels(SELF);
END on_load;
METHOD solve;
  EXTERNAL do_solve(SELF);
END solve;
METHOD adjust_pump_speed;
  EXTERNAL brent(SELF,dp,PU.N_r);
END adjust_pump_speed;
END pump_to_drum;

ADD NOTES IN pump_to_drum;
  'solver' name {QRSlv}
  'QRSlv' convopt {RELNOM_SCALE}
END NOTES;

(*
  The open loop model contains everything from the outlet of the steam drum
  around to the inlet to the steam drum.  
  *** ML1 -- PU -- CV -- PI -- ML2 -- AB -- ML3 -->
*)
MODEL clfr;

SE IS_A separator;
MI IS_A mixer;
OL IS_A pump_to_drum;
PU ALIASES OL.PU;
AG ALIASES OL.PW.AG;

h_err IS_A delta_specific_enthalpy;
h_err = inlet.h - outlet.h;

x_desired IS_A fraction;

OL.outlet, SE.inlet ARE_THE_SAME;
feedout ALIASES SE.outlet[1];
SE.outlet[2], MI.inlet[1] ARE_THE_SAME;
feedin ALIASES MI.inlet[2];

outlet ALIASES MI.outlet;
inlet ALIASES OL.inlet;
mdot ALIASES inlet.mdot;
x ALIASES AG.outlet.x;

x_err IS_A factor;
x_err = x - x_desired;

feedout.mdot = feedin.mdot;

dp IS_A delta_pressure;
inlet.p + dp = outlet.p; (* inlet to pump to exit of steam drum *)

Q_collected IS_A energy_rate;
Q_collected = feedout.mdot * feedout.h - feedin.mdot * feedin.h
 + outlet.mdot*outlet.h - inlet.mdot*inlet.h;

eta_ps IS_A fraction; (* power station thermal-to-electric efficiency for use in estimating pump
eta_col IS_A factor;
eta_coll = Q_collected / AG.Q_incident;

(* efficiency of gathering energy, including losses related to pump power *)
eta_sys IS_A fraction;
eta_sys = (Q_collected - PU.Wdot / eta_ps) / AG.Q_incident;

METHODS
METHOD specify;
RUN OL.specify;
RUN SE.specify; RUN MI.specify;
FIX eta_ps;
FIX x_desired;
END specify;
METHOD values;
RUN OL.values;
RUN SE.specify; RUN MI.specify;
eta_ps := 0.3;
x_desired := 0.8;
END values;
METHOD on_load;
EXTERNAL defaultself_visit_childatoms(SELF);
EXTERNAL defaultself_visit_submodels(SELF);
RUN reset; RUN values;

FIX inlet.p, inlet.T, inlet.mdot;
inlet.p := 42 (bar);
inlet.T := 175 (K) + 273.15 (K);
inlet.mdot := 7.735 (kg/s);

FIX feedin.mdot, feedin.T;
feedin.mdot := 7.5 (kg/s);
feedin.T := 150 (K) + 273.15 (K);

EXTERNAL solvernotes(SELF);
FREE feedin.mdot;

END on_load;
METHOD operatingpoint;
  FREE mdot;
  FREE PU.N_r;
  FIX AG.x;
  AG.x := 0.8;
  FIX dp;
  dp := 0 (Pa);
END operatingpoint;

METHOD solve;
  EXTERNAL do_solve(SELF);
END solve;
METHOD adjust_pump_speed;
  EXTERNAL brent(SELF,dp,PU.N_r);
END adjust_pump_speed;
METHOD adjust_inlet_enthalpy;
  FREE inlet.T;
  FIX inlet.h;
  inlet.h.lower_bound := 600 (kJ/kg);
  inlet.h.upper_bound := 900 (kJ/kg);
  EXTERNAL brent(SELF,h_err,inlet.h);
END adjust_inlet_enthalpy;
METHOD adjust_flowrate;
  mdot.lower_bound := 6 (kg/s);
  mdot.upper_bound := 15 (kg/s);
  EXTERNAL brent(SELF,x_err,mdot);
END adjust_flowrate;
METHOD adjust_all;
  RUN adjust_flowrate;
  RUN adjust_inlet_enthalpy;
  RUN adjust_pump_speed;
END adjust_all;
END clfr;

ADD NOTES IN clfr;
  'solver' name {QRSlv}
  'QRSlv' convopt {RELNOM_SCALE}
END NOTES;

(*-----------------*)