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Ultra-High Temperature
Concentrated Solar Thermal Energy

Tarek Ihab Abdelsalam

A thesis submitted for the degree of Doctor of Philosophy.

School of Engineering
The University of Edinburgh

2023
To everyone who has been, or will be, adversely affected by the climate change situation, whether through death, illness, displacement, loss of property, or loss of loved ones.
Abstract

Given the extremely high surface temperature of the Sun (~5778 K), solar radiation has the theoretical potential, in accordance with the second law of thermodynamics, to heat a receiver on Earth up to ultra-high temperatures (specified in this thesis as >1300 K). However, there is a gap between theory and practice, as contemporary solar thermal energy systems are still limited to temperatures below 900 K due to material and mechanical limitations. Running solar thermal energy at ultra-high temperatures promises greater energy conversion efficiencies for power plants by upgrading their basic cycles to include more advanced power cycles. Furthermore, the provision of solar thermal energy at ultra-high temperatures can unlock a wide range of energy-intensive industrial applications, including hydrogen and cement production, which can contribute to decarbonising sectors which are difficult to electrify.

This thesis proposes a novel concept of an ultra-high temperature solar cavity receiver based on an optically exposed liquid metal heat transfer fluid, which flows down a corrugated back plate. The concept is investigated using a quasi-steady-state analytical energy model, in addition to a radiation-coupled Computational Fluid Dynamics (CFD) solution. The developed analysis methods are tailored to the proposed class of receivers, nonetheless, they can be generalised for broad solar receiver analysis or for analysing similar problems involving volumetric radiation absorption in other thermal applications. The concept is shown implementable at its absorptive cavity configuration with an overall (optical and thermal) receiver efficiency exceeding 70%. The proposed concept is a step towards narrowing the technological mismatch, in terms of temperature and scale, between state-of-the-art thermal energy storage and concentrated solar thermal at ultra-high temperatures.

A characterisation of prospective ultra-high temperature receivers is presented, which involved a review of state-of-the-art solar thermal technologies with the purpose of identifying the existing challenges to operating at ultra-high temperatures. Based on this characterisation, the proposed receiver is designed to address the literature concerns. The proposed receiver concept involved novel engineering features, including the use of refractory containment materials and a transparent ceramic window to seal the aperture. Therefore, the conceptual investigation attempted to address possible concerns that might be introduced by the new features. Finally, the proposed receiver is demonstrated in a concentrated solar power plant application to emphasise, using quantitative terms, the benefits of operating the receiver at ultra-high temperatures for large-scale applications.
Lay Summary

Exploiting the abundant solar energy on Earth to generate electricity and operate industrial processes is one of the key elements to achieving sustainable development. The amount of solar energy received by the Earth in one hour exceeds the world’s total energy consumption for a year. Nevertheless, <5% of global energy consumption is sourced from solar energy systems due to their early stage of technological development, as they currently lack the economic feasibility edge in many applications against conventional fossil-based systems. The good news is that the research and development efforts over the past decade have succeeded to cut the costs of solar energy technologies more than any other renewable technology, including wind technologies. The rapid decline in the cost of solar energy indicates that with further research and development, solar projects may eventually mature and become financially feasible to replace conventional fossil-based systems. Therefore, this research aims to contribute to the ongoing development efforts in advancing the existing solar energy technologies. Solar energy technologies are classified into two general types: solar photovoltaic and solar thermal energy. This research is concerned with the development of solar thermal energy technologies.

This thesis discloses the technical evaluation of an original solar receiver technology, which is designed to operate at ultra-high temperatures. The provision of such a technology is crucial to improve the efficiency of concentrating solar power plants, facilitate dispatchable power generation from solar energy, and unlock the potential of solar thermal energy in decarbonising vital industrial sectors, such as hydrogen and cement production, which are currently largely reliant on the combustion of fossil fuels. The results and conclusions presented in this thesis pave the way towards the foundation of an ultra-high temperature storage-integrated solar thermal system, which is demonstrated to considerably benefit electricity production from solar power plants. This research also described computational methodologies to analyse the performance of the novel technology without the necessity of implementing costly and environmentally unfriendly preliminary experiments.
Declaration of Originality

I hereby declare that the research presented in this thesis, unless otherwise stated or referenced, are my own work and it was composed in the Institute for Energy Systems, School of Engineering at The University of Edinburgh.

Parts of the work outlined in Chapters 3, 4, 5, and 7 are published in the following journal article (a copy of the article is provided in Appendix 5):


Parts of the computational fluid dynamics work presented in Chapters 6 and 7 are submitted for publication in the following journal articles:


Tarek I. Abdelsalam
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## Nomenclature

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<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$A$</td>
<td>area ($m^2$)</td>
</tr>
<tr>
<td>$a$</td>
<td>amplitude ($m$)</td>
</tr>
<tr>
<td>$I$</td>
<td>incident radiation flux ($Wm^{-2}$)</td>
</tr>
<tr>
<td>$CR$</td>
<td>concentration ratio</td>
</tr>
<tr>
<td>$c$</td>
<td>specific heat capacity ($Jkg^{-1}K^{-1}$)</td>
</tr>
<tr>
<td>$C$</td>
<td>Courant number</td>
</tr>
<tr>
<td>$d$</td>
<td>diameter ($m$)</td>
</tr>
<tr>
<td>$E$</td>
<td>energy ($J$)</td>
</tr>
<tr>
<td>$F$</td>
<td>view factor</td>
</tr>
<tr>
<td>$g$</td>
<td>gravitational acceleration ($ms^{-2}$)</td>
</tr>
<tr>
<td>$G$</td>
<td>solar radiation ($J$)</td>
</tr>
<tr>
<td>$h$</td>
<td>convective heat transfer coefficient ($Wm^{-2}K^{-1}$)</td>
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<tr>
<td>$H$</td>
<td>height ($m$)</td>
</tr>
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<td>$I$</td>
<td>Irradiance ($Wm^{-2}$)</td>
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<td>$K$</td>
<td>turbulence kinetic energy ($J$)</td>
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<tr>
<td>$L$</td>
<td>length ($m$)</td>
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<tr>
<td>$m$</td>
<td>mass flow rate ($kgs^{-1}$)</td>
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<td>$\text{Nu}$</td>
<td>Nusselt number</td>
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<td>$P$</td>
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<td>pressure ($Pa$)</td>
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<td>$Q$</td>
<td>rate of heat transfer ($W$)</td>
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<td>$R$</td>
<td>thermal resistance ($KW^{-1}$)</td>
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<td>$R_{ij}$</td>
<td>Reynolds stresses ($Pa$)</td>
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<td>$t$</td>
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<td>temperature ($K$)</td>
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<tr>
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<td>width ($m$)</td>
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<tr>
<td>$\dot{W}$</td>
<td>mechanical power ($W$)</td>
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<tr>
<td>$v$</td>
<td>velocity ($ms^{-1}$)</td>
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### Subscripts

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<td>$\text{abs}$</td>
<td>absorption</td>
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<td>$\text{ap}$</td>
<td>aperture</td>
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<td>convection</td>
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<td>cavity</td>
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<td>$\text{cf}$</td>
<td>cavity fluid</td>
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<td>$\text{ch}$</td>
<td>characteristic</td>
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<td>$\text{conv}$</td>
<td>energy conversion</td>
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<td>focal</td>
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<td>heliostats</td>
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<tr>
<td>$\text{o}$</td>
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<td>$\text{p}$</td>
<td>at constant pressure</td>
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<td>radiation</td>
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<td>$\text{rv}$</td>
<td>recovered energy</td>
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<td>$\text{rec}$</td>
<td>receiver</td>
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<td>$\text{ref}$</td>
<td>reflection</td>
</tr>
<tr>
<td>$\text{sec, ref}$</td>
<td>secondary reflection</td>
</tr>
<tr>
<td>$\text{sol}$</td>
<td>solar</td>
</tr>
<tr>
<td>$\text{th}$</td>
<td>thermal</td>
</tr>
<tr>
<td>$\text{w}$</td>
<td>cavity walls</td>
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<td>$\text{win}$</td>
<td>window</td>
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### Greek and Coptic symbols

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<thead>
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<th>Symbol</th>
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<tr>
<td>α</td>
<td>optical absorptivity</td>
</tr>
<tr>
<td>β</td>
<td>volume fraction</td>
</tr>
<tr>
<td>Γ</td>
<td>mass flow rate per unit wetted perimeter (kgm(^{-1})s(^{-1}))</td>
</tr>
<tr>
<td>γ</td>
<td>surface tension (N.m(^{-1}))</td>
</tr>
<tr>
<td>Δts</td>
<td>time-step (s)</td>
</tr>
<tr>
<td>δ</td>
<td>thermal expansion coefficient (K(^{-1}))</td>
</tr>
<tr>
<td>ε</td>
<td>turbulence dissipation rate (m(^{2})s(^{-3}))</td>
</tr>
<tr>
<td>ε</td>
<td>emissivity</td>
</tr>
<tr>
<td>θ</td>
<td>aperture plane tilt angle (rad)</td>
</tr>
<tr>
<td>η</td>
<td>efficiency</td>
</tr>
<tr>
<td>λ</td>
<td>wavelength (μm)</td>
</tr>
<tr>
<td>μ</td>
<td>dynamic viscosity (kgm(^{-1})s(^{-1}))</td>
</tr>
<tr>
<td>ρ</td>
<td>density (kgm(^{-3}))</td>
</tr>
<tr>
<td>σ</td>
<td>Stefan-Boltzmann constant (Wm(^{2})K(^{-4}))</td>
</tr>
<tr>
<td>φ</td>
<td>inclination angle (rad)</td>
</tr>
<tr>
<td>ϑ</td>
<td>polar angle (rad)</td>
</tr>
<tr>
<td>ϕ</td>
<td>phase function</td>
</tr>
<tr>
<td>ψ</td>
<td>azimuthal angle (rad)</td>
</tr>
<tr>
<td>ω</td>
<td>specific turbulence dissipation rate (s(^{-1}))</td>
</tr>
<tr>
<td>Ω</td>
<td>solid angle (sr)</td>
</tr>
<tr>
<td>ξ</td>
<td>specific latent heat (Jkg(^{-1}))</td>
</tr>
<tr>
<td>†</td>
<td>fraction of input energy lost through aperture</td>
</tr>
</tbody>
</table>

### Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Definition</th>
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<tbody>
<tr>
<td>BSL</td>
<td>baseline model</td>
</tr>
<tr>
<td>CAD</td>
<td>computer-aided design</td>
</tr>
<tr>
<td>CCGT</td>
<td>combined cycle gas turbines</td>
</tr>
<tr>
<td>CFD</td>
<td>computational fluid dynamics</td>
</tr>
<tr>
<td>CPV-T</td>
<td>concentrated photovoltaic thermal system</td>
</tr>
<tr>
<td>CSP</td>
<td>concentrated solar power</td>
</tr>
<tr>
<td>CST</td>
<td>concentrated solar thermal</td>
</tr>
<tr>
<td>DNI</td>
<td>direct normal irradiation</td>
</tr>
<tr>
<td>DOM</td>
<td>discrete ordinates model</td>
</tr>
<tr>
<td>HTF</td>
<td>heat transfer fluid</td>
</tr>
<tr>
<td>LES</td>
<td>large eddie simulation</td>
</tr>
<tr>
<td>PV</td>
<td>photovoltaic</td>
</tr>
<tr>
<td>RANS</td>
<td>Reynolds-averaged Navier-Stokes</td>
</tr>
<tr>
<td>RSM</td>
<td>Reynolds stress model</td>
</tr>
<tr>
<td>SST</td>
<td>shear-stress transport</td>
</tr>
<tr>
<td>VOF</td>
<td>volume of fluid</td>
</tr>
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Chapter 1. Introduction

Sustainable development is a modern concept coined by the UN in its “Our Common Future” report (Brundtland, 1987) to establish transnational developmental goals which address contemporary global economic and social challenges, while preserving the environment and its natural resources for future generations. This abstract concept has expanded since the release of the report to include aspects of social inclusion and environmental sustainability for economic growth (Sachs, 2015). A central aspect of the environmental sustainability is preserving the availability and diversity of natural resources, which necessitates redesigning the consumptive activities of humans as they meet their needs. The UN’s Food and Agriculture Organisation (FAO) identified three key sectors for sustainable development: water, energy, and food (FAO, 2014). Achieving energy sustainability entails meeting the global energy demand without compromising the energy resources for future generations, in addition to protecting the environment against greenhouse gas emissions, which are the largest drivers for global warming and human-induced climate change (Olivier and Peters, 2020). One solution is shifting the current dependence on fossil fuels to renewable energy resources for supplying the energy requirements. However, such replacement must be facilitated with technologies that conserve the use of other natural resources, including water, land, and Earth materials.

1.1. Energy transition: challenges and solutions

In their first special report, which was commissioned as a part of the 2015’s Paris climate agreement, the Intergovernmental Panel on Climate Change (IPCC) announced that the world needs to limit the global temperature rise at 1.5 °C above pre-industrial levels and achieve a net zero global CO₂ emissions by 2050 to alleviate the catastrophic environmental consequences of the climate change (IPCC, 2018). Meeting this limit is projected to necessitate fundamental changes at revolutionary pace and scale to existing energy production and consumption systems, while the report indicated the world was completely off track at the time and heading towards a global temperature rise of 3 °C. One of the central targets proposed in the report is sourcing at least 70% of global electricity from renewables by 2050, nevertheless,
renewables share of global electricity production has only reached 28.3% by the end of 2021 (REN21, 2022). As illustrated in Figure 1.1, if the slow growth trend of renewable electricity during the 2010s continued, which increased by <10%, meeting the 70% target would be delayed by about a decade unless innovative solutions are put in place to overcome the challenges facing the penetration of renewables into the grid.

![Figure 1.1. Global power production by source (REN21, 2021). The black line represents the share of renewable electricity, which is scaled to the secondary y-axis on the right.](image)

Despite the unsatisfactory progress of green electricity, the current pace of the energy transition is furtherly lagging behind the electricity transition. In their 2021 review reports, REN21 and IRENA emphasised the importance of speeding up the green transition progress in thermal applications and transport, as policy makers still focus mainly on the power production sector (REN21, 2021, IRENA, 2021b). The reports demonstrated evidences that innovative solutions and systems can accelerate the transition processes; however, such solutions are advised to maintain a considerate and sustainable exploitation and management of rare earths and other minerals. As shown in Figure 1.2, the rapid expansion of renewables power is a key component in the first phase of IRENA’s strategy to meet the IPCC target.
1.1.1. Solar energy

Solar radiation is electromagnetic radiation emitted by the Sun, which is a combination of bright light and radiant heat. To appreciate the abundance of this energy, the Earth’s atmosphere receives about 174 PW worth of solar energy, which is roughly 10,000 times the total energy consumption by humans (Rhodes, 2010). There are two general methods for harvesting this clean and renewable energy: (1) conversion of sunlight into electricity using the photovoltaic effect or (2) harnessing the solar radiant heat to produce thermal energy contained in a heat transfer medium. Solar energy is utilised in a wide range of applications, including power production, water pumping, district heating, operating industrial processes, and space applications (IEA, 2011).
Nearly 99% of the solar energy is contained within the 0.15-4 μm wavelength band (Bhatia, 2014). Figure 1.3 displays the solar spectrum, at the surface temperature of the Sun (~6000 K), across the ultraviolet, visible, and infrared wavelengths. While solar power peaks at the visible range (~0.5 μm), more than half of the solar energy content is enclosed by the ‘near-infrared’ band (Akbari et al., 2006). A more realistic model of the solar irradiance will likely use a lower surface temperature of the Sun (<5800 K), which entails shifting the peak towards the infrared band and, in turn, growing the share of energy within the near-infrared band. Conventional PV cells have limited absorption ranges, particularly in the infrared band, compared to solar thermal absorbers as demonstrated in Figure 1.4. Furthermore, solar thermal offers greater energy conversion efficiencies, as PV is conceptually limited by the Shockley–Queisser efficiency limit (Shockley and Queisser, 1961).

Figure 1.3. Peak-normalised solar spectral power at sea level distributed among different wavelength bands (Akbari et al., 2006).

Figure 1.4. Operational wavelength ranges for different PV cells versus a solar thermal absorber (Fei Guo et al., 2014). The blue curve represents the solar spectrum. GaAs is Gallium arsenide; C-Si is crystalline silicon; A-Si is amorphous silicon.
In 2019, the International Energy Agency (IEA) presented a positive perspective on solar technologies based on their “unprecedented deployment and cost reductions” over the past decade (IEA, 2019). The report demonstrated the rapid penetration of solar thermal technologies into industrial applications and district heating networks. The report also highlighted the recent commercial interest in dispatchable power from mixed PV and concentrating solar power (CSP) plants, with projected shares displayed in Figure 1.5. The report, nonetheless, perceives the penetration of PV and CSP into the power market as complementary rather than mutually exclusive, given CSP’s potential for integration with TES technologies that enable shifting supply after sunset, as shown in Figure 1.6. If this scenario upholds, the dependence of PV growth on storage-integrated CSP may be a driving force for the latter’s growth.

Figure 1.5. Projected growths of PV and CSP in the global electricity production (IEA, 2019). The magnitudes of electricity production (TWh) are scaled based on the primary y-axis, while shares of solar sources are scaled based on the secondary y-axis.

Figure 1.6. A hypothetical daily electrical supply and demand profiles demonstrating the benefit of mixing electricity from PV and TES integrated CSP for medium-term (top plot) and long-term (bottom plot) shifting durations (IEA, 2019).
1.1.2. Variable renewable energy

As the renewables share grows, more changes to the system’s operational mechanisms become required, which is a central challenge to the growth of renewables. The natural intermittency of renewable resources results in variable renewable energy that is non-dispatchable by the system operators. Furthermore, these variations differ in frequency and periods from one renewable source to another. For example, hydropower, biomass, and geothermal variate on an annual or seasonal basis, while solar and wind variate on much smaller time scales (Bessa et al., 2014). While a small share of variable renewable energy in the system can be accommodated by the traditional one-way energy systems, a renewables-dominated energy system would typically require an interactive operational system through smart grids and demand-side management (Goulden et al., 2014) or integrated energy storage (Beaudin et al., 2015) to control the significant supply variability.

The intrinsic variability of renewable energy is independent of the demand profile, which means the peak supply from a renewable source does not naturally synchronise with the peak demand. This designates two system problems: an overload during higher supply than demand and a supply shortage during lower supply than demand. These problems are manifested in solar power systems, as the produced electricity typically peaks during the daytime, while the electricity demand usually peaks during the night. As a result, grid operators are sometimes forced to apply unfavourable measures, such as energy curtailment, discharge through dump loads, or running fossil-powered backup generators to maintain the balance of power supply and demand at every instance to sustain the supply quality (e.g. maintaining constant frequency and voltage in the grid). Such measures result in wasting generated or available green energy, which could have been reserved to supply future demand.

1.1.3. Energy storage

Meeting the IPCC’s demanding targets without compromising the energy stability and supply quality require the provision of feasible and reliable solutions to the intermittent nature of renewable energy. Innovative technological solutions accompanied by supportive forecasting and management tools can enable control over the rapid solar and wind power variabilities (Bessa et al., 2014). For example, network
interconnection of power networks and demand-side response management can manage the electrical intermittency in a grid without wasting the available energy, while adopting smart grids facilitate near real-time monitoring of supply and demand across different nodes in the network (Goulden et al., 2014). Nevertheless, energy storage remains the key solution for delivering a fully controllable solution for a total renewable energy system (Garvey et al., 2015, IRENA, 2017). Yet, the current global energy storage capacity (~160 GW) represents only 2.5% of global electricity production capacity (Datasa, 2021a).

Energy can be stored in various forms, as illustrated in Figure 1.7. Some storage systems may be more feasible than others depending on their stages of technological development, applicability to different energy sources, flexibility to different operating conditions, and availability of the resources of their components.

For electricity, pumped hydro is the dominating energy storage technology, accounting for ~96% of the total global storage capacity in 2020, while thermal energy storage (TES) and batteries constituted only 1.9% and 1.1% of the global storage capacity, respectively (Sandia, 2020). This leaves ~1% for all other energy storage technologies. However, the domination of pumped hydro is expected to decline over the next decade with the rapidly growing shares of competing technologies, driven by anticipated cost reductions, while the geographically limited sites for large-scale pumped hydro have already been exploited (IRENA, 2017, Deane et al., 2010). Battery storage is commonly associated with frequency regulation in smart grids for electricity sourced from wind and solar photovoltaics (PV) (Swierczynski et al., 2014, Kirli and
Kiprakis, 2020). The economic feasibility of batteries is more justifiable for off-grid renewable systems in remote locations (Corcuera et al., 2015, Abdelsalam et al., 2014); however, their characteristic degradation, use of costly materials, and environmental impact have limited their use in large-scale power plants (Yao et al., 2016).

The technical characteristics of different storage technologies are presented in Table 1. The different characteristics indicate that these technologies are not mutually exclusive; instead, they can form integrated solutions to address the variability of renewables for different scales and applications, as illustrated in Figure 1.8.

Table 1. Technical comparison between different energy storage technologies. Data obtained from the World Energy Council (2016).

<table>
<thead>
<tr>
<th>Storage Technology</th>
<th>Max. Power Rating (Mw)</th>
<th>Discharge Period</th>
<th>Lifetime/No. Of Cycles</th>
<th>Energy Density (Wh/Litre)</th>
<th>Efficiency (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pumped hydro</td>
<td>3,000</td>
<td>4-16 hours</td>
<td>30-60 years</td>
<td>0.2-2</td>
<td>70-85</td>
</tr>
<tr>
<td>Compressed air</td>
<td>1,000</td>
<td>2-30 hours</td>
<td>20-40 years</td>
<td>2-6</td>
<td>40-70</td>
</tr>
<tr>
<td>Thermal (molten salt)</td>
<td>150</td>
<td>4-18 hours</td>
<td>30 years</td>
<td>70-210</td>
<td>80-90</td>
</tr>
<tr>
<td>Lithium-ion batteries</td>
<td>Up to 8 hours</td>
<td>2-5 years</td>
<td>200-400</td>
<td>85-95</td>
<td></td>
</tr>
<tr>
<td>Lead-acid batteries</td>
<td>Up to 8 hours</td>
<td>3-15 years</td>
<td>50-80</td>
<td>80-90</td>
<td></td>
</tr>
<tr>
<td>Flow batteries</td>
<td>100</td>
<td>2-8 hours</td>
<td>12,000-14,000 cycles</td>
<td>20-70</td>
<td></td>
</tr>
<tr>
<td>Hydrogen</td>
<td>Up to a week</td>
<td>5-30 years</td>
<td>600 (at 200 bar)</td>
<td>25-45</td>
<td></td>
</tr>
<tr>
<td>Mechanical (flywheel)</td>
<td>20</td>
<td>&lt; 1 hour</td>
<td>20,000-100,000 cycles</td>
<td>20-80</td>
<td></td>
</tr>
</tbody>
</table>

Combining storage with generation, rather than operating them as standalone systems, improves the economic feasibility of both systems by sharing the infrastructure and eliminating unnecessary grid transmission and roundtrip energy transformations, with estimated throughput efficiency increases of up to 17.7% and reduced marginal system costs of 20-45% (Garvey et al., 2015). This marks a characteristic advantage for CSP-TES integrated systems over other renewables, which produce energy in a different form than storage.
1.2. Decarbonisation of thermal energy

Thermal energy is used in a wide range of applications and is commonly used as an intermedier in energy conversion systems. For example, when burning fossil fuels for power production, chemical energy is initially converted into thermal energy before being used to run a heat engine to generate electricity. Thermal energy is the second most consumed user-end form of energy after electricity (Alva et al., 2018). Therefore, the decarbonisation of thermal energy is crucial to the energy transition process.

Electricity is the globally preferred energy carrier, as it can transmit energy over long distances via electrical grids while incurring lower losses than heat transfer (Belyakov, 2019). However, energy cannot be stored in form of electricity. Additionally, electricity is expensive, as it is not directly available in nature but needs to be generated from other forms of energy, mainly thermal energy. Therefore, heat can be more practical for short-distanced or localised energy transport, such as transporting generated energy from CSP to a local TES system via heat transfer fluid (HTF). While heat transfer is subjected to conductive, convective, and radiative losses to the ambient, thermal losses can be mitigated by insulation and energy recovery.
1.2.1. Solar thermal energy

Solar thermal collectors are devices that collect energy by absorbing and converting the solar radiant heat into internal energy in an absorber material. The U.S. Energy Information Administration (EIA) classifies solar thermal collectors into three heat grades based on the temperature level (EIA, 2012). The low-temperature collectors use unglazed absorbers to deliver heat at temperatures <316 K for low-grade water and space heating applications, such as swimming pool heating and heating of ventilation air. Medium-temperature collectors use glazed flat-plates, or evacuated tubes, which enclose a flowing heat transfer fluid, typically water or air, to deliver heat at temperatures up to 363 K, which can be used for domestic water or space heating. The high-temperature collectors use concentrators to deliver heat at temperatures greater than the boiling point of water (373 K), which can be used to produce steam for power production and other utility-scale applications. Figure 1.9 displays the four main high-temperature solar thermal technologies. Further details about the technical aspects of highly concentrating collectors will be covered in Chapter 2.

![Figure 1.9. High-temperature solar thermal technologies categorised based on the type of concentration and mobility of their solar receivers (IEA, 2014).](image-url)
During the 2010-2020 decade, the growth of medium-temperature water heating collectors globally superseded the unglazed low-temperature collectors (REN21, 2021). As of 2020, the installed water heating capacity has exceeded the 500 GW\textsubscript{th} globally, which is equivalent to burning 239 million barrels of oil (REN21, 2021). In the power production sector, CSP did not display an equivalent growth. The two main market leaders of CSP, Spain and the United States, stalled their growing CSP capacities for six years after 2014, as shown in Figure 1.10. This was mainly attributed to the removal of financially incentive programmes, such as the termination of feed-in tariffs in Spain (Jason, 2017). However, this stall is likely to end in both countries with new projects being commissioned for the 2020s decade (Gonzalo, 2020, REN21, 2021). Furthermore, the rapid CSP growth in China, India, Middle East, and Latin America has started to join the market with greater commissions for the 2020-2030 decade (REN21, 2021).

During the 2010s decade, CSP technologies experienced a 68% cost reduction, which was the second largest cost reduction for all renewables technologies after solar PV, as illustrated in Figure 1.11 (IRENA, 2021a). The main factors that contributed to CSP’s substantial cost reduction included the advancement of CSP and TES technologies, increased competitiveness in the supply chain, and direction towards combined solar PV with TES-integrated CSP projects (Enkhardt, 2021, REN21, 2021). This cost-reduction trend is projected to prevail if new high-temperature HTFs are utilised in solar receivers and TES systems (Achkari and El Fadar, 2020).
Figure 1.11. Levelised cost of electricity for different utility-scale renewable power generation technologies commissioned during the 2010-2020 period (IRENA, 2021a).

1.2.2. Thermal energy storage

TES is perceived as the most suitable energy storage for thermal power plants, such as nuclear reactors and CSP plants (Denholm et al., 2012, Alva et al., 2018). TES is also used for seasonal energy storage, where excess or waste heat from any source is stored during summer months to be used in winter for district heating and other usages (Kalaiselvam and Parameshwaran, 2014b). For dealing with variable renewable energy, TES is predominantly used in capacity firming, energy time-shift and onsite generation shifting, as illustrated in Figure 1.5. The use of TES is currently dominated by CSP plants, as they store excess energy generated during nominal operation to then be dispatched during the evening or around the clock.

TES is a key element to the economic viability of large-scale solar thermal energy projects (Achkari and El Fadar, 2020). The 2017-2020 global CSP growth outside Spain and the United States synchronised with TES growth, as shown in Figure 1.13. Nevertheless, at least 54.5% of global operational CSP plants operate without an integrated TES system (Achkari and El Fadar, 2020). For district and space-heating solar collectors, it was found that seasonal TES could boost the solar fraction value, which is the share of demand met from solar energy, to up to 80% (Kalaiselvam and Parameshwaran, 2014b).
TES technologies are categorised based on the thermal storage medium. The main type is *sensible heat storage*, which uses a material (e.g. molten salts, oils, ceramics, rocks, or silicon) to store heat at a higher temperature utilising the medium’s high heat capacity. The second type is *latent heat storage*, which uses a phase-changing material (e.g. organic, inorganic, or eutectic alloys) to store heat utilising the medium’s high latent heat capacity. The third type is *thermochemical storage*, which uses pair of chemically reactive materials (e.g. silica-gel/water, magnesium-sulphate/water, or lithium-bromide/water) to undergo reversible endothermic and exothermic reactions to store the heat (Kalaiselvam and Parameshwaran, 2014a).
Due to the low overall efficiency and complexity of thermochemical storage technologies, such as the requirement for catalysts and safety measures, they are currently limited to small-scale applications (Wang, 2019). State-of-the-art latent heat storage systems, with their high energy densities, offer a promising potential for large-scale applications with the advantage of operating at nearly isothermal conditions with roundtrip thermal storage efficiencies ranging between 75-90% (Sarbu and Sebarchievici, 2018). However, various practical considerations has limited the spread of latent heat storage technologies, which are reviewed by Kousksou et al. (2014), Mohamed et al. (2017), Saha et al. (2021), and Sharma et al. (2009).

Metals are recently investigated as phase change storage mediums in the literature; however, the main limitation was their relatively small heat of fusion per unit weight compared to other materials (Saha et al., 2021). Another concern for using metals as storage mediums is the fact that metals naturally exist in form of complex compounds, or eutectic mixtures, with other materials, hence, requiring additional pre-treatment processes that typically produce CO$_2$. A study by Khare et al. (2012) evaluated the environmental credentials of using metals as phase change storage mediums, as shown in Figure 1.14. The study concluded that heavy metals and their alloys were still best candidates for high temperature (673-1023 K) phase change storage mediums, as they demonstrated a favourable compromise between thermal properties (e.g. heat of fusion, thermal conductivity… etc.) and life cycle environmental impact.

![Figure 1.14](image_url)

Figure 1.14. The environmental impact, in terms of produced quantities of CO$_2$, of using selected liquid metals and their alloys as phase change storage mediums (Khare et al., 2012). The evaluation of CO$_2$ production accounted for their manufacturing processes, working life, recyclability, and disposal.
Despite the promising potential of latent heat storage technologies, molten salt thermal storage remains the dominating TES technology, as it represents three quarters of the total installed TES capacity around the globe by 2017, as shown in Figure 1.15. Molten salts are extensively used in high temperature (573-838 K) energy applications, mainly with CSP, to utilise their favourable thermo-physical properties (e.g. volumetric capacity, boiling point, temperature stability, vapour pressure... etc.) compared to water (Caraballo et al., 2021). Their feasibility as a storage medium is also reinforced by their non-toxicity, non-flammability, relatively low-cost, and its potential to be used as a heat transfer fluid in the solar circuit compared to thermal oils (Gil et al., 2010). The main limitations of molten salts are their high viscosities, low thermal conductivities, and high boiling points (>473 K), which require anti-freeze measures to avoid solidification in circuit pipes (Alva et al., 2018).

Between the geographically restricted pumped hydro and batteries experiencing cyclic-induced degradation, TES has emerged as a more flexible and reversible alternative for renewable energy systems (Andrijanovits et al., 2012). However, they are currently hampered by their low round-trip efficiencies and energy densities, which limit their economic feasibility. Operating at ultra-high temperatures can overcome these limitations and unlock the true potential of TES technologies, while storage and energy recovery efficiencies can be maximised by increasing the size and temperature, respectively (Robinson, 2017).

![Figure 1.15. The installed capacities (as of 2017) of different TES, chemical, and mechanical storage systems (IRENA, 2017).](image)
1.2.3. Ultra-high temperature thermal energy solutions

Heat can be useless, regardless of its energy quantity, if the energy quality is low. This limitation is imposed by the second law of thermodynamics, which prohibits a complete conversion of “low-quality” energy into “high-quality” energy (Daintith, 2005). In Applied Physics and Engineering contexts, temperature is used a measure of the quality of thermal energy (Çengel, 2019, Oanhāra, 2009). The thermodynamic potential of a system to produce a useful output is defined based on the type of output. For example, enthalpy is the potential to do non-mechanical work and release heat, while Helmholtz energy is the potential to do mechanical and non-mechanical work (Silbey and Alberty, 2001). Another theoretical concept defining the useful fraction of energy in a thermodynamic system is Exergy, which is the maximum amount of work that can be extracted from a system before reaching equilibrium with its environment (Perrot, 1998). The thermodynamic potential and exergy are both proportional to the product of temperature and entropy. Therefore, to maximise the availability and usefulness of a thermal system, its temperature needs to be maximised, hence, the motivation for developing ultra-high temperature thermal energy systems.

Similarly to TES, solar thermal technologies also benefit from operating at ultra-high temperatures, which may sound counter-intuitive as thermal losses typically increase with temperature. Operating at ultra-high temperatures enable processing of energy-dense heat using compact solar receiver designs, which minimise the surface area to volume ratios, hence, minimise the gates for thermal losses. Compact receiver designs are particularly preferable for solar power towers for structural reasons, as will be discussed in Chapter 2.

Though not yet developed as solar thermal power production, solar thermochemical applications use similar solar concentrating concepts and benefit from operating at high temperatures (Steinfeld, 2005). Delivering the heat at ultra-high temperatures can unlock new applications for concentrating solar thermal (CST) systems. The prospective energy-intensive applications presented in Table 2 are currently powered by fossil fuel combustion and are hard-to-decarbonise. These applications are responsible for ~21% of global CO₂ emissions (IPCC, 2014). For instance, cement production is responsible for nearly 8% of global CO₂ emission (Ellis et al., 2020).
Table 2. Potential thermochemical applications for ultra-high temperature CST.

<table>
<thead>
<tr>
<th>Thermal Application</th>
<th>Operation Temperature (K)</th>
<th>Sources</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hydrogen production by thermochemical water-splitting</td>
<td>&gt;1200 (reduction process)</td>
<td>(Andrijanovits et al., 2012; Muhich et al., 2016; Zheng and Xu, 2018)</td>
</tr>
<tr>
<td>Cement production</td>
<td>1723 K</td>
<td>(Ellis et al., 2020)</td>
</tr>
<tr>
<td>Glass manufacturing</td>
<td>1273-1773 K</td>
<td>(Ahmad et al., 2017)</td>
</tr>
<tr>
<td>Processing of refractory materials (e.g. zirconium, niobium, tantalum…etc.)</td>
<td>&gt;2000 K</td>
<td>(Jin et al., 2019)</td>
</tr>
</tbody>
</table>

CSP technologies are largely disadvantaged by their low solar-to-electric energy conversion efficiencies against more mature renewables, such as wind and solar PV. The most recent status review of commercial CSP plants is provided by Imran Khan et al. (2023). In principle, the energy conversion efficiency of a power plant can be improved by maximising the heat addition temperature of its thermodynamic cycle (Çengel, 2019). Current CSP power plants, which operate in the solar-only mode (i.e. without hybridisation with a fossil fuel burner), can only operate basic power cycles, mainly the subcritical Rankine cycle, which limits the thermal conversion efficiencies, as illustrated in Figure 1.16. Increasing the temperature would enable the use of more advanced power cycles, which can improve the energy conversion efficiency by up to 50% (Glatzmaier, 2011, Stein and Buck, 2017, Dunham and Iverson, 2014).

Candidate power cycles include reheat supercritical Rankine cycles, which can boost the cycle efficiency by >40% and lower the levelised cost of electricity by >4%; however, this efficiency gain can only be promising at larger scales (>250 MW_{elec}) than current capacities of commercial CSP plants (Peterseim and Veeraragavan, 2015). Supercritical CO$_2$ Brayton cycles are compatible with high-temperature HTF, such as liquid metals in nuclear reactors (Cheang et al., 2015, Liu et al., 2019b), and they offer better performance than steam cycles, which suffer substantial cooling losses for condensing steam before pumping. Furthermore, gas turbine combined cycles offer the current highest cycle efficiencies (>60%) with a well-established experience in power production; however, they require ultra-high turbine inlet temperatures (>1273 K) to operate feasibly (Stein and Buck, 2017, Mohammadi et al., 2020).
The main concern for running an energy system at ultra-high temperatures is the resulting thermal losses, mainly radiative, to the colder ambient. Numerous engineering solutions have been developed over the years to minimise the thermal losses from energy systems, such as radiative barriers and advanced insulation materials (Hoffschmidt et al., 2012). Another approach is recovering the escaping thermal energy using PV (Weinstein et al., 2018) or heat pumps (Robinson, 2018).

The current global interest in ultra-high temperature technologies was manifested in the “First International Workshop on Ultra High Temperature Thermal Energy Storage, Transfer and Conversion”, which the author attended in November 2019. The proceedings of this workshop are published in (Datas, 2021a). This workshop was an outcome of the AMADEUS project, which aimed to create new materials and devices to facilitate TES operation at the temperature range of 1273-2273 K (Datas et al., 2020). Interest in ultra-high temperature applications was shown to expand beyond academia and even Earthly applications. At MIT, a new concept was proposed to thermally store grid electricity using a multi-junction PV being used as a heat engine integrated with silicon-based two-tank TES, which operated between 2173-2673 K (Amy et al., 2019). This concept was found to deliver a much cheaper (up to 100 times) storage operation than direct electricity storage, which outweighs the penalty of energy conversion. In South Australia, a company by the name “1414” has commercialised the first large-scale ultra-high temperature TES systems using silicon (Parham et al., 2003).
In space applications, where compactness and energy efficiency are prioritised over cost-effectiveness, ultra-high temperature CSP and TES systems are studied with a promising perspective for their use (Gilpin and Datas, 2021, Datas and Martí, 2017).

1.3. Research aim and motivation

In line with the growing interest and recent advancement of ultra-high temperature thermal energy solutions, this PhD thesis extends this momentum within the solar thermal energy field. Solar receivers in commercial CSP systems currently operate at temperatures <800 K. Reaching higher receiver temperatures is theoretically and practically attainable, as will be demonstrated in Chapter 2; however, conventional systems are constrained by the materials and mechanical properties of their components (Glatzmaier, 2011). This marks the gap between current solar thermal technologies and their theoretical potential, which the research here attempts to bridge.

This research aims to deliver an in-depth scientific understanding of how a solar thermal system can collect and process heat at ultra-high temperatures to, ultimately, enable integration with a state-of-the-art TES system. Such an integrated system can benefit the generation and storage systems, as illustrated in Figure 1.17, while cutting their inhibitive capital costs by sharing the infrastructure (Garvey et al., 2015). Given the proven feasibility of TES at ultra-high temperatures, developing a matching solar thermal concept is considered the next step towards the desired integrated system.

In this thesis, a novel solar thermal receiver technology is described and evaluated for the purpose of collecting, processing, and delivering energy at ultra-high temperatures. The developed technology introduces novel technical concepts, such as using an optically exposed liquid metal heat transfer fluid flowing over a corrugated plate, which are investigated using analytical and numerical methodologies. While tailored to the proposed receiver technology, the developed analysis methods can be utilised in analysing solar cavity receivers or applications involving similar flow and radiation configurations. While improving the energy conversion efficiency of CSP plants is a primary target of this work, a secondary motivation of this work is to enable the use of solar thermal energy to power hard-to-decarbonise thermal applications, such as hydrogen and cement production.
The objectives of this research are:

1. Establish theoretical limits of temperature for different working principles used by solar thermal energy technologies (Chapter 2);
2. Review the prospective technologies and nominate one to upgrade (Chapter 2);
3. Determine the essential technical characteristics for an ultra-high temperature operation by the solar receiver (Chapter 3);
4. Demonstrate the design features of the novel solar receiver (Chapter 4);
5. Develop a computationally feasible design tool using analytical and semi-empirical formulations to model the quasi-steady-state performance of the proposed receiver and study its characteristics (Chapter 5);
6. Develop a radiation-coupled CFD model to investigate the transient and specular characteristics of the proposed receiver (Chapter 6);
7. Evaluate the receiver performance and verify the analytical results (Chapter 7);
8. Demonstrate the proposed receiver in a power plant application to study the influences of receiver temperature and plant capacity on the overall performance of the application (Chapter 8).
Chapter 2. Conceptual Study on Ultra-high Temperature Solar Thermal Energy

The characteristics of heat transfer from the Sun to solar collectors are subject to a set of scientific laws, mainly the four laws of thermodynamics and the four laws of radiation. In light of these physical laws, the theoretical potential and constraints for delivering ultra-high temperature solar thermal energy are investigated in the first section of this chapter. The state-of-the-art concentrating solar thermal technologies are then reviewed, while following the process of elimination strategy to determine the prospective solar thermal energy technology with a potential for:

1. Delivering heat at ultra-high temperatures (specified here as >1300 K);
2. Compatibility with TES systems;
3. Applicability at large-scales (specified here as >5 MW\textsubscript{th} or >1 MW\textsubscript{elec}).

2.1. Fundamental principles for ultra-high temperature operation

The Photosphere, which is the visible surface of the Sun from Earth, radiates energy flux, in form of light and heat, towards the Earth at an annual average rate of 1361 Wm\textsuperscript{-2}. The Earth’s atmosphere then attenuates this flux via scattering and absorption, reducing the average incident irradiance reaching the Earth’s surface to ~1000 Wm\textsuperscript{-2} on a clear sunny day. This incident irradiation, however, varies considerably across the globe. For instance, the UK annual average solar irradiation ranges between 71.8 Wm\textsuperscript{-2} in the northwest of Scotland to 128.4 Wm\textsuperscript{-2} in the south of England (Burnett et al., 2014), while it exceeds 1000 Wm\textsuperscript{-2} in Northern Australia (ABARE, 2010). To account for the temporal variability of solar energy, prompted by recurrent night and cloudy periods, it is practical to represent the density of solar resources by the annual average incident insolation (direct normal irradiation), as shown in Figure 2.1.

\[1 \text{ This limit is based on existing commercial solar thermal technologies. The most recent breakthrough in the commercial setting was achieved by Bill Gates’ start-up, namely Heliogen, which recorded temperatures reaching 1273 K. Source: Nield, D. (2019) A Promising Solar Energy Breakthrough Just Achieved 1,000-Degree Heat From Sunlight. ScienceAlert, See https://www.sciencealert.com/ai-plus-sunlight-equals-hotter-solar-ovens-and-no-need-for-fossil-fuels (accessed 11-1-2022 2022).}

\[2 \text{ This limit is based on the definition of utility-scale solar determined by the Solar Energy Industries Association (SEIA). Source: Seia (2022) Major Solar Projects List, See https://www.seia.org/research-resources/major-solar-projects-list (accessed 31-1-2022 2022).}
2.1.1. Effect of concentrating solar radiation on the receiver temperature

In theory, a perfectly insulated blackbody receiver on Earth can reach temperatures up to the effective temperature of the radiation source (the photosphere), which is ~5,778 K (Mullan, 2010). However, the energy flux incident on the Earth is insufficient to sustain a real solar receiver at a temperature >364 K (Fletcher, 2001). Nevertheless, if the solar radiation received by a large area was condensed onto a smaller area, the resulting condensed flux can then facilitate greater receiver temperatures. This method is called concentrating solar radiation, which is a central concept for high-temperature solar applications. CSP and concentrated PV offer higher capacity factors than non-concentrated solar technologies; however, the cost of their indispensable tracking devices is a major limitation to their economic feasibility for small-scale (<1 MW_{elec}) power applications (Malik et al., 2010).

To quantify the impact of concentrating solar radiation on receiver temperature, we first define the geometric concentration ratio ($CR_g$), which is the area ratio of the concentrator aperture to the receiver. In practice, optical losses occur between the concentrator and receiver, which depend on the types of solar technology used, as will be explained in more detail in Chapters 3 and 5. To account for these optical losses, another dimensionless metric is defined as the optical concentration ratio ($CR_o$), which is the energy flux ratio perceived by the concentrator aperture to the receiver. The two concentration ratios, which are often confused in literature, are related by:
where \( \eta_o \) is the optical efficiency between the concentrator and receiver. When ideal optical conditions are assumed, \( \eta_o \) becomes unity; hence, both ratios become similar.

Assuming a perfectly insulated blackbody receiver with perfect optical properties, the ideal solar receiver efficiency \( \eta_{rc,\text{ideal}} \) can be expressed as:

\[
\eta_{rc,\text{ideal}} = \frac{I \, CR_g - \sigma T^4}{I \, CR_g}
\]

where \( I \) is the incident irradiance (taken here as 1000 Wm\(^{-2}\)), \( \sigma \) is Stefan-Boltzmann constant (evaluated as \( 5.67 \times 10^{-8} \) kg s\(^{-3}\) K\(^{-4}\)), and \( T \) is the temperature.

Using Equation (2), the variation of ideal receiver efficiency with its temperature is plotted in Figure 2.2. Expectedly, the receiver efficiency decreases as its temperature increases due to increased blackbody emissive loss. Accordingly, the theoretical maximum temperature that can be obtained under certain incident flux conditions is when the receiver efficiency reaches zero (i.e. when all energy received is lost through blackbody emission). Therefore, the maximum receiver temperature can be expressed in terms of the concentration ratio as follows:

\[
T_{\text{max}} = \frac{4 \sqrt{I \, CR_g / \sigma}}{}
\]

Figure 2.3 displays the proportional relationship between temperature and concentration ratio. It can be seen that reaching ultra-high temperatures (>1300 K) is theoretically conceivable with concentration ratios >500. In practice, nonetheless, achieving ultra-high temperatures requires concentrations >1500 to compensate the incurred optical and thermal losses from the receiver (Marugán et al., 2012). Owing to the fourth degree order of the blackbody emission term, the benefit of boosting the temperature by concentrating more irradiance weakens at high concentrations. For example, increasing the concentration ratio from 1 to 100 results in 216% increase in temperature, while increasing it from 10,000 to 40,000 results in only 41.4% increase in temperature.
Figure 2.2. Variation of the efficiency of an ideal blackbody solar receiver with temperature at various concentration ratios.

Figure 2.3. Effect of concentration ratio on the maximum temperature of a blackbody solar receiver.

The theoretical analysis presented in section 2.1.1 assumed perfect insulation and accounted only for the emissive losses. However, in practice, there can be other significant optical (e.g. reflection) and thermal (e.g. convection and conduction) losses from the receiver, which will be analysed in detail in Chapter 5. Delivering solar radiation at a higher concentration reduces the required receiver area, which minimises the thermal losses and, hence, higher receiver temperatures can be maintained.
2.1.2. Limits on concentrating solar radiation

There are two ways for concentrating the solar radiation: line-focus and point-focus. Each type has a maximum limit imposed by the second law of thermodynamics, which constrains the energy transfer between the Sun and the receiver (Smestad et al., 1990). The limits are also determined based on the geometry and displacement of the radiant source from the receiver.

If we consider the Sun as a Lambertian source (i.e. radiates energy according to Lambert’s cosine law), then the solar flux ($\phi_{cc}$) incident on the concentrator’s aperture would be described by:

$$\phi_{cc} = \int I A_{cc} \cos \beta \, d\Omega = \int_0^{\beta_{cc}} 2\pi I A_{cc} \sin \beta \cos \beta \, d\beta = \pi I A_{cc} \sin^2 \beta_{cc}$$

(4)

where $A$ is the aperture area, $\beta$ is acceptance angle, $\Omega$ is the solid angle, and subscript $cc$ denotes the concentrator’s aperture side. A similar expression can be applied for the receiver side (denoted with the subscript $rc$). Therefore, the concentration ratio can be expressed as follows:

$$CR \equiv (\phi_{rc} A_{cc})/(\phi_{cc} A_{rc}) = \frac{\sin^2 \beta_{rc}}{\sin^2 \beta_{cc}}$$

(5)

For an ideal geometrical system that conserves the solar flux, any contraction in the beams area must correspond to an equivalent divergence in the angle (Born and Wolf, 2019), which intrinsically constrains the limit to which non-parallel radiation can be concentrated. This principle is called the conservation of étendue, which is a central concept in non-imaging optics derived from the second law of thermodynamics (Chaves, 2008). Therefore, the maximum concentration ratio is when the numerator in Equation (5) is maximised (i.e. $\beta_{rc} = 90^0$) as follows:

$$CR_{max} = 1/\sin^2 \beta_{cc}$$

(6)

To verify the previous theoretical limit based on principles of radiative heat transfer, we can consider the Sun as a blackbody sphere with a radius $Z$ that radiates energy to an observer on the Earth at a distance $Z$, as displayed in Figure 2.4. The radiation received at distance $Z$ will be uniformly distributed across the spherical surface area.
$4\pi Z^2$ (Lovegrove and Pye, 2012). Therefore, the irradiance reaching the observer can be evaluated as:

$$I = \sigma T_{\text{sun}} \frac{z^2}{Z^2} = \sigma T_{\text{sun}} \frac{(Z \sin \beta_{\text{sun}})^2}{Z^2} = \sigma T_{\text{sun}} \sin^2 \beta_{cc}$$  \hspace{1cm} (7)

where $T_{\text{sun}}$ is the photosphere temperature and $\beta_{\text{sun}}$ is the half-angle subtended by the Sun. If we have an ideal concentrator collecting the solar irradiance and passing the flux to a smaller receiver, then the balance of radiative heat transfer becomes:

$$\sigma (T_{\text{sun}})^4 A_{cc} \frac{Z^2}{Z^2} = \sigma (T_{rc})^4 A_{cr} G_{rc-cc}$$  \hspace{1cm} (8)

where $G_{rc-cc}$ is the fraction of energy radiation from the receiver that reaches the Sun. According to second law of thermodynamics, the maximum limit of the receiver temperature ($T_{rc}$) is equivalent to the source’s temperature ($T_{\text{sun}}$), which occurs at a thermodynamic equilibrium between the receiver and the Sun ($G_{rc-cc} = 1$). Therefore, solving for the area ratio ($A_{cc} / A_{cr}$) in Equation (8) will result in Equation (5) as follows:

$$CR_{\text{max}} = A_{cc} / A_{rc} = Z^2 / z^2 = 1 / \sin^2 \beta_{cc}$$  \hspace{1cm} (9)

Figure 2.4. A simplified Sun-Earth geometrical model with the Sun as a spherically symmetric blackbody with a radius $z$ at and distance $Z$ from the Earth, while $\beta_{\text{sun}}$ is the half-angle subtended by the Sun and $T_{\text{sun}}$ is the temperature of the photosphere.
Similar analytical procedure can be repeated for a line-focus concentrator, which would result in a limitation in one dimension, hence, the irradiance will fall off at $1/Z$ instead of $1/Z^2$. Therefore, the maximum concentration ratio for linear concentration ($CR_{max,linear}$) will be:

$$CR_{max,linear} = 1/\sin \beta_{cc}$$ (10)

Note that maximising the concentration ratio is independent from the angular spread of beams on the receiver or the Sun source, as it is only a function of the concentrator’s acceptance angle $\beta_{cc}$. This indicates that narrowing the acceptance of the concentrator can result in extremely high concentration ratios; however, if it is much smaller than the angular spread of the solar beams, the concentrators will miss collecting a large share of solar radiation (Lovegrove and Pye, 2012). Contrariwise, a very wide $\beta_{cc}$ will capture all solar radiation but will result in a low concentration ratio. Therefore, $\beta_{cc}$ is usually designed to match the angular size of the Sun. Therefore, if we substitute $\beta_{cc}$ in Equations (6) and (10) with the value of the actual half acceptance angle (4.65 mrad), the maximum theoretical limits for point-focus and line-focus concentration ratios become 46,248 and 215, respectively. Reflecting these limits on the previous analysis in section 2.1.1, the maximum attainable temperatures by point and linear concentration can then be established as 5344 K and 1395 K, respectively. These theoretical limits institute the conceptual challenge to line-based concentration for ultra-high temperature applications.

Reflectors are more commonly used as concentrators than refractors, as the latter suffer from chromatic aberration, selective wavelength absorption and dispersion, which were found to limit point-focus Fresnel lenses to concentration ratios <1000 (Languy et al., 2011, Laine, 2013). Although this ratio can still be augmented, up to 8500, by attaching a diverging polycarbonate lens with a converging PMMA lens (Languy and Habraken, 2013), simple parabolic and flat mirrors are practically proven to be capable of delivering higher concentration ratios (Shanks et al., 2016). Accordingly, a more realistic approach to estimate of the peak optical concentration at the focal “point”, should account for the geometry of the reflector.
Considering a conventional parabolic reflector, which concentrates the intercepted solar beams by reflecting them onto a small flat spot, the optical geometry of this system is defined in Figure 2.5. Assuming a pill-box sun shape, the diameter of the focal spot \(d_F\) can be expressed as:

\[
d_F = \frac{2z\sin\beta_{sun}}{\cos\phi_{rim}}
\]  

(11)

where \(\phi_{rim}\) is the reflector’s rim angle. The theoretical concentration ratio of this system is a function of the reflector’s rim angle as follows (Steinfeld et al., 2003):

\[
C = \left(\frac{1}{\sin^2\beta_{sun}}\right) \sin^2\phi_{rim}
\]  

(12)

To find the rim angle that maximises the concentration ratio, we differentiate Equation (12) with respect to the rim angle and equate this to zero as follows:

\[
\frac{dC}{d\phi_{rim}} = \left(\frac{1}{\beta_{sun}^2}\right) \sin 2\phi_{rim} \cos 2\phi_{rim} = 0 \rightarrow \phi_{rim} = 0 \text{ or } 45^0
\]  

(13)

Therefore, for a typical dish or trough concentrator with a rim angle of 45\(^0\) and a cavity receiver (with a flat aperture), the maximum concentration ratio becomes 23,124, which is half the thermodynamic limit of point-focus concentration. This analysis ignores the optical inefficiencies associated with the concentrator and receiver, which will be covered in Chapter 5.

Figure 2.5. Concentrating solar radiation using an ideal parabolic reflector with a rim angle \(\phi_{rim}\), onto a circular receiver displaced at the focal distance \(L_F\) from the reflector. The focal point here is in fact a circle with a diameter \(d_F\).
2.1.3. **Exclusion of line-focus technologies**

In section 2.1.1, concentration of solar radiation was established as the key tool to attain thermal energy at ultra-high temperatures. However, some characteristic limits restrain the attainable degree of solar concentration on Earth. In section 2.1.2, the upper thermodynamic limit of linear concentration was established as 215, which can sustain a receiver temperature at 1395 K based on the theoretical analysis in section 2.1.1. This maximum temperature is anticipated to be considerably lower in practice due to optical and thermal losses associated with the employed solar energy collection technology. For that reason, line-focus concentrating systems are typically used for applications with temperatures <823 K (Lange and Pielke, 2013), which are lower than the range of temperatures (>1300 K) targeted by this research. Therefore, technologies that are based on linear concentration, such as parabolic troughs and compact linear Fresnel reflectors, are conceptually disqualified for the purposes of this research.

2.2. **Solar dish**

A solar dish, sometimes referred to as parabolic dish, consists of a set of curved mirrors arranged in a paraboloidal shape to intercept the incoming solar beams and reflect them towards a focal centre, where the receiver is positioned as illustrated in Figure 2.6. The parabolic reflector dish operates similarly to a satellite dish, hence, incorporates a two-axis sun tracking mechanism to retain coincident focal spot on the receiver throughout the daytime and across seasons. The applications of the solar dish can range from power production and co-generation to water desalination and solar cooking (Zayed et al., 2021, Coventry and Andraka, 2017).

![Figure 2.6. A simplified schematic of a solar dish (Kodama, 2003).](Image)
2.2.1. Applications

The receiver in a solar dish typically encloses a Stirling engine used to directly convert the collected heat into electricity. Therefore, solar-to-electricity generation can be handled completely by a single unit (dish). This modular design facilitates deployment flexibility for standalone small-scale installations as well as large-scale solar farms (Coventry and Andraka, 2017). Yet, solar dishes are more likely to be used in remote off-grid applications at power ratings ranging between 3-25 kW_{elec} (Hoffschmidt et al., 2012). Figure 2.7 demonstrates a conventional electricity production system for such applications, which incorporates a standalone battery storage system. These standalone systems can be economically feasible under certain circumstances, such as being distant from national grids, where the cost of grid connection can be more prohibitive, or if the load is situated at a natural protectorate where infrastructure installations for grid connection are prohibited (Abdelsalam et al., 2014).

![Figure 2.7. A schematic diagram of a conventional Sterling-based solar dish, with battery storage, in a standalone system (Kadri and Hadj Abdallah, 2016). PMSG stands for permanent magnet synchronous generator.](image)

When compared to other CSP technologies, Stirling engine based solar dishes have superior thermal-to-electricity efficiencies up to 45% (Roldán Serrano, 2017, Coventry and Andraka, 2017) with instantaneous solar-to-electricity efficiencies reached a record of 31.25% (Bataineh, 2015, Andraka, 2018). The superior efficiency of solar dishes is generally attributed to their potential to reach ultra-high concentration ratios (>>1000) enabling them to sustain high receiver temperatures as explained in Chapter 1. Lab-scale solar dishes were proven at temperatures up to 1600 K with actual thermal
efficiencies reaching 25% (Zayed et al., 2021, Mendoza Castellanos et al., 2017). Nevertheless, because of their early stage of development, commercial solar dish systems are yet to become economically feasible for applications at temperatures >1100 K (Roldán Serrano, 2017, Liu et al., 2016, Zayed et al., 2021).

2.2.2. Challenges of upscaling

The prohibitive capital cost, lack of commercial experience, and technical difficulty of integrating storage systems to solar dishes remain the main limitations restricting solar dishes from expanding to large-scale applications (Liu et al., 2016). At utility-scales, parabolic dish engines are usually connected directly to the grid without local storage systems, as shown in Figure 2.8, which is not ideal for the stability and overall feasibility of a future renewable-based grids (Garvey et al., 2015).

![Figure 2.8. An array of grid-connected modules of solar dish without a local storage system. (ESTELA, 2017).](image)

An alternative approach to direct electricity generation via Stirling engine is to hybridise the heat collected by solar dishes with fossil-based combustion, which operate a micro-turbine as demonstrated in Figure 2.9. This configuration is often used to supply combined heat and power loads with reported total efficiencies ranging between 18.35-26.48% (Giostri and Macchi, 2016, Le Roux and Sciacovelli, 2019, Mazzoni et al., 2018, Semprini et al., 2016). Stirling engine based solar dish were used also for cogeneration but resulted in lower total efficiencies ranging between 16-23% (Ferreira et al., 2016, Li et al., 2012, Moghadam et al., 2013).
2.2.3. Challenges of energy storage integration

The modularity of solar dishes represent a technical challenge to their integration with storage. As explained in section 1.2.3, TES benefits from increased the volume-to-surface area ratios, hence, increased capacities. However, the solar dish’s modularity restricts the use of large-scale storage systems. Furthermore, the continuous movement (for sun-tracking) of the receiver with the dish imposes mechanical constraints on attaching the receiver with a localised storage system.

Some attempts were made by Sandia and Infinia to integrate solar dishes with different storage systems as counter-weights to the reflectors as shown in Figure 2.10. During preliminary testing; however, they reported corrosion of containment materials at the target temperature (Andraka et al., 2015, White et al., 2013), which led stopping both research programmes before testing them on dishes (Coventry and Andraka, 2017). Despite the promising potential of solar dishes for small-scale applications, there are major challenges for conventional solar dish technologies to become feasible for storage integrated large-scale applications (Zayed et al., 2021, Abdelhady, 2021). A potential setup for large-scale application is presented in Appendix A2.
2.3. Solar furnace

Solar furnace is an upgrade technology to parabolic dishes, which tackles the previously discussed concerns about mobile receivers by decoupling the receiver from the mobile primary concentrator (heliostats). Here, sun-tracking is handled entirely by the heliostats, while the parabolic collector – including its parabolic mirrors – is held in a static position oriented towards the reflected beams as demonstrated in Figure 2.11. This technology is the first here to incorporate a secondary concentrator.

Another advantage of separating the primary concentrator and collector is allowing the installation of a ‘shutter’, in the between, to precisely control the amount of radiation reaching the collector as demonstrated in Figure 2.12.
The world largest solar furnace (thermal power output rated at 1 MW) is currently located in Odellio, France. The optical focus is a 40 cm diameter spot at which it receives an average flux of 10,000 suns, and peak flux reaching 13,500 suns (Vant-Hull, 2012). This furnace is primarily used to study materials at temperatures exceeding 3273 K (Spinner, 2001).
2.3.1. Applications

Unlike other solar thermal technologies, solar furnaces are not typically designed for power production. Instead, they are commonly used for material and thermochemical processing at ultra-high temperatures, or for testing new solar receivers of other technologies (Chandra and Dixit, 2018, Villafán-Vidales et al., 2019). A high-flux solar furnace in Mexico, with a peak optical concentration of 18,000 suns (Kraemer, 2019), has been developed at scales 10-25 kW to produce biochar by solar pyrolysis (Ayala-Cortés et al., 2019).

During the 2000s, an EU project, under the name SOLHYCARB, explored the potential of solar furnaces for hydrogen production at ultra-high temperatures (CNRS, 2010). Concentrated solar radiation was used to crack methane molecules at temperatures ranging between 1740–2070 K, with experimental results showing that higher temperatures enhanced the methane production (Rodat et al., 2011). However, there are various technical and economic concerns associated with upscaling solar furnaces to industrial scales (>10 MW), including carbon particles removal and low economic viability of catalysts (Abánades et al., 2012).

2.3.2. Challenges of upscaling

Solar furnaces have also been commonly used, at scales ≤100 kW, in scientific researches (Flamant et al., 2004, Marzo et al., 2014, Alonso et al., 2015); nevertheless, solar simulators have largely replaced solar furnaces in the recent years given their size, economical, and operational flexibility advantages (Gallo et al., 2017). Solar furnaces, similarly to solar dishes, are subjected to dynamic wind loads enforced upon their secondary (parabolic) concentrating area. Upscaling the system entails enlarged parabolic area and increased weight, which requires a stronger support structure to sustain the system mechanically as well as maintain accurate the optical focus on the target. Accordingly, it is yet not economically viable to build solar furnaces at scales >1 MW (Villafán-Vidales et al., 2019), which is the same concern raised more than 60 years ago when Odellio furnace was first assessed (Trombe, 1957).
2.3.3. Challenges of energy storage integration

In concept, solar furnaces are technically simpler than solar dishes to be integrated with a storage system given their static receiver. To the best of the author’s knowledge, there is no attempt to integrate, or tailor, an energy storage system to solar furnace. The reason for this absence in the literature is likely due to the limited use of solar furnaces in power production applications, where stabilising the output is a prime target. A potential TES integrated solar furnace setup is presented in Appendix A2.

2.4. Solar power tower

Despite their proven potential for delivering heat at ultra-high temperatures, the previously reviewed technologies based on parabolic reflectors are not yet feasible for large-scale applications. In principle, upscaling any solar system, regardless of the energy extraction technology or application, necessitates an increased aperture area of the primary concentrator to allow for more energy input into the system. However, increasing the aperture area of heliostats in non-modular technologies is confined by their optical contact with the receiver or secondary concentrator.

One way to expand this optical contact is by mounting the receiver or secondary concentrator on a towering structure, hence the name *solar power tower*, to facilitate an improved optical vision of the distant heliostats with minimal blocking from front heliostats, as illustrated in Figure 2.14. The tower substitutes the requirement for a sloping heliostat field used for the solar furnace technology. The tower height has a significant influence on optical efficiency and land utilisation, as demonstrated in Figure 2.15.

There are two setup concepts for solar power towers, as illustrated in Figure 2.16, depending on which component is held on top of the tower: (i) a receiver or (ii) a secondary reflector. In some literature contexts, which exclude the latter, the former is referred to as “solar power towers”. However, to avoid confusion, the former setup concept will be referred to here as *tower receiver*, while the latter will be referred to as *beam-down tower reflector*.

---

3 Also referred to as *central tower receiver*, which is not accurate, as towers are not always situated at the centre of the solar field, seeing that polar-type plants have towers at/near the edge of the field.
Figure 2.14. A demonstrative schematic showing the advantage of increasing the tower height to minimise (or eliminate) the optical blockage of subsequent rows of heliostats.

Figure 2.15. Effect of tower height on the solar field’s efficiency and size (Buck et al., 2018).

Figure 2.16. Schematic representations of the two setups for solar power towers. (a) Tower receiver. (b) Beam-down tower reflector.
2.4.1. Tower receiver

Mounting the receiver on top of a tower is the conventional setup for solar power towers. This technology has the theoretical potential to facilitate receiver temperatures >1300 K by maximising the number of heliostats in an optimised field layout which compromises between optical losses and land use (Sarbu and Sebarchievici, 2017). However, existing commercial power plants operate at temperatures much lower due to technical limitations, which will be discussed in detail in Chapter 3.

Although parabolic troughs control >80% of the current CSP capacity, tower receivers have the largest share in commissioned and under development power plants as shown in Figure 2.17. Tower receivers are anticipated to dominate the CSP market by the end of the next two decades (Insights, 2021), given their technological prospect of delivering a stable and reliable electricity, accessibility for hybridisation, and anticipated continuation of costs reductions (Achkari and El Fadar, 2020).

In CSP plants, heat collected by the tower receiver is inserted into a thermodynamic cycle, which is commonly the subcritical Rankine cycle at temperatures <823 K (Stein and Buck, 2017). The plant configuration commonly comprises two closed circuits, as illustrated in Figure 2.18(a), where a liquid HTF (e.g. molten salt or oil) circulates in the solar circuit separately from the steam generation circuit, while both circuits are linked via close-type heat exchanger. Alternatively, water/steam is used as the HTF in a single circuit, as shown in Figure 2.18(b). Common storage systems are the two-tank TES or direct steam storage in a pressurised tank. The latter is for short-term storage (<1 h), while the former can service longer (>10 h) storage periods (Bauer et al., 2021).
Figure 2.18. Schematics of the two most common types of tower receiver power plants. (a) Molten salt plant (Mehos et al., 2017). (b) Direct steam generation plant (Qazi, 2017).

Nearly 50% of capital costs and 40% of energy losses in tower receiver plants occur at the heliostats field (Rizvi et al., 2021). There are two general layout configurations for tower receivers: surround-type and polar-type. The heliostats in the surround-type are arranged in radially staggered rows around the tower, as displayed in Figure 2.19, to facilitate a complete $360^\circ$ optical contact with the elevated receiver. The heliostats in the polar-type fields are arranged to maintain a narrower optical contact with the receiver from a single North/South direction, as exemplified in Figure 2.20. The surround-type is more economical for large-scale applications, as heliostats can be compacted into a smaller land area around the tower, while an equivalent polar field
will need to fit into a one-sided conical area with a greater *slant range*, which is the distance between the heliostats and the receiver (Vant-Hull, 2012). Nevertheless, based on their compatibility with cavity receivers, as will be discussed in Chapter 3, polar fields are more associated with high-temperature applications and can be >50% more efficient than surround fields (Coventry et al., 2016), particularly for sites far from the Equator (Gadalla and Saghafifar, 2018).

Figure 2.19. Dubai’s surround-type tower receiver (100 MW<sub>elec</sub>), which has the current tallest tower at 262.44 m (DEWA). The rectangular patterned installations behind the circular heliostats are parabolic trough collectors (600 MW<sub>elec</sub>).

Figure 2.20. Spain’s PS20 (20 MW<sub>elec</sub>), which is among the largest polar-type fields. An older version PS10 (10 MW<sub>elec</sub>) is displayed at the back (Müller-Steinhagen, 2013).
The tower height is structurally and sometimes legally constrained, while expanding the heliostat field for the same tower height increases the spillage and shading losses (Li et al., 2016, Vant-Hull, 2012). As a result, the power scale of a single-tower plant is typically <200 MW; however, contemporary projects adopt multi-tower fields to increase the scale and circumvent this limitation (Breeze, 2019). This limitation is more rigid in polar-type fields than surround fields, as receivers in the former need to be mounted at higher altitudes to compensate for the increased slant range and cosine losses caused by the restricted optical acceptance (Falcone, 1986). Nevertheless, with the ongoing advancement in towering structures, this limitation is expected to be alleviated in the future (Jin and Li, 2019).

### 2.4.2. Beam-down tower reflector

The beam-down tower reflector setup was introduced by Rabl (1976) and is based on the Cassegrainian reflector principle, where an elevated hyperbolic secondary reflector reflects reflected beams from the heliostats towards a receiver placed at the ground level, as shown in Figure 2.21. The main advantage of mounting the secondary reflector on the tower instead of the receiver is to keep the latter near the ground, which simplifies maintenance and minimises the cost of vertical piping and pumping the HTF to high altitudes, where it is also prone to thermal losses and solidification (Vant-Hull, 2012). While tower receivers dominate for power production, beam-down tower reflectors are generally favoured for thermochemical applications, particularly hydrogen production, where products are more valued over power plant efficiency (Miller, 2017b, Steinfeld, 2005, Boretti et al., 2021).

![Figure 2.21. A schematic of the beam-down tower reflector (Epstein et al., 2014).](image)
Beam-down tower reflectors use shorter towers than tower receivers, as the hyperbolic secondary reflector needs to be positioned lower than the focal point of the heliostats (Vant-Hull, 1991, Miller, 2017b). As illustrated in Figure 2.22, the hyperbolic secondary concentrator can be brought closer to the ground; however, this would expand its size to intercept the converging beams from heliostats. A reasonable elevation of the secondary reflector elevation is found as ~2/3 of the vertical distance between the receiver and the primary focal point (Vant-Hull, 2012).

![Figure 2.22. Optical representation of the beam-down reflector system (Vant-Hull, 2012).](image)

There are multiple conceptual concerns over the feasibility of the beam-down tower reflector setup. Vant-Hull (2014) revealed that the costs of supporting and maintaining the secondary reflector, which requires a sturdy and wind-resistant support structure, can easily compromise the savings from eliminating the receiver tower and the vertical heat transport system. Additionally, since the reflector is placed at a lower distance away from the primary focal point, the image created at the heliostats is likely to be substantially magnified, which would necessitate an array of CPCs at close proximity to the receiver to restore some of the lost solar concentration. Furthermore, a real (non-ideal) secondary reflector will absorb a fraction of the intercepted radiation, which might not be sufficiently discharged via passive cooling by the wind, hence, require an active cooling mechanism. These costly auxiliaries can outweigh the benefits of beam-down tower reflector setup (Vant-Hull, 2014), which hampered its commercial applications.
Regarding the upscaling of beam-down setups, the secondary reflector restricts the heliostat field boundaries allowing for <50% power utilisation of an equivalent tower receiver field (Vant-Hull, 2014), requires active cooling, and subjected to destructive wind loads (Segal and Epstein, 2008). However, for large-scale tower receivers operating at temperatures >1100 K, secondary optics would be required to restore the lost magnification in the field; hence, beam-down tower reflectors are projected to deliver comparable optical performance (Segal and Epstein, 1999). Furthermore, the economic viability of the setup depends on the application (Kribus et al., 1998).

In Egypt, a 10 MW<sub>th</sub> beam-down solar facility was proposed to be installed as the first large-scale demonstration of this technology. The beam-down system was designed to facilitate high optical concentrations of up to 10,000 suns and operate a volumetric air receiver, sealed by a quartz window, at 1673 K. However, after construction in 2008, the project was halted due to non-technical reasons (Blackmon, 2008). Recently, a TES-integrated 50 MW<sub>elec</sub> beam-down plant was commissioned in Yumen, China (CSP-Focus, 2020), which is planned to operate using Sodium Nitrate as the HTF at temperatures up to 843 K (NREL, 2022c). Existing lab-scale (<1MW) prototypes include the Weizmann Institute’s 700 kW<sub>th</sub> facility, which facilitates an optical concentration of ~4000 suns using a 75 m<sup>2</sup> secondary reflector and an array of CPCs (Epstein et al., 2014). This facility is used to test a solarised gas turbine and operate a carbon-reduction process for ZnO at temperature of up to 1573 K.

Few literature studies attempted to address the conceptual concerns of beam-down tower reflectors raised by Vant-Hull (2014). In 2015, a novel concept was proposed by Li et al. (2015), in which a point-focus beam-down tower reflector is integrated with line-focus Frensel heliostats. This concept is proven to deliver greater wind resistance and cost-effectiveness than conventional beam-down setups (Dai et al., 2016). However, the predicted thermal efficiencies were found to range between 39% to 60% (Li et al., 2015), which are lower than state-of-the-art solar cavity receivers (Bellos et al., 2019). Although the optical performance of the concept was reported to be superior to conventional beam-down setups at the prototype scale (Dai et al., 2016), there is still no evidence that the stated optical advantage and practicality of the concept can stand in large-scale applications (Li et al., 2017).
An open loop receiver was incorporated in a 30 kW\textsubscript{th} TES integrated beam-down tower reflector plant at Niigata University, Japan (Nakakura et al., 2017). Instead of using a hyperbolic secondary reflector, an elliptical mirror was used, which can result in a better interception of incoming beams from the heliostats (Li et al., 2017). However, contrary to hyperbolic reflectors, elliptical mirrors need to be positioned after the focal point, as illustrated in Figure 2.23(a), which results in taller towers than with hyperbolic reflectors. According to Li et al. (2017), the overall optical performance of a beam-down tower system using a hyperbolic reflector is always better than that of using an elliptical reflector. The volumetric receiver here, displayed in Figure 2.23(b), incorporates a silicon carbide honeycomb heat exchanger to transfer heat to a flowing air up to 840 K (Nakakura et al., 2017).

Figure 2.23. 30 kW\textsubscript{th} beam-down receiver in Niigata University, Japan. (a) Schematic diagram of the plant. (b) The used solar receiver (Nakakura et al., 2017).
The silicon carbide volumetric receiver used in Niigata University’s beam-down system was later found to develop hot spots near the surface of the honeycomb heat exchanger due to poor thermal conductivity of the porous medium, which restricted its use under high radiative fluxes, as the hot spots were found to trigger substantial re-radiative losses from the receiver (Matsubara et al., 2014). Consequently, the receiver design evolved into an inner-circulating fluidised bed receiver, which showed an enhanced heat transfer by convective transport of the directly irradiated spherical non-reactive particles (Matsubara et al., 2014). A similar prototype used solid solar-absorbent sand particles to convey the reflected energy from the tower reflector to a heat transfer fluid in a cavity sealed by a quartz window, as demonstrated in Figure 2.24. One of the aims of this prototype was to reach temperatures >1573 K to supply a thermochemical water splitting cycle using redox cerium oxide (Kodama et al., 2017). While the fluidised bed temperature reached 1400 K, the temperature of the heat transfer fluid (air) was still below 1300 K (Kodama et al., 2017, Bellan et al., 2019). The overall (light-to-chemical) efficiency of a 5 kWth prototype used in coke gasification was found to range between 11% and 13.2% (Gokon et al., 2019).

Figure 2.24. A schematic of a fluidised bed receiver used to absorb the concentrated solar irradiance and transfer the energy to air (Bellan et al., 2018).
2.4.3. Solar-combustion hybrid power tower

The Hybridisation of solar thermal with fossil-based combustion is a compromise measure between low-carbon emission, cost-effectiveness, and energy dispatchability. In hybrid systems, sharing the infrastructure and the use of more efficient power cycles were found to significantly improve the economic viability (Nathan et al., 2018) and enable upscaling of CSP plants to >100 MW$_{elec}$ (Jin et al., 2012). Likewise, hybridised CST and chemical looping combustion can deliver a cost-efficient solution for mitigating the greenhouse gas emission (Hong and Jin, 2005). An integration with a CO$_2$ capture technology can also minimise the infrastructure costs, as the same components can be shared between CO$_2$ capture and TES while delivering hybrid chemical-thermal storage (Jafarian et al., 2013). For solar power towers, an integration with combined-cycle power plants, with optimally designed gas turbines, can reduce the CO$_2$ emissions and levelised cost of electricity by up to 32% and 22%, respectively, compared to conventional fossil-based power plants (Spelling et al., 2014a).

Tower receivers have commonly been hybridised with fossil-fired combustors to enable running advanced power cycles, such as the Combined Cycle Gas Turbines (CCGT), as illustrated in Figure 2.25. CCGT is one of the most efficient thermodynamic cycles, which can reduce electricity generation costs by up to 15% compared to the conventional subcritical steam Rankine cycle (Spelling et al., 2014b). The supplementary combustion assists in uplifting the upper temperature of the thermodynamic cycle beyond the minimum inlet temperatures (>$1273$ K) of state-of-the-art gas turbines (Puppe et al., 2015, Zohuri, 2021, Siros and Fernández Campos, 2017). However, the use of a gaseous (air) HTF in the solar loop has technical implications, as will be discussed in Chapter 3, which can considerably limit the solar share at large scales (Jin et al., 2012). Integration with a compatible and highly efficient TES can allow the solar share to increase by up to 50% (Spelling et al., 2014b). However, according to Giuliano et al. (2011), the economic advantage of solar-combustion hybrid power plants would be compromised at high capacities by the high capital cost of TES. The study also demonstrated the heavy environmental impact of solar-combustion CCGT plants compared to other types of solar power tower plants.
Varied topping and/or bottoming cycles may be used to address the aforementioned issues of solar-combustion CCGT. For example, the bottoming steam-based Rankine cycle may be replaced by the organic Rankine cycle. Organic fluids have high molecular masses with liquid-vapour phase-change occurring at lower temperatures than water-steam (Zhang et al., 2021a). Therefore, the organic Rankine cycle can enable power generation from a low-grade heat source, such as waste heat, geothermal, biomass products, or solar thermal (Tchanche et al., 2011). For CSP, a bottoming organic Rankine cycle was found to deliver higher energy conversion efficiencies; however, with higher levelised costs of electricity than of equivalent plants based on the steam Rankine cycle (Puppe et al., 2015). This was attributed to the high cost and lack of availability of organic Rankine cycle components for large-scale applications.

A variation to the topping cycle may include pressurised CO$_2$ as the HTF in a closed-loop Brayton cycle. The supercritical CO$_2$ cycle is emerging in the nuclear (Dostal et al., 2006) and solar thermal (Utamura et al., 2006) fields to utilise its high efficiency, as there is no condensation loss as in steam cycles, in addition to its ability to integrate with TES (Turchi et al., 2012). Moreover, CO$_2$ is characterised by its chemical stability and low cost (Iverson et al., 2013). Nevertheless, upscaling is limited by various technical challenges ranging from operational challenges, such as the heat exchange between CO$_2$ and secondary fluid loop (Iverson et al., 2013), and material challenges, such as corrosion-related issues at temperatures >873 K (Stein and Buck, 2017).
In a detailed study of six different CSP plants operating with a supercritical CO$_2$ Brayton cycle with auxiliary combustion chambers, the second highest rate of exergy destruction was found to occur in the combustion chamber despite assuming complete combustion, which reflects the high degree of irreversibility of combustion (Atif and Al-Sulaiman, 2017). Other working fluids were considered as substitutes for CO$_2$ in the topping cycle, such as supercritical ethane, which was found to chemically decompose at temperatures <673 K (Enríquez et al., 2015). Helium and nitrogen were also considered but were found less efficient than equivalent CO$_2$-based cycles (Dunham and Iverson, 2014, Forsberg et al., 2006). A study by Besarati and Yogi Goswami (2013) considered using a topping supercritical CO$_2$ cycle with a bottoming organic Rankine cycle. The overall combined cycle efficiency was found only 3-7% higher than conventional CCGT; thus, cost-benefit analyses are required to be implemented, particularly for applications at temperatures >1123 K, to assess the economic feasibility of such combined cycle configurations (Stein and Buck, 2017).

Conventionally, tower receivers are hybridised using separate fossil combustion chambers (Figure 2.26(a)) connected directly or indirectly to the solar HTF circuit via a heat exchanger. A novel design of a hybrid solar receiver combustor (Figure 2.26(b)) was proposed to combine the solar collection and combustion process in the solar receiver (Nathan Graham et al., 2017). The design incorporated CPC to facilitate high concentration ratios and lower radiant losses (Lim et al., 2016). Instead of a window, which was found problematic with particles released from combustion (Nathan Graham et al., 2017), a fluidic seal system, in form of a curtain made up of the exhaust gas, is used to restrict fluid flow through the aperture during the combustion process. This design promised reduced capital cost and levelised cost of electricity by up to 17%, in addition to reducing the net fuel consumption by up to 31%, compared to the conventional configuration (Chinnici et al., 2018). These benefits resulted from sharing the infrastructure, minimised land use, and reduced O&M costs due to minimised thermal cycling and thermal shock occurring in solar-only receivers when cavities cool during the night (Nathan et al., 2014). The potential of such a hybrid configuration was demonstrated for mineral processing and gasification (Nathan et al., 2017). Nevertheless, there is no information yet about the reliability of the used heat exchangers at high concentration ratios and temperatures.
Figure 2.26. Two configurations for solar-combustion hybrid power towers (Nathan et al., 2014). (a) A conventional configuration with separate solar and combustion components. (b) A novel solar receiver, which combines solar collection and combustion in the same component. EPGS is electrical power generating system.

Figure 2.27. Schematic diagram of the hybrid solar receiver combustor (Lim et al., 2016).
2.5. Concluding remarks on the candidate technologies

In view of the potential and challenges of the reviewed technologies, achieving ultra-high temperature by concentrating solar thermal energy is theoretically possible and has already been practically demonstrated at lab scales. Solar dish technologies have proven their potential to facilitate ultra-high temperatures and high energy conversion efficiencies. However, the early stage of their technological development and difficulty to integrate with storage systems are currently hindering their use for large-scale applications. Similarly, the solar furnace is practically proven to sustain receiver temperatures in excess of 3273 K; however, it is restricted to small-scale non-commercial applications. Solar power towers accumulated decades of commercial experience, with >20 TES integrated plants currently operating at capacities >1MW_{elec} (>5 MW_{th}). This progress, nonetheless, was solely reserved for the tower receiver setup, as the beam-down tower reflectors are not yet practically proven at large scales, mainly due to the unresolved conceptual concerns raised by Vant-Hull (2014). Further insights on the potential and challenges of large-scale beam-down tower reflectors are anticipated in the near future after accumulating operational experience from the 50 MW_{elec} Yumen Xinneng CSP plant (NREL, 2022c).

Tower receivers are the most promising CSP technology, with expectations to become the leading CSP technology with substantial reductions of their levelised cost of electricity within the upcoming decades (Achkari and El Fadar, 2020, Merchán et al., 2022). Tower receivers are practically proven to fulfil the upscaling and TES compatibility objectives of this research. They are also theoretically proven to facilitate stagnation temperatures >3000 K, allowing for energy conversion to thermal reservoirs at temperatures >2000 K (Steinfeld, 2002, Melchior et al., 2008). However, none of the currently operational commercial CSP plants operates at temperature >900 K due to mechanical and materials constraints (Miller, 2017a, Pacio and Wetzel, 2013, Silva-Pérez et al., 2017). The development of innovative solar receiver technologies, along with optimising the heliostat field layout based on the receiver technology, are the main areas for R&D to attain more feasible systems without hybridising with fossil-based combustion (Merchán et al., 2022). Accordingly, this thesis contributes to that effort by proposing a novel receiver technology for ultra-high temperature operation.

In the previous chapter, it was demonstrated that ultra-high temperatures can be attained by concentrating solar thermal energy using the tower receiver technology. However, some technical challenges have restricted state-of-the-art receivers from operating at ultra-high temperatures/solar fluxes. According to Glatzmaier (2011), the receiver temperature is constrained by the material and mechanical properties of receiver components, such as the HTF and solar absorbing material. In this chapter, five principal design characteristics are defined and assessed for the prospective ultra-high temperature solar receiver. These characteristics will then be used to outline the proposed novel solar receiver in the next chapter.

3.1. Receiver type

The mechanism for collecting solar thermal energy can take different forms in the solar receiver. Solar receivers can either be tubular or volumetric (CSTEP, 2014). Tubular receivers are usually used for compressed gas or liquid HTF, such as molten salts, oils, liquid metals, or water. Volumetric receivers typically operate on particles or unpressurised gaseous HTF, such as air, nitrogen, or CO₂. Further details about HTF in solar receivers will be covered in section 3.3.

Tubular receivers are the standard type for commercial CSP power plants, where concentrated solar radiation from the field impinges on a set of absorber tubes, which are typically coated with a suitable material to enhance their optical absorptivity. The absorbed heat is then transferred by convection and conduction to an internally flowing HTF, which circulates in and out of the tower receiver system to convey the collected energy to the storage system, a secondary circuit, or directly to the output application. Tubular receivers can either be fully exposed to the heliostat field or have restricted optical exposure (Merchán et al., 2022). The former type is called an external receiver, while the latter is called a cavity receiver. As demonstrated in Figure 3.1, the main design difference in the cavity receiver is that the absorber tubes are contained within an enclosure with an aperture through which the concentrated solar radiation can pass and reach the tubes.
External receivers benefit from the $360^\circ$ optical exposure to the heliostats, making them favourable for the surround-type tower receivers installed near the Equator. Nevertheless, external receivers are not recommended for operation temperatures $<1000$ K due to excessive thermal (mainly radiative) losses, which compromise their optical advantage over cavity receivers (Li et al., 2016). Furthermore, the restricted view of the cavity receiver could be compensated by increasing the tower height, as illustrated in Figure 3.2. Cavity receivers offer higher thermal efficiencies by entrapping, and recovering, re-radiation from the absorber (Becker and Vant-Hull, 1991). This advantage is augmented at ultra-high temperatures, where radiative losses dominate (Vant-Hull, 2021). The enclosure also seals the absorber from wind-induced convective loss and restricts it to an aperture, which can be sealed by fluidic seals or transparent windows (Clausing, 1983, Jilte et al., 2014, Harris and Lenz, 1985). However, cavity receivers tend to be heavier and mounted on taller towers; hence, their designs are required to be compact to alleviate the associated structural concerns.
At large scales, tower costs become substantial, while spillage and atmospheric attenuation dominate over the cosine losses; thereby, polar heliostat layouts, which are suitable for cavity receivers, become sub-optimal (Schmitz et al., 2006). This problem could be mitigated by expanding the optical acceptance of cavity receivers by incorporating two or more cavity receivers or a single cavity with multiple apertures on the same tower. Increasing the number of apertures can boost the power output and enable cavity receivers to operate at high capacities, which was demonstrated in the South African 50 MW_{elc} Khi Solar One plant displayed in Figure 3.3 (Mucci, 2015).

Figure 3.3. Tower receiver with multiple (three) apertures in South Africa (JG-Afrika, 2021).

The optimal power level for a single-aperture cavity receiver ranges between 25-50 MW_{th}, while multiple-aperture cavity receivers are recommended for capacities >50 MW_{th} (Schmitz et al., 2006). Based on the results presented in Figure 3.4, a single-aperture cavity receiver operating at 1600 K was found to deliver the highest solar-to-thermal efficiency compared to multi-aperture receivers at capacities <80 MW_{th}.

Figure 3.4. Annual solar-to-thermal efficiency ($\eta_{s-t}$) and net receiver power ($\dot{Q}_{\text{tot,net}}$) at different number of apertures ($N_a$) and temperatures ($T_{\text{rec}}$) (Li et al., 2020a).
In a study of large-scale CST with cavity receivers by Potter et al. (2020), a single-aperture cavity receiver was compared with an equivalent three-aperture cavity receiver at capacities of 50 MW\(_{th}\) and 500 MW\(_{th}\). The results displayed in Figure 3.5 reveal that the optical advantage of incorporating a multi-apertured receiver was manifested at extremely large capacities (\(>>100\) MW\(_{th}\)), which is justified by the lower optical tower height to heliostat field ratio in large-scale systems, thus, favouring a surround heliostat field (Figure 3.6(a)) than a polar field (Figure 3.6(b)). However, even this optical advantage tends to diminish at tower heights \(>200\) m, as illustrated in Figure 3.5(b), which are typically used in large-scale systems. Therefore, the financial feasibility of multi-apertured systems is yet to be investigated in more detail before being considered for large-scale ultra-high temperature systems.

![Figure 3.5](image1.png)  
**Figure 3.5.** Annual efficiency of heliostat field versus tower height for a single-aperture and three-aperture cavity receivers at (a) 50 MW\(_{th}\) and (b) 500 MW\(_{th}\) (Potter et al., 2020).

![Figure 3.6](image2.png)  
**Figure 3.6.** Layout of heliostat field for (a) single-aperture and (b) three-aperture cavity receivers (Potter et al., 2020).
3.2. Secondary optics

Maximising the concentration ratio is crucial to sustaining ultra-high receiver temperatures. Secondary optics is one way to keep the practical concentration ratios close to the established thermodynamic limit in section 2.1.2. Non-imaging secondary concentrators are proven to extend the range of concentration ratio up to 80-90% of the thermodynamic limit (Gordon, 2001). The conversation of the étendue principle permits further concentration by funnel-shaped reflector systems (Lovegrove and Pye, 2012), such as the compound parabolic concentrator (CPC), Trombe-Meinel cusp (Figure 3.7(a)), and Mouchot’s conical reflector (Figure 3.7(b)). CPCs were considered for various solar systems comprising cavity receivers to reduce their aperture size and minimise the thermal losses (Buck et al., 2018).

CPCs have specular reflective interiors to facilitate amplifying the solar flux received from primary concentrator, as illustrated in Figure 3.8. There are two types of CPCs, as shown in Figure 3.9. 2D CPCs produce linear concentrated radiation, while 3D CPCs produce a spot concentrated radiation. Linear and point CPCs augment the concentration ratio by factors of $1/\sin(2\beta_{CPC})$ and $1/\sin^2(2\beta_{CPC})$, respectively (Levêque et al., 2017).

(a)

Figure 3.7. Funnel-shaped reflectors. (a) A Trombe-Meinel cusp concentrator attached to a linear Fresnel system (Lovegrove and Stein, 2012). (b) A truncated cone dish engine developed by Augustin Mouchot in 1878, which was used for ice-making (Ragheb, 2014).
Figure 3.8. A schematic diagram of a compound parabolic concentrators (CPC) with a half acceptance angle $\beta_{CPC}$. The grey gradient represent radiative intensity (i.e. darker regions indicate higher radiative intensity).

Figure 3.9. Schematics of (a) linear and (b) point compound parabolic concentrators (CPC). The arrows signify incident irradiation from the primary concentrator(s) (Steinfeld, 2005).

The main limitation of using CPCs is their restricted acceptance angle, which narrows the optical view of the heliostats. For tower receivers, this limitation can be mitigated by raising the receiver higher; however, this approach is constrained by the towering structure (Buck et al., 2018). The financial and technical benefits of installing CPCs to tower receivers are justified only for receiver temperatures $>1200$ K (Li et al., 2020b). Furthermore, enabling the use of a highly efficient power cycle is projected to recover the optical losses introduced by the secondary concentrator (Schmitz et al., 2006). A 3D CPC was tested for a multi-source high flux (3000 suns) simulator, and it was found to increase the concentration ratio by 4.1 times at an optical efficiency of 85.4% and reduce the spillage loss from 78.9% to 32.1% (Li et al., 2019).
For the 600 kW<sub>th</sub> cavity receiver displayed in Figure 3.8, an aluminium CPC was installed at the receiver aperture to facilitate receiver temperatures up to 1573 K (Zhang et al., 2022). The aperture was sealed by a 3 mm quartz glass, which was passively cooled by the ambient air. Nevertheless, the CPC was found to require active (forced convection) cooling to sustain operation at ultra-high temperatures.

3.3. Heat transfer fluid

In Chapter 2, it was determined that material-related constraints restrain the receiver temperature. The HTF is a critical component which controls the performance and design of the receiver and the storage system (Vignarooban et al., 2015). The thermo-physical properties of the HTF determine the design and materials of the absorber, as they are typically required to be mechanically and chemically compatible. The stability of piping and containment materials, usually stainless steels and nickel-based alloys, while in contact with the HTF were found crucial for the longevity of CSP systems (Vignarooban et al., 2015). Therefore, whether solar absorption is handled directly by the HTF or indirectly via an intermediate solid absorber, the HTF poses a significant influence on the performance of a solar receiver (Vignarooban et al., 2015).

While the desired thermo-physical properties of the HTF may vary based on the technology and application, there are properties which are generally favourable in solar applications. Low melting/freezing temperature is one of the essential properties for an HTF, particularly for tower receiver systems, to reduce the risk of solidification and expansion in the piping system. High boiling temperature is also an important property for liquid HTFs to allow for greater operating temperature difference, hence, greater
energy transfer without boiling, which may complicate the design due to increased specific volume and deteriorated thermal properties at the gaseous state (Pacio and Wetzel, 2013). High thermal conductivity and high heat capacity are also favoured thermal properties, particularly if the HTF is used as a storage medium. Low viscosity is also a desired property to minimise the pressure drop and pumping power for circulating the HTF throughout the primary loop. At ultra-high temperatures, further properties may become vital, such as low corrosion, high thermal stability, and low vapour pressure (Vignarooban et al., 2015). A summary of the thermo-physical properties of some selected HTFs is presented in Table 3, while further details about the temperature-dependent properties of selected HTFs are presented in Appendix A1.

In addition to the thermo-physical properties, prospective HTFs are required to be cost-effective. For example, pure liquid metals may have superior thermo-physical properties to compressed air, however, for a small-scale application; it is usually more cost-efficient to use an air-based system than liquid-metal-based system (Vignarooban et al., 2015). In light of the sustainable development aim discussed in Chapter 1, the selection of the HTF and its containment materials should consider their abundance, circularity rates, and environmental costs of extraction and processing to its final form.

HTFs in solar applications are generally categorised as (i) water/steam, (ii) oils, (iii) molten salts, (iv) air and other ideal gases, (v) liquid metals, (vi) and organic fluids (Vignarooban et al., 2015, Qazi, 2017). Solid particles were used as heat transfer mediums in solar receivers (Ho, 2016). The suitability of a particular HTF depends on the application, scale, involved heat transfer mode(s), and operating temperature. Thermal oils and organic fluids are not considered here, as they boil at temperatures far below the temperature levels targeted by this thesis (Tian and Zhao, 2013); however, comprehensive reviews of their use in CSP applications are provided by Cabaleiro et al. (2012), Tian and Zhao (2013), and Vignarooban et al. (2015).

3.3.1. Water/steam

The conventional CSP plant type is the direct steam generator, where water/steam is used as the HTF and working fluid of the subcritical steam Rankine cycle. Direct steam generators were introduced in the 1980s as economic alternatives to oil-based

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Density (kg.m⁻³)</th>
<th>Specific heat capacity (J.kg⁻¹.K⁻¹)</th>
<th>Liquid range (K)</th>
<th>Thermal conductivity (W.m⁻¹.K⁻¹)</th>
<th>Dynamic viscosity (mPa.s)</th>
<th>Cost ($/kg⁻¹)</th>
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</thead>
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<tr>
<td>Aluminium (1600 K)</td>
<td>2192</td>
<td>1080</td>
<td>933-2743</td>
<td>237</td>
<td>~0.750</td>
<td>3.46</td>
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<tr>
<td>Tin (1600 K)</td>
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<td>505-2875</td>
<td>56.8</td>
<td>~0.605</td>
<td>15.9</td>
</tr>
<tr>
<td>LBEᵃ (1400 K)</td>
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<td>397-1943</td>
<td>21.7</td>
<td>~0.734</td>
<td>13</td>
</tr>
<tr>
<td>Sodium (1100 K)</td>
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<td>371-1156</td>
<td>63.3</td>
<td>0.181</td>
<td>3.43</td>
</tr>
<tr>
<td>Solar saltᵇ (800 K)</td>
<td>&lt;1500</td>
<td>533-838</td>
<td>0.53</td>
<td>1.69</td>
<td>0.5</td>
<td></td>
</tr>
<tr>
<td>FLiBeᶜ (1600 K)</td>
<td>1581</td>
<td>2678</td>
<td>732-1703</td>
<td>1.26</td>
<td>0.408</td>
<td>-ᵈ</td>
</tr>
<tr>
<td>Silicone oil (623 K)</td>
<td>900</td>
<td>2100</td>
<td>251-673</td>
<td>0.10</td>
<td>3.94</td>
<td>5</td>
</tr>
<tr>
<td>(1600 K)</td>
<td>0.226</td>
<td>1217</td>
<td>0.090</td>
<td>0.054</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Air (1300 K, 2.5 MPa)</td>
<td>2.78</td>
<td>1190</td>
<td>0.089</td>
<td>0.497</td>
<td></td>
<td></td>
</tr>
<tr>
<td>CO₂ (1273 K, 25 MPa)</td>
<td>10.34</td>
<td>1297</td>
<td>N/A</td>
<td>0.091</td>
<td>0.053</td>
<td></td>
</tr>
<tr>
<td>Steam (823 K, 25 MPa)</td>
<td>51.4</td>
<td>2162</td>
<td>0.074</td>
<td>0.031</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

ᵃ LBE is a metallic (lead-bismuth) eutectic alloy.
ᵇ Solar salt is the commercial name of an eutectic mixture of nitrate salts (60 wt.% of sodium nitrate and 40 wt.% of potassium nitrate).
ᶜ FLiBe is a molten salt made from a mixture of lithium fluoride and beryllium fluoride.
ᵈ It is difficult to accurately estimate the costs of enriched lithium used in FLiBe when produced in large quantities. A 100 m³ of FLiBe would contain over 30 tonnes (15.464% of FLiBe mass) of 99.995% ⁷Li with cost varying from $120/kg up to $800/kg.

parabolic troughs, as the former were found to reduce the levelised cost of electricity by 11% (Feldhoff et al., 2010). The receiver in such a system is typically a solenoid heat exchanger (Figure 3.11) topped with a steam drum (Figure 3.12). Currently, seven large-scale CSP plants are based on this technology type (SolarPACES, 2019), including the world’s largest CSP plant (392 MWₑₑₑₑelec Ivanpah solar power facility).
Figure 3.11. CSIRO’s 309 kWth supercritical steam generator receiver, which was demonstrated to generate steam at 23.5 MPa and 843 K (Gardner et al., 2012).

Figure 3.12. A tower receiver (PS10) operating as a direct steam generator (Solúcar, 2006).

In addition to the reduced costs and simplified system, direct steam generators benefit from the improved conversion efficiency through the elimination of the intermediate heat exchanger used between the solar circuit and power cycle (Karellas et al., 2019, Cundapí et al., 2017). Further advantages include minimised risk of leakage of toxic or flammable liquids contrary to multi-fluid-based systems. Nevertheless, such a system is limited to steam as the storage material, which is currently uncompetitive with state-of-the-art storage materials for high-capacity applications (Palacios et al., 2020). The scarcity of water in locations with high solar and land availabilities (e.g. desert regions) is another sustainability challenge to steam-operated CSP projects (Pistocchini and Motta, 2011). Additionally, the high specific volume and poor thermal properties of steam, as for most gaseous HTFs, limit the power capacity of a single-tower receiver, which led to the necessity of using multiple tower receivers to upscale the plant’s capacity.
With the advent of ultra-supercritical (>647.1 K) steam generators, there is a possibility for direct steam generators to facilitate large-scale CSP at ultra-high temperatures. In a thermodynamic analysis of a 1000 MW$_{\text{elec}}$ coal-fired plant running a reheat ultra-supercritical steam turbine at temperatures up to 883 K, it was shown that the power generation efficiency of the plant could reach up to 48.35% (Liu et al., 2018). However, the containment material to high-temperature steam is required to resist severe corrosive conditions and have sufficient creep strength (Liu et al., 2018). The chemical and mechanical damage mechanisms experienced at high steam temperature and pressure conditions are reported in detail by Nicol (2013), while corrosion of different alloys at high steam temperatures (>973 K) is extensively reported in the literature (Vignarooban et al., 2015, Otsuka and Fujikawa, 1991, Tan et al., 2008, Robertson, 1991). Therefore, new materials and innovative design measures are still required to enable the use of ultra-supercritical steam Rankine cycle at ultra-high temperatures (Nomoto and Tanuma, 2017). The thermomechanical integrity of the solenoid receiver is another challenge for direct steam generators operating at ultra-high temperatures (Elsafi, 2015, Karellas et al., 2019). Novel ultra-supercritical steam generators can be used at temperatures up to 1033 K, with power conversion efficiencies reaching 52% (Boretti and Castelletto, 2021, Weitzel, 2011).

### 3.3.2. Molten salts

Molten salts are currently the most used type of HTF in commercial CSP plants attributed to their favourable thermo-physical properties (refer back to Table 3) and thermal stability at temperatures up to 800 K without the requirement for pressurisation (Peng et al., 2010). Their high boiling points, low vapour pressures, stability at high temperatures, and superior volumetric capacities have promoted molten salts as more efficient alternatives to water/steam at high temperatures (Peng et al., 2008, Caraballo et al., 2021), while their non-toxicity, non-flammability, and relatively low cost provided them with a competitive edge against thermal oils (Gil et al., 2010). As discussed in Chapter 1, the main drive behind the spread of the use of molten salts in modern CSP systems is their potential to be used as a TES medium (Vignarooban et al., 2015, Caraballo et al., 2021), as exemplified in Figure 3.13. Nevertheless, their main limitations include high viscosities, low thermal conductivities, and high melting temperatures, typically >450 K (Alva et al., 2018).
Sodium and potassium nitrates in liquid forms are the common types of molten salts used in CSP applications. Figure 3.14 displays the temperature ranges for different molten salts. The eutectic molten salt mixture of 60% sodium nitrate ($\text{NaNO}_3$) and 40% potassium nitrate ($\text{KNO}_3$) is considered the most successful material for solar heat transport (Ushak et al., 2015), hence labelled as the solar salt. However, most molten salts suffer chemical decomposition at temperatures $<1000$ K, hence, prohibiting their use in ultra-high temperature applications (Zou et al., 2019).

Figure 3.14. Liquid temperature ranges of different molten salts, molten metals, a thermal oil, and an organic fluid (Vignarooban et al., 2015). Please note that the temperatures here are displayed in degree Celsius.
A prospective ultra-high temperate molten salt is the eutectic mixture of lithium fluoride (LiF) and beryllium fluoride (BeF₂), commonly known as FLiBe, which was used as a coolant and moderator in nuclear fission reactors at temperatures reaching 923 K (Serrano-López et al., 2013). The high boiling point (1703 K), chemical stability at high temperatures, and favourable thermal properties (Table 3) of FLiBe promoted its use in ultra-high temperature applications (Romatoski and Hu, 2017), including solar thermal generation and TES (Forsberg et al., 2006). However, the high melting temperature (732 K) of FLiBe may prohibit its use as an HTF in CSP (Srivastva, Malhotra and Kaushik, 2015). The toxicity of beryllium is another limitation of using FLiBe at large scales (Romatoski and Hu, 2017). Furthermore, the requirement to enrich lithium to at least 99.995% \(^7\)Li disadvantages the cost-efficiency of FLiBe (Leblanc, 2010). The high optical transparency of FLiBe, which is useful in nuclear reactors (Dolan, 2017), prohibits its use for direct solar absorption; hence, will typically necessitate an intermediate solid absorber.

### 3.3.3. Air and compressed gases

The abundance and high cost-efficiency of the air have encouraged its use in various energy applications, including solar thermal systems (Ávila-Marín, 2011). Air and other gases share the advantage of not boiling at ultra-high temperatures and are generally not environmentally hazardous by nature (Srivastva et al., 2015). Air and other gases, such as CO₂ and helium, are already proven feasible as working fluids operating at ultra-high temperatures in the Brayton and other gas-based cycles (Besarati et al., 2017, Peng et al., 2012, Poživil et al., 2014, Zhang et al., 2021b). Pressurised nitrogen was tested as an HTF in a 50 MW \(_{\text{e}}\) parabolic trough plant in Spain and was found to produce a similar performance to synthetic oil, however, with reduced flammability and environmental hazardousness (Biencinto et al., 2014).

A typical process flow for an air-based tower receiver system is demonstrated in Figure 3.15. A variation from the displayed system may include an open loop air/gas primary circuit, which is not efficient as the enthalpy of the return air would be lost. Another common variation is the system demonstrated in section 2.4.3, where the solar component can either operate or be used to preheat pressurised air in a solar-combustion hybrid CCGT cycle.
Similarly to steam, the thermal conductivities and volumetric heat capacities of air and other gasses are considerably lower than liquid HTFs (Liu et al., 2014). In a study of multi-cavity tubular receivers by Fleming et al. (2017), it was found that the convection coefficient by the operational HTF should be at least 500 Wm$^{-2}$K to achieve a theoretical thermal efficiency >90%, which is a limitation for many candidate gases, such as air and helium. However, the superior flow properties (e.g. dynamic viscosity) of air compared to liquid HTFs may somewhat compensate for the inefficient heat transfer performance (Vignarooban et al., 2015).

The high specific volumes of gaseous HTFs necessitate pressurisation to limit the sizes of equipment, which adds costs and engineering challenges, such as leakages from ducts and receivers (Srivastva et al., 2015). The feasibility of replacing synthetic oils with compressed gas HTFs was studied in Spain, where it was found that the substantial leakages from ball joints compromised the benefits of compressed gases (Muñoz-Anton et al., 2014). According to Ho and Iverson (2014), the design and practical requirements of using air as an HTF can be prohibitive in most cases, especially for large-scale applications. As a result, there is currently only one large-scale (1.5 MW$_{elec}$) air-based CSP plant located in Jülich, Germany, where air is heated up to 1000 K and then used to generate steam (Tian and Zhao, 2013). The oxidation of conventional containment materials for air can be problematic at temperatures >1000 K (Chen and Yeun, 2003, Klöwer, 1996).

Helium and supercritical CO$_2$ have been investigated in the literature as prospective HTFs for high-temperature systems (Vignarooban et al., 2015, Bian et al., 2022). Helium is relatively more affordable, as it is attained from the extraction processes of natural gas, and has already been used in high-temperature nuclear reactors (Xu et al.,
Helium can operate at ultra-high temperatures at competitive efficiencies; however, its poor thermal properties and heat transfer between the bulk of the fluid and the internal surfaces of the pipes necessitate high pressurisation and flow velocities, which raise the technical challenges of sealing and controlling the flow instabilities (Massidda and Varone, 2007, Xu et al., 2010). At present, supercritical CO$_2$ has a more positive outlook as an HTF and a working fluid in power cycles (Z. and S., 2011). For CSP applications, supercritical CO$_2$ can deliver higher efficiencies than steam Rankine or helium-based cycles; however, the requirement to operate at high pressures would limit supercritical CO$_2$ to solar power towers, as they are currently incompatible with parabolic trough systems (Vignarooban et al., 2015, Heller et al., 2022). Further research and development are still required to reduce the costs of supercritical CO$_2$ components by at least 30% for competing against commercial systems based on steam and molten salts (Heller et al., 2022).

Storage is the main limitation of gas-based CSP systems, as TES based on supercritical fluids are proven unviable (Kelly, 2010). Given the low volumetric heat capacities of gaseous HTFs, energy storage in gas-based systems usually rely on heating a secondary storage medium, such as solid particles, concrete, or PCM, by the heated gas (Ho, 2017). Nevertheless, the cost and heat transfer effectiveness between the gaseous HTFs and secondary storage mediums at ultra-high temperatures are current challenges which are yet to be resolved (Bashir et al., 2019, Ho, 2017).

3.3.3.1. Volumetric receivers

Volumetric receivers are common for unpressurised open-loop gas-based systems. As shown in Figure 3.16, the gaseous HTF is blown through porous structures, such as monolithic honeycombs or channels, which are directly irradiated by the concentrated sunbeams (Ho, 2017). The gas then extracts the energy from the structures via combined convection and conduction. Studies of different volumetric receiver designs found that the spots near or facing the aperture are usually the hottest, resulting in overheating and local failures, in addition to significant emissive losses to the ambient (Du et al., 2018, Ho, 2017, von Storch et al., 2015). Temperatures of volumetric receivers can reach up to 1273 K for metals, 1473 K for ceramics, and 1773 K for silicon carbide (Ávila-Marín, 2011); nevertheless, the outlet gas temperature is
typically lower due to the ineffective heat transfer between the gas flow and the solid absorber (Du et al., 2018, Sharma et al., 2015). Flow instabilities and non-uniform heating are primary limitations of volumetric receivers (Pitz-Paal et al., 1997, Kribus et al., 1996), which occur due to the high temperature dependence of viscosities and densities of gases (Ho and Iverson, 2014).

A volumetric receiver concept (Figure 3.17) was developed by Karni et al. (1997) to produce air at temperatures up to 1573 K. In this concept, ceramic fins called *porcupines* are used to absorb the solar flux and then transmit the energy to the air by conduction and convection. Under a solar flux of 5300 suns, the receiver was experimentally proven to deliver air at 20 bar and 1473 K, with a thermal efficiency peaking at 70% (Kribus et al., 2000). This design can reportedly achieve temperatures up to 1673 K by increasing the concentration ratio or lowering the HTF flow rate. However, the temperatures of the ceramic absorber and quartz window are the main two limitations, as they were required to be maintained at below 2073 K and 1073 K, respectively (Kribus et al., 2000). Despite the promising performance at the lab-scale, upscaling this design to a utility scale can be challenging due to technical limitations associated with the capped conical quartz window (Röger et al., 2006). In addition to requiring active cooling to sustain the window under the 1073 K temperature limit, quartz windows must be thin to minimise radiative attenuation, while sufficiently strong and durable to withstand high temperature and pressure (Poživil et al., 2014). Additionally, the aperture diameter of the window is limited to below 600 mm based on feasible contemporary manufacturing capabilities in the industry (Doron, 2020). These strict and contradicting requirements have limited the scale of this design to running microturbines up to 500 kW_{elec} (Doron, 2020).
In a study of the effect of prolonged exposure to concentrated solar irradiation (2000 suns) on porous multi-channelled monolithic silicon carbide honeycombs at temperatures up to 1673 K, it was found that the mechanical structures of honeycombs and porous structure were adversely affected (Agrafiotis et al., 2007). A visual representation of the effect after 94.2 hours of exposure is demonstrated in Figure 3.18. The study also found that the non-homogeneity of temperature distribution throughout the cross-section of the receiver created thermal stresses which mechanically degraded the receiver modules. Surface oxidation was also detected, which can be solved by using siliconised silicon carbide; however, this would lower the upper allowed temperature for the receiver. Further details about the degradation at microscopic levels are presented by Agrafiotis et al. (2007) and Licheri et al. (2019), while further details about the oxidation and mechanical failures of silicon carbide when used at other ultra-high temperature thermal reactors are documented by Liu (2018).
3.3.3.2. Tubular receivers

Pressurised fluids in closed-loop systems typically flow in tubes (Benoit et al., 2016). The external surfaces of the tubes are irradiated by the concentrated solar sunbeams, and the energy is then transmitted to the flowing gas via mixed conduction and convection. Tubular receivers have been in use in solar systems since the 1970s and have become the dominant type of receivers, given their simpler fabrication compared to volumetric receivers (Ho and Iverson, 2014, Zheng et al., 2020). Tubular receivers are verified for superheated CO$_2$ and hydrogen at high pressures of up to 25 MPa (Besarati et al., 2015, Ho and Iverson, 2014, Liao et al., 2013, Liu et al., 2019b), which may enable direct integration with Brayton cycles and deliver energy conversion efficiencies >50% (Ho, 2017, Turchi et al., 2013, Seidel, 2011). However, direct storage of pressured supercritical fluids is unlikely to be viable (Kelly, 2010).

The German Aerospace Centre (DLR) developed and tested several tubular receiver designs, as exemplified in Figure 3.19(a), for hybrid systems up to 1 MW$_{th}$ during the 2000s (Amsbeck et al., 2008, Heller et al., 2006, Heller et al., 2009b). The main design concept in DLR tubular receivers was using tubes with profiled multi-layer design, in which copper is sandwiched, by a hydroforming process, in between Inconel material, as shown in Figure 3.19(b). This tube design helps enhance radial and circumferential thermal flow distribution and heat transfer to the gaseous HTF (Ho and Iverson, 2014). To minimise the convective losses, the aperture of this cavity receiver was closed by segments of silica glass, as shown in Figure 3.19(c), which was found to boost the thermal efficiency of the receiver by 13% (Amsbeck et al., 2009a).

The heat transfer from the tube walls to the gaseous HTF is the main limitation of tubular receivers (Ho, 2017). Tubes at high temperatures are subjected to large transient thermomechanical loads, which can negatively affect their fatigue life (Ho and Iverson, 2014, Uhlig, 2011). In a numerical study of the thermal-fluid-mechanical characteristics of a tubular receiver operating on a pressurised gaseous HTF (superheated CO$_2$), the non-uniform solar flux in the cavity receiver resulted in a temperature gradient, which was found to play the dominant role in augmenting the stress and deformation of the tubular receiver (Wang et al., 2021a).
To combine the benefits of volumetric and tubular receivers, Heller et al. (2009a) proposed a pressurised air tubular receiver with the staggering volumetric arrangement of tubes displayed in Figure 3.20, so that the front tubes intercept the escaping radiation from the back tubes. Nevertheless, CFD simulations revealed poor heat transfer compared to volumetric receivers and lower outlet temperatures than the target temperature (Craig et al., 2014).
3.3.4. Liquid metals

Using liquid metals as HTFs has historically been driven by the nuclear power sector. Several studies (Frazer et al., 2014, Fritsch et al., 2015, Heinzel et al., 2017, Lorenzin and Abánades, 2016, Pacio et al., 2013) presented positive outlooks for using liquid metals as HTF in future CSP systems. Liquid metals are anticipated to replace molten salts at ultra-high temperatures, as the latter decompose at temperatures <850 K (Boerema et al., 2012). Such a replacement is projected to reduce the levelised cost of electricity production by up to 15% (Singer et al., 2010) and improve the receiver’s thermal efficiency by 20% under 2000 suns (Pacio et al., 2013). A cost analysis of a CSP plant operating at ~1473-1773 K using liquid metals as HTF and storage medium found that despite the increased costs of the components, the net cost reduction of the plant is expected to range between 20-30% depending on the heat engine efficiency (Wilk et al., 2018). The primary candidate liquid metals for use as HTFs in tower receiver systems at temperatures >1000 K are molten sodium, lead-bismuth, and tin based on various studies (Boerema et al., 2012, Ho, 2017, Coventry et al., 2015, Frazer et al., 2014, Kim et al., 2015, Romero and González-Aguilar, 2016).

Liquid metals are characterised by their wide liquid ranges (Figure 3.21), which facilitate single-phase operation across wide temperature ranges without concerns about volumetric expansion. Compared with gaseous HTFs in tubular receivers, a heat pipe with a liquid metal flow can deliver a greater internal heat transfer coefficient by an order of 100 (Bienert et al., 1979, Ho and Iverson, 2014). Accordingly, liquid metals are promising HTFs for ultra-high temperature CSP and TES applications.

Liquid metals are grouped into alkali, heavy, and fusible metals (Pacio and Wetzel, 2013). Alkali metals, which include lithium, sodium, and potassium, share similar properties, as they all have their outmost electron in an s-orbital (Mishra et al., 2003). They are characterised by their shininess, softness, high reactivity (highly susceptible to oxidation), low densities, relatively low boiling and freezing temperatures, high specific heat capacities and high thermal conductivities (Heinzel et al., 2017, Mishra et al., 2003). Sodium is the group’s most promising HTF, given its low reactivity and low cost compared to other alkali metals, in addition to its acquired practical experience of in nuclear power plants (Pacio and Wetzel, 2013).
Figure 3.21. Thermal properties of liquid metals compared to the solar salt (Fritsch et al., 2015). (a) Liquid temperature ranges. (b) Heat transfer coefficients.

Heavy metals\textsuperscript{4} are metallic elements with atomic numbers >20 and atomic densities >5000 kg m\textsuperscript{-3}, which include lead, iron, mercury, zinc, copper, silver, gold and platinum (Raychaudhuri et al., 2021). In addition to their high densities, they are characterised by their high boiling points and slow oxidation with air and water compared to alkali metals. However, their specific heat capacities and thermal conductivities are lower than alkali metals (Pacio and Wetzel, 2013). Moreover, they experience rapid volumetric expansions at ultra-high temperatures with direct-contact boiling. The toxicity of heavy metals, as well as their oxides, is another concern for their use. Lead-Bismuth Eutectic (LBE) alloy is one of the leading heavy liquid metals that have been used and tested as an HTF in various nuclear power reactors around the globe since the 1950s (Frazer et al., 2014, Morita et al., 2006) and has been proposed as a strong candidate for future CSP systems (Heinzel et al., 2017, Pacio et al., 2013). Accordingly, LBE will be the nominated liquid metal of the heavy metals group.

\textsuperscript{4} Tin also falls into the heavy metals group, but it is more prominent as a fusible metal.
Fusible metals – which include tin, gallium, and galinstan – are characterised by their low melting points (<550 K) and high boiling points (>2500 K) (Lorenzin and Abánades, 2016). The extremely low melting points of gallium (303 K) and galinstan (254 K) promote them as attractive candidates for HTF applications; however, their excessive corrosiveness and cost prohibit their extensive use (Pacio and Wetzel, 2013, Deng et al., 2021). Molten tin is the nominated metal from this group, given the established base of practical experience acquired from its use at ultra-high temperature conditions in the glass industry since the 1960s (Pilkington, 1969).

3.3.4.1. Sodium

Molten sodium is the only liquid metal to be used in a commercial CSP plant. The Australian plant (Jemalong Solar Thermal Station) is a 6 MW_{th} multi-tower plant constructed in 2021 (Vast-Solar, 2022). The plant comprises five towers, each with a flat billboard-shaped (external) tubular receiver (Figure 3.22) where sodium is heated from 543 K to 833 K under a flux equivalent to 1500 suns (Coventry et al., 2015).

Figure 3.22. One of the molten sodium based tubular tower receivers (1.2 MW_{th}) installed at Jemalong Solar Thermal Station (Coventry et al., 2015).

Liquid sodium receivers were tested in several setups, most notably the 2.85 MW_{th} facility constructed by Rockwell International in Alberquerque, New Mexico (Sandia, 1983), which was tested under a flux equivalent to 1530 suns and resulted in receiver efficiencies peaking at 90% ±10%. In Spain, liquid sodium was tested as a HTF in two different tubular receivers (Figure 3.23). The first was a semi-cavity receiver with a design comprising an octagonal aperture for the cavity with an internal absorbing surface of a 120° section of a vertical cylinder (Coventry et al., 2015, IEA, 1986). A serpentine arrangement of horizontal tubes filled the interior of the cavity and were
subjected to a peak flux equivalent to 630 suns to heat sodium from 543 K to 803 K with a daily average receiver efficiency of 66.7% ±5% (Baker, 1987). The second receiver was an external billboard receiver, with a design comprising five series bundles of 39 tubes operating at similar temperatures to the first receiver but at higher solar fluxes (peaking at 1400 suns), resulting in a daily average receiver efficiency of 79.1% ±5% (Baker, 1987). The lower efficiency of the semi-cavity receiver was attributed to the poor heat flux distribution inside the semi-cavity receiver and greater (twice) absorption surface area exposed to higher convective and radiative losses (IEA, 1986, Casal, 1987). These results highlight the significance of maximising the optical concentration ratio to achieve high receiver efficiencies.

Figure 3.23. Photographs and schematics of two molten sodium receiver setups constructed in Almeria, Spain (Baker, 1987, Casal, 1987, IEA, 1986). (a) 2.8 MW th semi-cavity receiver. (b) 2.5 MW th external receiver.
Liquid sodium is distinguished by its excellent specific heat capacity and thermal conductivity (Table 3), which impel better heat transfer with tube walls and enable direct use as storage material. Nevertheless, during experimental tests of the 2.5 MW\textsubscript{th} external receiver in Almeria, the tube surface temperature near to the peak solar flux was revealed to be substantially higher than the bulk temperature of the HTF, as demonstrated in Figure 3.24(a). Figure 3.24(b) displays the simulated tube temperature across the tube thickness, which reveals that the thermal conductivity through the tube, evaluated as 21.5 Wm\textsuperscript{-1}K\textsuperscript{-1} at 773 K (Atlas-Steels, 2011), is the main cause for the temperature difference between the tubes and bulk of the HTF. This temperature difference was predicted to augment at higher temperatures and concentration ratios, marking a significant drawback of tubular receivers for liquid metals. Coventry et al. (2015) claims that to utilise the full potential of molten sodium as a HTF, higher conductive tube and new absorber coating materials need to be developed to resist the thermal stresses and fatigue at high temperatures.

![Figure 3.24](image)

Figure 3.24. Temperature results of the external tubular receiver in Almeria, Spain (Schiel and Geyer, 1988). (a) Simulated and measured surface temperatures of tube, sodium temperature, and heat flux at the peak flux region on the receiver. (b) Simulated wall temperature across the tube’s circumferential direction. The lines represent the depth layer within the tube with uppermost resembling the outer surface of the tube, while the lowermost resembling the inner surface of the tube.

The low boiling temperature of sodium (1156 K) inhibits its use in ultra-high temperature applications. While the use of sodium as liquid-vapour phase change via heat pipes or pool boilers was considered by Coventry et al. (2015), no concepts for the receiver design have been proved feasible yet. The toxicity and safety concerns associated with sodium has been alleviated over decades of practical experience and improved safety measures in the nuclear power sector since the PSA’s fire accident in 1986 (Romero and González-Aguilar, 2016).
3.3.4.2. Lead-bismuth eutectic

Although not yet employed in commercial CSP plants, LBE has been proposed, recommended, and demonstrated as a viable HTF for solar thermal application (Ho, 2017, Deng et al., 2021, Wetzel et al., 2014, Frazer et al., 2014). Similarly to molten sodium, liquid LBE has been experimented in large-scale nuclear reactors in various countries, leading to an extensive build-up of operational experience (Subbotin et al., 2002). LBE has a higher boiling point (1943 K) than sodium, allowing its use at ultra-high temperatures. Its low specific heat capacity is compensated by its high density, resulting in a greater volumetric heat capacity than molten sodium by 30% and lower than solar salt by 45% (refer to density and specific heat capacity values in Table 3). Furthermore, as a liquid metal, the thermal conductivity of LBE is significantly higher (~40 times) than solar salt; however, it is three times smaller than molten sodium. In addition to being more costly than sodium, the main challenge of using LBE is its high corrosiveness and incompatibility with containment materials, which must be solved before use in long-term ultra-high temperature applications (Kim et al., 2015, Frazer et al., 2014, Pacio and Wetzel, 2013). LBE was tested as a HTF and storage material at 1043 K in a storage integrated tower receiver system by Kim et al. (2015), as shown in Figure 3.25. Although the study found that LBE, as a HTF, has delivered high efficiency heat transfer, LBE was not recommended as a storage material or primary HTF for high-capacity systems, and was proposed, instead, for short-term buffer storage. The study also asserted that there is currently no compatible metal-based material with LBE for long-term applications.

Figure 3.25. A storage integrated tower receiver based on LBE as a HTF (secondary to pressurised air) and storage medium in a two-tank sensible heat TES system (Kim et al., 2015).
3.3.4.3. Tin

Molten tin is characterised by its high boiling point (2875 K), low melting point (505 K), stability at ultra-high temperatures, non-toxicity, and low hazardousness compared to LBE and alkali metals (Lorenzin and Abánades, 2016). Molten tin has been used for decades in the *Pilkington process* (Pilkington, 1969), where glass sheets are manufactured by floating on a bed of molten tin at 1373 K to deliver sheets with uniform thicknesses and smooth surfaces. Therefore, a base of practical experience with using molten tin in static flow conditions has been developed in the process, including protecting molten tin from oxidation by enclosing the bath in a slightly pressurised atmosphere of Nitrogen and Hydrogen (Francis, 2016).

The specific cost of tin is higher than of the other candidate HTFs; however, in a study of large-scale (120 MW) tower receiver systems with different HTFs (Singer et al., 2010), liquid tin was found to result in close cost reductions to liquid sodium and LBE, as demonstrated by Figure 3.26. The rationale behind the economic advantage of using liquid tin was linked to the improved thermal conductivity of the HTF, which minimised the radiative losses from the receiver (Singer et al., 2010).

![Figure 3.26](image)

Figure 3.26. Cost reduction potential – displayed in terms of levelised electricity cost relative to that of the solar salt (NaNO₃-KNO₃) based system – of five different HTFs – two molten salts and three liquid metals – operating in a tower receiver system running an ultra-supercritical steam cycle with optimal number of panels and storage capacities (Singer et al., 2010). The darker (magenta) bars represent systems with equal storage sizes, while the lighter (lavender) bars represent systems with optimal storage sizes.
Using liquid tin or LBE as a direct storage medium in two-tank TES systems would burden the system financially due to their uncompetitive specific heat capacities, which are reflected in their elevated storage cost per unit of energy, as shown in Figure 3.27(a). According to Fritsch et al. (2015), a potentially feasible approach to use the liquid tin as a storage medium can be through a segmented storage system with a low-cost matrix (filler) material, such as quartzite or aluminium oxide, which was found to result in competitive storage costs per unit of energy, as shown in Figure 3.27(b). The three storage systems compared in this study are demonstrated in Figure 3.27(c).

![Figure 3.27. Storage costs (€/kWh$_{th}$) for solar salt and selected liquid metals in a 125 MW$_{elec}$ CSP system with a 10 h storage capacity (Fritsch et al., 2015).](image)

(a) Direct TES using the two-tank storage system. (b) All studied storage types with costs <£30/kWh$_{th}$. (c) Illustrative diagrams of the three studied storage systems: Two-tank direct storage (left); Thermocline storage with a floating baffle (middle); Segmented storage type with filler matrix (right).

Molten tin is chemically inert (Taylor, 2014); however, its high corrosiveness at ultra-high temperatures, which is shared with LBE, has been a major limitation for its use as a coolant in nuclear reactors for decades (Lorenzin and Abánades, 2016, Pacio and Wetzel, 2013). Conventional containment materials were found to degrade fast upon contact with molten tin due to the accelerated mass diffusion and reaction kinetics at ultra-high temperatures (Legkikh et al., 2016, Yvon, 2017). Various studies...

Oxygen content in some liquid metals, including molten tin, needs to be controlled (Agbede, 2014). If the oxygen content drops too low, this risks metal dissolution and the formation of intermetallic compounds at the surface, while a high oxygen content can instigate precipitation of oxygen in the colder regions, resulting in fouling in tubes of heat exchangers and pipes (Lorenzin and Abánades, 2016, Smith, 1966). Bolind et al. (2013) demonstrated techniques for controlling the oxygen content in the molten tin using commercially available and cost-efficient ceramic sensors while using argon and hydrogen for cleaning the tin at temperatures >873 K.

One of the main practical concerns in dynamic flow systems involving corrosive liquid metals at high temperatures is their direct physical contact with the moving parts in mechanical pumps, which causes mechanical degradation to these parts and induces thermal stresses and fatigue. Electromagnetic pumps offer an alternative to mechanical pumps, where electric and magnetic fields are used to drive the electrically conductive liquid metal by Lorentz force without any moving part. An electromagnetic pump was demonstrated to circulate molten gallium at 1773 K in cylindrical ducts (Ando et al., 2004). A magnetohydrodynamic generator, which generates electricity based on Faraday's law of electromagnetic induction, has also been recently proven to operate with molten gallium alloy at ultra-high temperatures and high Reynolds numbers (Namala and Krishna, 2021). Liquid metal magnetohydrodynamic pump and generator were proposed for a generic solar thermal system, demonstrated in Figure 3.28, to replace the conventional mechanical turbomachinery (Kaushik et al., 1995). While magnetohydrodynamic generators promised energy conversion efficiencies exceeding 60% (Baker and Tessier, 1987), such efficiencies are yet to be practically proven for large-scale systems, as well as their economic viability (Deng et al., 2021). For pumps, mechanical pumps remain superior in terms of the existing base of knowledge and practical experience. Efficiency-wise, electromagnetic pumps can pump alkali metals at <40% and heavy metals at <10%, while mechanical pumps typically have efficiencies ranging between 60-90% (Fritsch et al., 2015).
Figure 3.28. Using a magnetohydrodynamic (MHD) pump to drive two-phase liquid metal cycle with heat addition sourced from a solar thermal collector and a magnetohydrodynamic converter to generate electricity (Kaushik et al., 1995).

In 2017, a technological breakthrough transpired with the successful continuous pumping of molten tin at 1673 K using a mechanical (an external gear pump) ceramic pump (Amy et al., 2017), which involved a piping network as shown in Figure 3.29. The pump was made from Shapal Hi-M Soft, which is a machinable aluminium nitride composite characterised by its high thermal resistance and mechanical strength (Precision-Ceramic, 2022). The extremely low thermal expansion coefficient of the pump's material (5×10^{-6} K^{-1}) matches that of graphite, which was used for piping, joints, and seals (Amy et al., 2017). Outside the sealed sections, tungsten was used to withstand the tensile stresses.

The development of the ultra-high temperature mechanical pump revived the interest in using molten tin as HTF in several applications, particularly in solar thermal systems. For example, in a proposed hydrogen production system (Figure 3.30), molten tin was suggested as the HTF in a tower receiver integrated circuit to thermally crack methane in a bubble reactor (Zheng and Xu, 2018). In a different study, three candidate materials – graphite, silicon carbide, and mullite – were investigated for their suitability as containment materials for molten tin at temperatures >1573 K by Zhang et al. (2018b). Overall, all three materials exhibited negligible reaction or penetration by molten tin at 1623 K after direct and continuous contact for 100 hours. It worth noting that commercial grades of molten tin (≤98 wt% pure) were used in these experiments.
A sensitivity analysis was performed by DeAngelis et al. (2018) to determine the critical design factors that influence the performance of a liquid metal cavity receiver at 1623 K. The study revealed that cavity dimensions, natural convection inside the cavity, thickness of cavity wall insulation, and re-radiative energy loss at the insulation surfaces all showed negligible impact on the receiver performance. One of the main insights delivered by this study was that the emissivity of the cavity’s internal surfaces has minimal effect on the thermal efficiency of the receiver. The critical factors influencing the receiver performance were determined as the thermal conductivity of the absorbing materials in the receiver, wind-induced convection from the cavity, and the location of hot spots created within the cavity. The study recommended designing
the receiver in a way to keep the hot spots near the HTF outlet with minimised view factors with the aperture. Regarding the reliability of the receiver, the thermal stresses, induced by the substantial temperature differences inside the cavity, at the absorber tubes were found likely to induce thermomechanical failures, as they can easily exceed the fracture strengths of candidate containment materials (graphite was used as the containment material). An improved cup-cone design for the tubular receiver was proposed to reduce the induced thermal stresses compared to the basic U-tube design, as shown in Figure 3.31. However, under different and transient flux conditions, thermal stresses can still persist. Although graphite exhibits high creep ductility under irradiation, it does not creep significantly at ultra-high temperatures (Blackstone, 1977); fatigue may start to become a concern if stresses exceed half the fracture strength of graphite at ~43 MPa (Leichter and Robinson, 1970).

![Figure 3.31. Thermal stress distribution in two proposed designs for molten tin based tubular solar receiver (DeAngelis et al., 2018). (a) U-tube design. (b) Cup-cone design.](image)

### 3.4. Direct solar absorption

There are two mechanisms for solar absorption by the HTF. The first, and most commonly employed mechanism in solar thermal applications, is surface absorption, in which the solar irradiation is absorbed by an opaque solid absorber covered with a selective coating to maximise its absorptance. The absorbed energy is then conveyed to the HTF by combined conduction and convection, as illustrated in Figure 3.32(a).
The second approach is when the absorber material is eliminated or replaced by a transparent cover to allow for direct optical contact between the HTF and solar irradiation, as demonstrated in Figure 3.32(b). This absorption mechanism is called volumetric absorption, as solar beams penetrate through the fluid volume. Radiation is a volumetric phenomenon for fluids (Çengel, 2003) and is only applicable to fluids which are not entirely optically transparent. Detailed specifications of the volumetric absorption process will be provided in section 6.3.8.1.

![Diagram](a) Surface absorption by a conventional fluid. (b) Volumetric absorption by a nanofluid.

Figure 3.32. The two main mechanisms for solar absorption by a fluid (Khullar et al., 2017). (a) Surface absorption by a conventional fluid. (b) Volumetric absorption by a nanofluid.

The limitations associated with intermediate solid absorbers have increased the interest in volumetric absorption for high-temperature applications. Recently, a novel receiver considered volumetric absorption by molecular gases, such as water vapour and CO₂, combined with surface absorption by a blackened back-plate as shown in Figure 3.33 (Ambrosetti and Good, 2019). The concept is purely radiative and operates similarly to the greenhouse gas effect in glazed solar collectors. The rationale of this mode of operation is that the molecular gasses will absorb a significant fraction of the reradiated thermal radiation, while transmitting terrestrial solar radiation with shorter wavelength, hence reducing the radiative losses from cavity receivers at ultra-high temperatures (up to 1800 K). Nevertheless, the design did not consider the reliability issue of solid absorbers at ultra-high temperatures, which was reported to exceed 2000 K, and assumed uniform absorption by the back-plate.
3.4.1. Nanofluids

Water, air, and some molten salts are optically transparent near the solar peak wavelength. As a result, volumetric absorption is not viable by these fluids without optical manipulation. One way to improve the properties of fluid is by mixing a small amount (<10% by volume of the base fluid) of nanometre-sized particles with the desired properties (Taylor et al., 2013). These particles are usually made from metals, metal oxides, carbides, carbon nanotubes, or diamond nanoparticles (Buongiorno, 2005). The colloidal suspension of these particles in base fluids can be engineered, due to particle size dependence of optical properties, to improve the optical absorptance of the fluid while minimising the thermal emission (Otanicar et al., 2009).

The applicability of nanofluids in various CSP systems has been extensively explored in the literature (Nagarajan et al., 2014, Tembhare et al., 2022, Tyagi et al., 2009, Zeng and Xuan, 2022). For solar power towers, Taylor et al. (2010) showed that the use of a nanofluid-based receiver could improve the receiver efficiency by an order of 10% with minimal changes to the design and capital investment. The applicability of molten-salt-based nanofluids in CSP and TES was also evaluated in the literature, which is summarised by Muñoz-Sánchez et al. (2018). Furthermore, engineered nanofluids are practically proven as efficient direct solar absorbers in volumetric receivers at 573 K (Singh and Khullar, 2019) and solar fluxes up to 100 suns (Karim et al., 2019). Despite their potential in improving the efficiencies of solar receivers and TES systems, the practical applicability of engineered nanofluids has not yet been demonstrated at temperatures >900 K (Merchán et al., 2022, Palacios et al., 2020).
3.4.2. Solid particles

The high-temperature alternative to nanofluids is using solid particles as direct solar absorbers. The use of refractory absorptive solid particles can enable solar collection at temperatures in excess of 1273 K and fluxes up to 3000 suns (Ho, 2017). Utilising their inherent energy storage capacities, the heated particles, or pebbles, can then be used directly as a storage medium in a TES system (Al-Ansary et al., 2015). Typically, a secondary fluid, usually air, is used to extract heat from the particles through a heat exchanger. Figure 3.34 illustrates the basic configuration of a tower receiver system based on solid particles. Particle receivers are currently subjected to several technical challenges, including material constraints at high temperatures, non-uniformity of the mass flow rate, cyclic collisions of particles, and particle loss (Ho et al., 2019).

![Figure 3.34. A schematic of storage integrated particle tower receiver (Wu et al., 2014).](image)

The most basic design for particle receivers is the free-falling receiver, which was first proposed and built by Sandia National Laboratories in the 1980s (Falcone et al., 1985). The 1 MWth test facility (Figure 3.35(a)) comprised ceramic particles released from the top of the receiver through a discharge slit (Figure 3.35(b)). This gravity-driven flow creates an opaque thin (5-20 mm) curtain, which interjects the solar beams entering the cavity receiver from its aperture. In this facility, particles reach peak temperatures of 1173 K and bulk temperatures of 1073 K (Ho, 2016). Tests were carried out at 1000 suns with a reported particle heating rate reaching 300 K per metre drop distance and thermal efficiencies reaching 80% (Armijo and Shinde, 2016). The system was developed to run a supercritical CO₂ Brayton cycle for power production. Further details about the design of this receiver are described by Ho et al. (2016a).
The lack of control over the gravity-driven flow of particles in the free-falling designs led to some practical issues. The short residence time, induced by the gravity-driven acceleration of particles, was found to result in inefficient absorption of solar radiation and lower outlet temperatures (Ho et al., 2016b). Accordingly, recent developments focused on maximising the residence time of particles in the irradiated zone. The use of mechanical lifts and conveyance systems to recirculate the particles, as exemplified in Figure 3.35, has been a common feature in proposed particle receiver designs (Christian and Ho, 2014, Röger et al., 2011); however, this approach significantly increases the costs and structural complexity of particle receivers (Sment et al., 2022). An alternative approach to enable some degree of control over the gravity-driven flow of particles is by mechanically obstructing the flow with porous structures (Lee et al., 2015) or interruption by stages of solid obstacles (Khayyat et al., 2015). Nevertheless, later tests revealed that the solid structures used in obstructing the particle flow were subjected to overheating, oxidising, and degradation caused by the exposure to concentrated solar irradiation and wear from the physical contact with
hot particles (Ho et al., 2016b, Ho et al., 2017). To control the residence time of particles, while avoiding the technical issues of inserting solid obstructers at highly irradiated zones, particles can be fed into an axially rotating cylindrical receiver, so that the centrifugal force maintains the particles at the interiors of the receiver (Wu et al., 2014). In this design, particles enter one end of the cylindrical receiver, while concentrated solar irradiation enters the other end as demonstrated in Figure 3.36. Since gravity is the drive of the main (axial) flow of particles through the receiver, the receiver must either be facing the ground or inclined vertically up to $80^\circ$. Therefore, the residence time and outlet temperature are controlled through the rotational speed.

![Figure 3.36. A simplified schematic of a centrifugal particle receiver displaying main principles of operation (Wu et al., 2014).](image)

DLR carried out experimental and numerical tests on their novel receiver (Figure 3.37) and demonstrated average particle temperature and receiver efficiency reaching 1238 K and 75%, respectively (Wu et al., 2015a, Wu et al., 2015b). The 7.5 kW$_{th}$ prototype installed in Jülich is tilted $45^\circ$ to the ground, as shown in Figure 3.37(b), and uses spherical bauxite particles as the absorber (Reuschenbach and Buck, 2018). The design was successfully upscaled to 2.5 MW$_{th}$ and tested with reported maximum particle outlet temperature reaching 1263 K (Ebert et al., 2016, Amsbeck et al., 2018). Tests on the large-scale system exposed some practical issues, such as wind-driven particle loss through the aperture, particle loss in the junction between the rotational and stationary collector rings, clogging and flow issues related to particle impurities, overheating and excessive thermal expansion of wire mesh installed at the receiver interior surfaces (Ebert et al., 2019). It is anticipated that with further design improvements and upscaling to 20 MW$_{th}$, the system may become more economically viable for power production with a direct TES particle system (Frantz et al., 2020).
For open-aperture cavity receivers, wind-induced particle loss can be substantial, particularly for receivers with shallow cavity depths (Kim et al., 2010). Therefore, many proposed receiver designs incorporated either glazed (Amsbeck et al., 2009b) or aerodynamic (Tan et al., 2009) windows to seal the cavity from the wind effects. Despite their substantial benefits to the receiver (Uhlig et al., 2014), glazed windows risk solar beam blockages and mechanical wear by collisions and depositions of particles at the internal surface of the glazed window (Coventry et al., 2017, Shahabuddin, 2016). Therefore, air curtains and fluidic seals are considered more suitable for particle receivers to reduce particle losses; however, air curtains were found to have a minimal impact on improving the flow characteristics of particles and their loss in other scenarios than through the aperture (Ho et al., 2014).

Fluidised particles in solar receivers and TES has been extensively investigated for thermochemical and power production applications (Benoit et al., 2015, Falcone et al., 1985, Flamant, 1982, Steinfeld et al., 1992, Wang et al., 2016). Such systems use air-fluidised particles made from silicon carbide, zirconia, or silica sand, which can either be indirectly heated by the solar radiation, as demonstrated in Figure 3.38(a), or directly irradiated as illustrated in Figure 3.38(b). In Japan, directly irradiated fluidised particles were demonstrated at ultra-high temperatures (up to 1473 K) in a 100 kWth beam-down tower reflector system (Kodama et al., 2019). Yet, fluidised particles are limited to small scales, with doubts over their scalability (Tregambi et al., 2021).
Figure 3.38. Fluidised particle receiver concepts. (a) Fluidised particles flowing in a tubular enclosure of a TES integrated tower receiver system (Next-CSP, 2016). (b) Directly irradiated gas-fluidised particles bed in a beam-down tower system (Kodama et al., 2017).

The benefits and challenges of different particle receiver technologies are summarised in Table 4. Further to mentioned challenges, Siegel et al. (2014) found that the solar absorptance and stability of directly irradiated particles deteriorated more significantly when operating at 1273 K than at 973 K, which indicates that prolonged
exposures to air at ultra-high temperatures may result in chemical transformations of the particles. The study also found that the particles are more prone to oxidation at a temperature >973 K, which can be particularly problematic in TES-integrated systems where particles reside for prolonged periods at high temperatures.

### Table 4. Comparison between the five different particles receiver designs (Ho, 2017).

<table>
<thead>
<tr>
<th>Receiver design</th>
<th>Outlet temperature/thermal efficiency</th>
<th>Advantages</th>
<th>Challenges/research needs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Free-falling direct absorption particle receiver</td>
<td>&gt;700 °C/90% (modeling of ~100 MW, system), 50-80% (prototype testing ~1 MW)</td>
<td>Direct heating accommodates very large solar fluxes (~3000 sun); scalable to large capacities and particle mass flow rates</td>
<td>Increased convective heat losses from entrained airflow; particle loss through open aperture</td>
</tr>
<tr>
<td>Obstructed-flow Direct Absorption Particle Receiver</td>
<td>&gt;800 °C/60-90% (prototype testing ~1 MW)</td>
<td>Reduced velocity of particles (terminal velocity of ~0.5 m/s) resulting in increased residence time and heating; reduced entrainment of air flow and convective losses; potentially less particle loss</td>
<td>Potential overheating of obstruction materials with direct irradiance; additional cost of materials and complexity of installation</td>
</tr>
<tr>
<td>Centrifugal Direct Absorption Particle Receiver</td>
<td>~900 °C/75% (prototype testing ~15 kW)</td>
<td>Controlled particle residence time based on rotational speed of receiver</td>
<td>Large rotating receiver; scalability to larger systems (~100 MW)</td>
</tr>
<tr>
<td>Enclosed Falling Particle Receiver (Gravity Driven Flow)</td>
<td>&gt;700 °C/90% (modeling of enclosed receiver with aperture flux &gt; ~1000 kW/m²)</td>
<td>No particle loss and reduced convective loss</td>
<td>Large heat-transfer resistance between irradiated walls and particles; increased outer-wall surface temperature and potential for hot spots and limitations on solar flux</td>
</tr>
<tr>
<td>Enclosed Fluidized Particle Receiver</td>
<td>750 °C/Thermal efficiency to be determined</td>
<td>Increased particle-side heat transfer coefficient; no particle loss</td>
<td>Increased parasitics of fluidization; potential heat loss from fluidization gas; scalability of particle mass flow</td>
</tr>
</tbody>
</table>

#### 3.4.3. Liquid metals

There are limited options for direct solar absorber HTFs at ultra-high temperatures, as most liquid HTFs decompose and or evaporate, while gaseous HTFs are mostly optically transparent. As discussed in section 3.3.4, liquid metals are opaque materials which maintain their liquidity at ultra-high temperatures. Yet, most liquid metal-based receivers have considered the use of indirect heating of liquid metals via solid tubular absorbers (DeAngelis et al., 2018, Falcone, 1986, Pacio and Wetzel, 2013), which were found to impose critical design factors that impede the performance and reliability of the solar receiver at ultra-high temperatures (DeAngelis et al., 2018).

Tammen and Bobby (1984) considered using an optically exposed liquid mercury in their solar power system (Figure 3.39) as an HTF and direct storage medium in a two-tank TES system. At the solar concentrator, liquid mercury flowed within a transparent envelope to serve as the concentrating reflector and coolant.
The main challenge of using liquid metals as direct solar absorbers are their naturally high reflectance and low absorptance. However, at ultra-high temperatures, the reflectance of most metals drops substantially, as shown in Figure 3.40. While these drops may be substantial in the visible spectrum (Figure 3.40(a)), which carries the majority of the solar radiant energy, they are less steep at higher wavelengths (Figure 3.40(b)). According to Ujihara (1972), metals with lower reflectance at room temperature are likely to experience steeper drops in reflectance at high temperatures. Furthermore, the influence of a non-zero emissivity\(^5\) of the cavity interiors on the receiver’s thermal efficiency can be minimal (<5%), which was demonstrated for the cavity receiver at 1673 K (DeAngelis et al., 2018) and 623 K (Fang et al., 2014). Therefore, the optical properties of the liquid metals may not restrict their use as direct solar absorbers. The influence of the optical properties on the performance of the proposed receiver will be presented in Chapter 7.

While the solar absorptance of solid tin is <0.05 at room temperature, it can exceed 0.2 at liquid temperatures >1100 K (Greenstein, 1989). Solar absorptance of liquid metals can also be enhanced by colouration (Hou et al., 2018) or the addition of absorbing particles (Phelan et al., 2013).

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\(^5\) Emissivity is equivalent to optical absorptivity when the body is in a thermodynamic equilibrium according to Kirchhoff's law of thermal radiation
Figure 3.40. Reflectance drops with temperature in selected metals at wavelengths of (a) 690 nm and (b) 1060 nm (Ujihara, 1972). Though not plotted, tin trends are likely to be between the trends of Cu and Al based on their reflectance values at 300 K.

Oxidation of liquid metals can be another concern. Despite having moderate oxidation rates in its liquid state compared to other liquid metals, pure molten tin was found to exhibit exceptionally rapid oxidation and nearly linear reaction kinetics at temperatures >973 K (Yuan et al., 1999). Oxidation may be favourable for applications that use still liquid metal, such as TES (Jafarian et al., 2017, Silakhori et al., 2017). While oxidation can improve the optical absorptance of liquid metals (Bergström et al., 2007), this benefit might not be exhibited in dynamic flow systems, while some of the favoured thermal properties may be compromised by the oxidation (Dobosz et al., 2021, Sobolev et al., 2020). Therefore, oxidation prevention measures are likely to be required in liquid metal CSP systems. For molten tin, oxidation prevention measures may include enclosing the exposed flow by an optically transparent protective environment, as in the Pilkington process (Pilkington, 1969), or by doping the liquid metal with phosphorous (Xian and Gong, 2007).

3.5. **Flow configurations for an optically exposed liquid metal**

To the best of the author’s knowledge, there is no available design for liquid-metal-based receivers in the literature other than tubular receivers. Concepts used for particle receivers or exposed molten salts, nonetheless, may be utilised for liquid metals.
The ‘free-falling’ flow configuration can be applied to liquid metals to form an opaque curtain to interject the solar radiation, as illustrated in the external receiver displayed in Figure 3.41(a). Despite the promising results of preliminary tests at 973 K (Bohn, 1987), this external configuration is subjected to wind-induced disturbances to the HTF flow and substantial thermal losses, as described previously in section 3.1.

Other potential concepts include inclined HTF flow inside a cavity receiver, as demonstrated in Figure 3.41(b) and (c). In these configurations, a semi-transparent liquid HTF is infused from the top and flows radially down the interior walls of the cavity receiver, which are heated by the transmitted fraction of solar beams through the HTF film. This configuration was estimated to boost the annual electricity production by 14% (Tracey et al., 1989) and deliver higher outlet temperatures with 40% lighter and 30% less expensive compared to the conventional tubular receiver configuration (Tyner and Wu, 1988). The reported limitations, which included overheating and contamination of the HTF, when testing these receivers were mostly related to the thermo-physical characteristics of molten salts rather than to the flow configuration itself (Ho and Iverson, 2014). However, the stability of the film flow was also emphasised as a primary concern for this flow configuration. The rotation of the receiver displayed in Figure 3.41(c) was proposed to minimise the film flow instabilities in a similar technique to the centrifugal particle receivers discussed in section 3.4.2. It should be noted that most of these concerns are either not applicable or minimal to liquid metal HTFs, given their chemical passivity and favourable thermal and flow properties at ultra-high temperatures, as explained in section 3.3.4.

Figure 3.41(d) presents another receiver concept, in which the opaque walls are directly irradiated by the solar flux, while the HTF film is indirectly heated via conduction and convection. Although the solid absorber walls in this configuration are still susceptible to the thermomechanical-induced failure modes discussed previously, they are minimised by the enhanced heat transfer between the HTF film and walls, in addition to carrying numerous benefits to the energy collection performance compared to tubular receivers, as explained by Tracey et al. (1989). Furthermore, the design simplicity, due to the replacement of tube bundles with thin plates, can minimise the pumping requirement and reduce the weight and costs by about 6% (León et al., 1999). A potential upgrade to this configuration may be performed by replacing the opaque
absorber walls with transparent ceramic walls to enable direct absorption by the HTF film while mitigating the overheating of walls and the formation of hot spots on their exposed surfaces. Nevertheless, the chemical compatibility of transparent ceramics with molten metals, as well as their current forming and manufacturing techniques, are not yet demonstrated for such an application.

Figure 3.41. Design concepts for gravity-driven liquids in non-tubular flow configurations. (a) An external receiver for a surround heliostats field (Bohn and Green, 1989). (b) A cavity receiver for a polar (North-facing) heliostats field (Webb and Viskanta, 1985). (c) A cavity receiver for a surround heliostats field (Ho and Iverson, 2014). (d) A cavity receiver enclosing an indirectly heated gravity-driven flow suitable for either a surround or polar heliostats field (Ho and Iverson, 2014).

The free-falling liquid metal curtain (Figure 3.42) was initially considered to facilitate a compact cavity design; however, the shallow cavity depth may increases the thermal and optical losses, as will be explained in the next chapter. The main problem of this configuration for dense liquid metals is the excessive flow instability under gravitational acceleration. Computational fluid dynamics simulations were performed following the methodology which will be described in Chapter 6. The
resulted film was found to disintegrate into thinner shreds, as shown in Figure 3.43, with terminal shreds separating from the primary flow and splashes inside the cavity, leading to ineffective heat transfer and increased risk of corrosion and aperture blockage if a window is used to seal the aperture. Therefore, this flow configuration was discarded in this study. Using a solid secondary reflector inside the cavity was also considered; however, due to technical concerns of the beam-down configuration discussed in section 2.4.1, the use of a solid secondary reflector was avoided. Some of the considered concepts for the flow configuration are presented in Appendix A2.

Figure 3.42. Free-falling liquid metal curtain concept. The arrows represent incoming solar beams from the heliostat field and entering through the cavity’s circular aperture.

Figure 3.43. Volume fraction of the liquid tin curtain past the inlet slit at the receiver ceiling. The change in colour from red to yellow/green indicate thinning of the curtain, so that the mesh resolution across the thickness can no longer detect pure liquid tin at subsequent nodes, hence assuming a mixture of liquid tin and cavity fluid. The dark blue regions indicate absolute absence of liquid tin, which reflect the physical breakdown of the flow.
3.6. Summary

The main design characteristics of the prospective ultra-high temperature solar receiver are outlined to guarantee thermally efficient and reliable operation. The solar collection is recommended to be processed inside a cavity bounded by opaque walls to minimise thermal losses. Secondary optics are also found essential to minimise the radiative and thermal losses from the cavity aperture. Direct solar absorption by the HTF is established as a key solution to the reported thermomechanical and chemical limitations associated with the use of intermediate solid absorbers. Possible flow configurations for the optically exposed liquids inside the cavity receiver were reviewed while highlighting the limitations of each. Nevertheless, it was found that there is no material suitable for operation as a directly irradiated HTF and as a thermal storage medium, as the former requires different physical properties than the latter. Challenges associated with the operation at ultra-high temperatures were highlighted for different candidate HTFs. Liquid metals emerged as suitable HTFs for ultra-high temperature applications based on their suitable thermo-physical properties and existing base of practical knowledge of their use in other industrial applications. Liquid metals have been used for decades in the nuclear power sector as coolants at temperature exceeding 1000 K. Furthermore, liquid tin has been used for decades in glass manufacturing at ultra-high temperatures (>1300 K) to utilise its favourable thermo-physical properties, such as low melting point and high density. As a result, various materials have been verified for pumping, piping, sealing, and containment of liquid tin at ultra-high temperatures. In the following chapter, a novel receiver will be proposed based on the considerations brought from this chapter.
Chapter 4.  Design of a Novel Solar Receiver

A novel class of receivers can be proposed based on the identified central features presented in the previous Chapter. The receiver is described as ‘a cavity receiver with a compound parabolic concentrator at its aperture, which encloses a directly irradiated flow of a liquid metal HTF’. Molten tin is selected in this research to demonstrate the technical potential of the proposed receiver concept, based on the acquired practical knowledge and recent advancements in its pumping and containment, as discussed in section 3.3.4.3. In this Chapter, the design features will be described in detail for a 20 MW$_{\text{elec}}$ receiver system, which is based on the current largest commercial power plant operating with a cavity receiver. Specifications of PS20 are outlined in Table 5. The proposed receiver will be designed at this scale to heat molten tin from 800 K$^6$ to 1673 K, which can then either be used to run an ‘F-class’ CCGT cycle for power production (an example will be demonstrated in Chapter 8) or supply an equivalent thermal process. In practice, the lower temperature should be set with a reasonable tolerance above the freezing temperature of the liquid metal (molten tin freezes at 505 K) to avoid solidification inside the feed pipes. Ideally, heating and melting the tin from room temperature to the cycle’s low temperature (800 K) should occur only once before the nominal operation. During the diurnal cycle, molten tin is pooled into a collector tank and preserved above the melting temperature following the anti-freezing measures currently employed in liquid-metal-based nuclear power plants (Frignani et al., 2019) and molten-sodium-based CSP plants (Deng et al., 2021, Kotzé et al., 2011). The potential for utilising the energy content of the protective cavity environment to maintain the liquidity of the molten metal HTF will be explored in Chapter 7.

Table 5. Main specifications of the reference CSP plant – PS20 (NREL, 2022a).

<table>
<thead>
<tr>
<th>Annual Solar Resource (kWhm$^{-2}$)</th>
<th>Number of Heliostats</th>
<th>Heliostat Aperture Area (m$^2$)</th>
<th>Tower Height (m)</th>
<th>Rated Power (MW$_{\text{elec}}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2012</td>
<td>1255</td>
<td>120</td>
<td>165</td>
<td>20</td>
</tr>
</tbody>
</table>

$^6$ A ‘cold’ reservoir, with supplementary heating, for liquid tin at 800 K need to be used to maintain the low temperature and liquidity of tin.
4.1. Design considerations for the liquid metal flow

Based on the qualitative analysis of possible flow configurations presented in section 3.5, an inclined film flow over the back surface of the cavity receiver is adopted to support the liquid metal HTF flow, as shown in Figure 4.1. This surface is required to be chemically compatible and corrosion resistant to the hot liquid metal HTF. For molten tin at ultra-high temperatures, graphite is nominated as the containment material based on the compatibility verifications presented in section 3.3.4.3.

The inlet of the liquid metal HTF is located at the deepest – distanced away from the aperture – spot inside the cavity, so that the flow will be inclined towards the aperture, as shown in Figure 4.2. The HTF collector can be placed above the inlet for better control over the flow entering the receiver; however, this will require more complex insulation measures, increase the pumping load, and compromise the compactness of the receiver, which is vital for the aimed high tower heights as emphasised previously in section 3.1. Accordingly, the HTF collector is to be situated at the ground level before the tower piping. Therefore, the HTF flow control here will be implemented via the ceramic circulating pump and valves. For molten tin at temperatures up to 1673 K, the designs and materials of the circulating pump and values verified by Amy et al. (2017) and Henry et al. (2018) and will be used here.
Figure 4.2. Labelled sectional view of the CAD model of the proposed receiver module. This module is to be installed on top of a tower with a corresponding height to its power rating.

The inclined film configuration enables control over the gravity-driven liquid metal flow via the inclination degree. A secondary advantage to inclining the exposed liquid metal flow inside the cavity is the reduced view factor between the exposed surface and the aperture, which minimises the re-radiative energy losses through reflection or emission – re-radiated energy will be more likely to strike the cavity walls. The inclination degree, nonetheless, has implications on the size of the receiver. As illustrated in Figure 4.3, inclining the flow by an angle $\emptyset$ will shrink the projected HTF area facing the incident solar beams by a factor of $\cos \emptyset$. To compensate the compressed projection area, the length of the HTF film will need to increase, hence the depth of the receiver will expand by $\tan \emptyset$ multiplied by the original (vertical) length of the exposed flow. Although depth expansion is favourable in terms of minimising the thermal losses from the cavity receiver (Clausing, 1983, Sinha et al., 2019), the receiver size is constrained by towering structural limitations. Moreover, inclining the can depreciate land utilisation and augment spillage of reflected beams aimed at the receiver aperture. As shown in Figure 4.3, flow inclination will increase the optical ‘dead zone’ in front of the tower by a distance that is function of the tall tower height. It should be noted here that the first row of heliostats should be displaced at a distance where their reflected beams interject the highest (deepest) spot of the exposed flow. Therefore, HTF flow inclination is constrained by these limitations; hence, another approach should be used along with inclination to control the HTF film flow dynamics.
For ease of manufacturing, an inclination of $30^\circ$ (to vertical) was used for the support surface to the HTF film. However, this inclination angle was insufficient to prevent film disintegration under gravitational acceleration as demonstrated in Figure 4.4(a). An inclination of at least $68^\circ$ was found necessary to alleviate the gravitational component and preserve the HTF film continuity. Such inclination degree would expand the receiver to an impractical size. Therefore, surface corrugations were used to dampen and control the gravity-driven flow along the inclined surface. This approach was successful to preserve the continuity of the film as displayed in Figure 4.4(b). Wavy liquid films are proven to deliver higher heat transfer coefficients due to the mixing action and increase heat transfer area (Faghri and Zhang, 2006). The impact of shape profile of the surface corrugations on the dynamics of the film flow will be studied and analysed in the following chapters.
Other approaches were considered to control the dynamics of the gravity-driven film liquid metal flow; however, flow over a corrugated surface was established as the most feasible approach. One approach was utilising the magnetic properties of liquid metals by creating an electromagnetic field to control the velocity of the HTF film flow. However, the author believes this approach would be cost and weight prohibitive without further advancements of electromagnetic materials.

Reversing the film flow, to flow upwards against gravity, was considered. Apart from keeping the hottest HTF region away from the infiltrating airflow, flowing against gravity will eliminate the film continuity challenges caused by gravitational acceleration. However, after preliminary simulations of the upward film flow configuration, another problem was materialised. Given the small flow rates required for the liquid metal film to reach ultra-high temperatures, the upward flow failed to reach the top outlet at different scenarios, limiting control over outlet temperature via HTF flow rate. Additionally, the flow velocity and, in turn, the film thickness was severely non-uniform along the inclined surface, as the flow was fast and thin near the inlet which slows and thickens on the way upwards. This non-uniformity has adverse implications on the film stability and absorption of solar radiation as will be demonstrated in the Chapter 7. It is also preferable to position the hot outlet near the
bottom, which enjoys a smaller view factor with the aperture than the top outlet. Furthermore, Spirkl et al. (1997) found that it is more efficient to flow the volumetric HTF against the direction of radiation propagation. Accordingly, the upward film flow configuration was disregarded.

4.2. Cavity aperture size and orientation

As the gateway for the reflected solar beams from heliostats to the solar receiver, the aperture plays a critical role in the performance of a cavity receiver. The location, orientation, shape, and sealing technique of the aperture can determine the magnitudes of optical and thermal losses from the receiver as well as the oxidation rates of metallic components, most importantly the liquid metal HTF.

As discussed previously in section 3.2, the incorporation of a secondary concentrator is essential to achieve and sustain ultra-high receiver temperatures. Hoffschmidt et al. (2012) and Vant-Hull (2021) both advised that the aperture plane needs to be tilted towards the heliostats field, by approximately 30° to horizontal, to account for the restricted acceptance angle of the attached CPC. Inclined apertures can also help minimising the wind induced convective losses from the cavity (Flesch et al., 2014, Flesch et al., 2015).

Circular apertures are generally preferred over rectangular or square ones to avoid stress concentrations at the corners. Given the large capacities, in turn numbers of heliostats, targeted by this research, the image of a prospective heliostats field is likely to be in form of circular Gaussian on the aperture plane. Furthermore, the cavity is likely to contain internally pressurised atmosphere to protect the molten metal from oxidation as discussed in section 3.4.3; therefore, the window will then be subjected to wind load and pressure difference across its sides. Accordingly, minimising sharp corners is crucial to sustain the structural integrity of the window.

The size of the aperture was determined based on the solar flux required to achieve the target HTF outlet temperature. It should be noted that the received solar flux by the cavity receiver is not the same amount as the flux impinged the heliostats reflecting surfaces, as there are significant incurred optical losses between the heliostats and the

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7 Justification for Gaussian flux distribution for the proposed receiver will be provided in Chapter 6.
mounted receiver, which will be discussed in details in Chapter 5. The optimal
diameter of the aperture was estimated based on the receiver temperature and incident
solar flux in accordance to a semi-empirical method developed by Steinfeld and
Schubnell (1993), which is based on the assumption of Gaussian distributed solar flux.
This method has been validated, using the Monte-Carlo ray tracing for ultra-high
receiver temperatures and solar fluxes up to 12,000 suns.

A model of the 20 MW$_{elec}$ tower receiver plant was built using Energy3D, which is
a simulation-based engineering tool developed by Xie et al. (2018) that can be used to
build renewable energy power plants to estimate their performance and assist in
designing their layouts. As displayed in Figure 4.5, the layout of heliostats were used
to help with designing the aperture orientation and evaluate the radiant energy input to
the elevated receiver. A built-in model of the PS20 plant was modified using Energy3D
tools to account for the optical limitations of the CPC attached to aperture as illustrated
in Figure 4.5(a). The modifications were proposed by Segal (2012) and they include
layouts for larger multi-towered plants, as demonstrated in Figure 4.5(b) and (c).

![Figure 4.5. Layout of the 20 MW$_{elec}$ heliostats field (based on PS20 power plant) modelled by Energy3D. The model is used to determine the displacements of the first and last heliostats rows from the 165 m tower based on their corresponding reflected solar beams at nominal (STC at solar noon) conditions. The bottom image displays the solar irradiance reflected by each heliostat at nominal conditions.](image)
Figure 4.6. Heliostats field layouts for (a) a single, (b) three, and (c) six tower receivers incorporating secondary concentrators (Segal, 2012).

Recalling the analysis presented in Chapter 2 (section 2.1.1), the diameter of the Sun subtends a half angle ($\beta_{\text{sun}}$) of 4.65 mrad at the Earth surface. Therefore, it is expected that Sun image after reflection by a heliostat to magnify, illustrated in Figure 4.7. Therefore, it is possible to evaluate the size of the Sun image ($d_{\text{image}}$) at a given distance away from the heliostat using:

$$d_{\text{image}} = 2Z\sin(\beta_{\text{sun}})$$

(14)

where $Z$ is the distance between the heliostat and the image. Please note that this distance is not the horizontal field distance. Using the data from the Energy3D power plant model, the furthest heliostat location on the field was evaluated as 500 m away from the tower, the distance $Z$ is 525 m. Therefore, the calculated $d_{\text{image}}$ is about 4.8 m. This is establishes the maximum size (diameter) for the cavity aperture.
Figure 4.7. Magnification of the Sun image with distance from the reflector. Please note that the subtended angle is exaggerated to display the effect.

Future work may involve an optimisation of the heliostat layout based on the optical properties of the proposed receiver. Preliminary layouts may follow the guidance of Pitz-Paal (2011) and Segal (2012) to consider the effect of the restricted CPC acceptance.

4.3. Aperture sealing with a transparent ceramic window

For cavity receivers operating at ultra-high temperatures and containing exposed liquid metal is likely to necessitate sealing from the relatively cold and corrosive ambient environment. Wind-induced forced convection can have detrimental effects on the performance of open-aperture cavity receivers depending on the cavity depth, size and orientation of the aperture (Clausing, 1983, Sinha et al., 2019). The influence of wind on the performance of an open-aperture cavity is studied in a virtual wind tunnel analysis presented in Appendix A3.
The exposed liquid metal film flow is subjected to oxidation and dynamic interferences from the ambient air. For molten tin, a slightly pressurised atmosphere of nitrogen and hydrogen mixture\(^8\) will also need to be maintained inside the cavity to protect the exposed HTF film from oxidation. Therefore, the aperture is required to be sealed to isolate the internal cavity conditions from the ambient.

In the literature, apertures of pressurised ultra-high temperature solar cavity receivers are commonly sealed with transparent glazing materials, typically hard crystalline minerals, such as quartz and sapphire (Karni et al., 1998, Maag et al., 2010, Hughes et al., 2016). These materials are spectrally selective, ideally, in a way to transmit the incoming solar radiation with shorter wavelengths, and reflect the internal re-radiation with longer wavelengths (Maag et al., 2010). Quartz is made from crystalline silica, which has a melting point of 1980 K, while sapphire is a mineral corundum composing aluminium oxide, which has a melting point of 2290 K. Nevertheless, quartz windows are not recommended to operate at temperatures >1073 K to avoid degradation by the recrystallization induced by the thermal cyclic nature of CSP applications (Röger et al., 2006). While sapphire can withstand cyclic operation at ultra-high temperatures without the requirement of active cooling, it has higher reflectance at the visible spectrum, which carries the peak of solar energy, hence resulting in lower receiver temperatures and energy conversion efficiency of the receiver (Maag et al., 2010).

As discussed previously in section 3.4.2, fluidic seals and aerodynamic windows are preferred over glazed windows for particle receivers. However, it should be noted that some of reported issues in particle receiver, such as particle collisions and depositions on the internal surface of windows, are not applicable on the proposed flow configuration of liquid metal. The main advantage of the fluidic seals here will be the higher transmittance of incoming solar beams (Ho et al., 2014). However, glazed windows offer spectral selectivity of radiation to minimise re-radiative losses, no risk of fluidic interaction with the dynamics of the HTF film flow, and more

\(^8\) A \(\text{N}_2\)-\(\text{H}_2\) mixture containing >5.5 mol\% of \(\text{H}_2\) is considered flammable according to Airgas (2021) Safety data sheet, See https://www.airgas.com/msds/002119.pdf (accessed 18-10-2022). Although \(\text{H}_2\) concentration inside the cavity is above that limit, its concentration is likely to drop significantly below it in case of a leakage to the atmosphere. There is an existing base of practical experience in sealing this type of mixture from the ambient in the glass manufacturing industry.
protection against infiltrating external contaminants and oxidants (Tan et al., 2009, Hughes et al., 2016, Viegas et al., 2018). Additionally, fluidic seals require complex active control over their jet speed and direction based on the wind speed and attacking angles to avoid various potential failure modes, such as the mode displayed in Figure 4.8. The use of fluidic seals in cavity receivers is still underdeveloped but, with improved control, they might become a viable and more efficient alternative to glazed windows (Nathan et al., 2018, Calderón et al., 2018).

Transparent ceramics, which are most commonly used in optical windows for lasers, can offer a reliable solution to sealing the cavity aperture at ultra-high temperatures without the requirement of active cooling. Generally, transparent ceramics are characterised by their high optical transparencies coupled with high mechanical strengths, melting points, thermal shock resistances, and chemical stabilities (Kong et al., 2015).

In addition to the previously discussed sapphire, potential crystalline transparent ceramic candidates for the aperture window include magnesium aluminate spinel (MgAl₂O₄), aluminium oxynitride (Al₂₃O₂₇N₅, commonly abbreviated as AlON), and yttrium(III) oxide (Y₂O₃). The use of transparent ceramics in solar applications are not unprecedented. Erickson and Gavilan (2016) designed a tubular solar receiver with each tube enclosed within a concentric protective shell made from the mentioned candidate crystalline transparent ceramics, which was claimed to increase the electromagnetic to thermal energy conversion in the receiver by 12.5%. Furthermore, a transparent ceramic, yttrium aluminate (Y₃Al₅O₁₂), was found effective in converting near-infrared light into green light, where a dye-sensitised solar cell typically holds a high spectral response, hence, improving its efficiency (Liu et al., 2011). Yttrium aluminate was also found to deliver feasible and promising performance for solar-pumped solid-state lasers in aerospace applications, where it is required to operate under severe space environment, including ultra-high temperatures (Li et al., 2022).
Figure 4.8. Numerical analysis of a cavity aperture with an aerodynamic window demonstrating a failure mode of this sealing approach (Tan et al., 2009). Velocity vectors are displayed here for the case with a wind speed of 9 ms$^{-1}$, an attacking angle of 45°, and air jet speed of 8 ms$^{-1}$. The vectors are displayed on planes displaced by 0.5 m from the centre YZ plane (a) to the left and (b) right sides of the aperture.
In this study, magnesium aluminate spinel is selected to be the glazing material for the window. Spinel is a polycrystalline ceramic with superior mechanical and optical properties compared to other glazing materials. Compared with the multispectral grade of the polycrystalline Zinc Sulfide (Cleartran™), spinel is ten times harder with three times higher compressive strength and two times higher the fracture toughness (Sanghera et al., 2015). Spinel is also characterised by its superior transmission across the visible and mid-infrared ranges as demonstrated in Figure 4.9. Spinel is also characterised by its long-term resistance to chemical corrosion at temperatures up to 2473 K, high thermal shock resistance, and low coefficient of thermal expansion (Ghosh et al., 2015, Baudín et al., 1995, Bakker and Lindsay, 1967). Moreover, the mechanical properties and creep resistance at ultra-high temperatures can be considerably increased by doping with 2 wt% Y₂O₃ (Liu et al., 2019a), while the fracture strength can be improved by the application of a thin layer of silica sol–gel coating, which was reported to increase the strength by 20% and 30% at coating thicknesses of 32 nm and 72 nm, respectively (Tokariev et al., 2013).

![Figure 4.9](image)

**Figure 4.9.** The optical transmission superiority of spinel compared to (a) silica glass (Sanghera et al., 2011) and (b) other crystalline transparent ceramics (Sanghera et al., 2015).

The main limitation of using a spinel window is the high costs associated with its hot pressing and finishing. However, fabrication costs can be reduced by replacing the conventional hot pressing with a microwave sintering (Sanghera et al., 2015) or by polishing the spinel directly by the hot press to eliminate the finishing costs (Villalobos et al., 2012). Furthermore, manufacturing techniques for large-scale spinel windows and domes have been developed by the Materials and Electrochemical Research (MER) Corporation and the U.S. Naval Research Laboratory (NRL), which were
proven for sizes as large as 60 inches (1.52 m) with plans for further upscaling and manufacturing of “more complex 3-D shapes like spinel superdomes and tubes” (Goldman et al., 2017, Sepulveda et al., 2011, Sepulveda et al., 2013). Further improvements are anticipated in the future to optimise and enhance the sintering strategies for spinel (Shi et al., 2020).

The thickness of the window is a crucial factor determining the optical efficiency of the receiver. Ideally, the window should be thin enough to minimise the attenuation of the incoming solar radiation by reflection and absorption. However, it also needs to be thick enough to structurally sustain the wind load and thermal stresses induced by the 4-10% absorption of input concentrated solar radiation, in addition to the escaping radiation, through internal emission and reflection, incident on the internal surface of the window. Bopche and Kumar (2019) found that a glazing thickness of 4 mm delivers the optimum compromise between efficiency and structural reliability; however, this study considered cavity receiver at 453 K. For operation at ultra-high temperatures, quartz glass will need to be >10 mm to withstand the incurred thermal stresses, which can be prohibitive in terms of optical efficiency (Ambrosetti and Good, 2019, Becker et al., 2014). The superior mechanical and thermal endurance of spinel at ultra-high temperatures can compensate its slightly lower optical transmittance, across the visible spectrum, than quartz glass by reducing the glazing thickness. Additionally, the window can be protected from absorbing the internally re-radiated energy by using the developed ultra-high temperature infrared-reflective coating by Röger et al. (2009), which protects the window from radiation emitted at wavelengths equivalent to the range of temperatures inside the receiver. The application of this coating has reportedly reduced the window temperature by 174-184 K and receiver losses by 11% for a cavity receiver at 1373 K exposed to 570 suns (Röger et al., 2009).

Dust and depositions on the surfaces of windows present a practical challenge for glazed windows, as regular cleaning and maintenance can be costly for elevated tower receivers. However, there is an ongoing research effort to develop anti-soiling coatings for glazing materials used in various solar applications (Dahlioui et al., 2022, Huang et al., 2021, Ilse et al., 2019, Quan and Zhang, 2017, Zhang et al., 2019). Anti-soiling measures were projected to result in a conceivable gain in annual performance for CSP systems by 3% (Lorenz et al., 2014).
4.4. Cavity walls configuration and design

Cavity receivers are typically cylindrical or cuboidal. Although cylindrical cavities are easier to manufacture, cuboidal cavity receivers tend to trap secondary reflections and radiative emissions at their corners as explained by Lakshmipathy et al. (2020). In addition to this thermal advantage, cuboidal cavity receivers were found also to deliver higher optical performance, in terms of annual absorptance of reflected beams from the heliostats, than their cylindrical counterparts (Arrif et al., 2021).

Cavity walls, excluding the inclined surface upon which the liquid metal will flow, can play two roles in the solar energy collection process. The first configuration is a ‘reflective cavity’, where walls are to be plated with a highly reflective material to trap the collected solar beams inside the cavity by secondary reflection as illustrated in Figure 4.10. The hypothesis of this configuration is that the trapped beams will eventually be predominantly absorbed by the exposed liquid metal flow, assuming that the energy escaped through the aperture and energy absorbed by cavity walls are minimal. The lining material of the walls is likely to be metallic to withstand the thermal stresses at ultra-high temperatures. Porous yttria-stabilised zirconia, which is commonly used as a thermal coating in combustion chambers and, potentially, for turbomachinery blades, are not recommended as a reflective lining material here, as its reflectance was found to be significantly temperature-dependent close to the Sun’s peak wavelength (Nychka et al., 2006). The main candidate lining material for the reflective walls is silver, as it is the only material that can maintain an optical reflectance >0.95 at 1300 K across a wide range of visible and infrared wavelengths (Ujihara, 1972), while it is lower in cost than the noble reflective metals, such as gold and platinum group metals.
Figure 4.10. A simplified sketch showing the hypothetical entrapment and dissipation of a solar beam inside the cavity via multiple instances of secondary reflections and absorption.

The second configuration is the ‘absorptive cavity’, which allows the cavity walls to participate in the solar absorption process by preheating the liquid metal in a tubular radiant heat exchanger before starting the exposed film flow phase as demonstrated in Figure 4.11. This is can be considered as a hybrid of the reflective cavity configuration and the conventional tubular receivers. The premise of this configuration is that the fraction of energy absorbed by cavity walls can be significant due to the limited availability of highly reflective materials at ultra-high temperatures – it is most likely that the cavity walls will need to be lined with a layer of a costly noble metal, such as platinum or silver, to withstand the reduced reflectance of metals at ultra-high temperatures as discussed previously in section 3.4.3. This energy can then be utilised to improve the efficiency of the system and provide an active cooling for the walls as well. The concern over the development of hot spots on the wall absorber is less severe compared to conventional tubular receivers for three reasons:

1. Exposure to lower radiation. The tubular solar absorber, attached to cavity walls, is not directly exposed to the original concentrated solar beam, as it will be dampened first by the fractional absorption of the liquid metal film;
2. Radiation divided over a larger absorber area. Secondary reflections strike a larger absorbing area than the area receiving the original concentrated solar beam;
3. Smaller view factor with the aperture. Even in the case of hot spots development on the cavity wall surface, the associated radiant emissions will be more likely to strike the surface of the liquid metal film than escaping through the aperture.
The radiant energy absorbed by the cavity walls, in either configuration, can be transmitted through the wall thickness by conduction before being carried away from the external surface of the receiver by wind-induced forced convection. This energy loss mechanism is generally considered insignificant compared to radiative and convective losses at ultra-high temperatures (Becker and Vant-Hull, 1991, Harris and Lenz, 1985). The conductive loss can easily represent <1% of total energy losses from a cavity receiver with a suitable insulation layer (Hogan, 1994).

Figure 4.11. Simplified schematics illustrating the pathway of the liquid metal in the two studied cavity configurations: (a) reflective cavity; (b) absorptive cavity.

In the sensitivity study carried out by DeAngelis et al. (2018) of the 1623 K liquid metal based tubular receiver, a threshold for the insulation’s thermal conductivity of ∼0.1 Wm⁻¹K⁻¹ was found, beyond which the conductive losses through the walls can become a dominant loss mode. In the same study, a maximum threshold of ∼100
Wm\(^{-1}\)K\(^{-1}\) was also found for the thermal conductivity of tubular absorber material, as shown in Figure 4.12, beyond which it has minimal effect on improving the energy efficiency of the receiver. The thermal conductivity of graphite ranges between 25-470 Wm\(^{-1}\)K\(^{-1}\) depending on the grade (Mokhena et al., 2018); therefore, graphite can be used as the base material for the cavity wall absorber in the absorptive configuration. Graphite is also combines high thermal stability, exceptional corrosion resistance and mechanical properties (Ye et al., 2006), in addition to its compatibility with molten tin as discussed previously in section 3.3.4.3. Traditionally, synthetic graphite is traditionally formed by melting pure silica sand and fine coke in a furnace following the Acheson process (Cardarelli, 2018); however, more favourable thermo-physical properties, to solar energy applications, can be obtained by hot isostatic pressing (Olmec, 2020).

![Figure 4.12. Effect of thermal conductivity of insulation and tubular receiver on the efficiency of a liquid metal tubular receiver at 1623 K (DeAngelis et al., 2018).](image)

Based on the findings of the sensitivity study, cavity walls, in both configurations, were designed to deliver effective thermal conductivities <0.1 Wm\(^{-1}\)K\(^{-1}\). Alumina-silica (Al\(_2\)O\(_3\)-SiO\(_2\)) ceramic fibre based insulation is often used in high temperature applications, as it can operate at temperatures up to 1533 K; however, its thermal conductivity deteriorates substantially with temperatures, reaching 0.13 Wm\(^{-1}\)K\(^{-1}\) and 0.40 Wm\(^{-1}\)K\(^{-1}\) at 811 K and 1366 K, respectively (Headley et al., 2019, ZIRCAR, 2022). Recently, evacuated rectangular honeycomb structures, with thin metallic (10 μm stainless steel or 50 μm titanium alloy) walls, were found to outperform existing thermal insulations by delivering thermal conductivities <0.01 Wm\(^{-1}\)K\(^{-1}\) at 1600 K; however, manufacturing processes for such thin-walled structures are yet to be
developed (Desguers and Robinson, 2022). At present, microporous insulations can be a light and feasible solution for cavity wall insulation. Microporous insulations are composites made from small particles of inorganic oxides (usually silica), which result in narrow pores that restrict the thermal pathways within the material, and can also be equipped with thermally stable opacifiers to minimise the radiative heat transfer by scattering infrared radiation (Elmelin, 2019). For example, Excelfrax® microporous insulation is made from 80 wt.% silica, 15 wt.% silicon carbide, in addition to other supplementary materials, which can maintain a thermal conductivity of $0.030 \text{ Wm}^{-1}\text{K}^{-1}$ and $0.038 \text{ Wm}^{-1}\text{K}^{-1}$ at 873 K and 1073 K, respectively (Unifrax, 2018). Therefore, the reflective cavity wall, which aimed to be maintained at 800 K, can use this insulation material solely between the reflective lining and the stainless steel (grade 347H) shell as displayed in Figure 4.13. Coupling low thermal conductivity with low density ($230 \text{ kgm}^{-3}$) of the selected microporous insulation reinforces the compactness of the receiver design.

![Figure 4.13. Structure of the reflective cavity wall without active cooling. The thicknesses of the microporous insulation and stainless steel shell are 540 mm and 10 mm, respectively.](image)

The absorptive cavity walls, which will likely operate at temperatures >1600 K, cannot employ the microporous insulation on its own, as they are not recommended to operate for prolonged durations (>12h) at temperatures exceeding 1273 K (Elmelin, 2019, Unifrax, 2018). Therefore, another insulation material need to be sandwiched between the graphite interior and the microporous insulation, as illustrated in Figure 4.14, to cover the temperature range 1273-1623 K. Zirconium oxide fibre composite (85 wt.% zirconium dioxide, 10 wt.% ytrrium(III) oxide, and 5 wt.% silica) offers a light (density of $480 \text{ kgm}^{-3}$) and compatible material with graphite, which can maintain
its dimensional stability at temperatures up to 1923 K with thermal conductivities ranging between 0.125 Wm⁻¹K⁻¹ and 0.240 Wm⁻¹K⁻¹ at temperatures 1223 K up to 1923 K, respectively (Final-Advanced-Materials, 2021). This selected composite is commonly used in the insulation of induction and electric furnaces, as it is resistant to corrosion by molten metals and other oxide materials at ultra-high temperatures (Final-Advanced-Materials, 2021).

![Figure 4.14. Structure of the absorptive cavity wall displaying layers of graphite (210 mm), zirconium oxide fibres (180 mm), microporous insulation (150 mm), and stainless steel chassis (10 mm).](image)

To enhance the absorptance of the graphite tubular absorber at the internal surface of the wall, a compatible and durable absorptive coating will be applied on its interior surface. Recently, an absorptive coating made from iron–cobalt–chromite black spinel was found to be durable compatible with refractory materials, and was shown to deliver solar absorptance of 0.877–0.894 for 400 hours at 1523 K, followed by 200 hours at 1573 K (Wang et al., 2021b).

The reflective walls would also require active cooling to discharge the fraction of energy absorbed by their reflective lining, as other energy discharge mechanisms – forced convection from the ambient side, natural convection from the cavity side, and radiative emission – can be insufficient to facilitate thermodynamic equilibrium at the desired wall temperature. Unlike in the absorptive walls, the pipes/ducts will be imbedded inside the reflective walls to sustain the specularity of the reflective surfaces.
Active cooling can be performed either by the liquid metal HTF or by a secondary liquid or gas coolant as demonstrated in Figure 4.15. The first option might necessitate extra measures to guarantee leakage and corrosion proof pathways through the cavity walls for the liquid metal flow. However, given that the reflective cavity walls need to be maintained at relatively low temperatures, liquid metal coolants might be limited by their high freezing temperatures. An alternative approach is to use a secondary coolant with a lower freezing point and higher heat capacity. With a specific heat capacity of 4182 Jkg\(^{-1}\)K\(^{-1}\), liquid water at 293 K can extract >300 kW per 1 kg/s from the walls without changing phase. However, this energy will not then be transferred to the liquid metal circuit at high temperatures. Cooling of cavity walls using different coolants, and without, will be numerically investigated in Chapter 7.

The cavity fluid, which is a mixture gas (90 wt% nitrogen and 10 wt% hydrogen) might also serve as a feasible coolant, which additionally enables active control over the protective atmosphere in the cavity. The effective specific heat capacity of the cavity fluid is 2366.4 Jkg\(^{-1}\)K\(^{-1}\) (nitrogen is 1040 Jkg\(^{-1}\)K\(^{-1}\), hydrogen is 14,304 Jkg\(^{-1}\)K\(^{-1}\)), which is >7 times greater than liquid tin. The high specific heat capacity indicates that the increase in the cavity fluid temperature would be small (<140 K) enough to facilitate a thermodynamic equilibrium at a temperature below the melting point of the reflective silver lining. The effective thermal conductivity of the cavity fluid is 0.101 Wm\(^{-1}\)K\(^{-1}\) (nitrogen is 0.062 Wm\(^{-1}\)K\(^{-1}\) and hydrogen is 0.449 Wm\(^{-1}\)K\(^{-1}\)), which is >550 times more insulating than liquid tin.

Figure 4.15. Structures of actively cooled reflective walls by a (a) liquid or (b) gas coolant.
For higher temperature applications with wall temperatures beyond silver’s melting point (1234.8 K), cooling with the liquid tin will become feasible; hence, a wall design similar to the absorptive wall could be used as illustrated in Figure 4.16. The reflective lining material can be zirconium-platinum (ZrPt$_3$), which was initially proposed as a protective thermal coating for graphite, as it can maintain high infrared reflectance (>0.8) at ultra-high temperatures up to 2463 K (Alvey and George, 1991). The melting point of ZrPt$_3$ is 2523 K (Pan et al., 2014).

![Figure 4.16. An alternative wall structure for reflective walls at temperature >1200 K using graphite-compatible materials and liquid tin as the coolant.](image)

Regarding the manufacturability of the receiver from the proposed refractory materials, graphite, SiC, and mullite are commercially available in various forms and shapes, with reported prices for 1 in outer diameter (0.75 in inner diameter) of £102.4/m and £137.6/m for graphite and mullite tubes (Zhang et al., 2018b). Manufacturing the curved ‘forehead’ section of the receiver might be challenging given the high brittleness of ceramics. Nonetheless, fabrication of curved ceramic structure, such as SiC, were demonstrated using rapid prototyping and additive manufacturing techniques (Klosterman et al., 1999, Lakhdar et al., 2021). Additive manufacturing is also proven viable for producing complex shapes with graphite (Mohammadi et al., 2022).
At present, there is also an ongoing investigation on developing a fabrication technique for curved graphite structures in the University of Edinburgh. Nevertheless, while optimising the design is still to be implemented, the curved ‘forehead’ section of the wall may not be included in the later versions of the receiver. A trapezoidal cavity receiver is a potential option, which carries several benefits as will be discussed in Chapter 7. The insulation materials are also widely available in various shapes from global vendors, such as DBW Advanced Fiber Technologies (Germany), Final Advanced Materials (France), Hebei Suoyi New Material Technology (China), Savextherm (India), Stanford Advanced Material (USA), and Unifrax (UK).

The external dimensions of the receiver module are demonstrated in Figure 4.17. The internal width of the cavity (4.8 m) was determined based on the size of the aperture (~10% larger than aperture diameter to account for the optical deviations of secondary concentration), while the height and depth of the cavity were determined based on the inclination angle of the film flow and heliostats layout as explained previously in section 4.2. The inlet slit size was specified as 20 mm based on the flow rate and residence time required for the liquid metal HTF to achieve the target outlet temperature. In the next chapter, a mathematical model of the receiver will be described, from which the design parameters were estimated.

Figure 4.17. External dimensions of the cavity receiver module at the studied scale (20 MW_{elec}). This module is to be mounted on a 165 m towering structure.
Chapter 5. Quasi-Steady-State Analytical Analysis

In this chapter, the proposed receiver design described in the previous chapter is modelled analytically with the purpose of using the model to evaluate and compare the receiver performance at the two specified configuration, in addition to investigating the effects of different design parameters on the energy efficiency of the receiver. The analytical model will compose a combination of sub-models from the literature and tailored expressions to the design. Verification of the model against a numerical solution will be presented in Chapter 7.

5.1. Problem definition

Recalling the aim of the proposed design, which is to increase the temperature of a liquid metal HTF from its inlet temperature (800 K) to a desired ultra-high outlet temperature (1673 K). The input energy to the receiver module, which is the domain of the analysis, is sourced from heliostats reflections of incident solar energy on the solar field as displayed in Figure 5.1. The pathway of the HTF will involve a direct solar absorption phase and, in the absorptive cavity case, an indirect solar heating phase. Accordingly, the domain will involve interactions between fluid flow dynamics, heat transfer, and optics.

Figure 5.1. A simplified diagram displaying the main features of domain of the analytical model (bordered by the dashed rectangle) and its links to adjacent components in the system.
5.2. Assumptions

The analytical model will describe the quasi-steady-state energy flow throughout the receiver when all boundaries are at a thermal equilibrium with the ambient. Accordingly, the analytical model will not cover the transient developments of the liquid metal film flow and buoyancy-driven cavity fluid before reaching the steady-state conditions. Instead, the HTF film will be approximated as a boundary with prescribed optical and thermal properties.

All radiation instances, including emissive and reflective, are assumed diffuse. This assumption allows the use of area ratios and view factors to distribute the energy transfer on boundaries accordingly. The diffuse radiation assumption has been verified for graphite surfaces by (Wang et al., 2014).

About 99% of the energy in solar radiation is contained within the wavelength region from 150 nm to 4000 nm (Bhatia, 2014). This wavelength band is entirely lying within the transmitting range of the selected spinel material used for window (see Figure 4.9) with <2% deviations around the mean transmittance (~0.86) of the range. Consequently, the grey radiation assumption was employed to simplify the analysis. Nevertheless, the grey radiation assumption might not be valid for other glazing materials, including quartz glass (Sanghera et al., 2011, Zhang et al., 2018a). Accordingly, the analysis for other glazing materials would necessitate a non-grey (Khoukhi et al., 2003) or dual-banded grey (Craig et al., 2014) radiation model to evaluate the spectral effects of the window. In this analysis, the window material will be modelled without any enhanced optical features, such as spectral selectivity or internal infrared reflection.

The optical properties of the receiver materials were evaluated at the peak solar wavelength, which carries the highest radiation intensity (represented by the area under Planck’s curve at the solar temperature in Figure 5.2). Accordingly, the solar peak wavelength ($\lambda_{\text{sol,peak}}$) is calculated using Wein’s displacement law as follows:

$$\lambda_{\text{sol,peak}} = \frac{2898}{T_{\text{sun,bd}}}$$  \hspace{1cm} (15)
where $T_{\text{sun, bd}}$ is the solar blackbody reservoir temperature, which is treated here as a blackbody thermal reservoir at 5200 K, which is lower than the surface temperature of the Sun (5778 K) to account for atmospheric attenuation to solar beams (Lovegrove and Pye, 2012). Accordingly, the reference wavelength was evaluated as 0.5573 μm.

As discussed in section 4.4, at ultra-high temperatures, radiation and convection dominate as loss mechanisms over conduction unless the effective thermal conductivity of the walls exceeds 0.1 W.m$^{-1}$.K$^{-1}$ (DeAngelis et al., 2018). Accordingly, the conductive loss was ignored as the effective thermal conductivities of the reflective and absorptive cavity walls were evaluated as 0.031 W.m$^{-1}$.K$^{-1}$ and 0.092 W.m$^{-1}$.K$^{-1}$, respectively. The effective thermal conductivities ($k_{\text{eff}}$) were evaluated based on the insulation layers described in section 4.4 using:

$$k_{\text{eff}} = \sum_{i=1}^{i=z} \frac{x_i}{k_i}$$

where $z$ is number of insulation layers, $x_i$ and $k$ are the thickness and thermal conductivity of insulation $i$.

To confirm the insignificance of the conductive loss ($Q_{\text{conduction}}$), it was quantified using a simplified steady-state one-dimensional thermal resistance model as follows:

$$Q_{\text{conduction}} = \Delta T_w/R_{\text{eff}}$$

Figure 5.2. Planck’s curves (blackbody thermal emissions at designated radiation temperatures) with curve peaks highlighted to demonstrate the Wien displacement law (Colose, 2010). Please note that the Sun here is assumed as a blackbody source at 6000 K.
where $\Delta T_w$ is temperature difference across the wall and $R_{eff}$ is resultant thermal resistance evaluated based on the thermal circuit equivalence model. The resulted conductive losses were 12.055 kW and 96.579 kW in the reflective and absorptive receiver, respectively. These loss quantities represent <0.1% (reflective cavity) and ~0.11% (absorptive cavity) of input energy to the receiver, which are considered negligible in both cases.

The isothermal cavity walls were prescribed constant temperatures of 800 K and 1623.6 K for the reflective and absorptive cavities, respectively. The former prescribed temperature was based on the lowest (inlet) temperature of the HTF to the receiver, while the latter was prescribed 10% higher than the desired outlet temperature from the preheat phase. In practice, the reflective walls temperature should be kept as low as possible to maintain the reflectance of the metallic coating and minimise emissive losses, while the absorptive walls should be maintained at temperatures higher than the preheating liquid metal flow to account for the effectiveness of the wall preheater heat exchanger. The uniformity of temperature distribution along the wall will be numerically investigated using different wall insulations in Chapter 7. The emissivity of the reflective walls was prescribed as 0.1, assuming an effective reflectance of the metallic wall coating of 0.9, while the emissivity for the absorptive walls was 0.8 based on graphite’s approximate emissivity (Thorn and Simpson, 1953).

According to the data found in Assael et al. (2010) and Khvan et al. (2019), most the variations in thermo-physical properties of liquid tin at the studied range (800-1800 K) were found to be <5%. Accordingly, the properties in the analytical model were evaluated at the mean flow temperature of molten tin (1236.5 K). The dynamic viscosity and thermal conductivity were the two properties that vary most significantly (>30%) with temperature; however, their variations with temperature are linear in the studied temperature range (Assael et al., 2010, Assael et al., 2017, Giordanengo et al., 1999). Therefore, assuming their property values at the mean flow temperature would still be representative. Variations of thermos-physical properties of liquid tin with temperature are presented in Appendix A1.

Given that the system is analysed at its thermal equilibrium state, the optical properties of receiver materials are represented solely by their emissivities in
accordance with Kirchhoff’s law of thermal radiation. Similar to the thermo-physical properties, the variation of the optical absorption of tin with temperature is linear in the liquid phase as demonstrated by Figure 5.3. Therefore, the use of an emissivity value (0.2289) evaluated at the mean flow temperature is justified.

![Figure 5.3](image)

Figure 5.3. Variation of optical absorption of tin (solid and liquid) with temperature (Greenstein, 1989). (a) At a visible wavelength ($\lambda = 0.532$ μm). (b) At an infrared wavelength ($\lambda = 1.06$ μm).

The cavity fluid is modelled here as an incompressible ideal gas mixture of 9:1 N$_2$ and H$_2$ gases, respectively. The incompressibility assumption is justified by the cavity fluid’s small Mach number (buoyancy-driven flow in a sealed enclosure) and trivial pressure variations, compared to the temperature variations, across the cavity.
5.3. Actual receiver efficiency

The ideal receiver efficiency was defined in section 2.1 in terms of input solar irradiation, geometric concentration ratio, and receiver temperature. In reality, there are various types of energy loss mechanisms, which would reduce the actual energy efficiency of the receiver ($\eta_{rc}$). Therefore, Equation (2) should be modified to account for the optical and thermal losses incurred between the incident solar energy on the field and the collected thermal energy by the HTF as follow:

$$\eta_{rc} = \frac{\dot{Q}_{abs}}{P_{sol}} \quad (18)$$

where $\dot{Q}_{abs}$ is absorbed energy carried by the HTF and $P_{sol}$ is rate of concentrated solar radiation impinging the external surface of the aperture window with an area $A_{ap}$.

5.4. External optical losses

The radiative transfer between the sun and the receiver can be modelled using the laws of geometrical ray optics (Modest and Mazumder, 2022). Accordingly, the total spectral $P_{sol}$ quantity can be expressed as follows (Li et al., 2016):

$$P_{sol}(\vec{r}) = \int_{\lambda=0}^{\infty} \int_{\Omega=0}^{2\pi} I_{\lambda,rc}(\vec{r}, \lambda, \vec{s}) |\vec{\theta} \cdot \vec{n}| d\Omega dA d\lambda$$

$$= A_{ap} \int_{\Omega=0}^{2\pi} I_{rc}(\vec{r}, \vec{s}) |\vec{\theta} \cdot \vec{n}| d\Omega \bigg|_{\lambda=\lambda_{sol,peak}} \quad (19)$$

where $\Omega$ is solid angle and $I_{\lambda,rc}(\vec{r}, \lambda, \vec{s})$ is spectral rate of intensity from direction $\vec{s}$ and intercepted by an arbitrary area element $dA$ on the aperture plane with a position vector $\vec{r}$ and normal vector $\vec{n}$.

The $P_{sol}$ quantity can be evaluated from the available solar resource on the heliostats field $I_{\lambda,sol}(\vec{r}, \lambda, \vec{s})$ by accounting for the series of optical losses, demonstrated in Figure 5.4, which attenuate the solar beams reaching the receiver aperture as follows:
where $A_{helio}$ is total area of heliostats and $\eta_0$ is optical efficiency of the solar field, which is a resultant of the cosine ($\eta_{\cos}$), shading ($\eta_{shading}$), reflection ($\eta_{ref}$), blocking ($\eta_{blocking}$), atmospheric attenuation ($\eta_{atm}$), and spillage ($\eta_{spillage}$) efficiencies as follows:

$$\eta_0 = \eta_{\cos} \eta_{shading} \eta_{ref} \eta_{blocking} \eta_{atm} \eta_{spillage}$$  \hspace{1cm} (21)

Figure 5.4. Illustration of the optical losses incurred in a solar tower field.

The cosine loss is the reduction in reflecting area receiving the incoming solar beams with incidence not in-line with the reflector surface’s normal as displayed in Figure 5.5. Sun-tracking mechanisms are used to maintain the reflector surface’s normal at a bisecting orientation between the incident and reflecting beams. The shading and blocking losses occur due to the use of multiple heliostats in the field, where shading is the interception of solar beams reaching the reflector surface by other
heliostat(s), while blocking is the interception of reflected beams travelling towards the receiver by the back(s) of other heliostat(s). The reflection loss of the heliostat is to account for its <1 reflectance. The atmospheric attenuation is usually accounted for in large-scale solar fields, where dust and other particulates interact with the long-distance travelling beams towards the receiver by scattering and absorption. The spillage loss is when some of the reflected beams do not strike the aperture due to optical inaccuracy or enlargement of the reflected heliostats image (i.e. loss of optical concentration) beyond the receiver aperture. A schematic demonstration of the seven optical losses in a solar tower field is displayed in Figure 5.6. An optical assessment of these component optical efficiencies of a PS10-like plant was performed by Rinaldi et al. (2014) using DELSOL3 (Kistler, 1986), which is a computer code developed to evaluate the optical performance of solar tower systems. The estimated component efficiencies are tabulated in Table 6.

![Figure 5.5. A simplified schematic of an incident solar beam on a reflector to demonstrate the cosine effect (Haitao et al., 2017). The loss in this diagram is equals to 1-cosα.](image)

![Figure 5.6. A demonstration of the optical losses in a solar tower system (Falcone, 1986).](image)
Table 6. Estimated values for the optical efficiency and its component efficiencies for a PS10-like solar power plant (Rinaldi et al., 2014).

<table>
<thead>
<tr>
<th>Efficiency</th>
<th>Estimated Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \eta_{cos} )</td>
<td>84.4%</td>
</tr>
<tr>
<td>( \eta_{shading} )</td>
<td>96.6%</td>
</tr>
<tr>
<td>( \eta_{ref} )</td>
<td>88%</td>
</tr>
<tr>
<td>( \eta_{blocking} )</td>
<td>99.1%</td>
</tr>
<tr>
<td>( \eta_{atm} )</td>
<td>99.4%</td>
</tr>
<tr>
<td>( \eta_{spillage} )</td>
<td>95.5%</td>
</tr>
<tr>
<td>( \eta_o )</td>
<td>67.5%</td>
</tr>
</tbody>
</table>

From Equation (20), the solar flux reaching the receiver aperture (\( p_{sol} \)) can be expressed in terms of the optical properties of the field as follows:

\[
p_{sol} = \frac{P_{sol}}{A_{ap}} = \eta_o \frac{A_{hetia}}{A_{ap}} \left[ \int_{\Omega=0}^{2\pi} I_{sol}(\hat{r}, \hat{s}) |\hat{n}.\hat{s}|d\Omega \right]_{\lambda=\lambda_{sol,peak}} \\
= \eta_o CR_o \left[ \int_{\Omega=0}^{2\pi} I_{sol}(\hat{r}, \hat{s}) |\hat{n}.\hat{s}|d\Omega \right]_{\lambda=\lambda_{sol,peak}} \\
\approx \eta_o CR_o l \tag{22}
\]

where \( l \) is average incident solar flux on the field.

### 5.5. Receiver aperture

The input solar energy to cavity is subjected to the optical transmittance of the window (\( \tau_{win} \)). Therefore, the transmitted rate of energy to the cavity (\( \dot{Q}_{in} \)) is expressed as:

\[
\dot{Q}_{in} = \tau_{win} P_{sol} \tag{23}
\]

The volumetric attenuation of the concentrated beams through the window thickness will be analysed numerically using a radiation model as will be described in the next chapter. In the analytical model, this attenuation is modelled based on experimentally measured surface properties of the window material. Therefore, the \( \tau_{window} \) here is assumed constant and evaluated at \( \lambda = \lambda_{sol,peak} \) as 0.86 for a sample with the same thickness (Sanghera et al., 2011). The 0.14 aperture loss fraction is distributed on reflection (\( = 1 - \tau_{window} - \epsilon_{window} \)) and absorption (\( \sim \epsilon_{window} \)) losses by the window based on the emissivity of spinel (\( \epsilon_{window} \)), which was evaluated as 0.0765 (Harris et al., 2013, Sako et al., 2021).
5.6. Energy flow inside the receiver

The transmitted energy into the cavity will be absorbed by the HTF film and cavity walls, while the remaining energy will be lost through multiple thermal loss mechanisms. Accordingly, the $\dot{Q}_{in}$ can be expressed as follows:

$$\dot{Q}_{in} = \dot{Q}_{abs} + \sum \dot{Q}_{losses} = \dot{Q}_{abs} + \dot{Q}_e + \dot{Q}_{ref} + \dot{Q}_c + (1 - \eta_{rv})\dot{Q}_w$$

(24)

where $\dot{Q}_{abs}$ is rate of useful absorbed/collected energy, $\dot{Q}_e$ is emissive loss, $\dot{Q}_{ref}$ is reflective loss, $\dot{Q}_c$ is convective loss, $\dot{Q}_w$ is rate of total absorbed energy by cavity walls, and $\eta_{rv}$ is fraction of energy recovered from wall absorption. Note here that the $\dot{Q}_{abs}$ constitutes two components: direct absorption by the HTF and indirect heating through cavity walls. Therefore, the $\dot{Q}_{abs}$ can be expressed as:

$$\dot{Q}_{abs} = \dot{Q}_{abs, htf} + \eta_{rv}\dot{Q}_w$$

(25)

where $\dot{Q}_{abs, htf}$ is rate of directly absorbed energy by the HTF. The $\eta_{rv}$ factor depends on the applied cooling/preheating method and efficiency of cavity walls heat exchanger. For an uncooled – or cooled without energy recovery from the coolant to the HTF – reflective cavity, $\eta_{rv}$ will be zero and, hence, $\dot{Q}_{abs}$ will only constitute the direct absorption phase. In the case study presented here, no energy recovery is considered for the reflective cavity; however, the impact of energy recovery will be investigated. For the absorptive cavity case, $\eta_{rv}$ is prescribed at 0.9 based on reported efficiencies of tubular liquid metal (sodium) solar receivers (Sandia, 1983, Schiel and Geyer, 1988). An illustration of the rate of energy breakdown inside the receiver at each case is displayed in Figure 5.7.
5.7. Radiative analysis

At ultra-high temperatures, radiative losses dominate over other thermal modes. Radiative losses include two main components: reflective and emissive. The reflective component occurs since the cavity is not a blackbody absorber (i.e. the overall reflectance of its internal components is non-zero). Therefore, internal reflectance of the concentrated input beams will occur and a percentage of this reflected energy will escape the cavity through the aperture to the ambient. The emissive component occurs since the internal components of the receiver are at temperatures >0 K; hence, emissions will occur and a percentage of that energy will also eventually escape through the aperture.

5.7.1. Reflective loss

The reflective loss from a cavity receiver is conventionally expressed as follows:

$$\dot{Q}_{ref} = \hat{\tau}_{ref} \dot{Q}_{in}$$  \hspace{1cm} (26)

where $\hat{\tau}_{ref}$ is the fraction of input energy escaping the cavity through the aperture. The simplest model uses the reflectance of the absorber, or main receiver material, as $\hat{\tau}_{ref}$ (Kolb et al., 1989). However, this approach does not account for the geometry and structure of the cavity. Li et al. (2010) uses the area ratio ($A_{ap}/A_{cav}$) as a correction factor multiplied by the reflectance; however, this accounts only for the blockage of reflection instead of reflection exchange between different cavity surfaces. Ray tracing (Fang et al., 2011) can solve this issue; however, it is computationally expensive.
Here, $\tau_{ref}$ will be expressed in terms both geometrical and optical properties of the cavity as follows (Duffie et al., 1985, Zou et al., 2017):

$$\tau_{ref} = 1 - \frac{\epsilon_{cav}}{1 - (1 - \epsilon_{cav})(1 - A_{ap}/A_{cav})} \quad (27)$$

where $\epsilon_{cav}$ is the effective cavity emissivity, which is function of the emissivities of HTF ($\epsilon_{htf}$) and cavity wall lining ($\epsilon_w$). Utilising the diffuse radiation assumption, $\epsilon_{cav}$ can be calculated by breaking the internal cavity area ($A_{cav}$) into a finite number ($n$) of planar surfaces, each with a labelled index $i$ as follows:

$$\epsilon_{cav} = (1/A_{cav}) \sum_{i=1}^{n} \epsilon_i (A_i F_{i-ap})$$

$$= (1/A_{cav}) \left[ \epsilon_{htf} A_{htf} F_{htf-ap} + \epsilon_w \sum_{i=1}^{n-1} (A_i F_{i-ap}) \right] \quad (28)$$

where $A_{htf}$ is surface area of the HTF film, $F_{htf-ap}$ is the view factor from HTF film surface to the aperture, and $F_{i-ap}$ is view factor from surface $i$ to the aperture.

### 5.7.2. View factors evaluation

The view factors of each surface inside the cavity with the aperture are evaluated using a MATLAB algorithm (script of the used functions is presented in Appendix A4). The algorithm is based on Stokes’ theorem and employs the contour double integration method to calculate the view factors between arbitrary surfaces, while accounting for the shading of obstructions (Francisco et al., 2014). The coordinates of the surfaces were obtained from the 3D CAD model and supplied to the MATLAB algorithm to calculate their corresponding view factors with the aperture. The contour integral method was proven for different enclosures and provided more accurate results than conventional techniques, such as the area-integration method (Ambirajan and Venkateshan, 1993, Bopche and Sridharan, 2009, Rammohan Rao and Sastri, 1996).
5.7.3. Components absorption and secondary reflections

The main radiation absorbing components inside the cavity are the HTF film and cavity walls. The total power absorbed by each components ($\dot{Q}_{abs,j}$) constitutes of two parts as follows:

$$\dot{Q}_{abs,j} = \dot{Q}_{initial,j} + \dot{Q}_{secondary,j} \tag{29}$$

where $\dot{Q}_{initial,j}$ and $\dot{Q}_{secondary,j}$ are rates of energy absorption by component $j$ during the first instance of beams striking its surface and secondary instances, respectively.

The secondary reflections are an important feature of the cavity receiver, which determines the energy capturing performance of the receiver. In this context, secondary reflections are defined as the reflectance instances, which occur after the original input beams strike the HTF film and cavity walls once. Based on this definition, the rate of total energy absorption from secondary reflections ($\dot{Q}_{abs,ref}$) can be calculated as follows:

$$\dot{Q}_{abs,ref} = \sum \dot{Q}_{abs,j} - \sum \dot{Q}_{initial,j}$$

$$= [\dot{Q}_w + \dot{Q}_{abs,htf}] - [\epsilon_{htf} - x\epsilon_{htf} + \epsilon_w - \epsilon_w\epsilon_{htf} + x\epsilon_w\epsilon_{htf}]\dot{Q}_{in} \tag{30}$$

where $x$ is percentage of original solar beam that did not strike the HTF film first (i.e. beams which strike the side walls first).

Exploiting the diffuse radiation assumption, the $\dot{Q}_{abs,ref}$ can be distributed over the internal components (HTF film and cavity walls) using surface area and emissivity ratios. By applying this on Equation (29) for the two components, the total absorption by each component can be expressed as:

$$\dot{Q}_{abs,htf} = (1 - x)\epsilon_{htf}\dot{Q}_{in} + \left(\frac{\epsilon_{htf}A_{htf}}{\epsilon_{htf}A_{htf} + \epsilon_wA_w}\right)\dot{Q}_{abs,ref} \tag{31}$$

$$\dot{Q}_w = x\epsilon_w\dot{Q}_{in} + \epsilon_w(1 - \epsilon_{htf})(1 - x)\dot{Q}_{in} + \left(\frac{\epsilon_wA_w}{\epsilon_{htf}A_{htf} + \epsilon_wA_w}\right)\dot{Q}_{abs,ref} \tag{32}$$
5.7.4. Emissive loss

Given that the cavity walls in both configurations are isothermal, the emissive power loss from the receiver can be evaluated from Stefan-Boltzmann law as follows:

\[
\dot{Q}_e = \epsilon_{eff} \sigma A_{ap} (T_r^4 - T_a^4)
\]

where \(T_a\) is ambient temperature, \(T_r\) is cavity radiation temperature, \(\epsilon_{eff}\) is effective emissivity of an apertured cavity, which can be approximated in terms of the emissivity \((\epsilon_{abs})\) and area \((A_{abs})\) of all solar absorbing surface(s) according to Duffie et al. (1985):

\[
\epsilon_{eff} \approx \frac{\epsilon_{abs}}{1 - (1 - \epsilon_{abs})(1 - \frac{A_{ap}}{A_{abs}})}
\]

The representative radiation temperature \((T_r)\) of the cavity can be estimated by breaking down the internal cavity area into elemental planer surfaces, similarly to evaluating the representative emissivity of the cavity \((\epsilon_{cav})\) as follows:

\[
T_r = \frac{1}{A_{cav}} \sum_{i=1}^{n} T_i (A_i F_i_{-ap})
\]

\[
= (\frac{1}{A_{cav}}) \left[ T_{htf} A_{htf} F_{htf_{-ap}} + T_w \sum_{i=1}^{n-1} (A_i F_i_{-ap}) \right]
\]

where \(T_{htf}\) is the mean surface temperature of HTF film, \(T_w\) is cavity wall temperature, and \(F_{htf_{-ap}}\) is the view factor from the inclined flow surface to the aperture.

5.8. Convective loss

There are two modes for the convective heat loss from a cavity receiver: forced and natural convection. In the wind tunnel tests presented in section 4.3, it was shown that the external wind-induced flow has minimal convective effects on the proposed cavity design. Moreover, it was also decided that the aperture to be sealed from the ambient by a transparent ceramic window. Therefore, for an adequately insulated cavity with a
high temperature difference across its domain, it is expected that natural convection to
dominate the convective heat transfer in the receiver (wind-induced convection will
only be considered here as a cooling mechanism for the window). For a cavity without
aperture sealing, this assumption can also be valid if the Richardson number was >>1.
Richardson number \((Ri)\) is the ratio between natural convection (buoyancy term) and
forced convection (flow shear term) calculated as follows:

\[
Ri = \frac{\rho}{g} \left( \frac{\partial \rho}{\partial z} \right)^2 = \frac{g \delta (T_{\text{hot}} - T_{\text{ref}}) L_{ch}}{v_{ch}^2} = \frac{Gr}{Re^2}
\]  

(36)

where \(\rho\) is density, \(g\) is gravitational acceleration, \(z\) is domain depth, \(T_{\text{hot}}\) is hot wall
temperature, \(T_{\text{ref}}\) is reference temperature, \(\delta\) is thermal expansion coefficient (for
ideal gases: \(\delta = 1/T_{\text{ref}}\)), \(L_{ch}\) is characteristic length, \(v_{ch}\) is characteristic velocity, \(Gr\)
is Grashof number, and \(Re\) is Reynold’s number. \(Gr\) is a dimensionless ratio of
buoyancy to viscous force acting on the fluid, while \(Re\) is a dimensionless ratio of
inertial to viscous forces in the fluid. These dimensionless ratios are expressed as
follows:

\[
Gr = \frac{g \delta \rho^2 (T_{\text{boundary}} - T_{\text{bulk}}) L_{ch}^3}{\mu^2}
\]

\[
Re = \rho v_{ch} L_{ch} / \mu
\]

(37) (38)

where \(T_{\text{boundary}}\) is boundary temperature, \(T_{\text{bulk}}\) is bulk cavity fluid temperature, and
\(\mu\) is dynamic viscosity.

The convective heat transfer loss is expressed in terms of the geometrical and
thermal properties of the radiation absorbing surface(s) as follows:

\[
\dot{Q}_c = \sum_{i=1}^{n_{abs}} A_{abs,i} h_{abs,i} (T_{abs,i} - T_{cav})
\]

(39)

where \(T_{cav}\) is average cavity temperature, \(n_{abs}\) is number of absorbing surfaces, \(A_{abs,i}\),
\(h_{abs,i}\), and \(T_{abs,i}\) are surface area, convective heat transfer coefficient, and temperature
of the absorbing surface \(i\), respectively. Accordingly, \(\dot{Q}_c\) per configuration can be
calculated as follows:
\[
\dot{Q}_{c,\text{reflective}} = A_{htf} h_{htf} (T_{htf} - T_{cav})
\]  
(40)

\[
\dot{Q}_{c,\text{absorptive}} = A_{htf} h_{htf} (T_{htf} - T_{cav}) + A_w h_w (T_w - T_{cav})
\]  
(41)

where \(h_{htf}\) and \(h_w\) are convective heat transfer coefficients at the HTF film top surface and cavity walls surfaces, respectively. Using the area weighted average method, \(T_{cav}\) is calculated as:

\[
T_{cav} = (1/A_{cav})(T_{htf}A_{htf} + T_wA_w + T_{win}A_{ap})
\]  
(42)

where \(T_{win}\) is window temperature, which is estimated by applying energy balance on the window as will be demonstrated in section 5.10.

The natural convection transfer coefficient on a surface inside the cavity is given by:

\[
h = Nu \frac{k_{cav}}{L_{ch}}
\]  
(43)

where \(k_{cav}\) is thermal conductivity of the cavity fluid, \(L_{ch}\) is characteristic length, and \(Nu\) is Nusselt number. \(Nu\) of buoyancy-driven flows are expressed in terms of Rayleigh number (\(Ra\)), which is a dimensionless number resembling that characterises the flow regime. \(Ra\) is defined as:

\[
Ra = Gr Pr
\]  
(44)

where \(Pr\) is Prandtl number, which is a dimensionless property of the fluid defined as a ratio of the momentum diffusivity to thermal diffusivity as follows:

\[
Pr = \frac{c_p \mu}{k}
\]  
(45)

where \(c_p\) and \(k\) are specific heat capacity and thermal conductivity, respectively.

\(Nu\) is correlated to \(Ra\) through the following generic expression:

\[
Nu = c(Ra)^n
\]  
(46)
where $c$ is a constant describing the geometry and orientation of the hot surface(s), and $n$ is a constant describing the flow regime. Several values are available in the literature for different geometries, surfaces orientations, flow regimes, and fluids (Aydin and Guessous, 2001, Becker and Vant-Hull, 1991, Bennett, 1974, Tsuji and Nagano, 1988).

$Ra$ values are evaluated as $256 \times 10^9$ and $1.24 \times 10^7$ for the reflective and absorptive cavities, respectively. Krishnamurti (1973) defined the flow regimes of buoyancy-driven flows based on the fluid’s $Pr$ and $Ra$ as illustrated in Figure 5.8. Accordingly, the buoyancy-driven flow of the cavity fluid (assumed as N$_2$ gas, which has $Pr \sim 0.76$ at 1236.5 K) is considered fully turbulent at both cavity wall configurations.

![Figure 5.8. Flow regimes of buoyancy-driven flows with different Prandtl numbers (Krishnamurti, 1973).](image)

In a numerical analysis of turbulent ($Ra = 10^9$ to $10^{12}$) buoyancy-driven flows inside rectangular enclosures by Velusamy et al. (2001), the interaction effects between surface radiation and turbulent natural convection were found to diminish with the aspect ratio of tall rectangular enclosures as demonstrated in Figure 5.9. Therefore, given that the aspect ratio of the proposed receiver is $>$3, the variation between radiation coupled and uncoupled $Nu$ values will be $<$1%; hence, $Nu$ can be evaluated using a decoupled correlation.
Since no constants values, for Equation (46), were empirically generated for the proposed receiver design, approximate estimate values were generated using Catton (1978) correlation for a rectangular enclosure with a tilted surface, where $c = 0.18$ and $n = 0.29$. In Chapter 7, different correlations will be tested against the developed numerical solution to examine their applicability in this analysis.

For cavity receivers with open apertures, a survey of natural convection correlations were performed by Samanes et al. (2015) for PS10-like receivers. The survey recommends the correlation by Clausing (1983), as it has the advantage of accounting for the wind effects and orientation of each internal surface, and can be used to evaluate the heat transfer on each individual surface.

5.9. Hydrodynamics of the gravity-driven film

A simplified illustrative schematic of the HTF film flow is displayed in Figure 5.10 as guidance for this analysis. The outlet temperature of the directly irradiated HTF film ($T_{out}$) can be controlled via controlling its mass flow rate ($\dot{m}$) as:
\[ T_{out} = \frac{\dot{Q}_{abs,htf}}{\dot{m} \cdot c_p_{htf}} + T_{in} \]  (47)

where \( T_{in} \) is inlet HTF temperature to the receiver. Assuming maintained film continuity, the mass flow rate of the film is function of the velocity and volumetric film thickness \( (t_y) \) at any point in the flow as follows:

\[ \dot{m} = \rho_{htf} w(t_y)@distance \text{ x from inlet} \]  (48)

where \( w \) is internal width of the cavity receiver.

Figure 5.10. A simplified schematic of the directly irradiated HTF film flow over an inclined corrugated surface. The amplitude and wavelength of the corrugations are not displayed to scale, while the displayed film thickness is constant, which is not the accurate representation of the flow.

Steady laminar liquid film flows over inclined surfaces can be analytically modelled by Nusselt’s solution (Nusselt, 1916). Nusselt’s solution provides exact solutions to the Navier–Stokes equations – momentum and energy conservations – for smooth gravity-driven liquid films without considering the vapour shear stress or unsteady flow features, such as interfacial waves. The velocity profile at location \( x \) from inlet
across the z direction (reference coordinates are defined in Figure 5.10) according to Nusselt’s solution is described as:

\[ \nu(x, z) = \frac{\Delta t_v(x)}{2\mu_{htf}} \left[ \frac{2z}{t_v(x)} - \left( \frac{z}{t_v(x)} \right)^2 \right] \]

\[ \to \nu(x, z) = \frac{\rho_{htf} g t_v^2(x) \cos\phi}{2\mu_{htf}} \left[ \frac{2z}{t_v(x)} - \left( \frac{z}{t_v(x)} \right)^2 \right] \]

where \( \Delta \) is the wall shear stress \( [\rho_{htf} g t(x) \cos\phi] \), while the Nusselt film thickness is expressed as:

\[ t_{v,Nusselt}(x) = \frac{3\mu_{htf} m}{\rho_{htf}^2 g \cos\phi} \]

Various solutions varied in the literature challenged the accuracy of Nusselt’s quasi-steady approximation for predicting the velocity profiles of unsteady and turbulent film flows over wavy surfaces (Moran et al., 2002, Tseluiko et al., 2013, Yih, 1963). Currently, there is no exact steady-state solution liquid films flowing with a Reynolds number greater than a critical value\(^9\), which is established as \((4/5)\tan\phi\) (Tseluiko et al., 2013, Yih, 1963), where the unidirectional Nusselt flow becomes vulnerable to infinitesimal perturbations (Benjamin and Fiszdon, 1965). The Reynolds number for the liquid metal film flow here is significantly greater than that critical value. The Reynolds number of a gravity-driven film down an inclined surface is defined as:

\[ Re_{film} = 4\Gamma/\mu_{htf} \]

where \( \Gamma \) is mass flow rate per unit wetted perimeter.

To the author’s knowledge, no empirical correlation has been demonstrated on gravity-driven liquid metal film flows. A semi-empirical model was developed by Mudawwar and El-Masri (1986) was developed for free-falling turbulent liquid film flows undergoing heating or evaporation. Their study established that it is unlikely to

\(^9\) The critical value is a fraction multiplied by the cotangent (or tangent) of the inclination angle. The fraction depends on the definition of the used arbitrary Reynolds number, which varies among literature sources.
develop a universal correlations for different fluids that are solely function of Reynolds and Prandtl numbers for film flows with $Re < 10^4$. The reason was due to the dependence of heat transfer on Kapitza number ($Ka$), which is a dimensionless ratio of surface tension to inertia forces used as an indicator of the hydrodynamic wave regime of gravity-driven liquid films. $Ka$ is given as:

$$Ka = \left(\frac{\gamma_{htf}}{\mu_{htf}}\right)^{4/3}\left(\frac{\rho_{htf}}{g \cos \theta}\right)^{1/3}$$

(52)

where $\gamma_{htf}$ is surface tension of HTF. The significance of $Ka$ lies in the role played by surface tension forces in supressing deformations of the film free surface caused at laminar flows. At laminar to turbulent transition, the gravity-driven inertia induces surface wavy fluctuations, with geometries and scale determined by the surface tension forces. However, this effect becomes negligible at fully turbulent flows (Mudawwar and El-Masri, 1986). The critical Reynolds number ($Re_{crit}$) for this flow type is given as:

$$Re_{crit} = 97/Ka^{0.1}$$

(53)

The calculated $Re$ values of the HTF film flow inside the cavity receiver were 270,950 and 621,186 for the reflective and absorptive cavity configurations, respectively. These values are sufficiently higher than the calculated $Re_{crit}$ (56,871) to confidently consider the HTF film as a fully turbulent flow. Based on the Mudawwar and El-Masri (1986) model, the mean $t_\nu$ and $v$ can be expressed as:

$$t_{\nu,mean} = 0.145(Re \cos \theta)^{0.58}\mu_{htf}^{2/3}/(g^{1/3}\rho_{htf}^{2/3})$$

(54)

$$v_{mean} = 1.72(Re \cos \theta)^{0.42}\left(g\mu_{htf}/\rho_{htf}\right)^{1/3}$$

(55)

Nevertheless, this model is yet to be experimentally validated for gravity-driven liquid metal films. Therefore, in the present analytical model, the HTF film is modelled as a continuous frozen, solid-like, volume with opaque surface optical properties. Volumetric and turbulence modelling of the film flow will be numerically implemented in the next chapter.
5.10. Window steady-state energy balance

The proposed transparent ceramic window is a novel addition to solar cavity receivers. The main purpose of the window is to seal the cavity from ambient oxidants and wind-induced convection. The aperture window marks the entry of concentrated solar radiation and exit of all radiative losses, reflective and emissive. Therefore, the window is subjected to large amounts of radiative energy, which, even with a small optical absorptance, could increase its temperature beyond the maximum allowable limit. The window is also subjected to wind-induced pressure and pressure difference across its sides. Nevertheless, the pressure difference can be trivial compared to the strength of the yttria doped magnesium aluminate spinel. The wind-induced stress can be evaluated as follows:

\[
\text{Dynamic pressure} = \frac{\text{Wind load}}{A_{ap}} = \frac{0.5 \rho_{amb} v_{\text{wind}}^2 \cos \theta}{A_{ap}} \tag{56}
\]

where \( \rho_{amb} \) is density of air at tower height elevation, \( v_{\text{wind}} \) is horizontal component of wind velocity directed towards the aperture, and \( \theta \) is tilt angle of the aperture plane to vertical. At \( v_{\text{wind}} \) of 30 ms\(^{-1}\), the dynamic pressure of wind striking the window would be <500 Pa. The cavity atmosphere is can be slightly pressurised at 100 Pa. At present, there is no available information in the literature about the fracture toughness of yttria doped spinel at temperatures >2073 K. However, the fracture toughness of undoped magnesium aluminate spinel is typically >1.1 MPa.m\(^{0.5}\) at temperatures up to 1973 K (Cao et al., 2013, Miller et al., 2018). Therefore, the concern about the wind load and internal pressure on the window can be neglected.

The main threat to the mechanical integrity of the aperture window is thermal stresses and phase change. The phase-change temperature of MgAl\(_2\)O\(_4\) spinel is 2408 K (Ganesh, 2013, Kong et al., 2015, Meng et al., 2018). Nevertheless, the window is also subjected to cyclic thermal loading from repeated cooling and reheating from ultra-high temperatures down to the ambient temperature. One way to enhance the thermal resistance of MgAl\(_2\)O\(_4\) spinel is by optimising its microstructural design. For example, micro-crack toughening is a process where the MgAl\(_2\)O\(_4\) spinel composite is cooled from 2000 K down to the room temperature, so that the mismatch between the
thermal coefficients of the material’s phases, MgO matrix grains and MgAl$_2$O$_4$ particles, would induce large tensile hoop stresses and micro-crack development at the grains boundaries (Aksel et al., 2002). This practice is found to reduce the tensile strength and Young’s Modulus of spinel; however, it also impedes the accumulation of strain energy required to propagate the thermally induced micro-cracks (Aksel et al., 2004) and enable high resistance to rapid temperature changes (Gruber et al., 2016, Harmuth and Tschegg, 1997). This practice of hindering the crack propagation is preferred over preventing cracks initiation, as the latter is difficult to achieve in applications involving high stresses.

The steady-state window temperature ($T_{\text{win}}$) is calculated when the window reaches a thermodynamic equilibrium with its ambient, so that the following energy balance will stand:

$$ \sum Q_{\text{win, gain}} = \sum Q_{\text{win, loss}} $$

$$ \rightarrow \epsilon_{\text{win}}(P_{\text{sol}} + Q_{\text{ref}} + Q_{e}) = A_p(\dot{q}_{\text{in}} + \dot{q}_{\text{out}}) + \dot{Q}_{\text{cond}} $$

(57)

where $\sum \dot{Q}_{\text{win, gain}}$ and $\sum \dot{Q}_{\text{win, loss}}$ are total rates of energy gain and loss by the window, respectively. $\epsilon_{\text{win}}$ is window material emissivity, $\dot{Q}_{\text{cond}}$ is conductive loss through the lateral sides of the window, $\dot{q}_{\text{in}}$ and $\dot{q}_{\text{out}}$ are thermal fluxes discharged from the window towards the inside and outside the cavity, respectively.

A simplified thermal resistance circuit model is presented in Figure 5.11 for the window. At the ambient/external side of the window, $\dot{q}_{\text{out}}$ encounters two thermal resistances: $R_1$, which resembles the resistance to wind-induced forced convection, and $R_2$, which resembles the resistance to the radiative emission from the window to the sky. Similarly, at the internal side of the window, $\dot{q}_{\text{in}}$ encounters two thermal resistances: $R_3$, which resembles the resistance to natural convection by the buoyancy-driven cavity fluid, and $R_4$, which resembles the resistance to the radiative emission from the window to the internal cavity surfaces. The conductive loss through the lateral area of the window ($\dot{Q}_{\text{cond}}$) pass through $R_5$, which is the overall thermal resistance between the window and cavity walls. The window lateral sides are assumed to be perfectly insulated; hence, $\dot{Q}_{\text{cond}}$ is assumed to be negligible.
Figure 5.11. Thermal circuit model for the aperture window.

The five thermal resistances are expressed as:

\[ R_1 = \frac{1}{h_{wind}} \]  \hspace{1cm} (58)

\[ R_2 = \left[ \varepsilon_{win} \sigma (T_{win}^2 + T_a^2)(T_{win} + T_a) \right]^{-1} \]  \hspace{1cm} (59)

\[ R_3 = \frac{1}{h_{win}} \]  \hspace{1cm} (60)

\[ R_4 = \left[ \varepsilon_{win} \sigma (T_{win}^2 + T_r^2)(T_{win} + T_r) \right]^{-1} \]  \hspace{1cm} (61)

\[ R_5 = f(T_w) \rightarrow \infty \]  \hspace{1cm} (62)

where \( h_{wind} \) is the wind-induced forced convection heat transfer coefficient and \( h_{win} \) is the buoyancy-driven natural convection heat transfer coefficient. \( h_{win} \) is calculated from Equation (43), while \( h_{wind} \) is evaluated from:
\[ h_{\text{wind}} = N u_{\text{win}} \frac{k_a}{d_{\text{win}} \cos \theta} \]  

where \( k_a \) is ambient air thermal conductivity and \( d_{\text{win}} \) is window diameter. The Nusselt number here \((N u_{\text{win}})\) is evaluated from the following correlation for turbulent wind flow interjecting a flat surface (Lienhard, 2016):

\[ N u = 0.037 Re^{0.8} Pr^{0.333} \]  

where \( Re \) and \( Pr \) are evaluated based on a wind velocity with a horizontal component of 3 \( \text{ms}^{-1} \) and ambient air temperature of 300 K with thermo-physical properties evaluated at an altitude of 165 m above the ground.

### 5.11. Summary

In this chapter, the thermal performance of proposed solar receiver is mathematically described using a set of simplifications and approximations. This model represents the receiver performance at its nominal state of operation and does not include the effects of transient features of its involved flows or their interactions with radiation. Nevertheless, the developed model can be used as a computationally inexpensive design tool to provide quantitative predications, which can be used in parametric studies or to investigate the technical aspects of the proposed concept. In the next chapter, an alternative, numerical, analysis methodology will be described, which will then be used in Chapter 7 to examine the accuracy of the assumptions and sub-models employed in the analytical model.
Chapter 6. Numerical Analysis

In the previous chapter, the proposed receiver was modelled using an approximate quasi-steady-state analytical model, which is developed as an inexpensive design tool to help evaluate the performance and run parametric studies. This analysis tool is required to be verified against an accurate solution or validated experimentally. A comprehensive experimental validation is cost prohibitive, given the size and temperature level of the proposed receiver. It should be noted that the concept is unlikely to be viable for small-scale applications or at temperatures <1000 K, as conventional systems would operate more feasibly at such conditions (Chapter 2). Therefore, a numerical solution is developed and used to analyse the receiver and verify the analytical model discussed in Chapter 5. The main modelling methods of this numerical approach will then be validated against experimental and computational data in Chapter 7.

This chapter starts with a brief description of the computational fluid dynamics analysis tool, followed by a characterisation of the two involved fluid flows in the proposed receiver, which are the gravity-driven HTF film and buoyancy-driven cavity fluid. Subsequently, an in-depth description of the developed numerical solution, including descriptions of sub-models and solving methods, is presented for the involved fluid flows. Finally, steady-state analyses of the cavity wall and ceramic windows are described as separate supplementary simulations at the end of the chapter.

6.1. Computational fluid dynamics

*Computational Fluid Dynamics* (CFD) is a numerical methodology employed in this study to analyse the dynamics of the involved fluid flows and their interactions with heat transfer and the solid boundaries of the proposed receiver. In this method, the domain of the studied case is simulated computationally by numerically solving a set of differential equations describing the fluid dynamics and heat transfer mechanisms occurring between the *boundaries* of the domain (Chung, 2010).

6.1.1. Main governing equations

The fundamental physical principles governing any fluid flow are mass conservation, Newton’s second law, and energy conservation. These principles are
expressed in a set of partial differential equations, known as *Navier-Stokes* equations, which are employed in CFD to describe the motion of Newtonian fluids. These equations relate the density, temperature, pressure, and velocity of the fluid based on principles of mass, momentum, and energy conservation laws. In this analysis, unsteady (time-dependent) and three-dimensional (3D) forms of the equations are used. Accordingly, the unsteady 3D *continuity* (mass conservation) equation for a point in a compressible fluid is expressed as:

\[
\frac{\partial \rho}{\partial (t)} + \nabla \cdot (\rho \vec{v}) = 0 \tag{65}
\]

where \(t\) is time. \(\nabla\) is a vector of partial derivatives, which allows for the calculation of gradient, curl, and divergence of a vector field – for a field defined using the Cartesian coordinate system, \(\nabla = \frac{\partial}{\partial x} \hat{i} + \frac{\partial}{\partial y} \hat{j} + \frac{\partial}{\partial z} \hat{k}\), where \((x, y, z)\) are the positional coordinates. \(\vec{v}\) is the velocity vector \((\vec{v} = u \hat{i} + v \hat{j} + w \hat{k})\), where \((u, v, w)\) are the velocity coordinates. Similarly, the momentum and energy conservation equations can be expressed using a generalised compact form for an arbitrary conserved property \(\phi\):

\[
\frac{\partial \rho \phi}{\partial (t)} + \nabla \cdot (\rho \vec{v} \phi) = \nabla \cdot \left( - j_{\text{diff}}^\phi \right) + S^\phi \tag{66}
\]

where \(j_{\text{diff}}^\phi\) is the diffusion flux of \(\phi\), and \(S^\phi\) is the source of \(\phi\) per unit volume. In this context, Equation (66) is called a *transport equation* for the property \(\phi\). The detailed derivation and alternative expression forms of the governing equations are provided by Versteeg et al. (2007). The *convection term* describes properties transportation by the ordered fluidic motion, while the *diffusion term* describes properties transportation by the random fluidic motion. Diffusion terms are function of the shear stress tensor, as viscosity works as a diffusion of momentum. Therefore, turbulence and creation of boundary layers are produced from the diffusion in the flow (Temam, 1984).
6.1.2. Reynolds-averaged Navier–Stokes equations

The six independent variables in the governing equations are present in all governing equations. Therefore, all five equations, in addition to an applicable equation of state (e.g. ideal gas equation), need to be solved simultaneously; hence the descriptive term coupled system for solving these equations. Direct Numerical Simulation (DNS) is a high-fidelity method to resolve turbulence at its entire temporal and spatial scales without any modelling. However, the computational cost of DNS is proportional to $Re^3$ (Pope, 2000), which prohibits its use in most engineering applications involving turbulent fluidic flows. Accordingly, Reynolds (1895) developed a mathematical method known as Reynold’s decomposition to separate a quantity’s average expected value from its fluctuations. This method delivers averaged solutions to the Navier-Stokes equations while using knowledge of the turbulence properties to model the whole spectrum of turbulence scales. The resulting approximated equations are known as Reynolds-averaged Navier–Stokes (RANS) equations. For example, Reynolds decomposition can be used to split the velocity field into a mean component ($\overline{v_i}$) and a fluctuating component ($v_i'$) as follows\textsuperscript{10}:

$$v_i = \overline{v_i} + v_i' \tag{67}$$

where $v_i$ is the component of velocity vector in the $x_i$ direction.

There are three ways for Reynolds averaging: temporal, spatial, and ensemble (Blazek and Blazek, 2005). The ensemble-averaging method used for generic turbulence that vary temporally and spatially, which is the case for the two fluid flows here. All three averaging approaches are used to eliminate the mean fluctuating components of the flow variables (e.g. $\overline{\nu_i}$).

The force fluxes subjecting the flow due to turbulent fluctuations are called Reynolds stresses. This stress tensor is important when modelling turbulence (turbulence modelling will be discussed within the context of this study’s application in section 6.3.7). For a homogenous fluid, the 3D Reynolds stress tensor ($R_{ij}$) is defined as:

\textsuperscript{10}Mean and fluctuating components of any scalar $\varphi$ are denoted in this study as $\overline{\varphi}$ and $\varphi'$, respectively.
\[ R_{ij} = -\rho \overline{v_i v_j'} = -\rho \begin{bmatrix} (v_1')^2 & v_1' v_2' & v_1' v_3' \\ v_2' v_1' & (v_2')^2 & v_2' v_3' \\ v_3' v_1' & v_3' v_2' & (v_3')^2 \end{bmatrix} \] (68)

RANS equations can then be obtained by substituting all flow variables in their split forms into the continuity and momentum equations. However, to close the momentum equations, \( R_{ij} \) requires turbulence modelling, as will be discussed in section 6.3.7.

6.1.3. Discretisation of the governing equations

Solving the partial differential equations of fluid dynamics and heat and mass transfer can be implemented by discretisation, which means approximating them into algebraic equations with a finite number of unknowns. The three most commonly used discretisation techniques in CFD are the *Finite element method*, *finite difference method*, and *finite volume method*. The finite element method is commonly used in structural analysis of solids, which can analyse conduction in defined solid volumes, while convection at the boundaries requires empirical or analytical algorithms to solve. The finite difference method has historically dominated CFD applications, as it was simpler to program and less computationally expensive than the finite element method (Anderson, 1995). Nevertheless, the finite difference method is limited to simplified geometries, as it cannot conveniently be applied to 3D unstructured meshes (Chakraborty, 2008). Currently, the finite volume method is the most common method used in CFD and heat transfer, as it is can be applied to complex shapes and can be used to solve nonlinear problems. While the finite element method provides approximate solutions based on local data and the finite difference method provides approximate derivatives using nodal values, the finite volume method employs conservation laws to provide exact expressions for cell-average values and then use it to approximate the values within the cell (Ranganayakulu and Seetharamu, 2018).

In the finite volume method, the domain of the cavity is discretised into a finite number of non-overlapping volumes (cells), where conservation laws are applied on each cell, so that the total flux entering a cell is equal to the total flux leaving adjacent cells. The key feature of the finite volume method is the use of integral formulation of the conservation laws, which is advantageous in the case of flow discontinuities (Moukalled et al., 2015). Integrating the partial differential equations over each
volume/cell transforms them into algebraic equations, which can be solved for the values of the dependent variable. The finite volume method integrates Equation (66) over each arbitrary control/cell volume ($V_{cell}$) in the domain before converting the divergence terms (volume integrals of the convection and diffusion terms) into surface integrals using Gauss’s theorem (Gauss, 1877). Detailed information about the linearisation and numerical solving procedures for fixed and spatially/temporally variating meshes are provided by Moukalled et al. (2015).

**6.1.4. Properties of the numerical solver**

Ansyl Fluent 19.1 (Fluent, 2016) is the CFD software used in this study. Fluent is a general-purpose code based on the finite volume method with a cell-centred formulation (Feng, 2020). The software is commonly used to model fluid flows, turbulence, heat and mass transfer, and chemical reactions in various applications spanning from blood flow in medical applications to aerodynamic analysis of vehicles and aircraft wings. A major advantage of fluent is its high scaling potential using parallel processing, which supports the use of high-performance computing to facilitate solving large-scale and complex CFD problems.

Fluent comprises two numerical solvers: *density-based* and *pressure-based*. The density-based solver computes the density field from the continuity equation, while the pressure field is evaluated from the energy equation and equation of state. The pressure-based solver evaluates the pressure field through an equation manipulated from the continuity and momentum equations. The density-based solver is predominantly used for compressible flows at high Mach numbers, while the pressure-based is commonly used for low-speed incompressible flows (Versteeg et al., 2007). In this study, the pressure-based solver is used, given the low Mach number of the involved fluid flows, which will be evaluated in the next chapter to validate this selection. The justification is that at low Mach numbers, any slight variation in the density would result in large pressure variations and solution instabilities, which can significantly reduce the solution convergence rate (Darwish, 2000). Please note that the employed *Volume of Fluid* method in this study (method will be explained in section 6.3.5) is not compatible with density-based solvers.
There are two types of algorithms in Fluent’s pressure-based solver: segregated and coupled. The difference lies in whether the non-linear momentum and pressure-based continuity equations are segregated and solved sequentially for the solution variables in a loop methodology to converge the solution, or the equations are coupled and solved simultaneously. The main advantage of segregated algorithms is their lower memory requirements, as only a single equation needs to be stored in the memory at a time. Nevertheless, Darwish et al. (2015) argued that the segregated algorithms are highly dependent on the numerical grid, or CFD mesh, as decoupled pressure and velocity would require higher under-relaxation and an increased number of iterations to converge problems with denser meshes. Therefore, their advantage of requiring lower memory than the coupled algorithm diminishes with increased geometrical complexity and size of the computational domain (Van Doormaal et al., 1987). Furthermore, the coupled algorithm delivers more robust and accurate solutions for free-surface flows with large body forces, large energy sources (e.g. radiation), and buoyancy-driven flows with high-Rayleigh numbers (Heining et al., 2009, Vakilipour and Ormiston, 2012). It is recommended to use the coupled algorithm for meshes with poor qualities, which are typically resulted from complex geometries, and for transient flows to allow for larger and more practical time-stepping (Mangani et al., 2014). Accordingly, the coupled algorithm is used in this study to analyse the two fluid flows.

Fluent is a cell-centred software based on the collocated grid storage scheme, so that all flow variables are stored in cell centres only. However, the discretised momentum equations contain terms which are functions of pressure values at the cell boundaries. Therefore, a pressure interpolation scheme is used to compute the face values based on the cell centre values. There are only two pressure interpolation schemes in Fluent which are compatible with the Volume of Fluid method: The Pressure Staggering Option (commonly known as PRESTO!) and the body-force-weighted scheme. The body-force-weighted is used here for the gravity-driven HTF flow, given the significant influence of the gravitational force on this flow. The PRESTO! scheme is used for the cavity fluid flow, as it is recommended for curved domains with buoyancy-driven flows at high Rayleigh numbers (Hirsch, 2007). Similarly, the discretised continuity equations contain terms function of velocity values at the cell boundaries, which need to be related to the cell values. Instead of
linear interpolation of velocity values, which would result in unphysical checker-board oscillations of the pressure values, an alternative velocity interpolation method developed by Rhie and Chow (1983) was used, in which the face velocity values are momentum-weighted averaged. Finally, density face values are interpolated using the first-order upwind scheme (Karki and Patankar, 1989), which is stable and provides accurate results for shock-free flows.

Gradients of flow variables are required to be evaluated at the cell centre to construct their values at the cell faces and compute secondary diffusion terms and velocity derivatives. Calculating the gradients of temperature is particularly important in this study, as it is required by the radiation model imbedded in the source term of the energy equation. Using the Green-Gauss theorem, the gradient of a variable $\varphi$ at the cell centre is computed using the following discrete form:

$$\nabla \varphi = \left( \frac{\partial \varphi}{\partial x}, \frac{\partial \varphi}{\partial y}, \frac{\partial \varphi}{\partial z} \right) = \frac{1}{V_{cell}} \sum_{f}^{N_f} \varphi_f \cdot \hat{A}_f$$  \hspace{1cm} (69)

The value of $\varphi_f$ is calculated using a suitable gradient evaluation scheme. In this study, the Green-gauss node-based method (Holmes and Connell, 1989; Rauch et al., 1991) was selected to evaluate all gradients, as it is reported to deliver accurate results, which are insensitive to the generated unstructured mesh (Delis et al., 2011); however, the main limitation of this method is its higher computational expense than cell-based methods. In this method, the $\varphi_f$ is evaluated using the arithmetic average values at the nodes ($\varphi_n$) of the face as follows:

$$\varphi_f = \frac{1}{N_{n,f}} \sum_{n}^{N_{n,f}} \varphi_n$$  \hspace{1cm} (70)

where $N_{n,f}$ is number of nodes on face $f$. $\varphi_n$ is evaluated from the weighted average values of cells surrounding the node.

6.2. Characterisation of the involved fluid flows

There are two flow types that are crucial to the operation of the proposed receiver concept. The main flow is a gravity-driven liquid metal film flowing over a corrugated
and inclined surface. The secondary flow is a buoyancy-driven cavity fluid flowing inside the bulk of the cavity. As per the analysis in Chapter 5, both flows are fully turbulent and involve transient features, which require different treatments. In this section, an overview of the flow characteristics of the two flow types will be presented, while emphasising the limitations of conventional solving methods.

6.2.1. Non-linear unsteady liquid metal film flow

The HTF film flow here is treated as an open-channel flow, meaning it flows within a conduit with a free surface. Based on the flow characterisation presented in section 6.2.1, this open-channel flow is classified as an unsteady and rapidly-varied flow, which indicate that the liquid level is not temporally or spatially fixed along its path with flow involving frequent hydraulic jumps and drops caused by the corrugations.

The stability classification of flow type can be quantified by evaluating the Froude number ($Fr$), which is a dimensionless ratio of inertia to gravity defined as follow:

$$Fr = \frac{v}{\sqrt{g \sin \phi}}$$  \hspace{1cm} (71)

The evaluated $Fr$ values here were $>$1 at both cavity configuration cases. Thus, this flow is labelled ‘supercritical’, which indicate that the generated surface waves will be carried downstream by the faster flow. This characterisation is useful when selecting the turbulence model of the flow, as will be discussed in section 6.3.7.5.

No steady-state solution is available for a liquid film flow over a corrugated surface as discussed in section 5.9. The literature contains numerous modelling approaches for this type of flow using partial differential equations, which can accurately predict the velocity fields and film thicknesses, while accounting for linear flow instabilities – a comprehensive review of the models is provided by Craster and Matar (2009). Nevertheless, there are reservations over the validity of these models at high Reynolds’ numbers (Oron and Gottlieb, 2002, Pumir et al., 1983, Scheid et al., 2005).

Tseluiko et al. (2013) found that this type of flow at Reynolds numbers exceeding a critical value (defined in section 5.9) would involve transient non-linear instabilities, which necessitate time-dependent computation of Navier-Stokes equations. At high Reynolds number, kinematically and inertially induced eddies starts to develop near
the corrugations troughs (Pozrikidis, 1988, Zhao and Cerro, 1992). In turbulent film flows, the eddies are 3D (i.e. unsteady in the spanwise direction) flow features and were found to significantly affect the film continuity, heat and mass transfer (Negny et al., 2001). These eddies were found to disappear as Reynolds number increases, and then reappear at much higher Reynolds number, leaving a window of vortex-free turbulent film flow (Nam et al., 2009). Before the disappearance of the eddies, the film’s free surface displays hydraulic jump features, such as anharmonicity with the corrugations, which transforms into harmonic as the Reynolds number increases for a range of values before reappearing again as demonstrated in Figure 6.1. Within this range, the liquid’s inertia as it falls from the corrugations crests are adjusted, so that they penetrate through the eddies and suppress them. This phenomenon has been experimentally and numerically observed and investigated in numerous studies (Heining et al., 2009, Trifonov, 1999, Wierschem and Aksel, 2004a).

Figure 6.1. Experimental pathlines and numerical streamlines (overlaid) generated for an inclined (8°) flow of oil to demonstrate the eddies generation phenomenon (Wierschem et al., 2010). a) No eddie at low Reynolds number (Re=8). b) Inertia-induced trough eddie at a higher Reynolds (Re=16). c) Suppression of the eddie by the higher inertia (Re=31). d) Flow separation and reappearance of the eddie at higher Reynolds number (Re=48). Please note that the Reynolds numbers here are based on an arbitrary definition.

Previous studies of this flow type established that the corrugations sizing have a significant influence on stabilising the film flow (Ern et al., 2011, Trifonov, 2014). An experimental study of inclined film flow over sinusoidal corrugations by Wierschem
et al. (2002) detected high disparity between the experimentally evaluated velocity fields and their corresponding predicted fields by the Nusselt’s solution near the regions where the film is thickest. Later experiments found that as the corrugation amplitude increased, the measured velocity fields diverged considerably from theoretical predictions (Wierschem and Aksel, 2003, Wierschem and Aksel, 2004b).

A critical design characteristic of the periodic corrugations profile is the amplitude-to-wavelength ratio \( (a/\lambda) \). This ratio controls the shape and magnitude of trough eddies (Gu et al., 2004, Schörner et al., 2015). At the same Reynolds number, this ratio can control the appearance of the trough eddies as demonstrated in Figure 6.2. In their study, Gu et al. (2004), water films were found to break at the convex-to-concave transition points, which were found to be mitigated by using longer corrugation wavelengths. While this disintegration mechanism depends on the surface tension of the fluid, it will be shown to prevail in liquid metal films in the next Chapter. The rationale behind the impact of the amplitude-to-wavelength ratio on the film continuity lies in the build-up of stagnating fluid regions, which can block the mainstream flow and separate its streamlines around them, creating dry patches past these regions. Therefore, the presence of a trough eddie and separation indicate the existence of a stagnant liquid upstream. However, this separation could be avoided if the mainstream flow possesses a sufficient inertia to destroy the stagnant built-ups, while the \( a/\lambda \) ratio is shown to control this mechanism (Gu et al., 2004). The influence of the \( a/\lambda \) ratio on the flow instabilities was numerically studied by Ern et al. (2011) using a direct numerical solution based on a finite-element Lagrangian Eulerian approximation of the free-surface Navier–Stokes equations. This study found that longer amplitude and shorter wavelengths induce a stabilising effect on the film.

The corrugations profile is described using the following sinusoidal function:

\[
y_{cr} = a(t_v, v)\cos(2\pi x_{cr}/\lambda(t_v, v))
\]

where \((x_{cr}, y_{cr})\) are positional coordinates of the corrugations. The amplitude and wavelength of the corrugations here are scaled based on the inlet slit thickness and inlet velocity, which both represent the inlet volumetric conditions of the film flow.
6.2.2. Turbulent buoyancy-driven cavity fluid flow

In the previous chapter, the buoyancy-driven flow of the cavity fluid was analytically analysed. Based on the evaluated Rayleigh numbers from this analysis, the flow was determined as fully turbulent in at the two studied cavity wall configurations. This turbulence is justified by the high temperature difference, which is significantly greater than the pressure difference, across the domain.

The buoyancy-driven flow of the cavity fluid presents thermal, convective and conductive, impact on the HTF film and cavity walls. In theory, the flow also might pose a shearing effect on the gravity-driven film, which may be in form of instigating, or supressing, surface waves at the interface (Isaenkov et al., 2020, Segin et al., 2006). However, given the small density ($\ll 0.001$) and viscosity ($<0.05$) ratios between the cavity fluid and liquid tin, such interfacial shearing effects would require much greater momentum from the cavity fluid to become significant, which is unlikely in the case of a passively, buoyancy-driven, flow (Vempati et al., 2010).

Numerical modelling buoyancy-driven flows in closed domains necessitate information about the mass inside the domain. The most common approach to model natural convection in closed domains is through steady-state calculations using the Boussinesq approximation, where density differences are ignored unless when they
are multiplied by the gravitational acceleration to evaluate specific weights\(^{11}\) then the following approximation is used to eliminate the density \(\rho\) from the buoyancy term:

\[
\text{specific weight} = (\rho - \rho_{op}) g = -\rho_{op} \beta (T - T_{op})
\]

(73)

where \(\rho_{op}\) and \(T_{op}\) are the operating density and temperature, respectively. The prescribed constant \(\rho_{op}\) provides information about the mass; hence, mass is specified. However, this approach is not applicable when temperature differences are large within the domain [when \(\beta(T - T_{op}) \geq 1\)] (Bejan, 1993). Although this approximation has been frequently employed to model natural convection in solar cavity receivers (Facão and Oliveira, 2011, Pye et al., 2003, Sahoo et al., 2012), the validity of their results were later challenged due to lack of verification to the use of the Boussinesq approximation (Moghimi et al., 2015). Generally, it is recommended to use unsteady calculations for turbulent buoyancy-driven flows (Pope, 2000).

For CSP cavities, which typically exhibit large temperature differences, the ideal gas assumption is found to deliver more accurate results than the Boussinesq approximation for modelling natural convection in cavities (Moghimi et al., 2015). Accordingly, the cavity fluid is modelled as an incompressible ideal gas, which is justified by the trivial pressure differences, compared to temperature differences, across the closed domain and small Mach number of the buoyancy-driven flow, which will be verified using CFD simulations in the next chapter (section 7.2.6).

### 6.3. Solution description

In this section, a CFD solution is described for the proposed receiver. The solution couples radiation with fluid dynamics to evaluate the receiver’s energy performance, while investigating the influence of the flow characteristics discussed in section 6.2 on the feasibility of the proposed receiver concept. The CFD solution analyses different physical aspects of the proposed receiver and their interactions, which were not possible to consider in the analytical model described in Chapter 5. The addressed problem here is composed of volumetric radiation by the HTF film, specular

\(^{11}\) The rationale is that differences in inertia is insignificant but the gravity amplifies these differences when computing the specific weights.
reflections inside the cavity, advection and diffusion of properties by the HTF film hydrodynamics and the dynamics of the buoyancy-driven cavity fluid. The present CFD solution also accounts for the transient progressions of the two fluid flows. While this numerical approach is computationally intensive, its results will be used to verify the accuracy of the more computationally feasible analytical approach of Chapter 5.

6.3.1. Assumptions

The thermo-physical properties of the HTF and cavity fluid are made temperature-dependent instead of using constant values evaluated at the mean flow temperatures. The cavity fluid is modelled as a pure mixture of 90 wt% N\textsubscript{2} and 10 wt% H\textsubscript{2}, and assumed to be free of particulates; hence, posing no scattering or attenuation effects on radiation. The HTF is modelled as a semi-transparent fluid with opaque temperature dependent optical properties to simulate its volumetric absorption of radiation.

As in the analytical model, cavity walls are modelled as isothermal boundaries with prescribed temperatures, and the absorptive cavity’s wall heat exchanger is modelled using the same efficiency used in Chapter 5. The absorptive cavity walls are also assumed diffuse to account for their surface irregularities when simulating radiation. Emitted and reflected radiation beams on the window and cavity walls of the reflective receiver are assumed to be totally, dust-free mirror-like, specular. The significance of radiation specularity will be investigated in the next chapter. Similarly to the analytical model, the numerical analysis here also employs the grey radiation assumption for the justifications stated in section 5.2. Therefore, all optical properties are evaluated at the peak solar wavelength.

6.3.2. Computational domain

The 3D domain of the CFD solution shown in Figure 6.3 is imported from the SolidWorks CAD model of the receiver to Ansys DesignModeler software, which is used to assign the domain as a fluid volume. The displayed domain represent the volume occupied by the two involved fluids, while the solid boundaries, such as walls and window, and their characteristics are prescribed as boundary conditions.
Modelling of the sinusoidal corrugations is computationally challenging, as it necessitates a precise method for discretising small-scale features to maintain their curvatures, which was performed using a high order non-uniform rational basis spline (NURBS). Nevertheless, running a CFD analysis on a control volume modelled with high order NURBS typically results in increased mesh sizes and, in turn, increase computational expense. An attempt was performed to use Fluent’s wall roughness model to virtually model the corrugation effects, without modelling them physically, using a *User Defined Function* that describes the corrugation profiles as a roughness profile. However, flattening the corrugations was found to result in considerable shrinkage (10.6%) of the absorption surface area of the frozen HTF. Therefore, the corrugations had to be physically modelled. Importing the CAD model of the receiver from SolidWorks to Ansys DesignModeller required the use of STEP 214 (ISO 10303-21) file format, which was found to preserve the profile of the corrugations. Other CAD formats, including the commonly used Initial Graphics Exchange Specification (IGES), which were found to distort, or skip instances of, the corrugation features.

The use of symmetry boundary condition was considered to simplify the domain geometry and reduce the computational expense of simulations. A “slice” model (Figure 6.4(a)) was initially considered, which consisted of the middle 100 mm section of the receiver with symmetry boundary condition imposed at the section’s width sides. However, this width-wise simplification was found to overestimate the outlet film temperature by 11.45% and 42.9% for the reflective and absorptive cavities,
respectively. This was due to the non-uniformity of the inlet radiation to the receiver and circular shape of the aperture. A “half” model (Figure 6.4(b)) was later considered, in which half of the geometry’s width is modelled with a symmetry boundary condition imposed at the sectioned surface. However, this symmetry boundary condition risks interfering with the computation of radiative intensities near the boundary, as they are not only functions of 3D spatial coordinates but also directional ones; hence, variations in the latter complicate the plane symmetric conditions (Cai et al., 2016). Moreover, any geometrical simplification would also risk misrepresenting the 3D film flow features should they appear near the symmetry boundary. Therefore, the computational domain of the study was decided to include the full geometry of the receiver, displayed in Figure 6.4(c), with no symmetry boundary conditions.

![Figure 6.4](image)

Figure 6.4. Geometrical simplifications of the domain. (a) Slice model. (b) Half model (symmetry surface is highlighted with green). (c) Full model. Please note that the displayed receiver design is of an earlier version, which has later been modified.

6.3.3. Solution strategy

Based on the characterisation of the two involved fluid flows in section 6.2, it was determined that the physical features of each flow occur at different time and length scales (Ledesma-Aguilar et al., 2010, Shimizu and Loper, 1997). For example, the transient features in the gravity-driven HTF film flow occur in the order of milliseconds, while they take hundreds of milliseconds in the buoyancy-driven cavity fluid flow. To accurately capture the transient flow features of interest in complex turbulent flows, the simulation time scale is required to be sufficiently smaller (<50%)
than the physical time scale of the formation and development of those flow features (Choi and Moin, 1994). Accordingly, coupling both fluids to the same time-advancement strategy would necessitate unfeasible time-stepping to converge the buoyancy-driven flow. Similarly, the length scale of the HTF film flow is controlled by the small wavelength of a single corrugation, while the depth of the whole cavity determines the length scale of the buoyancy-driven flow. Accordingly, each flow requires different spatial discretisation properties to guarantee accurate solutions. Therefore, the flows are solved in two loosely coupled simulation sets, allowing for the use of suitable temporal and spatial discretisation, as well as suitable turbulence modelling, for each flow. The only limitation of using this approach is neglecting the viscous interaction between the two flows; nevertheless, such interfacial shearing interactions in the studied cases were demonstrated as negligible in section 6.2.2. Please note that the buoyancy-driven cavity fluid does not participate in radiation, hence, does not influence the volumetric radiation absorption by the HTF.

The strategy for running the CFD simulations is demonstrated in Figure 6.5. The strategy comprises three simulation sets: a transient set for each involved flow and a steady-state set to combine the results. The first set is the ‘film simulations’, in which the HTF film flow is transiently simulated using a suitable multiphase model, while the buoyancy-driven movements of the cavity fluid is disabled by prescribing the latter with constant temperature and thermo-physical properties. A radiation model is used in this set to simulate the volumetric absorption of radiation by the HTF film. The main purposes of this set is check the continuity of the HTF film from its inlet to outlet, which is essential for the validity of the analytical model, and to compute the surface temperature of film, which will be prescribed in the next two simulation sets. The second set is the transient buoyancy simulations, in which the HTF film is modelled as frozen material with a prescribed surface temperature distribution imported from the film simulations. The cavity fluid here is prescribed with temperature-dependent thermo-physical properties to instigate buoyancy effects. The main purpose of this set is to evaluate the natural convection heat transfer coefficients on the internal surfaces.

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12 Values of the thermo-physical properties are acquired at the prescribed temperature, which was the evaluated $T_{cav}$ from the analytical model.
including HTF film, cavity walls, and window. The final set is the *steady-state simulations*, which imports the pseudo-steady-state results of the previous transient sets as boundary conditions to combine their effects and run steady-state energy analysis to evaluate the energy performance of the receiver. This set also involves two secondary simulation sets, which are used to evaluate the thermal integrity of the ceramic window and cavity wall structures proposed in Chapter 4.

Figure 6.5. Solution strategy for the CFD simulations.

Parallel processing of the simulations was facilitated by two main computational resources. The University of Edinburgh’s research computer cluster (Eddie Mark 3 – Scientific Linux 7 Operating System) was used to run the film simulations. This cluster provides over 8000 cores with up to 3 TB of memory per node. Each film simulation used 288 CPU processor cores – Intel® Xenon® Gold 6138 – each with a RAM capacity of 32 GB. The buoyancy and steady-state simulations were operated on a pool of servers called ‘VLX’, where simulation was to run on a server comprising of 16 CPU processor cores – Intel(R) Core(TM) i7-10700 – each with a RAM capacity of 16 GB. Sample scripts written to run Fluent jobs on Eddie are presented in Appendix A4. To compare the computational expense of each simulation set at the specified resources, the wall-clock time used to converge the solution of the absorptive cavity case at each set is quoted as follows: 1287h (film); 63.61h (buoyancy); and 1.198h (steady-state).
The convergence criteria for film and steady-state simulation sets were set at $10^{-8}$ for the residuals of the energy and radiation equations and $10^{-4}$ for the residuals of all other governing equations. The convergence criteria for the buoyancy simulations followed special criteria, which will be specified in the next section.

6.3.4. Temporal discretisation

To account for the transient flow features, each term in the governing equations is discretised temporally over a specified number of time-steps ($\Delta t_s$), which can either be performed using an implicit or explicit formulation. The method used by Fluent’s pressure-based solver for temporal discretisation is the implicit time integration. In this method, the convection, diffusion, and source terms are evaluated at the fields from a future time level ($t + \Delta t_s$). The method is called implicit, as the future value of the variable $\phi^{t+\Delta t_s}$ in a cell is related to its present value $\phi^t$ in adjacent cells through the mentioned spatially discretised terms $F(\phi^{t+\Delta t_s})$ as follows:

$$\phi^{t+\Delta t_s} = \phi^t + \Delta t_s \cdot F(\phi^{t+\Delta t_s})$$  \hspace{1cm} (74)

Equation (74) is solved at each time-step before advancing to the next time-step, with initial property value ($\phi^{t=0}$) prescribed by the user. Recall the requirement to provide information about the initial mass to numerically simulate the buoyancy-driven flow, the initial density is computed from the initial operating temperature and $p_{op}$ using Equation (117). Subsequently, the implicit transient solution is applied, and as it progresses, the conserving initial mass is checked. The main advantage of this scheme is its unconditional stability to the size of $\Delta t_s$ (Versteeg et al., 2007).

There are no available direct solution for turbulent buoyancy-driven flows at high Rayleigh numbers, as discussed in section 6.2.2. A transient solution procedure developed by Henkes and Hoogendoorn (1994) is adopted to generate the required pseudo-steady-state results for the cavity fluid flow. The procedure starts with running the simulation as a steady-state problem at the same Rayleigh number before running the transient solution. The transient solution is implemented using a fixed time-step estimated using the following formula, which is recommended to avoid divergence (Bejan, 2013, Henkes and Hoogendoorn, 1994):

$$\Delta t_s = 0.25L_c/\sqrt{g\delta \Delta T_{domain} L_{ch}}$$  \hspace{1cm} (75)
where $\Delta T_{\text{domain}}$ is temperature difference in the domain. Consequently, the calculated $\Delta t_s$ were 0.377s and 0.75s for the reflective and absorptive cavities, respectively. Convergence to pseudo-steady-state is then achieved when the frequency of solution oscillations ($f_{osc}$) decays to the following condition (Henkes and Hoogendoorn, 1994):

$$4f_{osc}\Delta t_s = [0.05 - 0.09]$$

(76)

The values in Equation (76) depend on the turbulence model and values of Rayleigh and Prandtl numbers in this study. The essence of this condition is to ensure that the internal gravity waves have decayed. Upon reaching this criterion, the values of the heat transfer coefficients on the walls are monitored to guarantee no significant changes (to the sixth significant digit) would occur as time progresses further.

6.3.5. Mesh generation

All simulations used the same computational 3D domain described in section 6.3.2 and spatially discretised using unstructured tetrahedral cells\(^\text{13}\) to delineate the curvatures of the corrugations (Loseille, 2017). The initial fixed mesh (Figure 6.6(a)) was generated using with a fine span angle centre of $\theta^0$ to accurately discretise the corrugations. Instead of generating a fixed mesh for the film simulations with a fully refined mesh near the corrugations boundary, a coarser mesh was generated using the ANSYS 3D Mesh software and the Proximity and Curvature sizing feature of the software so that the corrugation amplitude is filled with at least three tetrahedral cells. The dynamic gradient adaption method is then applied to the initial mesh to track and refine the interfacial regions of HTF film, to maintain their sharpness, as the flow progresses towards the outlet as demonstrated in Figure 6.6(b). Once a refined region is no longer at, or near to, the interface, it will be locally coarsened to reduce the number of cells and save memory for the newly refined regions. Nevertheless, it should be noted that as the HTF flow extends towards the outlet, its upper interfacial region increases extends and, hence, the net number of refined cells increases.

\(^\text{13}\) For receivers with flat surfaces and less complex geometries (no high order NURBS are used to model the geometry), it is advised to use hexahedral meshes to reduce the memory and CPU time required for simulations.
Figure 6.6. Generated mesh for CFD simulations. (a) Fixed mesh generated from ANSYS 3D Mesh software. (b) Dynamically adapted mesh used in the film simulations (colour code is the volume fraction of the HTF).

The initial mesh consisted of 6,257,137 cells and was used for both cavity wall configurations; however, grew at different rates due to the different film flow velocities in each configuration. Accordingly, the mesh of the reflective cavity case required more cells, given its slower flow rate, as it was more susceptible to develop local stagnant regions and flow separation as discussed in section 6.2.1. Since variable time-stepping is used in the film simulations, the transition from fine to coarser volumetric cells away from the corrugations needed to be done gradually, so that the mesh skewness and cell aspect ratios are maintained below 0.95 and 5, respectively. This requirement is advised by Fluent to guarantee a stable solution convergence when using the volume of fraction method, which will be discussed in the next section.
There are several methods for adapting a computational mesh – comprehensive reviews of the methods are provided in (Dannenhoffer and Baron, 1985, Versteeg et al., 2007, Warrem et al., 1991). The selected dynamic gradient adaption method in the film simulations performs mesh adaption automatically at a prescribed time-stepping frequency (for every 20 time-steps here) based on an algorithm which tracks the Euclidean norm of the gradient of a selected solution variable multiplied by a length function (Dannenhoffer and Baron, 1985). This algorithm tracks the volume fraction of the HTF and evaluates its gradients to adapt the mesh based on prescribed thresholds values for refining and coarsening. Mesh cells which are at gradient regions above the prescribed refine threshold are refined, while those at gradient regions below the coarsen threshold are coarsened. Gradient values here are normalised by their maximum values in the whole domain to facilitate solution independence (Geßner and Kröner, 2001); thus, all gradient and threshold values are between 0 and 1.

6.3.6. Multiphase and surface tension modelling

Modelling the HTF film flow in the film simulations requires a multiphase model to differentiate its properties from cavity fluid properties. Based on the characterisations presented in section 6.2, the multiphase flow case here is classified as a stratified/free-surface flow with the two fluids separated by a defined sharp interface. The validation of the selected models will be presented in section 7.2.2.1.

6.3.6.1. Volume of fluid method

The most suitable modelling method available in Fluent for the classified case is the Volume Of Fluid (VOF) method (Hirt and Nichols, 1981), which is a simplified Eularian model used to simulate continuous-to-continuous phase interactions. VOF captures the immiscible interface by solving the following continuity equation for the volume fraction of the HTF ($\beta_{htf}$), while the volume fraction of the cavity fluid (denoted with the subscript $cf$) is the complementary value:

$$\frac{\partial}{\partial t} (\beta_{htf} \rho_{htf}) + \nabla \cdot (\beta_{htf} \rho_{htf} \vec{v}_{htf}) = m_{cf\rightarrow htf} - m_{htf\rightarrow cf}$$  \hspace{1cm} (77)

This volume fraction equation is constrained by the following boundedness condition:

$$\beta_{htf} + \beta_{cf} = 1$$  \hspace{1cm} (78)
For the two immiscible fluids, there should be zero net mass transfer at their interface, hence, the right-hand side in Equation (77) is set to zero. Therefore, the volume fraction equation reduces to:

$$\frac{\partial \beta_{hf}}{\partial t} + \nabla \cdot (\beta_{hf} \mathbf{\tilde{v}}_{hf}) = 0$$  \hspace{1cm} (79)

The effect of the different phases are accounted for in the rest of transport equations through the thermo-physical and optical properties as follows:

$$\kappa = \beta_{hf} \kappa_{hf} + (1 - \beta_{hf}) \kappa_{cf}$$  \hspace{1cm} (80)

where $\kappa$ is the considered property. This approach of solving for $\beta_{hf}$ and the mixture (denoted with the subscript $mix$) is applied to all transport equations, so that solving a transport equation $\kappa$ for a scalar $\varphi^\kappa$ will be implemented as:

$$\frac{\partial}{\partial t} (\beta_{hf} \rho_{hf} \varphi_{hf}^\kappa) + \nabla \cdot (\beta_{hf} \rho_{hf} \mathbf{\tilde{v}}_{hf} \varphi_{hf}^\kappa - \beta_{hf} \mathbf{\tilde{v}}_{hf} \varphi_{hf}^\kappa) = S_{hf}^\kappa$$  \hspace{1cm} (81)

$$\frac{\partial}{\partial t} (\rho_{mix} \varphi_{mix}^\kappa) + \nabla \cdot (\rho_{mix} \mathbf{\tilde{v}}_{mix} \varphi_{mix}^\kappa - f_{mix} \mathbf{\tilde{v}}_{mix} \varphi_{mix}^\kappa) = \nabla \cdot S_{mix}^\kappa$$  \hspace{1cm} (82)

No simultaneous coupling of velocity, pressure, and volume fraction equations is used, as this approach is still underdeveloped and unfeasible for practical problems.

6.3.6.2. Implicit formulation

In section 6.3.4, it was established that the implicit scheme is can reinforce the solution stability. This is particularly important when using transient VOF for modelling open-channel flows to mitigate its associated convergence instabilities and enable the use of practical under-relaxation and time-step sizes (Patil et al., 2009). Thus, Equation (77) is temporally discretised using the following implicit formulation:

$$\frac{V_{cell}}{\Delta t(i)} (\beta_{hf}^{i+1} \rho_{hf}^{i+1} - \beta_{hf}^i \rho_{hf}^i) + \Sigma_f (\rho_{hf}^{i+1} u_f^{i+1} \beta_{hf,f}^{i+1})$$

$$= V_{cell} (\dot{m}_{cf \rightarrow hf} - \dot{m}_{hf \rightarrow cf})$$  \hspace{1cm} (83)

where $i$ is the index for previous time-step, $u_f$ is volume flux through the face, and subscript $f$ is face value.
The *Implicit Body Force* treatment feature of Fluent is used for improved convergence when body forces, such as gravity, are significant. One of the limitations of the implicit formulation is generating numerical diffusion at the interface, which is mitigated here using an anti-diffusion method developed by So et al. (2011).

6.3.6.3. Interface capturing and discretisation schemes

To preserve the sharpness of the interface, a suitable spatial discretisation scheme is required for the convective term \( \nabla \cdot (\beta_{htf} \vec{v}_{htf}) \) of Equation (79). The *Compressive* interface-capturing scheme (Ubbink and Issa, 1999) is selected based on its suitability for modelling multiphase flows which comprise fluids with a large viscosity difference at a lower computational expense than the *High-Resolution Interface Capturing* (HRIC) scheme (Barral-Jr. et al., 2019).

Initially, before the HTF film reaches the outlet, first-order upwind schemes are used for the momentum and energy equations to sustain the convergence stability of the solution. Upon reaching the outlet and after the equations exhibit a stable trend towards convergence, the discretisation schemes are switched to second-order schemes for accurate results and reduced numerical diffusion at the interface. Please note that the second-order central differencing scheme is unbounded, hence, is susceptible to value oscillations and violating Equation (78). Temporal discretisation in the film simulations is implemented by the bounded second-order implicit time integration throughout the simulations, as recommended for liquid flows with variable densities (Hirsch, 2007).

6.3.6.4. Surface tension

The continuum surface force method (Brackbill et al., 1992) is used to model the surface tension and wall adhesion of the HTF film through the source term of the momentum equation. However, the surface tension effects are expected to be insignificant for a gravity-driven dense liquid metal compared to its inertia. For film flows, the significance of surface tension is assessed using a dimensionless number called *Weber number*, which is a ratio of the drag to cohesive forces of the fluid. This value is evaluated here as >>1, which confirms the insignificance of surface tension...
effects. The expression for the Weber number \( (We) \) for an inclined gravity-driven film flow is given as (Takeshi, 1999):

\[
We = \frac{\rho_{htf} g \cos \theta t_v^2}{\gamma_{htf}}
\]  

(84)

where \( \gamma_{htf} \) is surface tension of the HTF.

### 6.3.7. Turbulence modelling

The two involved fluid flows were established as fully turbulent in Chapter 5. Turbulence here indicates that the velocity fields in the domain are fluctuating, while the transported quantities of the governing equations mix and fluctuates as well. In typical engineering applications, these fluctuations are of high frequency and small scale, which would be computationally unfeasible to solve directly without modelling (Pope, 2000). Turbulence modelling address these problems solving the Navier-Stokes equations using ensemble-averaged values, following the RANS approach discussed in section 6.1.2, and manipulating the transport equations to eliminate the small-scale resolution. Since each flow is simulated separately, as discussed in section 6.3.3, a suitable turbulence modelling approach is used for each flow.

#### 6.3.7.1. Large Eddie Simulation model

As discussed in section 6.1.2, the feasible alternative to DNS is RANS. However, there is a third approach developed by Smagorinsky (1963), called Large Eddie Simulation (LES), in which large eddies are resolved directly, without modelling, on the premise that they carry transported quantities which are less isotropic and more influenced by the macroscopic features, such as the prescribed boundary conditions, while smaller eddies are less problem-dependent and, hence, can be modelled by a universal model (Pope, 2000). Accordingly, instead of ensemble-averaging, the Navier-Stokes equations are manipulated mathematically to “filter” out the small eddies. However, the computational cost of LES is typically higher than the RANS approach, as the former requires resolving the governing equations for a turbulent flow in both time and space domains. Therefore, LES is only used here to generate a reference case to verify the selected RANS-based turbulence model.
6.3.7.2. Two-equation eddy-viscosity turbulence models

The most commonly used RANS models in a wide-range of engineering applications are the $K - \varepsilon$ turbulence models. In these models, turbulence is modelled using two partial differential transport equations: one for turbulence kinetic energy ($K = 0.5 \upsilon_t' \upsilon_t'$) and another for the dissipation rate of its kinetic energy ($\varepsilon$) as follows:

$$\frac{\partial (\rho K)}{\partial t} + \frac{\partial (\rho K v_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \frac{\partial K (\mu + \frac{\mu_t}{Pr_K})}{\partial x_j} \right] + G_K + G_b - \rho \varepsilon - Y_M + S_K \tag{85}$$

$$\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho \varepsilon v_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \frac{\partial \varepsilon (\mu + \frac{\mu_t}{Pr_\varepsilon})}{\partial x_j} \right] + \frac{C_{1\varepsilon} \varepsilon (G_K + C_{3\varepsilon} G_b) - C_{2\varepsilon} \rho \varepsilon^2}{K} + S_\varepsilon \tag{86}$$

where $\mu_t$ is turbulent viscosity, $C_{1\varepsilon}$, $C_{2\varepsilon}$, and $C_{3\varepsilon}$ are constants. $Pr_K$ and $Pr_\varepsilon$ are turbulent Prandtl numbers for the $K$ and $\varepsilon$, respectively. $S_K$ and $S_\varepsilon$ are source terms for each equation. $Y_M$ is dilatation dissipation term, which accounts for the compressibility of flows at high Mach numbers – this term is neglected here, given the low Mach numbers of the involved flows. $G_K$ and $G_b$ represent the generation of turbulent kinetic energy due to average velocity gradients and buoyancy, respectively, and are expressed as:

$$G_K = R_{ij} \frac{\partial v_j}{\partial x_i} \tag{87}$$

$$G_b = \delta g_i \frac{\mu_t}{Pr_t} \frac{\partial T}{\partial x_i} \tag{88}$$

where $R_{ij}$ is the Reynold stress tensor (defined in section 6.1.2), $g_i$ is the gravitational acceleration component towards the $x_i$ direction, and $Pr_t$ is turbulent Prandtl number for energy (taken as 0.85). The turbulent viscosity ($\mu_t$) is defined as follows:

$$\mu_t = \rho C_\mu K^2 / \varepsilon \tag{89}$$

where $C_\mu$ is a constant. All constants and turbulent Prandtl numbers are obtained from (Launder, 1972), which are experimentally validated for water, air, and ideal gases (please recall that the cavity fluid is modelled as an ideal gas).
There are three main the $K - \varepsilon$ turbulence models: standard (Launder, 1972), ReNormalisation Group (RNG) (Orszag et al., 1993), and realisable (Shih et al., 1995). The main differences across the three models lie in the calculation of $\mu_t$, $Pr_K$, $Pr_\varepsilon$, and the generation/destruction terms in Equation (86).

The applicability and limitations of the $K - \varepsilon$ models are well understood in the literature. Generally, the standard $K - \varepsilon$ model is experimentally validated for modelling natural convection on an isothermal vertical surface; however, the model was reported to overestimate the velocity within the boundary layer and inaccurately model peaks of $K$ and its fluctuations near the wall (Plumb and Kennedy, 1977). This marks the main limitation of $K - \varepsilon$ models, as they poorly resolves the viscous layer. An alternative modelling approach was developed by Wilcox (1998), which is best suited for near-wall turbulence modelling. In this approach, the specific dissipation rate ($\omega$), which can be considered as a ratio of $\varepsilon$ to $K$, as is used instead of $\varepsilon$; hence, this approach is widely referred to as $K - \omega$ model. Furthermore, The evaluation of $\mu_t$ accounts for low Reynolds number effects and shear flow spreading. The two transport equations in this model are expressed as:

$$\frac{\partial (\rho K)}{\partial t} + \frac{\partial (\rho K v_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \frac{\partial K}{\partial x_j} \left( \mu + \mu_t \right) \right] + G_K - Y_K + S_K \tag{90}$$

$$\frac{\partial (\rho \omega)}{\partial t} + \frac{\partial (\rho \omega v_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \frac{\partial \omega}{\partial x_j} \left( \mu + \frac{\mu_t}{Pr_\omega} \right) \right] + G_\omega - Y_\omega + S_\omega \tag{91}$$

Note that the buoyancy generation term is missing in the $K - \omega$ equations.

6.3.7.3. Modelling near-wall flow

To investigate the applicability of different RANS turbulence models, the behaviour of turbulent fluid flow near wall boundaries is reviewed first, as the fidelity of numerical simulations of turbulent flows are highly dependent on near-wall modelling (Pope, 2000). Preserving the no-slip condition – viscous fluids adjacent to solid boundaries share the same velocities – indicates that there is a region attached to the wall, called the boundary layer, where the computed variables have steep gradients, which require a suitable representation by the turbulence model. A common
representation of this boundary layer is the law of the wall (Von Kármán, 1931), which relates the mean velocity of a point to its logarithmic distance from the wall. Nevertheless, only one sublayer of the boundary layer obeys this law. To formularise a universal law for near-wall flow behaviour, the distance of a point on an arbitrary layer from the wall ($y$) and its velocity ($v$) are normalised into $y^+$ and $v^+$ as follows:

$$y^+ = \frac{\rho v^+ y}{\mu}$$  \hspace{1cm} (92)

$$v^+ = \frac{v}{v^*}$$  \hspace{1cm} (93)

where $v^*$ is shear/friction velocity, which is defined as follows:

$$v^* = \sqrt{\frac{\tau}{\rho}}$$  \hspace{1cm} (94)

where $\tau$ is shear stress at that arbitrary layer, which is expressed as follows:

$$\tau = 0.5 \rho C_{f,w} v^2$$  \hspace{1cm} (95)

where $C_{f,w}$ is skin friction coefficient.

As demonstrated in Figure 6.7, the wall boundary layer is be divided into three sublayers. The first and closest to the wall boundary (at $y^+ < 5$) is the viscous sublayer. The fluid within the viscous sublayer is dominated by the viscous stresses, while turbulent stresses, such as $R_{ij}$, are assumed negligible; therefore, $y^+$ and $v^+$ are linearly related. The second sublayer (at $5 < y^+ < 30$) is the buffer layer, which is a transition region between the viscous-dominated region and turbulence dominated region. The buffer layer is complex without a well-defined profile, as viscous and turbulent stresses becomes equal in magnitude; hence, it is a conventional practice to avoid locating the first cell centre within this layer. The third layer ($y^+ > 30$) is the logarithmic layer (log-law layer), in which the velocity profile obeys the law of the wall, as turbulence stresses dominate over viscous stresses. The extend of the log-law layer depends on the value of $Re$ but is roughly ~300 distance units. Overall, the velocity profile of the boundary layer can be expressed as follows:

$$v^+ = \begin{cases} 
    \frac{y^+}{2.439 \ln(y^*) + 5.2} & (y^+ < 5) \\
    (y^+ \geq 30)
\end{cases}$$  \hspace{1cm} (96)
Figure 6.7. Components of the wall boundary layer. Figure adapted from (Fluent, 2016).

According to the guidance provided by Fluent (2016), $K - \varepsilon$ models with standard wall functions should avoid positioning the first wall-adjacent cell centres outside of the log-law region ($30 \leq y^+$), as the models are reportedly less accurate at modelling flow at viscosity-dominated regions. Nevertheless, to account for complex near-wall flow features, Fluent offers the Enhanced Wall Functions option, in which the linear and logarithmic trends are modelled separately within the model by splitting the domain into two layers, hence named the two-layer approach, based on the value of an arbitrary near-wall Reynolds number ($Re_y$) defined as follows:

$$Re_y = \frac{\rho y \sqrt{K}}{\mu} = \begin{cases} Re_y \geq 200 & \text{turbulence layer} \\ Re_y < 200 & \text{viscous layer} \end{cases}$$  

(97)

The model then solves the “turbulence layer” using the selected $K - \varepsilon$ model, while the viscous layer is modelled using a single-equation model developed by Wolfshtein (1969). The Enhanced Wall Functions then applies a blending function developed by Kader (1981) to blend the three sublayers functions into a single law function. Accordingly, the wall-adjacent cell centres should be positioned within the viscous sublayer ($y^+ < 5$) when the Enhanced Wall Functions option is used. Since the $K - \omega$ model resolves the flow at the viscous sublayer, same cell-centre positioning guidance to Enhanced Wall Functions should be followed.
Despite its proven accuracy in modelling flow fields and skin friction at near-wall regions, from the viscous sublayer up to the logarithmic part of the boundary layer, the $K - \omega$ model is notorious for the sensitivity of $\omega$ to the freestream conditions (Pope, 2000). Accordingly, to combine the advantages of both models, Menter (1994) proposed to use $K - \omega$ for modelling turbulence within the boundary layer, while switching to the standard $K - \varepsilon$ to model the free-shear regions. This model is commonly known as the Shear-Stress Transport (SST) model. The two transport equations are similar as in Wilcox’s $K - \omega$ with the difference is including an additional cross-diffusion term to Equation (91), which is used to blend the two models. Details about this term and its derivations are provided by Menter (1994).

6.3.7.4. Turbulence in the buoyancy simulations

In a numerical study by Vieira et al. (2012) of modelling natural convection inside an isothermal cubical cavity at comparable flow conditions ($Pr=0.6, Ra=10^6-10^{11}$) to the buoyancy-driven cavity fluid flow, different RANS turbulence models were compared. The SST model was found to deliver a more robust and accurate solutions than the other models. Accordingly, the SST model was chosen to model turbulence in the buoyancy simulations.

Utilising the ideal gas assumption for the cavity fluid, the value of $\delta$, which is used in Equation (88) to evaluate the $G_b$ term of the $K - \varepsilon$ equations, is evaluated in the buoyancy simulations using the following:

\[
\delta = -\frac{1}{\rho} \left( \frac{\partial \rho}{\partial T} \right)_p
\] (98)

To verify the near-wall mesh resolution of the nominated mesh grade in section 7.2.1.2, the $y^+$ was evaluated at all the wall boundaries as displayed for the reflective and absorptive cavities in Figure 6.8 and Figure 6.9, respectively. The $y^+$ values are shown to be maintained within the viscous sublayer ($y^+ \leq 4.766$) at the corrugated surface, and log-layer ($y^+ \leq 225.4$) at the other boundaries.
6.3.7.5. Turbulence in the film simulations

In highly turbulent flows (\( Re >> Re_{crit}, Fr >> 1, \) and \( We >> 1 \)), gravity and surface tension effects fail to confine the fluctuating turbulence. Therefore, a complex – 3D with inherent nonlinear instabilities – surface wave phenomenon occurs, as discussed in section 6.2.1. The surface waves are found to correlate strongly with scalars transport across the interface in turbulent open-channel flows (Freeze et al., 2003). For instance, surface waves of a turbulent film were found to increase the heat transfer coefficient at the interface by \( >40\% \) (Kutateladze, 1982). Freeze et al. (2003)
characterised the influence of $Fr$ value on turbulent surface waves and heat transfer of inclined open-channel flow. In this study, the mean film thickness was estimated using the standard $K - \varepsilon$ model and compared to experimental measurements, which revealed a significant disparity across the results. This disparity was found to increase with the $Fr$ value and surface waviness, as $K - \varepsilon$ model failed to accurately model the turbulent interaction at the free surface; hence, prohibiting its use for modelling supercritical ($Fr \gg 1$) film flows. The $K - \omega$ model offers a more suitable alternative to capture near-wall features; however, it would require finer mesh (as discussed in section 6.3.7.3), while the sensitivity of $\omega$ to the freestream conditions can destabilise the solution convergence.

One of the main limitations of the $K - \varepsilon$ and $K - \omega$ models is that they are based on the eddy-viscosity hypothesis, which assumes an isotropic turbulent eddy viscosity ($\mu_t$). This hypothesis was proposed by Boussinesq (1877) to relate the turbulent stresses to the mean flow to close the Navier-Stokes equations. This approach assumes linear proportionality between the turbulent shear stresses and the mean strain rate with $\mu_t$ as the proportionality constant, which can misrepresent complex 3D flows with anisotropic turbulence, such as flows with large streamline curvatures or flow separations (Di Piazza and Ciofalo, 2010, Sleiti and Kapat, 2006, Yang and Tucker, 2016). Open-channel flows over surfaces with sinusoidal corrugations involve nonlinear anisotropic turbulence as explained by Assato and de Lemos (2009). Therefore, the two-equation eddy viscosity turbulence models are not suitable to model the HTF flow in the film simulations.

To achieve a closure to the Navier-Stokes equations at a higher (second) order, Reynolds Stress Model (RSM) is selected here to model the turbulence of the HTF film. This model is originated from the mathematical works of Chou (1945) and Rotta (1951) then furtherly developed by multiple authors; an extensive review of the development of RSM is provided in (Launder et al., 1975). This model is considered the most complete classical turbulence model (Ye et al., 2021). In this approach, all components of $R_{ij}$ are directly computed depending only on exact transport equations for them, which can account for directional effects and other complex interactions on turbulence. The exact transport equations for $R_{ij}$ are expressed as:
\[
\frac{\partial R_{ij}}{\partial t} + C_{ij} = -D_{t,ij} + D_{t,ij} - P_{ij} - G_{ij} + \phi_{ij} - \epsilon_{ij} - F_{ij} + S_{ij}
\]  

(99)

The equation terms here are defined as follows: \(C_{ij}\) is convection, \(D_{t,ij}\) is turbulent diffusion, \(D_{t,ij}\) is molecular diffusion, \(P_{ij}\) is stress generation, \(G_{ij}\) is buoyancy generation, \(\phi_{ij}\) is pressure-strain, \(\epsilon_{ij}\) is dissipation, \(F_{ij}\) is generation from system rotation, and \(S_{ij}\) is user-defined source term. The modelling equations of these terms are provided by Davidson (2021). The \(D_{t,ij}\), \(G_{ij}\), \(\phi_{ij}\), and \(\epsilon_{ij}\) terms need modelling to close Equation (99). In Fluent, \(D_{t,ij}\) is modelled using a generalised gradient-diffusion model developed by Daly and Harlow (1970) with a simplified scalar turbulence diffusivity proposed by Lien and Leschziner (1994) to minimise the numerical instabilities. The \(\epsilon_{ij}\) is modelled in accordance to a model developed by Sarkar and Lakshmanan (1991). The \(\epsilon_{ij}\) is modelled in a similar manner to Equation (88) as follows:

\[
G_{ij} = -\delta \left( g_i \frac{\mu_t}{Pr_t} \frac{\partial T}{\partial x_j} + g_j \frac{\mu_t}{Pr_t} \frac{\partial T}{\partial x_i} \right)
\]

(100)

Note that in film simulations, using Equation (98) to evaluate \(\delta\) is inapplicable.

Modelling the pressure-strain term (\(\phi_{ij}\)) depends on the selected scale equation. Fluent offers \(K - \varepsilon\) and \(K - \omega\) models as scale equations for evaluating this term. The selected scale equation for the HTF film flow is an evolved version of the \(K - \omega\) the Baseline (BSL) model developed by Menter (1994), which is the origin of the same author’s SST model. As of 2022’s Fluent version 19.1, the SST model is not included as a scale equation for the RSM model. The only difference between the SST and BSL is the former’s improved ability to predict the shear stress in adverse pressure gradient boundary-layer flows, which are significant in aerodynamic applications. However, since the modelled flow here is of a compressed liquid, this additional ability is not useful. Similar to SST, the BSL is characterised by eliminating the sensitivity of \(\omega\) to the free-stream conditions. Verification of the selected turbulence modelling approach for the HTF film flow will be presented in section 7.2.2.2.
6.3.8. Radiation modelling

Radiation is the primary mode of heat transfer in the studied problem, as the energy input to the receiver is in form of concentrated thermal radiation. Moreover, the domain encloses boundaries and fluids at elevated temperature, while Stefan-Boltzmann law institutes that radiant heat flux is proportional to the fourth-order of the absolute temperature. Therefore, it is anticipated that re-radiation would present a significant fraction of heat exchange between the boundaries.

Recalling the energy equation in section 6.1.1, it can be re-written in terms of temperature as follows:

\[
\frac{\partial}{\partial t}(\rho E) + \nabla \cdot (\tilde{\nu} \rho E + p) = \nabla \cdot \left( k_{\text{eff}} \nabla T - \sum_{j} h_j \tilde{j}_j + (\tilde{\zeta}_{\text{eff}}, \tilde{\nu}) \right) + S_E \tag{101}
\]

where \( k_{\text{eff}} \nabla T \) is conduction term and \( S_E \) is volumetric heat source term, which is used here to introduce the radiation transfer effects into the CFD simulations.

6.3.8.1. Volumetric absorption theory

Before describing the radiation model, the theoretical behaviour of light beams transmitted through a medium is explained, which is essential for understanding the modelling of volumetric absorption of radiation. The exponential attenuation of a light beam as it travels through a medium with a thickness of \( t_\nu \) (illustration displayed in Figure 6.10) is described by Bouguer’s, or Lambert’s, law as follows:

\[
-A \int_{0}^{t_\nu} C \, dx = \int_{I(0)}^{I(t_\nu)} \frac{dI(x)}{I(x)}
\]

\[
\rightarrow I(x) = I(0)e^{-(AC)x} \quad [x = 0 \rightarrow t_\nu] \tag{102}
\]

where \( x \) is the travelled distance by the light through the medium. \( AC \) is the absorption coefficient of the medium, which is an important parameter used by the radiation model to define the interactions of the window and HTF film with radiation. \( AC \) is a dimensional (per unit path length) representation of how far the light beam can penetrate through a medium before it is completely absorbed.
In the literature, sometimes $AC$ is confused with the **attenuation coefficient**. The attenuation of a light represents the total loss of a beam intensity due to two processes: absorption and scattering. Thus, the attenuation coefficient ($AT$) is defined as:

$$AT = AC + SC$$

where $SC$ is the *scattering coefficient*, which represents the redirection of light towards random direction(s), which converts the beam into a diffuse light. Note that the units of all terms in Equation (105) are per unit length ($m^{-1}$). The values of $AC$ and $SC$ for liquid tin and MgAl$_2$O$_4$ spinel are acquired from literature experimental measurements, as will be discussed in section 6.3.9.3.

$AC$ should not be confused with the *logarithmic absorbance*\(^{14}\), which is a dimensionless volumetric property of materials, which represents the exponential attenuation of transmitting light through a medium by absorption, reflection, and scattering. Please note that the absorbance is a volumetric property, while absorptivity/absorptance (symbolised here by $\alpha$) is a surface property. *Beer’s law* can be used to relate different volumetric optical properties of the material as follows:

$$\tau = 10^{-\alpha} = e^{-t_o}$$

where \( \tau \) is the optical transmittance (\( \tau = \frac{l_{\text{transmitted}}}{l_{\text{incident}}} \)). \( t_o \) is the optical thickness, or depth, which is the dimensionless natural logarithm of the ratio of incident to transmitted radiation through a medium. The optical thicknesses of the HTF film and window are evaluated here to characterise the radiation problem, which will be used to select the radiation model and solving schemes. It is related to the geometrical thickness through:

\[
 t_o = AT \ t_V \tag{105} 
\]

The propagation of electromagnetic radiation waves through a material is also specified by the material’s refractive index, which is a frequency-dependent complex value with the real component representing refraction and imaginary component representing the attenuation (commonly referred to as the extinction coefficient). Equation (102) describes the beam attenuation through \( t_V \) without accounting for the refractive attenuation effects, as if the optical path length is equivalent to the thickness of the medium. For a more realistic representation, consider an incident beam that is not perpendicularly interjecting the HTF-cavity-fluid interface or window surface. Therefore, in accordance to Snell’s law of refraction, the refracted angle (\( \theta_{\text{refr}} \)) will be different from the incident angle (\( \theta_{\text{inc}} \)) based on the ratio of refractive indices across the interface/surface as follows:

\[
 n_1 \theta_{\text{inc}} = n_2 \theta_{\text{refr}} \tag{106} 
\]

where \( n \) is refractive index, subscripts 1 and 2 denote sides of the interface/surface of the semi-transparent medium. The reflectance and refraction across at the interface between the HTF film and cavity fluid is demonstrated in Figure 6.11. Accordingly, the reflectance (\( R \)) of the interface (from the cavity fluid side) can be defined as follows:

\[
 R_{cf}(\delta) = \frac{1}{2} \left[ \frac{(n_{cf}\cos\theta_{\text{refr}} - n_{hft}\cos\theta_{\text{inc}})^2}{(n_{cf}\cos\theta_{\text{refr}} + n_{hft}\cos\theta_{\text{inc}})^2} + \frac{(n_{cf}\cos\theta_{\text{inc}} - n_{hft}\cos\theta_{\text{refr}})^2}{(n_{cf}\cos\theta_{\text{inc}} + n_{hft}\cos\theta_{\text{refr}})^2} \right] 
\]
For $\theta_{\text{inc}} > 0$, the geometrical path line of the beam to transmit through the medium would be longer than $t_V$. Note also that the speed of light ($c$) through a medium varies based on the refractive index as follows:

$$c = c_{\text{vacuum}}/n$$  \hspace{1cm} (108)

Accordingly, the optical path length ($t_{o,PL}$) can be defined in terms of the geometrical path length ($t_{\text{geo,PL}}$) as follows:

$$t_{o,PL} = n(t_{\text{geo,PL}})$$  \hspace{1cm} (109)

From Equations (102) to (109), a more realistic value for $AC$ can be expressed in terms of $\alpha$ as follows (Hu et al., 2002):

$$AC = a \ln(10)/t_{o,PL}$$  \hspace{1cm} (110)

Another useful expression for $AC$ in terms of the overall optical absorptance ($\alpha$) is obtained from Incropera (2002):

$$AC = -\ln(1 - \alpha)/t_{o,PL}$$  \hspace{1cm} (111)

6.3.8.2.  Radiation transfer equation

Unlike heat transfer by conduction and convection, radiative heat transfer is function of seven independent variables: three spatial coordinates (for 3D problems), two directional angles, wavelength (for non-grey problems), and time (for transient problems). Consequently, radiative heat transfer cannot be simply represented using
partial differential equations, as was the case for conduction and convection in Equation (101), as it involves a set of directional and positional dependent processes, including absorption, emission, and scattering. The interactions of these processes are described by an integro-differential equation namely the Radiative Transfer Equation (RTE). This equation is solved to retain the intensity field \((I)\) as follows:

\[
\frac{\partial I_\lambda}{\partial t} + I_\lambda - (1 - \mathbf{\chi}_\lambda) I_{b\lambda} = \frac{\mathbf{\chi}_\lambda}{4\pi} \int_0^{4\pi} I_\lambda(\vec{r}, \vec{s}') \phi_\lambda(\vec{s}, \vec{s}') d\Omega' \tag{112}
\]

where \(I_{b\lambda}\) is the blackbody intensity at wavelength \(\lambda\). \(\mathbf{\chi}_\lambda\) is the single-scattering albedo, which represents the absorption and scattering processes (i.e. it is a ratio of scattering efficiency to total attenuation by scattering and absorption). \(\vec{r}, \vec{s},\) and \(\vec{s}'\) are position, direction, and scattering direction vectors, respectively. \(\phi\) is the scattering phase function, which describes the redistribution of an approaching radiation from a specified direction to the other directions. \(\Omega'\) is solid angle. The subscript \(\lambda\) denotes a spectral quantity. The integral term \(\frac{\mathbf{\chi}_\lambda}{4\pi} \int_0^{4\pi} I_\lambda(\vec{r}, \vec{s}') \phi_\lambda(\vec{s}, \vec{s}') d\Omega'\) represents the scattered radiation from the surroundings. Due to the complexity of this scattering term, there are no analytical solutions available for most engineering applications.

To simplify the RTE, the local thermal equilibrium assumption is used to express the absorption/emission coefficients in terms of temperature, which can be defined in accordance to the Zeroth Law of Thermodynamics. Utilising the employed grey radiation assumption stated in section 6.3.1, the spectral effects can be neglected. The time-dependent effects can also be neglected here, as the timescales of the transient simulations of this study are significantly greater than of transient radiation effects, which occur in the order of femtoseconds to picoseconds (Liu et al., 2017). Therefore, the RTE can be reduced into the following grey quasi-steady-state form:

\[
\frac{dI(\vec{r}, \vec{s})}{ds} + (AC + SC) I(\vec{r}, \vec{s}) = AC n^2 \frac{\sigma T^4}{\pi} + \frac{SC}{4\pi} \int_0^{4\pi} I(\vec{r}, \vec{s}') \phi(\vec{s}, \vec{s}') d\Omega' \tag{113}
\]

Equation (113) is then solved using spatially discretised transport equations (Chui and Raithby, 1993) in a similar manner as solving the governing equations.
Equations (101) and (113) are often coupled and solved simultaneously to accelerate the convergence. This approach is called the Coupled Ordinates METHOD (COMet), which was proposed by Mathur and Murthy (1999) and validated for optically thick mediums (>10). However, given the small optical thicknesses of both the HTF film and window (<10), the energy and RTE equations are solved sequentially using a finite-volume scheme validated for unstructured meshes (Murthy and Mathur, 1998). The energy equation is solved for 10 iterations for every iteration of the RTE, which was found to be suitable to maintain a coordinated convergence for both equations. Solving for less iterations can delay the convergence, while solving for more iterations displayed trivial impact on reducing the numerical residuals.

6.3.8.3. Radiation model selection

Fluent offers six different radiation models to solve Equation (113), which are compared in Table 7. Only one model is capable of processing specular reflections and modelling radiation in participating media within enclosures (Raithby and Chui, 1990), which is required to model the volumetric radiation absorption in the liquid metal HTF film and window. This model is called the Discrete Ordinates Model (DOM), which was originally proposed by Chandrasekhar (1950) to solve radiative transfer in astrophysical problems. A comprehensive description of the DOM model and its applications is provided by Chui and Raithby (1993).

To account for the non-homogenous radiative flux distributed across the cavity domain and through the aperture, it is recommended to use a “high-accuracy” radiation model, identified in the literature (Li et al., 2020c, Pitz-Paal et al., 1997) as either DOM or Monte Carlo’s ray tracing (MCRT). Craig et al. (2016) compared MCRT against the Ansys Fluent’s finite-volume-based DOM for modelling radiation in solar cavities. Several procedural advantages of using DOM over MCRT were highlighted in the study, such as the treatment of scattering in semi-transparent media and the integration of heat and mass transfer with optics in one software. The main disadvantages raised in this study were DOM’s higher computational expense and its additional requirement to perform angular discretisation independence tests. Nevertheless, the computational expense limitation is considered a temporary drawback, which can diminish with further development of computer processing technologies. Mecit and Miller (2014)
Table 7. Comparison of the radiation models available in Fluent 19.

<table>
<thead>
<tr>
<th>Radiation Model</th>
<th>Advantages</th>
<th>Limitations</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>Discrete Transfer Radiation model (DTRM)</td>
<td>• Simple model</td>
<td>• Not compatible with parallel processing</td>
<td>(Carvalho et al., 1991)</td>
</tr>
<tr>
<td></td>
<td>• Applicable to a wide-range of optical thicknesses</td>
<td></td>
<td></td>
</tr>
<tr>
<td>P-1</td>
<td>• Low CPU requirement</td>
<td>• Not compatible with parallel processing</td>
<td>(Cheng, 1964)</td>
</tr>
<tr>
<td>Rosseland</td>
<td>• Fast</td>
<td>• Limited to optically thick media</td>
<td>(Siegel and Howell, 1992)</td>
</tr>
<tr>
<td>Surface-to-Surface (S2S)</td>
<td>• Best results and performance for enclosures without participating media</td>
<td>• Incompatible with periodic boundary conditions</td>
<td></td>
</tr>
<tr>
<td>Discrete Ordinates Model (DOM)</td>
<td>• Only model that can account for scattering and radiation in fluids</td>
<td>• Can be CPU-intensive for fine degrees of angular discretisation</td>
<td>(Chui and Raithby, 1993)</td>
</tr>
<tr>
<td></td>
<td>• High-accuracy</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>• Not restricted to grey radiation</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Monte Carlo (MC)</td>
<td>• Highest accuracy</td>
<td>• Highest computational cost</td>
<td>(Mayles et al., 2007)</td>
</tr>
<tr>
<td></td>
<td>• Applicable to a wide-range of optical thicknesses</td>
<td>• Does not support mesh adaption</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Does not support hanging nodes</td>
<td></td>
</tr>
</tbody>
</table>

demonstrated the feasibility and flexibility of using DOM for modelling cavity receivers with complex geometries, with validation against experimental results acquired from a lab-scale simulator. An important advantage of DOM is the ease of building complex arbitrary geometries on CFD software, such as Fluent, attributed the graphical user interface feature. Numerous studies (Craig et al., 2014, Khalsa and Ho, 2011, Khivsara et al., 2014, Kribus et al., 2014, Pitz-Paal et al., 1997) verified the accuracy of DOM for modelling volumetric and particle receivers. Furthermore, DOM has been verified in various studies (Coelho, 2007, Liu and Chen, 1999, Silvestri et al., 2018) as a robust and compatible approach with fluid dynamics solvers, for solving turbulence-radiation interaction with an emphasis on its applicability on participating media with small optical thicknesses (0.1-10). Verification of DOM for modelling radiation in isothermal cavities will be presented in Chapter 7.
6.3.8.4. Discrete ordinates model

As the name implies, DOM approximates the RTE by discretising the spatial domain – 3D Cartesian domain – and the entire angular domain ($\Omega = 4\pi$) into a finite number of discrete angular intervals (solid angles) and corresponding weight factors. Accordingly, the RTE is converted from an intego-differential form into a set of partial differential transport equations for each discrete direction. Each solid angle is assigned with a fixed direction vector ($\hat{s}$) within the domain, so that the model would solve the transport equations for each directional $\hat{s}$. Therefore, the scattering integral over the entire angular domain is approximated (discretised) into a linear system of equations using the numerical quadrature scheme as follows:

$$\int_0^{4\pi} I(\bar{r}, \bar{s}') \phi(\bar{s}, \bar{s}') d\Omega' = \sum_{i=1}^n w_i I(\bar{r}, \bar{s}_i) \phi(\bar{s}_i, \bar{s}_i)$$

(114)

where $w_i$ is a weight coefficient for the directional vector $i$. The polar coordinates are defined using the polar ($\theta$) and azimuthal ($\psi$) angles, which themselves are defined in the solver with respect to the global Cartesian system as illustrated in Figure 6.12(a). Accordingly, for 3D problems, each octant of the $4\pi$ angular domain is discretised into eight solid angles (will be referred here as control angles), each represented by user prescribed numbers of polar ($N_\theta$) and azimuthal ($N_\psi$) angles as illustrated by Figure 6.12(b). Each control angle ($\Omega$) is expressed in terms of the angular coordinates using the following surface integral:

$$\Omega = \iint_{A_{s,oct}} \sin\theta \, d\theta \, d\psi$$

(115)

where $A_{s,oct}$ is the oriented surface of an octant. Accuracy of the solution depends on user-prescribed $N_\theta$ and $N_\psi$, which are proportional to the computational expense. For example, (2×2) angular discretisation would result in 2×2×8=32 equations, while (5×5) angular discretisation would result in 5×5×8=200 equations. Note that these equations are solved in addition to other CFD governing equations, while the memory overhead from storing the transported values of the fine mesh is already high. Solution independence from the prescribed $N_\theta$ and $N_\psi$ will be presented in section 7.2.1.3.
Given the complex receiver geometry, walls are normally not aligned with the global spatial reference frame. Thus, when unstructured meshes are used for spatial discretisation, the control volume faces are likely to misalign with the global angular discretisation, resulting in unevenly divided control angles by the walls, as illustrated in Figure 6.13(a). This problem is called the control angle overhang. A representation of this problem in the 3D domain is displayed in Figure 6.13(b).

![Image](a)

Figure 6.12. Demonstration of the angular discretisation used by the DOM. (a) Definition of the computational domain using the angular reference system (represented by the angles \( \theta \) and \( \psi \)) within the spatial coordinate system (represented by the Cartesian coordinates \( x, y, z \)). (b) Angular discretisation of a quadrature of the domain using \( N_\theta = N_\psi = 5 \) angles.

![Image](b)

Figure 6.13. Control angle overhang demonstration. (a) A wall splitting the control angle \( \Omega_m \) (Coelho, 2012). (b) A control volume face with an overhang control angle, which is split by the face into the incoming and outgoing directions in a 3D domain (Fluent, 2016).

This problem is solved by pixelation of overhanging control angles into a number of polar and azimuthal pixels, as illustrated in Figure 6.14. This solution was proposed by Murthy and Mathur (1998) and employed by Fluent, so that each pixel on the
overhanging control angle is assigned either to the incoming direction or the outgoing from the face. Generally, a pixelation of 1×1 is sufficient for most problems; however, for problems involving specular reflections or semi-transparent media, Fluent recommends a pixelation of 3×3 up to 5×5. However, higher pixelation can significantly increase the processing cost with minimal improvement in the results (Fluent, 2016). Therefore, a pixelation of 5×5 was used for the overhanging control angles in all film and steady-state simulations.

![Pixelation of an overhanging control angle](image)

**Figure 6.14.** Pixelation of an overhanging control angle (Fluent, 2016).

### 6.3.9. Boundary conditions

The CFD problem is defined with geometry, volume properties, initial and boundary conditions. Boundary condition types include flow inlet/outlet, walls, and symmetries. As discussed in section 6.3.3, no symmetry boundary condition was used in this study. The main boundaries of the problem are displayed in Figure 6.15.

![Main boundaries of the problems](image)

**Figure 6.15.** Main boundaries of the problems.
In this problem, there is a single flow inlet and outlet for the HTF film. The inlet slit was defined as a velocity inlet (prescribed with a fixed inlet velocity) with prescribed scalar properties, such as temperature and volume fraction of liquid tin (prescribed as 1). The outlet slit is defined as a pressure outlet (prescribed with the same cavity pressure) with a prescribed properties for the backflow (outlet temperature at 1673 K and volume fraction of liquid tin of 1). The use of a pressure outlet here rather than an outflow boundary is due to the latter’s inclination to divergence during when during a backflow, which occurs during the transient phase when the HTF film has not reached the outlet. The aperture is defined as a semi-transparent wall with prescribed optical and thermo-physical properties of MgAl$_2$O$_4$ spinel at the solar peak wavelength, while the cavity walls are defined as isothermal wall boundaries with optical and thermo-physical properties depending on the wall material.

The two cavity wall configurations are defined using boundary conditions, while both used the same computational domain discussed in section 6.3.2 and geometrical properties described in Chapter 4. For example, the prescribed inlet values for the velocity and temperature of the HTF film varied across the configurations. The inlet velocity and temperature of the HTF, as well as the isothermal temperature of the cavity walls, in each configuration used similar values to those estimated in the analytical model. The resulted temperature distribution, per configuration, from the film simulations set are imported to the subsequent sets as an imposed mapped boundary condition, so that temperature cell-centred are translated, by interpolation, to the new meshes. A summary of the boundary conditions prescribed in different simulation sets per cavity walls configuration is presented in Table 8.

6.3.9.1. Liquid tin properties

Utilising the thermodynamic equilibrium assumption and the correlation between the absorption coefficient and absorptivity in Equation (111), the absorption coefficient can be expressed in terms of temperature-dependent emissivity as follows:

$$AC(T) = -\ln[1 - \epsilon(T)]/t_{o,PL}$$  \hspace{1cm} (116)

where $\epsilon(T)$ is temperature dependent emissivity, which can be obtained for liquid tin
Table 8. Summary of boundaries conditions prescribed per cavity wall configuration.

<table>
<thead>
<tr>
<th>Boundary Condition</th>
<th>Reflective Cavity</th>
<th>Absorptive Cavity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ambient temperature, $T_a$ (K)</td>
<td>300</td>
<td></td>
</tr>
<tr>
<td>Operating pressure (Pa)</td>
<td>101,425</td>
<td></td>
</tr>
<tr>
<td><strong>HTF</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cavity fluid</td>
<td>90 wt% N$_2$ + 10 wt% H$_2$</td>
<td></td>
</tr>
<tr>
<td><strong>Material</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Aperture</td>
<td>MgAl$_2$O$_4$ spinel</td>
<td></td>
</tr>
<tr>
<td><strong>Wall lining</strong></td>
<td>Silver</td>
<td>Iron–cobalt–chromite black spinel</td>
</tr>
<tr>
<td>HTF inlet temperature, $T_{in}$ (K)</td>
<td>800</td>
<td>1448.68*</td>
</tr>
<tr>
<td>Cavity walls temperature, $T_w$ (K)</td>
<td>800</td>
<td>1623.6*</td>
</tr>
<tr>
<td>Cavity walls emissivity, $\varepsilon_w$</td>
<td>0.1</td>
<td>0.8</td>
</tr>
<tr>
<td>Cavity walls diffuse fraction</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>Window diffuse fraction</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Window refractive index, $n_{win}$</td>
<td></td>
<td>1.721</td>
</tr>
<tr>
<td>Window absorption coefficient, $AC_{window}$ (m$^{-1}$)</td>
<td>0.655</td>
<td>(Fernelius et al., 1982)</td>
</tr>
<tr>
<td>Window scattering coefficient, $SC_{window}$ (m$^{-1}$)</td>
<td>0.336</td>
<td>(Dericioglu et al., 2005)</td>
</tr>
</tbody>
</table>

Prescribed only in film simulations

| Cavity temperature, $T_{cav}$ (K) | 908.41*          | 1623.57*          |
| Inlet HTF velocity, $v_{htf,in}$ (m/s) | 0.230*          | 0.533*           |
| HTF refractive index, $n_{htf}$ | 2.16             | (Golovashkin and Motulevich, 1964) |

*Values estimated from the analytical model of Chapter 5.

from experimental measurements by Greenstein (1989). These experiments were performed at two wavelength levels: 0.532 μm and 1.06 μm. Since the model is using the grey radiation assumption, the $\varepsilon(T)$ value was interpolated at the solar peak wavelength (0.5573 μm). The measurements were evaluated across the temperature range solid tin at 300 K to liquid tin 1400 K. Due to the absence of available measurements in the literature, values for $\varepsilon(T)$ at higher temperatures were extrapolated from these measurements, as the trend was approximately linear throughout the whole measured temperature range within the liquid phase. Variation
of the liquid tin emissivity throughout the temperature range 800-1400 K was evaluated as 21.7%. This variation is likely to become more substantial in other liquid metals, such as aluminium and copper, based on their greater dependence of their optical properties on temperature at ultra-high temperatures (Ujihara, 1972). The \( AC(T) \) values were computed at 20 different temperatures (100-2000 K) and imported into Fluent using a piecewise-linear function to linearly interpolate the values between the measurements. Same approach was used for the temperature-dependent thermo-physical properties of liquid tin, including density, viscosity, surface tension, thermal conductivity, and specific heat capacity. The experimental measurements of liquid tin’s thermo-physical properties were acquired from (Assael et al., 2010, Cahill and Kirshenbaum, 1964, Giordanengo et al., 1999, Khvan et al., 2019, Yuan et al., 2002).

6.3.9.2. Cavity fluid properties

The buoyancy flow was disabled in the film and steady-state simulations sets using constant prescribed thermo-physical properties for the cavity fluid as discussed in section 6.3.3.

In the buoyancy simulations set, the density of the cavity fluid was modelled using the incompressible ideal gas as justified in section 6.2.2. Therefore, the density \( \rho_{cf} \) is solved using:

\[
\rho_{cf} = \frac{p_{op} M_W}{RT}
\]

where \( p_{op} \) is prescribed internal cavity pressure, \( M_W \) is molecular weight of cavity fluid, and \( R \) is the universal gas constant. The other thermo-physical properties were made temperature-dependent using piecewise polynomial functions of the 5th degree. These functions are built-in functions developed for \( \text{N}_2 \) gas at 1 atm and 288 K (Fluent, 2016). Accordingly, the temperature-dependent dynamic viscosity \( \mu_{cf} \), specific heat capacity \( c_{cf} \), and thermal conductivity \( k_{cf} \) are expressed in terms of temperature \( T \) as follows:

\[
\mu_{cf} = (7.473 \times 10^{-6}) + (4.084T \times 10^{-8}) + (-8.245T^2 \times 10^{-12}) + (1.306T^4 \times 10^{-15}) + (-8.178T^5 \times 10^{-20})
\]
\[ c_{cf} = \begin{cases} 
979 + 0.418T + (-1.176 \times 10^{-3})T^2 \\
+ (1.674 \times 10^{-6})T^4 \\
+ (-7.256 \times 10^{-10})T^5 \\
868.6 + 0.442T + (-1.687 \times 10^{-4})T^2 \\
+ (2.997 \times 10^{-8})T^4 \\
+ (-2.004 \times 10^{-12})T^5 
\end{cases} \quad T<1000 \text{ K} \quad (119) \\
\]

\[ k_{cf} = (4.737 \times 10^{-3}) + (7.272 \times 10^{-5})T + (-1.122 \times 10^{-8})T^2 + \\
(1.455 \times 10^{-12})T^4 + (-7.872 \times 10^{-17})T^5 \quad (120) \]

6.3.9.3. Aperture boundary conditions

The aperture window in the film and steady-state simulations was treated as a specular semi-transparent wall boundary through which the concentrated solar radiation transmits. A demonstration for a generic irradiation incident on that boundary type is displayed in Figure 6.16. Nevertheless, the specular fraction impose on the window was 1; thus, no diffuse reflection or transmission occur at the window. Since no radiation modelling was processed in the buoyancy simulations, the aperture was defined as an isothermal wall boundary.

![Figure 6.16. A generic incident (at an angle $\theta$) irradiation ($q_{\text{irrad}}$) at the semi-transparent window boundary (Fluent, 2016). $n_a$ and $n_b$ are refractive indices of the cavity fluid ($n_a=1$) and window boundary, respectively.](image-url)
Dispersion of light through the window material is described by Sellmeier equation (Sellmeier, 1872), which is an empirical expression correlating the refractive index of the semi-transparent medium with the light wavelength (in vacuum) as follows:

\[ n^2(\lambda) = 1 + \sum_i \frac{B_i \lambda^2}{\lambda^2 - C_i} \]  

(122)

where \( B_i \) and \( C_i \) are Sellmeier coefficients, which need to be experimentally evaluated. Tropf and Harris (1995) evaluated the Sellmeier coefficients for MgAl_2O_4 spinel, so that Equation (122) can then be expressed at the peak solar wavelength (\( \lambda_{sol,peak} \)) for spinel as follows:

\[ n_{\text{spinel}} = \sqrt{1 + \frac{1.8938(\lambda_{sol,peak})^2}{(\lambda_{sol,peak})^2 - (0.09942)^2} + \frac{3.0755(\lambda_{sol,peak})^2}{(\lambda_{sol,peak})^2 - (15.826)^2}} \]  

(123)

Given the targeted large-scale application, the number of input ray sources (primary reflector/heliostats) is anticipated to be >>200 (PS20’s solar field encompasses 1255 heliostats). Therefore, the non-uniform distribution of the solar flux intercepted by the receiver aperture can be modelled as a 2D Gaussian distribution with the highest peak flux density at the aperture centre. This approximated flux distribution has been validated for PS10 and PS20 power plants, in addition to several other large-scale tower receiver plants, using various simulation codes (Besarati et al., 2014, Farges et al., 2015, He et al., 2013, He et al., 2019, Salomé et al., 2013, Yao et al., 2009) and experimental work (Ho and Khalsa, 2012, Litwin and Pacheco, 2002, Reilly and Kolb, 2001, Röger et al., 2014). Li et al. (2019) found that the use of a 3D CPCs significantly reduce the flux non-uniformity at the aperture.

The Gaussian flux distribution can either be circular or elliptical on the aperture boundary. For receiver apertures aligned horizontally, the elliptical Gaussian distribution would more accurately represent the projection of the solar flux to account for the incident angles of beams reflected from heliostats, or ~2 m above, the ground level. However, circular Gaussian distribution is justified for apertures, which are tilted with an angle equals, or close, to the incidence angle of the resultant (flux-weighted) incoming beams from the heliostats (Huang and Sun, 2016). In a CFD study
of a PS10-like cavity receiver by Craig et al. (2014), it was found that the flux distribution on the aperture boundary have trivial effect on the temperature results and overall thermal efficiency of the receiver.

The radiative flux distribution on the aperture surface (displayed in Figure 6.17) was compiled to Fluent using a user-defined function (script provided in Appendix A4) using the following 2D Gaussian distribution equation:

$$I(x, y) = I_{\text{max}} \exp \left[ -1 \left( \frac{(x - x_{cc})^2}{2s_x^2} + \frac{(y - y_{cc})^2}{2s_y^2} \right) \right]$$  \hspace{1cm} (124)

where \((x_{cc}, y_{cc})\) are spatial coordinates of the aperture centre on the aperture plane, \(s_x\) and \(s_y\) are standard deviations in the \(x\) and \(y\) directions, respectively. The \(I_{\text{max}}\) is the peak intensity at the aperture centre. This generated distribution resulted in an area-weighted average flux equals to the evaluated mean flux in the analytical model \(p_{sol}\), which was defined in section 5.4.

![Figure 6.17. Contour plot displaying the prescribed Gaussian intensity flux (W.m\(^{-2}\)) distribution at the aperture surface.](image)

6.3.10. Secondary steady-state simulations

In addition to the main simulations discussed in the previous section, two secondary simulations were processed using resulted data from the main simulations as boundary conditions. The first is used to model the steady-state energy balance from and to the window material to evaluate its temperature, while the second is used to evaluate the temperature distribution across the thickness of the cavity walls. These simulations are
used to examine the design properties of the ceramic window and cavity walls discussed in Chapter 4.

6.3.10.1. Window temperature

The steady-state temperature of the 4-mm thick window was numerically evaluated using the model presented in section 5.10. However, utilising the defined receiver geometry, the conductive loss \( \dot{Q}_{\text{cond}} \) from the window is not ignored. The thermal fluxes, \( \dot{q}_{\text{in}} \) and \( \dot{q}_{\text{out}} \), discharged internally and externally are modelled in Fluent using the mixed convection-radiation thermal boundary condition. This simulation set was integral to the steady-state simulations set, as it required input radiation (overall incident radiative flux map on the internal surface of the window) and convection (internal heat transfer coefficient on the aperture surface), which are obtained from the film and buoyancy simulations. In the film and buoyancy simulations, the window temperature was prescribed with the value evaluated in the analytical model for each configuration. In section 7.2.8, the window temperature results will be compared across the modelling method.

The mesh in this simulations set is generated using the ‘Proximity and Curvature’ feature with mesh-inflation towards the wall boundary, as shown in Figure 6.18(a), and a map-meshed thickness with >30 cells, as displayed in Figure 6.18(b). Results from the mesh-independence and angular-discretisation-independence tests of the window temperature simulations are displayed in Table 9 and Table 10 respectively. Based on the tabulated results, mesh grade number 2 and angular discretisation of 3x3 were used in all window temperature simulations.
Figure 6.18. Generated mesh for the window temperature simulations. (a) Aperture front view. (b) Side (thickness) view.

Table 9. Results of the mesh-independence tests performed on the window temperature simulations. Simulations were run using angular discretisation of 3×3.

<table>
<thead>
<tr>
<th>Mesh Grade</th>
<th>Number of Cells</th>
<th>Volume-Weighted Average Temperature (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Reflective</td>
</tr>
<tr>
<td>1</td>
<td>3185</td>
<td>1928.1766</td>
</tr>
<tr>
<td>2</td>
<td>25,480</td>
<td>1943.6877</td>
</tr>
<tr>
<td>3</td>
<td>203,840</td>
<td>1943.6893</td>
</tr>
</tbody>
</table>

Table 10. Results of the angular-discretisation-independence tests performed on the window temperature simulations. Simulations were run using mesh grade 2.

<table>
<thead>
<tr>
<th>Number of Solid Angles ((N_\theta \times N_\phi))</th>
<th>Volume-Weighted Average Temperature (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Reflective</td>
</tr>
<tr>
<td>1×1</td>
<td>2998.0825</td>
</tr>
<tr>
<td>3×3</td>
<td>1943.6877</td>
</tr>
<tr>
<td>4×4</td>
<td>1943.6823</td>
</tr>
<tr>
<td>5×5</td>
<td>1943.6821</td>
</tr>
</tbody>
</table>

6.3.10.2. Cavity walls

Different cavity wall structures were proposed in section 4.4 for each of cavity wall configuration. To guarantee the insulating and lining materials are maintained at temperatures below their corresponding maximum allowed temperatures (specified in section 4.4), the temperature distribution across the cavity wall thickness was numerically evaluated using Fluent. The mean radiation flux absorbed by the cavity walls in the film simulations are imported here as a surface, heat flux, boundary
condition and applied to the cavity-side boundary wall. Therefore, no radiation modelling was used in this simulations set, given its opaque structure. Additionally, the mean heat transfer coefficient on the cavity walls evaluated from the buoyancy simulations were applied as a boundary condition on the cavity-side boundary wall. Similar ambient conditions, including ambient temperature and wind velocity, were used as in sections 5.10 and 6.3.10.1 and applied on the ambient-side boundary wall.

Four different models were prepared for these simulations. As displayed in Figure 6.19, the models utilised the symmetry boundary condition to analyse a single 50-mm width element of the wall. The first model is of a simple stratified structure for the reflective cavity walls without any active cooling. The second and third are actively cooled structures for the reflective cavity walls using a duct enclosing a gaseous coolant and a pipe enclosing a liquid coolant, respectively. These three models are used to study the effectiveness of cooling for the reflective cavity walls. The fourth model is of the absorptive cavity, which includes the embedded preheating graphite pipe for the liquid tin HTF. The detailed design of each wall structure, including layer thicknesses and material properties, was provided in section 4.4.

Please note that this simulations set is a preliminary investigation on cavity wall temperature by analysing an elemental section on the wall using average energy flux quantities without accounting for the non-uniform radiative flux distribution on the walls. Transient operational effects are also not accounted for, as the cavity wall model is assumed to be in a thermodynamic equilibrium with its surroundings.
Figure 6.19. The four numerically studied structures of the cavity walls (all wall thicknesses are set as 0.55 m as described in section 4.4). (a) Reflective cavity wall without any active cooling. (b) Gas-cooled reflective cavity wall. (c) Liquid-cooled reflective cavity wall. (d) Absorptive cavity wall comprising the liquid tin preheater.

The structured meshes were generated for the four models using quadrilateral cells as displayed in Figure 6.20. Mesh-independence tests were performed for each model following the similar procedure adopted in the previous simulations. Mesh grade number 2 of each model was guaranteed to deliver grid-independent solutions as shown from the results in Table 11.
Figure 6.20. Generated mesh for the four cavity wall models. (a) Reflective cavity wall without any active cooling. (b) Gas-cooled reflective cavity wall. (c) Liquid-cooled reflective cavity wall. (d) Absorptive cavity wall comprising the liquid tin preheater.

6.4. Summary

In this chapter, the novel receiver was modelled numerically using five different sets of CFD simulations to evaluate its energy performance and inspect the feasibility of its main design features. The numerical modelling strategies and methods were described in depth, to some extent, with reference to the characteristics of the involved fluid flows to emphasis and justify the suitability of the modelling selections for the developed receiver. Solution independence tests from spatial, temporal, and angular
Table 11. Results of the mesh-independence tests performed on the cavity walls simulations.

<table>
<thead>
<tr>
<th>Cavity Wall Configuration</th>
<th>Model</th>
<th>Mesh Grade</th>
<th>Number of Cells</th>
<th>Mass-Weighted Average Static Temperature of the Domain (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reflective</td>
<td>No active cooling</td>
<td>1</td>
<td>31,090</td>
<td>1155.8717</td>
</tr>
<tr>
<td></td>
<td></td>
<td>2</td>
<td>74,244</td>
<td>1155.1253</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3</td>
<td>426,044</td>
<td>1155.0976</td>
</tr>
<tr>
<td></td>
<td>Gas-cooled using cavity fluid</td>
<td>1</td>
<td>30,688</td>
<td>684.18997</td>
</tr>
<tr>
<td></td>
<td></td>
<td>2</td>
<td>74,319</td>
<td>677.07947</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3</td>
<td>439,523</td>
<td>677.04074</td>
</tr>
<tr>
<td></td>
<td>Liquid-cooled using liquid tin</td>
<td>1</td>
<td>45,974</td>
<td>495.04835</td>
</tr>
<tr>
<td></td>
<td></td>
<td>2</td>
<td>84,551</td>
<td>495.05185</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3</td>
<td>495,017</td>
<td>495.04994</td>
</tr>
<tr>
<td>Absorptive</td>
<td>Liquid tin preheater</td>
<td>1</td>
<td>60,842</td>
<td>770.76697</td>
</tr>
<tr>
<td></td>
<td></td>
<td>2</td>
<td>97,647</td>
<td>770.64842</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3</td>
<td>568,971</td>
<td>770.41406</td>
</tr>
</tbody>
</table>

discretisation of the computational domain were presented for each simulations set to guarantee an acceptable level of fidelity for energy and temperature results from the model. The developed computational model addressed several aspects of the solar receiver concept, which were either simplified or neglected in the analytical model described in the previous chapter. Among these aspects were accounting for the transient development of the gravity-driven HTF film and buoyancy-driven cavity fluid prior to reaching pseudo-steady-state conditions. The developed CFD solution also modelled the interaction between radiation and the complex film flow by coupling DOM with the transient CFD solution. The model also featured accounting for specular reflections and volumetric absorption by semi-transparent media, such as the ceramic window and liquid metal HTF. Moreover, temperature-dependent optical and thermo-physical properties were computed using experimentally measured values.

Secondary steady-state simulation sets were also described to investigate the thermal feasibility of the suggested ceramic window and cavity walls designs in Chapter 4.

In the next chapter, the developed models in Chapters 5 and 6 will be used to demonstrate performance of the novel receiver at the 20 MW$_{elec}$ scale. Validation of the employed numerical models for the studied case will also be presented, while results from both models will be compared to study the applicability of the developed analytical model and sensitivity of its results to the employed assumptions.
Chapter 7. Performance Evaluation of the Proposed Receiver

In this chapter, the performance of the proposed receiver will be evaluated at the scale of 20 MW_{elec} CSP power plant using the developed analytical and numerical methods described in the Chapters 5 and 6. The overall receiver efficiency, defined in Chapter 5, will be used as the performance metric to study the influence of selected design parameters in a set of parametric studies and sensitivity analyses. The significance of the results will be discussed to deliver insights about the potential and limitations of the proposed receiver concept at its two cavity wall configurations, in addition to scrutinising the modelling methodologies developed in this thesis.

7.1. Analytical results

The analytical model described in Chapter 5 is used to investigate the performance of the receiver and study its properties. The model is solved sequentially, without using any iterative method, to compute the quasi-steady-state energy breakdown inside the receiver. The computed collected energy at each case is then used to evaluate the overall receiver efficiency using Equation (18). This solution procedure is iterated at different values of selected parameters – including the concentration ratio, convection heat transfer coefficient, and optical properties – to study their influences on the overall receiver performance and draw conclusions on the feasibility of the proposed concept.

7.1.1. Energy performance

The rates of energy breakdowns inside the receiver at both configurations are demonstrated by the Sankey diagrams in Figure 7.1. From the evaluated rates of energy absorption/collection at the given receiver temperature (1673 K), the absorptive cavity showed a superior overall efficiency of 73% compared to the reflective cavity at only 32%. Another general observation is that the reflective loss is greater than the emissive loss in the reflective cavity, while it is the opposite in the absorptive cavity. This is explained by the higher effective reflectance and lower radiation temperature of the reflective cavity than of the absorptive cavity.

Although the reflective cavity captured >80% of the transmitted concentrated solar radiation through the window, about 55.6% of the captured energy was absorbed by
the reflective walls. The reason behind this significant loss mechanism is large area ratio between the cavity wall and HTF film surface of (~4.66), while the emissivity of the liquid tin is only 2.289 times greater than the cavity walls. This has resulted in the majority (67.1%) of the secondary reflections being absorbed the cavity walls. In the absorptive cavity, >90% of the transmitted solar radiation was captured; however, only 8% of the captured energy were lost through the cavity walls.

![Sankey diagram](image)

Figure 7.1. Power flow through the receiver from the analytical model at the specified solar input and receiver temperature (1673 K). (a) Reflective cavity. (b) Absorptive cavity. Please note that in these Sankey diagrams, the window loss represents the attenuation, by external reflection and absorption, of the concentrated input solar radiation. Energy collection routes are marked with green; energy loss routes are marked with red.

The significant wall absorption by the reflective walls marks a conceptual challenge to the reflective cavity, as active cooling in this case would be necessary. Theoretically, a full recovery of this wall loss would increase the efficiency up to 71.5%, which is still lower than the absorptive case by ~1.5%. Recovering the energy from the walls is more challenging in the reflective cavity than in its absorptive counterpart. Unlike the absorptive cavity, the reflective walls should be maintained at a lower temperature to maintain their reflectance, making it more challenging to use liquid metals as the coolant given their relatively high freezing temperatures. Additionally, the
containment materials of liquid metals, such as graphite, are typically absorptive with rough surface finish, which may require further treatment to sustain the specular reflectiveness of the interior walls. The coolant ducts/pipes will also need to be fully immersed inside the walls to preserve the flatness and specularity of the internal wall surfaces. Therefore, it is more likely that a secondary fluid, such as the cavity fluid, will be used for cooling the reflective walls; hence, recovering the coolant energy back to the hot HTF is conceptually less efficient than in the absorptive cavity. A numerical study of different wall cooling options will be presented in section 7.2.8.

Instead of recovering the energy from the reflective walls ($\dot{Q}_w$) directly during the nominal operation, it is more likely to utilise its energy to maintain the liquidity of the HTF during the diurnal cycle. For instance, if we assume an 80% efficient heat transfer from the reflective walls to the cavity fluid ($\eta_{w\rightarrow cf}$) for a nominal duration of six hours ($t_{sun}$), followed by an 80% efficient heat transfer to the liquid tin ($\eta_{cf\rightarrow htf}$) for a nighttime duration of 18 hours ($t_{night}$), then the resultant energy would be sufficient to cover a temperature drop ($\Delta T$) of up to 225 K at the studied scale, as estimated from:

$$\eta_{w\rightarrow cf} \eta_{cf\rightarrow htf} \dot{Q}_w t_{sun} = (\dot{m}c_p\Delta T)_{htf} t_{night}$$  \hspace{1cm} (125)

Mitigating this conceptual limitation of the reflective cavity could be implemented by enhancing the solar absorptance of the liquid metal HTF or by using a highly reflective and thermal resistant lining material, which can maintain reflectance >0.95 at ultra-high temperatures. The former approach is possible by employing colouration of the liquid metal (Hou et al., 2018) or addition of ceramic absorbing particles (Phelan et al., 2013). Enhancing the solar absorptance of liquid tin from 0.2289 to 0.85 was found to reduce the wall absorption loss in the reflective cavity by 76.5% and improve the capturing of solar radiation by 9.5% as demonstrated in Figure 7.2. However, this performance is still lower than of an equivalent absorptive cavity without any manipulation of the HTF optical properties. A third approach is optimising the cavity geometry with minimal cavity walls to HTF surface area ratio, while maintaining a sufficient cavity depth to minimise the other thermal losses escaping through aperture.
Recalling the importance of concentrating the solar radiation from the theoretical analysis presented in section 2.1.1, it was concluded that higher concentration ratios are fundamental to sustain the receiver at ultra-high temperatures, whilst maintaining a high power output. Here, the influence of the concentration ratio is studied for the proposed receiver at its two configurations. As demonstrated in Figure 7.3, the energy efficiency gains from increasing the concentration ratio is not linear, as the curve plateau towards ultra-high concentrations. Therefore, there is a concentration ratio level for each receiver, beyond which further concentration may not be economically justifiable. Temperature exhibited negligible influence on the trend of the reflective cavity as illustrated by the almost coinciding curves in Figure 7.3(a). However, the curves of the absorptive cavity were found to plateau at greater concentration ratios when the temperature level increased as illustrated in Figure 7.3(b). This reflects the significance of the emissive losses in the absorptive cavity configuration. This also delivers an important insight about the theoretical feasibility of facilitating high solar concentration, typically by incorporating secondary optics, for cavity receivers, as augmenting the concentration ratio beyond 5000 suns may only deliver reasonable gains for absorptive cavities at ultra-high temperatures.
7.1.3. Significance of natural convection

The impact of the buoyancy-driven flow of the cavity fluid on the heat transfer at the internal surface of the cavity is studied here using the semi-empirical analysis presented in section 5.8. The evaluated mean temperatures and convective heat transfer coefficients are presented in Table 12.
The convective effect of the buoyancy-driven cavity fluid is found to be different per configuration. In the reflective cavity, the walls temperature (800 K) is considerably lower than the HTF film temperature (1236.5 K), which resulted in a mean cavity temperature (910.4 K) lower than the absorber/film temperature. Consequently, the convective effect of the cavity fluid flow results in extracting heat from the HTF film and, hence, is considered as an energy loss mechanism. However, the absorptive cavity walls are included as an absorber, while maintained at a marginally higher temperature (1623.6 K) to the mean HTF film (1560.84 K). This small temperature difference allowed the hot window to drive the mean cavity temperature (1628.7 K) to be slightly higher than the absorber surfaces (Table 12). Therefore, the convective effect of the cavity fluid here would generally transfer heat from the window to the absorber surfaces and, hence, is considered an energy recovery, and passive window cooling, mechanism. Nevertheless, this favourable heat transfer mechanism is trivial at the given scale to carry out substantial benefits to the receiver. In section 7.2, these steady-state results will be evaluated using the numerical solution described in Chapter 6 and compared to verify the drawn conclusions here.

Table 12. Computed steady-state mean temperatures and heat transfer coefficients from the analytical solution.

<table>
<thead>
<tr>
<th>Computed Quantity</th>
<th>Cavity Wall Configuration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Absorber temperature, ( T_{abs} ) (K)</td>
<td>Reflective</td>
</tr>
<tr>
<td>Window temperature, ( T_{win} ) (K)</td>
<td>1236.5</td>
</tr>
<tr>
<td>Cavity temperature, ( T_{cav} ) (K)</td>
<td>910.4</td>
</tr>
<tr>
<td>Convection coefficient at the absorber surface(s), ( h_{abs} ) (W.m(^{-2}).K(^{-1}))</td>
<td>1.1577</td>
</tr>
<tr>
<td>Convection coefficient at the window, ( h_{win} ) (W.m(^{-2}).K(^{-1}))</td>
<td>2.5172</td>
</tr>
</tbody>
</table>

*The negative sign indicates an opposite heat transfer (energy gain) due to \( T_{cav} > T_{abs} \).

The effect of convection on the energy efficiency of the receiver is studies by varying the value of the convection coefficient at the absorber surface(s) up to 100 Wm\(^{-2}\)K\(^{-1}\). According to DeAngelis et al. (2018), the convection coefficient value for a sealed cavity can range between 0-15 Wm\(^{-2}\)K\(^{-1}\). This value may become slightly higher in the proposed concept when accounting for the shearing effects at the interface between the liquid metal film and cavity fluid. The interfacial viscous effects are
ignored in the analytical analysis but will be investigated by the numerical model and proved to pose minimal impact on natural convection in section 7.2.5. For open cavities, the effect is likely to be considerably higher depending on the cavity design and aperture orientation and location as discussed previously in sections 4.2 and 4.3.

The results of the convection sensitivity study is presented in Figure 7.4. Since the temperature difference $T_{abs} - T_{cav}$ is the main drive for the buoyancy-driven flow in the cavity, the convection effect (i.e. slopes in Figure 7.4) is more significant in the reflective cavity than in the absorptive cavity. This loss mechanism in the reflective cavity is shown to become more consequential at higher temperatures, particularly for open cavities. In contrast, the favourable energy recovery mechanism in the absorptive cavity was shown to diminish at high temperatures before converting into a loss mechanism at temperatures higher than the window temperature. Nevertheless, for a window-sealed cavity, the convective effect of the buoyancy-driven flow are shown to pose negligible influence on the energy efficiency of the cavity receiver.

### 7.1.4. Significance of the optical properties

The optical properties of the liquid metal and cavity walls lining material are studied here to investigate their influence on the performance of the cavity receiver at its two wall configurations. As shown in the plotted results in Figure 7.5, the emissivity of the exposed liquid metal showed a crucial impact on the performance of the reflective cavity, as the energy conversion efficiency was estimated to increase by 348% when the HTF emissivity increases from 0.1 to 0.9. This sensitivity of receiver performance to the liquid metal’s emissivity was found insignificant in the absorptive cavity, as the energy conversion efficiency was found to increase by only 8.3% the HTF emissivity increases from 0.1 to 0.9. These results are compared with literature data evaluated by DeAngelis et al. (2018) for a liquid tin based tubular receiver. The literature data matched the general trends of the analytically evaluated data, while the absorptive case presented a comparable efficiency; however, with a reduced concern over the development of hot spots on the absorber surfaces. The presented results settle the contradicting conclusions in the literature about the importance of absorber emissivity on the efficiency of a cavity receiver, as it is now clarified that the emissivity would only pose a significant influence when the cavity walls are reflective.
Figure 7.4. Influence of the convection coefficient ($h_{abs}$), at various receiver temperatures, on the overall efficiency of the proposed receiver at its (a) reflective and (b) absorptive cavity wall configurations. $h_{abs}$ is displayed here as an absolute quantity. Thus, the lines of the absorptive cavity show positive slopes, as heat is transferred to, instead of from, the HTF.

The reflectance of the cavity wall lining material is also studied with results presented in Figure 7.6. The effect of wall reflectance was found to be trivial (<3 pp.) at low reflectance up to 0.5 for both configurations. At higher reflectance, the receiver performance change more considerably based on the configuration; when wall reflectance increases from 0.5 to 1, the receiver efficiency is estimated to increase by 216% in the reflective cavity and decrease by 34.6% in the absorptive cavity. The plotted trends reveal a hypothetical case of the reflective cavity being more efficient than its absorptive counterpart when both have a walls reflectance close to unity. This confirms the conceptual dependence of the reflective cavity’s feasibility on developing lining materials with reflectance close to unity at ultra-high temperatures. The
reflectance of silver, the most reflective material across the visible spectrum, was reported by Ujihara (1972) to decrease from 0.972 to 0.939 when temperature increases from 800 K to 1234 K (just before its melting point) at a spectral wavelength of at 0.69 μm. This decrease corresponds to a receiver efficiency reduction of ~8%. For other metallic lining materials, this reduction is expected to become much more significant due to their higher dependence of reflectance on the temperature as discussed previously in section 3.4.3. This emphasises the importance of maintaining the walls at the lowest possible temperature.

![Graph](image1.png)

**Figure 7.5.** Influence of liquid metal emissivity ($\varepsilon_{htf}$) on the receiver efficiency at its (a) reflective and (b) absorptive cavity wall configurations. Data is compared with a literature tubular cavity receiver based on liquid tin (DeAngelis et al., 2018); hence, the reported literature emissivities are of the receiver tubes instead of the liquid metal itself.
In view of the results from the optical properties parametric studies, it is concluded that the absorptive cavity, which operate at low cavity wall reflectance, is characteristically insensitive to the optical properties of its opaque internal surfaces, while the optical properties are of paramount importance to the efficiency of the reflective cavity. This conclusion will be examined against numerical simulations of the receiver at each configuration in section 7.2.7.

![Figure 7.6. Influence of the cavity walls reflectance \((1 - \varepsilon_w)\) on the overall efficiency of the proposed receiver at its two cavity wall configurations. The typical reflectance values are highlighted for absorptive and reflective metallic materials.](image)

7.2. Numerical results

Pseudo-steady-state results from the CFD simulations described in Chapter 6 are presented here to scrutinise the conclusions drawn from the analytical model and verify its validity as a computationally inexpensive tool to study the proposed cavity receiver. Validation of the employed sub-models is also presented to verify guarantee the high-fidelity of the numerical results.

7.2.1. Solution independence study

The independence of the developed numerical solution from spatial and temporal discretisation are checked by running time-step and mesh sizing independence tests. The angular discretisation of the domain used by the DOM is also checked using a separate independence test.
7.2.1.1. Time-step independence

To check the independence of the numerical solution from the prescribed fixed $\Delta ts$ values, the buoyancy simulations were iterated at different $\Delta ts$ sizes, while the resulted average speed of the cavity fluid and heat transfer coefficient on the inclined surface are evaluated and compared as shown in Table 13. As shown in tabulated results, the prescribed $\Delta ts$ values – 0.377s for the reflective cavity and 0.75s for the absorptive cavity – guaranteed deviations in results $<$0.5% from smaller time-step sizes.

Table 13. Results of the time-stepping-independence tests performed on the buoyancy simulations. Same mesh grid (defined in the next section as grade 4) is used for all simulations.

<table>
<thead>
<tr>
<th>$\Delta ts$ (s)</th>
<th>Average Speed of Cavity Fluid (m.s$^{-1}$)</th>
<th>Average Heat Transfer Coefficient on the Inclined Surface (W.m$^{-2}$.K$^{-1}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Reflective</td>
<td>Absorptive</td>
</tr>
<tr>
<td>0.150</td>
<td>0.6071</td>
<td>-</td>
</tr>
<tr>
<td>0.377</td>
<td>0.6069</td>
<td>0.2125</td>
</tr>
<tr>
<td>0.750</td>
<td>0.6215*</td>
<td>0.2115</td>
</tr>
<tr>
<td>1.000</td>
<td>-</td>
<td>0.2131</td>
</tr>
</tbody>
</table>

*To prevent divergence whilst solving the momentum and energy equations across large $\Delta ts$, conservative under-relaxation factors were employed to dampen the convergence instabilities.

Time-stepping in the film simulations was made variable to enable automatic control over the numerical instabilities as the solution progresses. The $\Delta ts$ interval is controlled through prescribed values for the Courant number ($C$) (Courant et al., 1928). $C$ is a dimensionless parameter, which represents the average time for a travelling information in the transport equations to stay in a single mesh cell. Thus, $C$ is expressed as follows:

$$C = \frac{v \Delta t s}{x_{cell}}$$  \hspace{1cm} (126)

where $x_{cell}$ is the cell length. If an explicit method is used, then $C$ should not exceed unity to avoid resulting in a negative numerical viscosity. However, the used implicit scheme is characteristically less sensitive to numerical instabilities, so $C$ values can safely exceed unity without divergence. Still, the resulting time-step are restricted to sizes close to the Kolmogorov time scale (Kolmogorov, 1941) to preserve the small flow turbulence (Choi and Moin, 1994). The Kolmogorov time scale is defined as follows:
\[ \Delta t_{\text{Kolmogorov}} = \sqrt{\frac{\mu}{\rho \overline{\varepsilon}}} \]  

(127)

where \( \overline{\varepsilon} \) is the average rate of dissipation of turbulent kinetic energy per unit mass, which can be evaluated for a given flow field, as will be explained in section 6.3.7. Please note from Equation (126) that at a given \( \varepsilon \) value, \( \Delta t \) will variate with the dynamic mesh, which is employed only in the film simulations as will be discussed in the next section. Solution independence from the prescribed \( \varepsilon \) value was checked for the film simulations as presented in Table 14. Solution independence was guaranteed at \( \varepsilon \) values of 20 and 10 for the reflective and absorptive cavities, respectively, as lower \( \varepsilon \) values were shown to change the HTF temperature at the outlet by <0.2%.

Table 14. Results of the time-stepping-independence tests performed on the film simulations. Information about the dynamic mesh adaption employed in these simulations: Mesh coarsening threshold of 0.3 and refining threshold 0.5 were applied here for all simulations.

<table>
<thead>
<tr>
<th>Courant Number</th>
<th>Mean Time-Step Size (ms)</th>
<th>Outlet HTF Temperature (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Reflective</td>
<td>Absorptive</td>
</tr>
<tr>
<td>30</td>
<td>4.753</td>
<td>-</td>
</tr>
<tr>
<td>20</td>
<td>2.814</td>
<td>2.786*</td>
</tr>
<tr>
<td>10</td>
<td>1.566</td>
<td>1.338</td>
</tr>
<tr>
<td>5</td>
<td>-</td>
<td>0.174</td>
</tr>
</tbody>
</table>

To prevent divergence whilst solving the momentum and energy equations across large time-steps, conservative under-relaxation factors were used to dampen the convergence instabilities.

7.2.1.2. Mesh independence

To test the solution independence from the generated mesh, the film simulations were iterated using different refine and coarsen thresholds for the dynamic mesh adaption tool. As shown in Table 15, a coarsen threshold of 0.3 and refine threshold of 0.5 are demonstrated to guarantee independence of outlet HTF temperature from the adapted mesh size with a deviation of <0.2% from the finer mesh.

Table 15. Results of the mesh-independence tests performed on the film simulations. The Courant numbers used in these runs were 20 (reflective cavity) and 10 (absorptive cavity).

<table>
<thead>
<tr>
<th>Mesh Coarsen and Refine Thresholds</th>
<th>Final Mesh Size (Cells)</th>
<th>Outlet HTF Temperature (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Reflective</td>
<td>Absorptive</td>
</tr>
<tr>
<td>(0.4,0.6)</td>
<td>8,164,112</td>
<td>6,445,087</td>
</tr>
<tr>
<td>(0.35,0.55)</td>
<td>44,147,080</td>
<td>8,006,066</td>
</tr>
<tr>
<td>(0.3,0.5)</td>
<td>88,772,269</td>
<td>29,026,429</td>
</tr>
<tr>
<td>(0.2,0.4)</td>
<td>105,786,889</td>
<td>51,051,859</td>
</tr>
</tbody>
</table>
Mesh independence was also performed on the fixed meshes used in the buoyancy and steady-state simulations. Five mesh grades were generated using the ‘Proximity and Curvature’ size function in ANSYS Mesh software before being exported to Fluent. This sizing function is used to prioritise mesh refinement near the boundaries and corrugations. Grid-independence of buoyancy films simulations is determined for mesh grade 4, as it resulted in results deviations <2% from the finer grade as shown in Table 16. Nevertheless, mesh grade 3 was sufficient to deliver grid-independent solutions for the steady-state simulations as illustrated in Table 17.

Table 16. Results of the mesh-independence tests performed on the buoyancy simulations. The sizes of time-steps used in these runs were 0.377s and 0.75s for the reflective and absorptive cavities, respectively.

<table>
<thead>
<tr>
<th>Mesh Grade</th>
<th>Number of Cells</th>
<th>Average Cavity Fluid Speed (m.s(^{-1}))</th>
<th>Average Heat Transfer Coefficient on the Inclined Surface (W.m(^{-2}).K(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Reflective</td>
<td>Absorptive</td>
<td>Reflective</td>
</tr>
<tr>
<td>1</td>
<td>631,243</td>
<td>0.5339</td>
<td>0.2611</td>
</tr>
<tr>
<td>2</td>
<td>986,467</td>
<td>0.5768</td>
<td>0.2137</td>
</tr>
<tr>
<td>3</td>
<td>2,367,986</td>
<td>0.6388</td>
<td>0.2113</td>
</tr>
<tr>
<td>4</td>
<td>4,366,271</td>
<td>0.6069</td>
<td>0.2115</td>
</tr>
<tr>
<td>5</td>
<td>7,822,798</td>
<td>0.6005</td>
<td>0.2118</td>
</tr>
</tbody>
</table>

Table 17. Results of the mesh-independence tests performed on the steady-state simulations.

<table>
<thead>
<tr>
<th>Mesh Grade</th>
<th>Number of Cells</th>
<th>Total Absorbed Radiation (MW)</th>
<th>Total Reflection by the Inclined Surface (MW)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Reflective</td>
<td>Absorptive</td>
<td>Reflective</td>
</tr>
<tr>
<td>1</td>
<td>631,243</td>
<td>75.393</td>
<td>82.873</td>
</tr>
<tr>
<td>2</td>
<td>986,467</td>
<td>75.244</td>
<td>82.860</td>
</tr>
<tr>
<td>3</td>
<td>2,367,986</td>
<td>74.727</td>
<td>82.548</td>
</tr>
<tr>
<td>4</td>
<td>4,366,271</td>
<td>74.783</td>
<td>82.713</td>
</tr>
<tr>
<td>5</td>
<td>7,822,798</td>
<td>74.665</td>
<td>82.643</td>
</tr>
</tbody>
</table>

7.2.1.3. Angular discretisation independence

To examine the solution independence from the user-prescribed \(N_\theta \times N_\Phi\) used by the DOM, the steady-state simulations were iterated using different degrees of angular discretisation as shown in Table 18. All runs used the same pixelation 5×5 for the overhang control angles and same mesh size (defined in 7.2.1.2 as “Grade 3”). From the tabulated results, angular discretisation of 5×5 was employed in the simulations, as finer degrees showed deviations <1%. Same angular discretisation was used in the
film simulations, as processing finer degrees \( (5 \times 5) \) were found computationally prohibitive\(^\text{15}\). Angular-discretisation-independence tests of the film simulations were performed at \( \leq (5 \times 5) \) as shown in Table 19, which exhibited \(<1\%\) deviation in results when moving from \( (4 \times 4) \) to \( (5 \times 5) \); hence, it is likely to for the results to deviate less when using finer angular discretisation. Fluent (2016) guidance on angular discretisation specifies a minimum of \( (3 \times 3) \), and up to \( (5 \times 5) \), to be used for sufficiently reliable results for most problems, as finer degrees can be substantially CPU-expensive without significant improvement in results accuracy. In a comparable CFD study of radiative losses from a tubular cavity receiver, angular discretisation of DOM finer than \( (3 \times 3) \) showed \(<0.7\%\) variations in computed heat loss and \(<0.04\%\) in the computed outlet temperature (Craig et al., 2020).

Table 18. Results of the angular-discretisation-independence tests performed on the steady-state simulations. The mesh used in all runs was “Grade 3” as defined in section 7.2.1.2.

<table>
<thead>
<tr>
<th>Number of Solid Angles ( (N_\theta \times N_\phi) )</th>
<th>Total Absorbed Radiation (MW)</th>
<th>Total Surface Reflection on the Inclined Surface (MW)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Reflective</td>
<td>Absorptive</td>
</tr>
<tr>
<td>1\times1</td>
<td>77.485</td>
<td>81.064</td>
</tr>
<tr>
<td>3\times3</td>
<td>75.534</td>
<td>82.394</td>
</tr>
<tr>
<td>5\times5</td>
<td>74.727</td>
<td>82.548</td>
</tr>
<tr>
<td>7\times7</td>
<td>75.559</td>
<td>82.830</td>
</tr>
<tr>
<td>9\times9</td>
<td>75.005</td>
<td>82.642</td>
</tr>
</tbody>
</table>

Table 19. Results of the angular-discretisation-independence tests performed on the film simulations. The Courant numbers used in these runs were 20 and 10 for the reflective and absorptive cavity, respectively. Mesh coarsening threshold of 0.3 and refining threshold 0.5 were applied here for all simulations.

<table>
<thead>
<tr>
<th>Number of Solid Angles ( (N_\theta \times N_\phi) )</th>
<th>Volume-Weighted Average Temperature (K)</th>
<th>Total (Volume-Integral) Emission (MW)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Reflective</td>
<td>Absorptive</td>
</tr>
<tr>
<td>1\times1</td>
<td>905.68209</td>
<td>1587.9148</td>
</tr>
<tr>
<td>3\times3</td>
<td>904.84233</td>
<td>1597.4581</td>
</tr>
<tr>
<td>4\times4</td>
<td>903.82068</td>
<td>1621.3099</td>
</tr>
<tr>
<td>5\times5</td>
<td>903.73731</td>
<td>1627.5532</td>
</tr>
</tbody>
</table>

\(^\text{15}\) To illustrate the increase in computational cost with finer angular discretisation, the convergence computation time, using the described computational resources, for a single steady-state simulation takes 0.662h, 1.198h, and 5.782h for \( (1 \times 1) \), \( (5 \times 5) \), and \( (8 \times 8) \), respectively.
7.2.2. Verification and validation of numerical models

No replicable case was found in the literature comprising a turbulent liquid film flowing over an inclined corrugated surface. Accordingly, the validation of the multiphase and surface tension modelling is done separately from turbulence modelling. Validation of the DOM in modelling radiation inside a cavity is also implemented.

7.2.2.1. Multiphase and surface tension models

The multiphase and surface tension methods described in section 6.3.6, are validated here for modelling a gravity-driven film flowing over a corrugated surface. The modelling methods are used to iterate two literature cases to evaluate the volumetric film thickness profile for comparison with the literature results.

The first reference case (Case A) comprises experimental measurements of the thickness of a vertically flowing silicon oil film over a corrugated surface using the setup demonstrated in Figure 7.7 (Zhao and Cerro, 1992). The sinusoidal properties of the corrugations are displayed in Figure 7.8. The flow conditions of the replicated case are specified by Zhao and Cerro (1992) and denoted in the reference as case number “3S”. The Nusselt film thickness of the flow is 0.75 mm, which is defined as follows:

\[ t_{Nu} = \sqrt{\frac{3\mu \Gamma}{\rho g}} \]  

(128)

The properties of silicon oil are obtained from the reference case, and are given as follows: density (\( \rho \)) = 970.4 kgm\(^{-3}\); dynamic viscosity (\( \mu \)) = 0.1941 kgm\(^{-1}\)s\(^{-1}\); and surface tension (\( \gamma \)) = 0.0214 Nm\(^{-1}\). The measured profile of the film thickness is displayed in Figure 7.9. A mathematical solution based on the perturbation theory is also plotted against the measured values to emphasis the invalidity of analytical approaches in modelling gravity-driven film flows over corrugated surfaces.
Figure 7.7. Experimental apparatus used to measure the volumetric film thickness in the reference Case A (Zhao and Cerro, 1992).

Figure 7.8. Geometric properties of Case A validation model.

Figure 7.9. Reference film thickness profile for Case A. Lines marked with rectangular markers are experimentally measured values by Zhao and Cerro (1992), while lines marked with crosses are theoretically calculated values by Shetty and Cerro (1993) using a perturbation analysis. Film thicknesses are non-dimensionalised here by dividing their values by the Nusselt film thickness.
The second reference case (Case B) is a CFD solution provided by Gu et al. (2004) of a glycerol film flow over a corrugated inclined surface with geometrical properties demonstrated in Figure 7.10. The inlet flow conditions ($\Gamma =34.083 \text{kgm}^{-2}\text{s}^{-1}$) of the replicated case are specified by Gu et al. (2004) and denoted in the reference as case number “Wavy Plate 1”. The selected fluid for comparison is named ‘Glycerol 2’ in the reference case, which refers to 2:1 glycerol-to-water volume ratio. The properties of the fluid are determined in the reference as follows: density of 1227 kgm$^{-3}$; dynamic viscosity of 0.0195 kgm$^{-1}$s$^{-1}$; and surface tension of 0.0671 Nm$^{-1}$. This CFD solution was validated by Gu et al. (2005) using experimental measurements from Zhao and Cerro (1992) and Shetty and Cerro (1993). The computed film thickness from the reference case is displayed in Figure 7.11.
To guarantee solution independence from spatial and temporal discretisation, mesh-independence and time-stepping-independence test simulations were implemented for the two validation cases with results shown in Table 20 and Table 21, respectively. Similar dynamic gradient adaption method, based on gradients of volume fraction of the liquid film, was employed in both validation cases. Accordingly, the refine and coarsen thresholds and Courant number were used to represent the dynamically adapting mesh and variable time-stepping, respectively.

The free surface (interface) position of the liquid film is evaluated relative to centreline of the sinusoidal corrugations profile. Due to the numerical diffusion at the interface, the free surface is tracked at a volume fraction of 0.5. Therefore, uncertainty values are presented next to the computed free surface positions to represent the thickness of the interfacial numerical diffusion layer – defined here as the thickness of the interfacial region with volume fraction values ranging from 0.1 to 0.9. Mesh-size-independence was guaranteed for both cases at normalised coarsen and refine thresholds of 0.3 and 0.5, respectively, as lower thresholds resulted in film thickness deviation of <1%. Time-step-size-independence was also guaranteed for both cases at Courant number of 20.

The results of the validation cases are compared to their corresponding reference solutions in Table 22. The free surface position of the liquid film is again used as response quantity, and was evaluated at three different locations on the corrugations resembling the crest, trough, and at the centreline. Generally, the results followed the same trends of the reference with deviations <10%.

Table 20. Results of the mesh-independence tests performed on the multiphase validation cases. The Courant number used in these runs was 20 all simulations.

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Mesh Coarsen and Refine Thresholds</th>
<th>Final Mesh Size (Cells)</th>
<th>Liquid Free Surface Position at 20 mm from Inlet (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>(0.4,0.6)</td>
<td>140,952</td>
<td>2.1522 ±0.201</td>
</tr>
<tr>
<td></td>
<td>(0.3,0.5)</td>
<td>165,746</td>
<td>2.1383 ±0.158</td>
</tr>
<tr>
<td></td>
<td>(0.2,0.4)</td>
<td>180,509</td>
<td>2.1234 ±0.135</td>
</tr>
<tr>
<td></td>
<td>(0.4,0.6)</td>
<td>42,081</td>
<td>0.725 ±0.061</td>
</tr>
<tr>
<td></td>
<td>(0.3,0.5)</td>
<td>129,651</td>
<td>0.727 ±0.057</td>
</tr>
<tr>
<td></td>
<td>(0.2,0.4)</td>
<td>157,497</td>
<td>0.725 ±0.057</td>
</tr>
</tbody>
</table>
Table 21. Results of the time-stepping-independence tests performed on the multiphase validation cases. Normalised coarsen threshold of 0.3 and refine threshold 0.5 were applied here for all simulations.

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Courant Number</th>
<th>Mean Time-Step Size (ms)</th>
<th>Liquid Free Surface Position at 20 mm from Inlet (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>A</td>
<td>30</td>
<td>1.314</td>
<td>2.2129 ±0.260</td>
</tr>
<tr>
<td>A</td>
<td>20</td>
<td>0.921</td>
<td>2.1383 ±0.158</td>
</tr>
<tr>
<td>A</td>
<td>10</td>
<td>0.585</td>
<td>2.1429 ±0.202</td>
</tr>
<tr>
<td>B</td>
<td>30</td>
<td>3.004</td>
<td>0.727 ±0.058</td>
</tr>
<tr>
<td>B</td>
<td>20</td>
<td>2.854</td>
<td>0.727 ±0.057</td>
</tr>
<tr>
<td>B</td>
<td>10</td>
<td>1.629</td>
<td>0.726 ±0.057</td>
</tr>
</tbody>
</table>

Table 22. Comparison of results from the multiphase validations cases with their corresponding values from the reference solutions. The study model runs were implemented using a Courant number of 20, coarsen and refine thresholds of 0.3 and 0.5, respectively.

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Longitudinal Distance from Inlet (mm)</th>
<th>Location on the Corrugation</th>
<th>Liquid Free Surface Position (mm)</th>
<th>Relative Discrepancy</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Study Model</td>
<td>Reference Solutions*</td>
</tr>
<tr>
<td>A</td>
<td>15.8750</td>
<td>Midway</td>
<td>1.179 ±0.158</td>
<td>1.218</td>
</tr>
<tr>
<td>A</td>
<td>17.4625</td>
<td>Trough</td>
<td>1.121 ±0.158</td>
<td>1.199</td>
</tr>
<tr>
<td>A</td>
<td>20.6375</td>
<td>Crest</td>
<td>2.161 ±0.158</td>
<td>2.206</td>
</tr>
<tr>
<td>B</td>
<td>20</td>
<td>Midway</td>
<td>0.727 ±0.057</td>
<td>0.773</td>
</tr>
<tr>
<td>B</td>
<td>25</td>
<td>Crest</td>
<td>0.608 ±0.057</td>
<td>0.664</td>
</tr>
<tr>
<td>B</td>
<td>31</td>
<td>Trough</td>
<td>0.760 ±0.057</td>
<td>0.746</td>
</tr>
</tbody>
</table>

*Experimental measurements from (Zhao and Cerro, 1992) and CFD solution from (Gu et al., 2004) were used as reference for Cases A and B, respectively.

7.2.2.2. Turbulence model

In section 6.3.7.5, the Reynolds Stress Model (RSM), based on $K - \omega$ scale equations to solve the pressure-strain term ($\bar{\phi}_{ij}$) in Equation (99), was selected to model the turbulence of the gravity-driven HTF film flow. To verify this selection, a reference solution is generated using the \textit{algebraic Wall-Modelled Large Eddie Simulation} (WM-LES) method described by Shur et al. (2008), which is a modification of the LES approach discussed in section 6.3.7.1 to enable solving wall-bounded flows at high Reynolds numbers that otherwise would be computationally prohibitive. This approach has already been validated and recommended for unsteady flows over periodic channel flows, which comprise 3D turbulent flow features (Bae et al., 2019, Carton de Wiart et al., 2018).
The verification simulations used the same multiphase and surface tension modelling described in the main study and same corrugation profile used in the proposed cavity receiver. The verification case here is similar to the transient film simulations without the radiation and energy modelling to minimise the computational cost. The simulations were iterated using the WM-LES methods along with a set of Reynolds-averaged Navier-Stokes turbulence models, including the RSM $K - \omega$ based model. The inlet velocity of the liquid film was prescribed as 0.23 ms$^{-1}$, while each simulation is executed for 3 simulation seconds, which is a sufficient period for the film flow to fully develop past the inlet slit – velocity profile of the film becomes equivalent, per sinusoidal cycle, for the subsequent corrugations downstream. The mean velocity of the film, which controls the residence time of the exposed liquid metal inside the cavity in the main simulation, is evaluated at each case and compared as shown in Table 23. The results demonstrate that the RSM with $K - \omega$ scale equations generated the closest mean velocity to the reference case with a discrepancy of <4%. The results also confirm the general superiority of $K - \omega$ based models over of $K - \varepsilon$ based models, which is in line with observations of a relevant CFD study by Rastan and Sohankar (2018) of a half-corrugated channel flow at Reynolds number of 10,000. In this study, the standard $K - \varepsilon$ model was found to ignore the impeding effects of trough eddies on the main flow, as demonstrated in Figure 7.12, leading to overestimated film speeds. This CFD study used a WM-LES solution, prepared by Mirzaei et al. (2014), as the reference case.

Table 23. Verification using different RANS turbulence models against a reference WE-LES solution. Same Courant number and mesh size are used in all simulations.

<table>
<thead>
<tr>
<th>Turbulence Model</th>
<th>Mean Film Velocity (m/s)</th>
<th>Deviation from WM-LES (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>WM-LES</td>
<td>0.29354529</td>
<td>0.000</td>
</tr>
<tr>
<td>$K - \varepsilon$ (standard)</td>
<td>0.53923001</td>
<td>83.70</td>
</tr>
<tr>
<td></td>
<td>0.34944489</td>
<td>19.04</td>
</tr>
<tr>
<td>$K - \varepsilon$ (RNG)</td>
<td>0.35332170</td>
<td>20.36</td>
</tr>
<tr>
<td>$K - \omega$ (Wilcox’s)</td>
<td>0.33819747</td>
<td>15.21</td>
</tr>
<tr>
<td>SST</td>
<td>0.34637262</td>
<td>17.99</td>
</tr>
<tr>
<td>RSM ($K - \varepsilon$ based)</td>
<td>0.40275774</td>
<td>37.20</td>
</tr>
<tr>
<td></td>
<td>0.34685606</td>
<td>18.16</td>
</tr>
<tr>
<td>RSM ($K - \omega$ based)</td>
<td>0.30290528</td>
<td>3.190</td>
</tr>
</tbody>
</table>
Figure 7.12. Relative pressure (Pa) streamlines of the wavy channel flow modelled using (a) $K - \omega$ and (b) $K - \varepsilon$ models (Rastan and Sohankar, 2018).

For the buoyancy-driven cavity fluid flow, the employed solution procedure, described in section 6.3.4, was validated by Henkes and Hoogendoorn (1995) for turbulent ($Ra = 5 \times 10^{10}$) natural-convection flows in enclosures. The employed SST model, in addition to its two sub-model components (standard $K - \varepsilon$ and $K - \omega$ models), have all been validated in the literature for turbulent natural convection in rectangular cavities, including ones with an inclined surface at various inclination angles (Dworkin et al., 2004, Henkes and Hoogendoorn, 1995, Rundle and Lightstone, 2007, Sharif and Liu, 2003, Walsh and Leong, 2004).

7.2.2.3. Radiation model

The DOM radiation model described in section 6.3.8.4 is validated here for modelling radiation inside a cavity. The references cases are obtained from two solutions. The first is a finite-element solution provided by Mei et al. (2017), which was prepared using Ansys Mechanical – Steady-State Thermal (ANSYS, 2013). The second solution was processed using the Monte Carlo method by Prokhorov and
Hanssen (2004). These reference cases are of an isothermal cylindrical cavity with an inclined base as demonstrated in Figure 7.13. Cases here are defined based on the height of the cylindrical cavity as follows: Case 1 is 4 unit length; Case 2 is 8 unit length. The radius of the cylinder in either case is set as 1 unit length. These cases were iterated here using the DOM integrated Ansys Fluent solver used in the main study. Since Fluent is based on the finite-volume method, the domain had to be filled with a fluid. To disable the fluid’s radiative effects, its absorption and scattering coefficients were prescribed as zeros, while its refractive index is set as unity. The convective effects of the fluid were deactivated by setting its thermo-physical properties at constant values. Moreover, the thermal conductivity of the fluid was set at a negligible, yet a non-zero, value to suppress thermal conduction through fluid. As a result, the heat transfer inside the domain was restricted to the geometry and boundary conditions.

Figure 7.13. Generic geometrical properties of the references cases used to validate DOM in modelling radiation inside an isothermal cavity.

Solution independence from mesh resolution and angular discretisation were performed on the created validation model. Each validation case was iterated using four different meshes (Table 24) and three different degrees of angular discretisation (Table 25). The resulting effective emissivities from all iterations are compared to guarantee mesh independence at 80,883 cells and 88,614 cells for Cases 1 and 2, respectively. Solution independence from angular discretisation was similarly guaranteed at 5×5 for both cases. The effective emissivity ($\epsilon_{eff}$) is evaluated for each case as follows:
\[ \epsilon_{eff} = \frac{\int_0^{A_w} q_{abs} \, dA}{\int_0^{A_{apt}} q_{in} \, dA} \]  

(129)

where \( q_{abs} \) is rate of energy flux absorbed by the cavity walls, \( q_{in} \) is rate of input energy flux to the cavity, \( A_w \) and \( A_{apt} \) are surface areas of cavity walls and aperture, respectively. Note that the evaluated wall emissivity here is independent of temperature due to the isothermality of the cavity.

Table 24. Results of the mesh-independence tests performed on the radiation validation cases. The angular discretisation was fixed at 5×5 in all simulations.

<table>
<thead>
<tr>
<th>Case no.</th>
<th>Cavity Length (Unit Length)</th>
<th>Emissivity of Cavity Walls</th>
<th>Mesh Size (Cells)</th>
<th>Effective Emissivity</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4</td>
<td>0.7</td>
<td>2,093</td>
<td>0.95292</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>17,094</td>
<td>0.95679</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>80,883</td>
<td>0.95791</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>130,315</td>
<td>0.95791</td>
</tr>
<tr>
<td>2</td>
<td>8</td>
<td>0.9</td>
<td>5,133</td>
<td>0.99347</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>26,239</td>
<td>0.99838</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>88,614</td>
<td>0.99856</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>140,383</td>
<td>0.99856</td>
</tr>
</tbody>
</table>

Table 25. Results of the angular-discretisation-independence tests performed on the radiation validation cases. The mesh sizes were fixed 80,883 cells and 88,614 cells for cases 1 and 2 simulations, respectively.

<table>
<thead>
<tr>
<th>Case no.</th>
<th>Cavity Length (m)</th>
<th>Emissivity of Cavity Walls</th>
<th>Number of Solid Angles ((N_\theta \times N_\phi))</th>
<th>Effective Emissivity</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4</td>
<td>0.7</td>
<td>3×3</td>
<td>0.95700</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>5×5</td>
<td>0.95791</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>7×7</td>
<td>0.95841</td>
</tr>
<tr>
<td>2</td>
<td>8</td>
<td>0.9</td>
<td>3×3</td>
<td>0.99833</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>5×5</td>
<td>0.99856</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>7×7</td>
<td>0.99853</td>
</tr>
</tbody>
</table>

The resulted effective emissivities are compared with values obtained from the two reference solutions as shown in Table 26. The DOM approach is considered valid for modelling radiation in isothermal cavities, as it showed <0.5% discrepancies from corresponding references values.
Table 26. Comparison of results between the created validation cases with the corresponding cases from two references.

<table>
<thead>
<tr>
<th>Cavity Length (Unit Length)</th>
<th>Emissivity of Cavity Walls</th>
<th>Effective Emissivity</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>DOM in Fluent (Mei et al., 2017)</td>
<td>Relative Discrepancy</td>
<td>Prokhorov and Hanssen, 2004</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>0.7</td>
<td>0.95791</td>
<td>0.95576</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>0.97531</td>
<td>0.97349</td>
</tr>
<tr>
<td></td>
<td>0.9</td>
<td>0.98896</td>
<td>0.98792</td>
</tr>
<tr>
<td></td>
<td>0.7</td>
<td>0.99813</td>
<td>0.99459</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>0.99838</td>
<td>0.99682</td>
</tr>
<tr>
<td></td>
<td>0.9</td>
<td>0.99856</td>
<td>0.99856</td>
</tr>
</tbody>
</table>

7.2.3. Film continuity

The continuity of the simulated gravity-driven HTF film in the film simulations is visually examined here using volume fraction contours. Recalling the characterisation analysis discussed in section 6.2.1, the corrugation profile is designed to facilitate a continuous film flow within the window between disintegration by excessive acceleration and by stagnation. To emphasise the influence of the corrugations profile on the continuity of the HTF film, the film simulation case of the reflective cavity was iterated using two corrugation profiles. The first profile has a small amplitude-to-wavelength ratio ($a/\lambda = 0.0318311$), which caused the gravity-driven film to disintegrate into separated thinner threads as demonstrated in Figure 7.14(a). These threads are shown to reattach at further points downstream of their separation points, resulting in uncovering longitudinal, aligned with the primary flow direction, oval dry patches between the successive separation and reattachment points. The second profile has a large amplitude-to-wavelength ratio ($a/\lambda = 0.1193662$), which is found to result in creating stagnant fluid regions around the peaks, where the fluid builds up and blocks the decelerated main flow. This is shown in Figure 7.14(b) to results in separating the flow; however, the separated threads are shown to then reattach as the flow accelerates down the corrugations curve, leaving behind oval dry patch that are smaller than the corrugations period as demonstrated by Figure 7.14(c). These observations exhibit the two extreme flow conditions for the gravity-driven film that should be avoided.
Figure 7.14. Volume fraction contour plots displaying two types of film discontinuity over two different corrugation profiles with amplitude-to-wavelength ratios \( a/\lambda \) of (a) 0.0318311 and (b) 0.1193662. (c) Velocity vectors on a single corrugation displaying the creation of an oval dry patch at \( a/\lambda = 0.1193662 \).

The corrugations profile in the reflective cavity case was prescribed with \( a/\lambda = 0.0596831 \), which facilitated a continuous film flow from inlet to outlet as shown in Figure 7.15(a). The absorptive case was prescribed with a larger ratio \( a/\lambda = 0.1094190 \) to correspond to its faster flow, which is shown to successfully maintain the film continuity as shown in Figure 7.15(b). This marks another conceptual limitation for the reflective cavity over its absorptive counterpart, as smaller \( a/\lambda \) translates into a smaller available heat transfer surface area for the direct solar absorption process by the HTF. This is caused by the requirement to reside the exposed HTF inside the cavity for longer times to reach the target temperature than in the absorptive cavity. In this design, at the studied scale, the heat transfer surface area of the exposed film was greater in the absorptive cavity by 21.63\% than in the reflective cavity.
Figure 7.15. Volume fraction contour plots displaying continuous films over the inclined corrugated surface. (a) Reflective cavity with $a/\lambda = 0.0596831$. (b) Absorptive cavity with $a/\lambda = 0.1094190$. The yellow/orange spots (marking regions with volume fractions between 0.7 and 1) are resulted from mesh resolution inaccuracy, which is more detectable in the reflective cavity given its slower flow and requirement for finer mesh across the film thickness. However, the mesh independence tests presented in section 7.2.1.2 guaranteed minimal impact on the temperature results.

As HTF film flows past the inlet, its velocity overshoots to a maximum before stabilising and developing into a fixed cyclic profile over the corrugations. The developing range of the flow depends on the inlet velocity. Consequently, the developing range in the reflective cavity was shorter (~4 corrugation periods) than in the absorptive cavity (~5.5 corrugation periods) as demonstrated in Figure 7.16.
Figure 7.16. Velocity vectors of the HTF film past the inlet slit and along the inclined corrugated surface. (a) Reflective cavity. (b) Absorptive cavity.
The main limitation of numerically modelling the gravity-driven film flow is the unavoidable creation of a numerical diffusion layer at the interface between the HTF and cavity fluid, which is marked by the green to orange regions in Figure 7.17. The interface here is defined using an isosurface of the cells with volume fraction values of 0.5. The thickness of the diffusion layer is minimised using the anti-diffusion method developed by So et al. (2011), while the face fluxes in the implicit VOF formulation, Equation (83), are interpolated using the compressive scheme (Ubbink and Issa, 1999), which is a high-resolution scheme with 2nd order reconstruction.

Figure 7.17. Volume fraction contour plot of the HTF film displaying the thickness of the numerical diffusion layer at the interface.

The spots marked with light-blue/green/yellow/orange in the presented contour plots indicate numerical diffusion resulted from insufficient mesh resolution. This can be moderated by improving the mesh resolution; for example, by lowering the refinement threshold of the dynamic mesh adaption. However, excessive refinement of the film would make the film simulations computationally prohibitive. Nevertheless, the mesh-independence study of the film simulations, presented in section 7.2.1.2, showed that further refinement does not substantially change the
results. To visualise the influence of mesh resolution on the continuity of the HTF film, the reflective cavity film simulation is iterated using a coarser mesh – the normalised coarsening and refinement thresholds of the dynamic gradient adaption were 0.4 and 0.6, respectively. As shown in Figure 7.18, the resulting adjacent HTF layer to the corrugations showed several regions with numerical diffusion resulted from the poor mesh resolution, which did not exist in Figure 7.15(a). Note that the affected areas are predominantly between the crest-to-trough regions, where the film accelerates under the gravitational force.

![Figure 7.18. Volume fraction contour plot (with displayed mesh grid on the surface) showing an example of discontinuous film flow in the reflective cavity due to poor mesh resolution.](image)

The viscous shearing effect of the buoyancy-driven flow on the film is neglected in the film simulations. The verification for ignoring this effect will be explained in section 7.2.6.

### 7.2.4. Temperature distributions

As explained in section 6.3.3, the main purpose of the film simulations is to evaluate the pseudo-steady-state temperature distribution on the HTF film surface produced after undergoing volumetric radiation inside the cavity receiver. In the previous section, the liquid tin film was shown to cover the corrugated inclined surface completely. The thickness of the film was sufficient to be optically opaque and entirely attenuate the incident radiation on its surface, by reflection and volumetric absorption,
before reaching the solid inclined surface. The temperature distribution on the inclined surface was equivalent to the HTF film, given graphite’s high thermal conductivity and presumed perfect insulation. As displayed in Figure 7.19, the temperature distribution revealed no localised hot spots, with temperatures exceeding the HTF film, on the surface; thus, verifying the hypothesis of protecting the solid surface from high radiative flux induced overheating by covering its surface with a liquid metal film.

In Figure 7.20, the mass-weighted average temperature of the HTF film is evaluated and displayed along the inclined surface. The outlet HTF temperatures are evaluated as 1644.7 K and 1677.4 K in the reflective and absorptive cavities, respectively. Accordingly, the results obtained from the film simulations proved that the proposed receiver at this scale was successful to achieve the target outlet temperature (1673 K) with ±5% deviations. The temperature distribution of the HTF film along the inclined surface shown a more uniform gradient in the reflective cavity, which is likely caused
under the effect of the stronger secondary reflections than in the absorptive cavity. The absorptive cavity displayed a more centrally oriented temperature gradient, as shown in Figure 7.19(b), with a greater peak temperature reaching 1860 K, compared to only 1789 K in the reflective cavity. Additionally, the temperature of the film in the absorptive cavity was dropped by ~41 K after flowing past the inlet. This is explained by the low incident radiation on the near inlet region, where the film lost more energy through radiative emission than gained by absorption. This drop in temperature is repeated in both cavities at the near outlet region, as average film temperature dropped before reaching the outlet by 115 K and 182 K in the reflective and absorptive cavities, respectively. These temperature drops may be mitigated by covering their regions with reflective surfaces, which is justifiable given their exposure to relatively low incident radiation as will be demonstrated in section 7.2.5.

The resulted temperature distribution of the cavity fluid, in each cavity configuration, from the buoyancy simulations are displayed in Figure 7.21. Overall, the absorptive cavity displayed a more homogenous distribution close to the analytically estimated $T_{cav}$ more than the reflective cavity. The evaluated volume-weighted average temperature of the cavity fluid in the reflective cavity was 930.28 K, while this figure was 1631.26 K in the absorptive cavity. Comparing these values with the analytically estimated $T_{cav}$ in Table 12, the deviations across the numerical and analytical solutions were 2.14% and 0.16% for the reflective and absorptive cavities, respectively. The higher deviation in the reflective cavity is a result of the greater temperature difference between the walls and the HTF film in the reflective cavity ($\Delta T \approx 363.5$ K) than in the absorptive cavity ($\Delta T \approx 62.76$ K). The hottest surface in the domain, in both receiver configurations, was the passively cooled window. The implications of these temperature distributions inside the cavity on the convective heat transfer by buoyancy-driven cavity fluid flow will be examined in section 7.2.6.
Figure 7.20. Mass-weighted average temperature of the HTF film along the inclined surface. (a) Reflective cavity. (b) Absorptive cavity. The y-axis temperature values are average temperatures across the width of the inclined surface (i.e. average of the xz plane).
Figure 7.21. Temperature (K) distribution of the cavity fluid in the (a) reflective and (b) absorptive cavity. The temperature scales here are set within ±50 K bounds of the analytically evaluated $T_{\text{cav}}$ (marked by the green colour) to display the regional deviations from the estimated average.

7.2.5. Radiation distribution

The pseudo-steady-state results from the film simulations are used here to investigate the radiative flux distributions in the proposed cavity receiver at each configuration. The distributions of the resultant radiation intensity inside the reflective and absorptive cavities are displayed in Figure 7.22. The effect of the greater secondary reflections in the reflective cavity can be visualised by enclosing greater illumination than in the absorptive cavity.
Figure 7.22. Resultant radiation intensity (Wm$^{-2}$) inside the (a) reflective and (b) absorptive cavities. Note that the displayed intensity in these contour plots are the scalar summation of radiation from all sources towards different directions.

In the previous section, it was suggested to cover the terminals of the inclined surface with reflective covers to minimise the emissive losses from the HTF film. From the presented contour plots in Figure 7.23, it can be seen that the outlet terminal region of the inclined is still subjected to a high incident surface radiation (>1 MWm$^{-2}$). However, the inlet region in the absorptive cavity displayed a relatively lower incident surface radiation (<1 MWm$^{-2}$), which may be safely covered to minimise the initial radiative loss, particularly in the absorptive cavity. The flux displayed in Figure 7.23(a) is shown to be wider than in Figure 7.23(b). Therefore, the design of the reflective cavity need to be wider, around the focal point of the concentrated input, than its absorptive counterpart to protect the sidewalls from exposure to excessive incident radiation. This marks a compactness advantage for the absorptive cavity over the reflective cavity.

Figure 7.24 displays the incident radiative flux on the cavity walls. The side regions are found to be exposed to high radiative fluxes, which may require modifications to the cavity design; for example, using a trapezoidal or wider cavity. Apart from this side regions, the rest of the absorptive cavity walls were subjected to radiative fluxes below 1 MWm$^{-2}$, which are at least four times less concentrated than the incident radiative flux on the back surface. This verifies the hypothesis that the walls of the absorptive cavity are subjected to lower fluxes than of the conventional configuration.
Figure 7.23. Distribution of surface incident radiation on the HTF film interface. (a) Reflective cavity. (b) Absorptive cavity. Note that the displayed intensity in these contour plots are the scalar summation of ‘one-way’ reflections and emissions received by the HTF film surface (i.e. reflections and emissions by the film are not accounted for here).
Figure 7.24. Distribution of surface incident radiation on the cavity walls. (a) Reflective cavity. (b) Absorptive cavity. Note that the displayed intensity in these contour plots are the scalar summation of ‘one-way’ reflections and emissions received by the HTF film surface (i.e. reflections and emissions by the film are not accounted for here).
In the analytical model, reflections at the wavy surface of the HTF film were assumed to be diffuse. To examine the validity of this assumption, the steady-state simulation of each cavity configuration was iterated twice using a flat (un-corrugated) inclined back surface with a diffuse fraction of 1 (fully diffuse) and another with a diffuse fraction of 0 (fully specular). The resulting incident radiation on cavity walls are displayed in Figure 7.25. When compared with corresponding plots from the film simulations (Figure 7.24), it was observed that both diffuse cases displayed closer flux distributions than the specular cases.

The specular cases, in Figure 7.25 (b) and (d), display higher incident radiative fluxes on the receiver’s ‘forehead’, or curved wall section, which may initiate thermal stresses on its surface. Therefore, facilitating a diffuse reflection at the HTF film surface is crucial to expand the concentrated solar radiation inside the cavity. Nevertheless, the maximum evaluated flux on the absorptive cavity walls was still ~1 MWm\(^{-2}\), while the incident radiative flux exceeded 2 MWm\(^{-2}\) on the reflective cavity walls. The main contributing factor for the diffusiveness of the film is its surface waviness, which are caused by the flow instability described in section 6.2.1 and flowing over a corrugated surface. This represents another advantage of using surface corrugations, in addition to preserving the film continuity and increasing the heat transfer surface area. This founds an important insight when designing the corrugations, as their profile should be optimised by maximising the amplitude-to-wavelength ratio, to maximise the surface diffusiveness, within the window that preserves the continuity of the film, which was described in section 7.2.3.

Surfaces of metals and still liquids are characterised by their high specularity. However, the specular reflectance of metals, except silver, drop substantially with temperature (Ujihara, 1972). Moreover, reflections from a rippled or wavy liquid surface, such as ocean water surfaces disturbed by the wind, are not specular and are better represented by the diffuse reflection, as they are scattered in a wide range of angles (Fougnie et al., 1999, Lillesand et al., 2015, Marghany, 2020). Nakakubo et al. (2021) engineered a technique to convert specular liquid metals into diffusive media by using electrochemical deposits. Another approach to increase the diffusiveness of liquid metals can be done by adding ceramic particles (Phelan et al., 2013).
Figure 7.25. Distribution of surface incident radiation on the cavity walls using the frozen-approximated HTF film. (a) Reflective cavity with diffuse film. (b) Reflective cavity with specular film. (c) Absorptive cavity with diffuse film. (d) Absorptive cavity with specular film. Note that the displayed intensity in these contour plots are the scalar summation of ‘one-way’ reflections and emissions received by the HTF film surface (i.e. reflections and emissions by the film are not accounted for here).

The area-weighted average incident radiation on the cavity walls is evaluated for each cavity configuration from the film simulations and the diffuse steady-state simulation to quantify the impact of the diffuse HTF film assumption on the radiation results. As shown in Table 27, the discrepancy in the results were <5%. Therefore, the diffuse approximation is considered applicable for the wavy surface of the liquid tin film at the given operating conditions, design, and scale.
It can be noticed that the incident radiation on the cavity walls was slightly underestimated by the diffuse steady-state simulations. A possible reason for this discrepancy is the accounting for the temperature-dependent optical properties (absorption coefficient) of the HTF in the film simulations, so that the volumetric absorption varied spatially along and across the film, while in the steady-state simulations, the surface emissivity of the frozen film was prescribed with a constant value evaluated at the mean film temperature. A possible improvement to the steady-state simulations is to define the surface emissivity by a user-defined function, which correlates the emissivity/solar absorptance of liquid tin to the temperature map produced from the film simulations.

Table 27. Comparison of radiation results between the film simulations and steady-state simulations with different radiation boundary conditions for the frozen film surface.

<table>
<thead>
<tr>
<th>Source of Results</th>
<th>HTF Film Radiation Boundary Condition</th>
<th>Area-Weighted Average Surface Incident Radiation on Cavity Walls (Wm(^{-2}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Film simulations</td>
<td>Volumetric Radiation</td>
<td>Reflective: 1,054,209, Absorptive: 604,701.4</td>
</tr>
<tr>
<td>Steady-state simulations</td>
<td>Diffuse Surface Radiation</td>
<td>Reflective: 1,016,965, Absorptive: 579,750.1</td>
</tr>
</tbody>
</table>

7.2.6. **Natural convection**

The cavity fluid was modelled as an incompressible ideal gas to allow variation of its density with temperature and instigate the buoyancy-driven flow inside the cavity to evaluate the resulting natural convection on the internal surfaces of the cavity. The velocities of the cavity fluid in the buoyancy simulations were sufficiently low (<3.42 ms\(^{-1}\)) to result in Mach numbers <0.01, which are substantially below the compressible threshold at Mach number = 0.3 (Christopher and Sudarshan, 2012). Accordingly, the incompressible ideal gas assumption is considered valid at this scale and temperature difference (Panton, 2013). Generally, the resulted velocity magnitudes from the buoyancy simulations were higher in the reflective cavity than in the absorptive cavity. This is expected due to the higher temperature difference in the reflective cavity (\(\Delta T_{cav,max} \approx 1362\) K) than in the absorptive cavity (\(\Delta T_{cav,max} \approx 673\) K).
From the plotted velocity vectors displayed in Figure 7.27, it was found that the general flow direction of the cavity fluid was opposite in each configuration: anti-clockwise in the reflective cavity and clockwise in the absorptive cavity. Note that the displayed velocities in Figure 7.27 are driven only by the temperature differences, illustrated in section 7.2.4, across the boundaries of the domain, while the interfacial effects from the HTF film flow is ignored. Therefore, as the surface with the highest temperature in the domain, in addition to being located at the bottom of the cavity, the window was a primary driver for buoyancy inside the cavity regardless of its wall configuration. Regarding the implication of this temperature-driven flow on the HTF film and its continuity, the absorptive cavity has an advantage of facilitating a favourable co-current direction at the interface, which contributes in stabilising the gravity-driven film flow (Kapitza, 1965). However, the momentum of the cavity fluid at the interface, in either cavity configuration, is trivial to pose any significant shearing effects on the gravity-driven liquid metal flow; hence, ignoring the interfacial effects in the film simulations is justified.

The viscous effect of the HTF film on the cavity fluid flow is likely to dominate at the interface in favour of the former’s, given its greater momentum and viscosity. Nevertheless, due to the small viscosity of the cavity fluid, this interfacial shearing effect is likely to dominate only at the interface between the fluids. Furthermore, in the absorptive cavity, the temperature-driven buoyancy flow resulted in a cavity fluid flowing co-currently to the HTF film at comparable velocities. Therefore, ignoring the interfacial shearing effects is particularly justified for the absorptive case.

To investigate the effect of interfacial shearing on the buoyancy simulations, the frozen film boundary is modelled as a translationally moving wall with an imposed no-slip condition. The absolute velocity boundary condition of the moving wall was prescribed with a values obtained from the film simulations. The velocity values are extracted at the film interface, which is defined by an iso-surface of regions with a volume fraction = 0.5. The resulted velocity vectors in each configuration are presented in Figure 7.27. When comparing the velocity vectors in Figure 7.27 and Figure 7.27, it can be observed that the general flow directions and velocity magnitudes are still similar with minor differences, including a slight shifting on the
saddle point away from the inclined surface. For a more detailed quantitative analysis of the significance of the interfacial shearing effect, the volume-weighted average cavity fluid velocity and area-weighted average convection heat transfer coefficient on the inclined surface are evaluated and compared for the stationary and moving inclined wall simulations. The results presented in Table 28 show close results with relative discrepancies <6%. In view of these results and the minor impact of natural convection in the energy transfer inside the proposed receiver, the shearing effect on the cavity fluid can be neglected. However, accounting for this shearing effect may be worthwhile at considerably greater relative velocity between the two fluid flows.

Figure 7.26. Velocity vectors of the buoyancy-driven cavity fluid with a stationary frozen film boundary. (a) Reflective cavity. (b) Absorptive cavity.
Table 28. Comparison of buoyancy simulations results between using a stationary and moving inclined wall boundary condition.

<table>
<thead>
<tr>
<th>Inclined Wall Boundary Condition</th>
<th>Average Speed of Cavity Fluid (m.s$^{-1}$)</th>
<th>Average Heat Transfer Coefficient on the Inclined Surface (W.m$^{-2}$.K$^{-1}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Reflective</td>
<td>Absorptive</td>
</tr>
<tr>
<td>Stationary wall</td>
<td>0.6069</td>
<td>0.2115</td>
</tr>
<tr>
<td>Moving wall</td>
<td>0.6364</td>
<td>0.2244</td>
</tr>
</tbody>
</table>

Figure 7.27. Velocity vectors of the buoyancy-driven cavity fluid with a moving frozen film boundary. (a) Reflective cavity. (b) Absorptive cavity.
7.2.7. Verification of the analytical analysis

In this section, the results from the numerical simulation are compared with corresponding values from the analytical model to investigate the latter’s suitability and accuracy in modelling the proposed receiver. The results of the power breakdown inside the receiver are summarised per model in Table 29. Overall, the relative discrepancies between the results of the two models were <10% with the exception of the convective losses, which were of trivial quantities to present any impact on the energy efficiency of the receiver. The analytical model is found to overestimate the reflective losses in both configurations, which is likely a result of the employed diffuse assumption. The significance of the diffuse assumption in contributing into the disparity of results across the two models will be demonstrated in the sensitivity analyses presented later in this section.

Table 29. Summary of results (rates of energy) from the analytical and numerical models.

<table>
<thead>
<tr>
<th>Thermal Power Quantity (MW)</th>
<th>Reflective</th>
<th>Absorptive</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Analytical</td>
<td>Numerical</td>
</tr>
<tr>
<td>Transmission through window</td>
<td>87.416</td>
<td>88.423</td>
</tr>
<tr>
<td>Reflective loss</td>
<td>14.171</td>
<td>13.163</td>
</tr>
<tr>
<td>Emissive loss</td>
<td>0.382</td>
<td>0.398</td>
</tr>
<tr>
<td>Convective loss</td>
<td>0.033</td>
<td>0.135</td>
</tr>
<tr>
<td>Wall absorption</td>
<td>40.458</td>
<td>43.533</td>
</tr>
<tr>
<td>HTF absorption</td>
<td>32.371</td>
<td>31.194</td>
</tr>
<tr>
<td>Recovery from walls</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td><strong>Overall Receiver efficiency</strong></td>
<td>31.82%</td>
<td>30.56%</td>
</tr>
</tbody>
</table>

*This is a one-way attenuation of solar beams by external reflection and material absorption. The rate of absorbed energy by window from the incident radiation on its internal surface is not counted in this value, as they are already accounted in the reflective and emissive losses.

The highest discrepancy across the energy models was the natural convection components as illustrated in Table 29. A comparison of the evaluated convection heat transfer coefficients are presented in Table 30, which reveals that the analytical model has underestimated the natural convection coefficients, at the internal surfaces of the cavity, by up to 14.3%.

240
Table 30. Comparison of the evaluated convection heat transfer coefficients from the analytical and numerical (buoyancy simulations) solutions.

<table>
<thead>
<tr>
<th></th>
<th>Reflective cavity</th>
<th>Absorptive cavity</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Analytical</td>
<td>Numerical</td>
</tr>
<tr>
<td>Mean cavity temperature, $T_{cav}$ (K)</td>
<td>910.36</td>
<td>930.28</td>
</tr>
<tr>
<td>Absorber surfaces, $h_{abs}$ (Wm$^{-2}$K$^{-1}$)</td>
<td>1.1577</td>
<td>1.2939</td>
</tr>
<tr>
<td>Window, $h_{win}$ (Wm$^{-2}$K$^{-1}$)</td>
<td>2.5172</td>
<td>2.5808</td>
</tr>
</tbody>
</table>

The negative sign indicates an opposite heat transfer to the absorbing surfaces.

The most likely reasons behind the inaccuracy of the analytical model in estimating the natural convection coefficients are the literature values used for semi-empirical coefficients in Equation (43). To check the validity of the selected literature correlation by Catton (1978), a set of literature correlations for natural convection inside rectangular enclosures were used to evaluate the heat transfer coefficient. The results are compared in Table 31. The results of the literature correlations show significant disparities with results from Catton’s correlation being the closest to the numerical solution. Although a new correlation might need to be developed for the proposed receiver, which is to be tailored to its design and flow configuration, the insignificance of natural convection compared to other heat transfer mechanisms permits the use of literature correlations to provide rough estimates for natural convection.

Table 31. Comparison of evaluated convection heat transfer coefficients using different semi-empirical literature correlations. The correlation used in the analytical model is Catton (1978).

<table>
<thead>
<tr>
<th>Correlation Source</th>
<th>Reflective $h_{abs}$</th>
<th>Reflective $h_{win}$</th>
<th>Absorptive $h_{abs}$</th>
<th>Absorptive $h_{win}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(Wm$^{-2}$K$^{-1}$)</td>
<td>(Wm$^{-2}$K$^{-1}$)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(Baïri, 2008)</td>
<td>1.0367</td>
<td>2.2541</td>
<td>-0.0094</td>
<td>0.1472</td>
</tr>
<tr>
<td>(Berkovsky and Polevikov, 1974)</td>
<td>1.2965</td>
<td>2.5586</td>
<td>-0.0895</td>
<td>0.1660</td>
</tr>
<tr>
<td>(Catton, 1978)</td>
<td>1.1577</td>
<td>2.5172</td>
<td>-0.0105</td>
<td>0.1642</td>
</tr>
<tr>
<td>(Kuyper et al., 1993)</td>
<td>1.4373</td>
<td>2.8423</td>
<td>-0.0086</td>
<td>0.1767</td>
</tr>
<tr>
<td>(Markatos and Pericleous, 1984)</td>
<td>1.5787</td>
<td>3.1431</td>
<td>-0.0097</td>
<td>0.1961</td>
</tr>
<tr>
<td>(Seki et al., 1978)</td>
<td>0.8414</td>
<td>1.6588</td>
<td>-0.0050</td>
<td>0.1030</td>
</tr>
<tr>
<td>(Velusamy et al., 2001)</td>
<td>3.7865</td>
<td>7.3303</td>
<td>-0.0206</td>
<td>0.4508</td>
</tr>
</tbody>
</table>

The sensitivity analysis presented in section 7.1.4 for influence of the optical properties were repeated using the numerical model to examine the analytical results.
First, the influence of the HTF emissivity on the receiver efficiency is studied by iterating the steady-state simulations using varied values for the surface emissivity of the frozen film from 0 to 1 with results shown in Figure 7.28. Second, the influence of the cavity reflectance on the receiver efficiency is studied in the same way with results shown in Figure 7.29. From the plotted results in the two figures, it can be seen that the trends generally match with maximum discrepancy of 5.7%. The analytical model was found to slightly overestimate the efficiency of the reflective cavity and underestimate the efficiency of the absorptive cavity. This is resulted from underestimating the wall absorption by the analytical model. To study the effect of the diffuse radiation assumption employed in the analytical model, the optical study simulations of the reflective cavity were repeated using a diffuse frozen film surface, which resulted in a closer trend to the analytical model. Therefore, the diffuse assumption is considered the main source for the discrepancy between models. Moreover, this discrepancy was relatively smaller in the absorptive cavity and was maximum when the wall reflectance peaked as illustrated in Figure 7.29(b).

To check the validity of evaluating the physical properties of the liquid metal HTF at the mean flow temperature, the mass-weighted average values of the HTF’s optical and thermo-physical properties were computed from the film simulations and compared to their corresponding constant values used in the analytical model. The inspected thermo-physical properties included the specific heat capacity, density, dynamic viscosity, thermal conductivity, and surface tension. Overall, the disparities in the evaluated values for the thermo-physical properties did not exceed the 1% from the values at the mean flow temperature. However, the disparity across the solutions was 2.64% for the absorption coefficient. This indicates a stronger temperature-dependence of the optical properties than the thermo-physical properties. It was also found that the analytical property values would result in a considerably closer results to their corresponding numerical values, with disparities below 1% in all properties, when they are evaluated at the representative radiation temperature (defined in section 5.7.4 as $T_r$) rather than the mean film flow temperature. The analytical model should be adjusted accordingly, as radiation dominates the heat transfer at ultra-high temperatures.
Figure 7.28. Influence of HTF emissivity on the overall receiver efficiency using the analytical and numerical models. (a) Reflective cavity. (b) Absorptive cavity.
Figure 7.29. Influence of cavity reflectance on the overall receiver efficiency using the analytical and numerical models. (a) Reflective cavity. (b) Absorptive cavity.

7.2.8. Window steady-state temperature

The steady-state temperature of the aperture window ($T_{\text{win}}$) is evaluated here using the simulations described in section 6.3.10.1. The results are presented in Table 32 for three different passive-cooling mechanisms for the window. The wind-induced convection heat transfer coefficient ($h_{\text{wind}}$) imposed as a boundary condition on the external surface of the window is evaluated from Equation (63) as 6.481 Wm$^{-2}$K.

<table>
<thead>
<tr>
<th>Cooling Mode</th>
<th>Reflective Average</th>
<th>Reflective Maximum</th>
<th>Absorptive Average</th>
<th>Absorptive Maximum</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{\text{win}}$ (K) Emission, natural and wind-induced convection</td>
<td>1943.69</td>
<td>2161.91</td>
<td>1909.54</td>
<td>2121.35</td>
</tr>
<tr>
<td>$T_{\text{win}}$ (K) Emission and natural convection</td>
<td>2206.24</td>
<td>2388.85</td>
<td>2179.35</td>
<td>2355.37</td>
</tr>
<tr>
<td>$T_{\text{win}}$ (K) Emission only</td>
<td>2224.10</td>
<td>2404.98</td>
<td>2184.86</td>
<td>2361.04</td>
</tr>
</tbody>
</table>

The distribution of the window’s steady-state temperature at each configuration is displayed in Figure 7.30. These contour plots are displayed for the case with all passive (emissive and convective) cooling mechanisms.
Figure 7.30. Temperature distribution on the window subjected to passive (emissive and convective) cooling mechanisms. (a) Reflective cavity. (b) Absorptive cavity.

The results show that the window can achieve a thermodynamic equilibrium at temperatures below the phase-change temperature of its material (2408 K) at both cavity configurations even without any convective cooling effects. The window temperature is always higher in the reflective cavity than in an equivalent absorptive cavity due to the significantly higher reflective losses in the former, which transmit to the ambient through the window. While the internal natural convection is found to pose negligible cooling effect on the window, the wind-induced convection is showed
substantial cooling effects at a wind speed of 3 ms\(^{-1}\). Although the emissive cooling mechanism seems sufficient at the studied scale to preserve the window’s thermomechanical integrity, this analysis does not account for transient operation and optical inaccuracies, which may lead to overheated regions on the window that exceed its maximum allowable temperature. Since the wind-induced cooling is unreliable, an active cooling, potentially using the cavity fluid, might need to be incorporated in practice to maintain the window temperature at a safe level with a sufficient tolerance to counter any operational uncertainties. At larger scales, or higher concentration ratios, the wind-induced and emissive cooling effects will not scale up equally with the increased incident radiation on the window; the use of an active cooling mechanism will then be necessary to sustain the window against failure.

7.2.9. Cavity walls steady-state temperature distribution

The results of the steady-state temperature distribution across different designs of cavity walls, which were described in section 4.4, are evaluated using the simulations set described in section 6.3.10.2. At the ambient-side boundary, the external surface of the wall is prescribed with a wind-induced convection heat transfer coefficient of 14.85197 Wm\(^{-2}\)K and ambient temperature of 300 K. The evaluated mean wall absorption were prescribed as heat flux sources at the cavity-side boundary as follows: 98.486 kWm\(^{-2}\) for the reflective and 156.718 kWm\(^{-2}\) for the absorptive cavity.

Simulations on the elemental wall model were iterated using a combination of material and temperature of the wall coolant fluid. The resulted mean and maximum temperatures at the internal surface of the wall are presented in Table 33 for each case.

Apart from being a substantial energy loss, the results also show that the reflective wall necessitates active cooling, as its steady-state temperature is likely to exceed the 3000 K as demonstrated in Figure 7.31.
Table 33. Summary of results of the cavity walls temperature simulations. Coolant ducts and pipes were displaced 0.5 m apart in these simulations.

<table>
<thead>
<tr>
<th>Cavity Wall Configuration</th>
<th>Coolant Case Material</th>
<th>Coolant Case Temperature (K)</th>
<th>Cavity-Side Wall Temperature (K)</th>
<th>Average</th>
<th>Maximum</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reflective</td>
<td>No Coolant</td>
<td></td>
<td>3092.06</td>
<td>3092.07</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Molten tin</td>
<td>800.00</td>
<td>1214.59</td>
<td>1381.89</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>505.15</td>
<td>919.69</td>
<td>1087.11</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>293.00</td>
<td>708.14</td>
<td>875.77</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>363.00</td>
<td>778.11</td>
<td>945.72</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Water</td>
<td></td>
<td>600.00</td>
<td>839.89</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>839.89</td>
<td>934.38</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Cavity fluid</td>
<td>800.00</td>
<td>1039.89</td>
<td>1134.37</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>950.00</td>
<td>1189.88</td>
<td>1284.36</td>
<td></td>
</tr>
<tr>
<td>Absorptive</td>
<td>Molten tin</td>
<td>800.00</td>
<td>855.83</td>
<td>935.43</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>1124.34</td>
<td>1179.05</td>
<td>1259.07</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>1448.68</td>
<td>1502.4778</td>
<td>1582.844</td>
<td></td>
</tr>
</tbody>
</table>

Figure 7.31. Temperature distribution within the reflective wall without active cooling. The structure of this wall was illustrated in Figure 6.19(a).

As demonstrated in Figure 7.32, molten tin at 800 K was shown to be ineffective in maintaining the surface temperature of the internal reflective wall below silver’s melting point (~1235 K). Therefore, the operational temperature range for liquid tin in this case would be from its melting point (505.15 K) and below 652 K. Even without considering practical tolerances from these limits, this leaves a maximum temperature difference of 150 K, which is equivalent to a cooling effect of only 7.24 MW, which is <20% of the reflective wall absorption. This emphasizes the conceptual challenge of directly preheating the liquid metal HTF as in the absorptive configuration.
Figure 7.32. Temperature distribution across the reflective cavity wall with liquid tin as a coolant at (a) 800 K and (b) 505.15 K. The black circles mark the inner surfaces of the coolant pipes. The structure of this wall was illustrated in Figure 6.19(c).

A 137.6 kgs\(^{-1}\) flow of liquid water can provide sufficient cooling for the reflective wall with a temperature difference of only 70 K. While recovering this energy back to the HTF is unlikely, it could possibly be used as a by product for a secondary application, such as for preheating water for steam generation. A demonstration of using liquid water coolant from 293 K to 363 K is displayed in Figure 7.33.

A more promising coolant alternative for the reflective wall is the cavity fluid, as it could be used to preheat the liquid metal HTF before entering the direct absorption phase. A 48.85 kgs\(^{-1}\) flow of cavity fluid, which represents third of the cavity volume per second, could cool the walls by 350 K. A demonstration of a case of cooling the reflective wall from 600 K to 950 K is displayed in Figure 7.34.
Figure 7.33. Temperature distribution across the reflective cavity wall with liquid water as a coolant at (a) 293 K and (b) 363 K. The black circles mark the inner surfaces of the coolant pipes. The structure of this wall was illustrated in Figure 6.19(c).
Figure 7.34. Temperature distribution across the reflective wall cooled with the cavity fluid at (a) 600 K, (b) 800 K, and (c) cavity fluid at 950 K. The black rectangles mark the inner surfaces of the coolant ducts. The structure of this wall was illustrated in Figure 6.19(b).

While the demonstrated coolant configuration based on the cavity walls maintained the average temperature of the internal surface below 1200 K, Figure 7.34(c) reveals that the maximum surface temperature would exceed silver’s melting temperature by ~50 K. This could be mitigated by slightly reducing the distance between the coolant ducts from 50 cm to 45 cm, as illustrated in Figure 7.35, which is found to reduce the average and maximum surface temperature to 1156.08 K and 1219.48 K, respectively.

Regarding the absorptive cavity wall, the results of preheating the liquid tin from 800 K to 1448.68 K are presented in Figure 7.36. As no reflective lining is required to be cooled here, the main purpose of the absorptive wall simulations is to ensure that the suggested design and composition materials of the wall are operating at below their maximum allowable, which were specified in section 4.4. The microporous insulation layer, second top layer, is shown to be maintained at temperatures below 1273 K, while the zirconium oxide layer, third top layer, is shown to be maintained at temperatures
Figure 7.35. Temperature distribution across the reflective cavity wall cooled with cavity fluid at 950 K with ducts displaced at 0.45 m apart. The black rectangles mark the inner surfaces of the coolant ducts. The structure of this wall was illustrated in Figure 6.19(b).

below 1923 K. Therefore, the specified materials and thicknesses in section 4.4 are verified for this application. The spacing between the HTF pipes has a significant effect on the temperature distribution, as a 1 m spacing can lead to insulation layers exceeding their maximum allowable temperature (results presented in Appendix A3).

In practice, the spacing between the HTF pipes should be specified based on the incident flux, so that it should be smaller in regions subjected to higher fluxes.
Figure 7.36. Temperature distribution across the wall sections (black lines mark the borders between different materials) of the absorptive cavity cooled by molten tin at (a) 800 K, (b) 1124.34 K, and (c) 1448.68 K. The structure of this wall was illustrated in Figure 6.19(d).
7.3. Summary

The performance of the proposed receiver was investigated using the analytical and numerical models described in Chapters 5 and 6. The results revealed several conceptual challenges to the feasibility of the receiver at its reflective configuration, including its high sensitivity to the optical properties of its components and requirements for costly and complex measures to achieve comparable energy efficiencies to its absorptive counterpart. The absorptive cavity showed superior energy performance, with considerably higher insensitivity to the optical properties, which resulted in comparable efficiencies (>70%) to low-temperature cavity receivers, which range from 45% up to 80% (Le Roux et al., 2014, Gallo et al., 2015, Gueguen et al., 2020, Maurya et al., 2022). Accordingly, it is concluded that the proposed receiver concept is likely to proceed in the future with its absorptive configuration. Although the absorptive cavity still includes indirect heating of the HTF through an intermediate solid absorber, the incident radiative flux on the preheater walls are shown to be considerably lower than in the conventional receiver configuration, which minimises the risk of developing hot spots on its surfaces.

Liquid tin, which was used as the representative liquid metal HTF, displayed promising performance, as it successfully achieved the target outlet temperature at both cavity configurations. The main concern of the directly illuminated HTF film flow is its susceptibility to disintegrate under the gravitational driving force. The use of sinusoidal-corrugated inclined flow was demonstrated as a viable approach to preserve the film continuity, which is crucial in providing an opaque protection layer for the corrugated solid surface from potential thermomechanical damage caused by exposure to highly concentrated solar radiation. The amplitude-to-wavelength ratio of the corrugation profile is marked as a key design parameter in sustaining the continuity of the gravity-driven film, as it was demonstrated to control two film disintegration scenarios: over-acceleration and stagnation.

The buoyancy-driven flow was shown to pose trivial thermal and shearing effects on the HTF, as natural convection inside an enclose cavity is demonstrated to be negligible compared to the radiative heat transfer mechanisms occurring at ultra-high temperatures. Moreover, the convective effect is shown to be trivial to maintain the
interior surfaces of the reflective walls at a temperature below the melting point of its prospective lining material (silver). Accordingly, the reflective cavity walls might require active cooling to sustain the structural integrity and reflectance of the reflective walls. The cavity fluid is demonstrated to be a potential coolant for the reflective cavity walls. Furthermore, the suggested insulation materials, for each cavity configuration, were verified using steady-state simulations.

The proposed transparent ceramic window was shown to reach thermal equilibrium at a temperature lower than its maximum allowable temperature. However, in practice, the transient and cyclic nature of its operation may necessitate active cooling particularly at larger capacities or higher concentration ratios. The pressure loads on the ceramic window is found to be negligible compared to its fracture toughness.

The analytical model, which was developed as a quick and computationally feasible design tool for the proposed receiver was verified using the more computationally intensive CFD model. The CFD model was used to simulate each heat transfer mechanism to evaluate their components and display the distributions of temperature and radiation inside the cavity. Various assumptions made in the analytical model were verified using the numerical results, while the main sub-models of the numerical solution were validated using experimental and numerical data from the literature. The accuracy of the analytical model in modelling the proposed receiver was found sufficient to draw reliable conclusions on the characteristics of the receiver, which was manifested in the parametric studies, such as studying the influence of optical properties. The main weakness of the analytical model is finding suitable coefficients for the semi-empirical correlation used to evaluate the natural convection coefficient. Nevertheless, the insignificance of natural convection in this application moderates the impact of this limitation. In the next Chapter, the analytical model will extend to include a CSP application to exhibit a broad view for the benefits brought to the overall system by operating the solar receiver at ultra-high temperatures.
Chapter 8. Ultra-high Temperature Concentrated Solar Power Plant Demonstration

In theoretical analysis presented in Chapter 2, the significance of maximising the solar concentration along with increasing the receiver temperature was demonstrated, while in Chapter 7, the impact of their interaction was investigated per configuration of the proposed cavity receiver. However, these analyses were focused only on the receiver performance and on minimising its associated energy losses during the solar collection process. However, the benefits of the developed ultra-high temperature receiver can only be presented when it is utilised in an application and compared to conventional systems. Therefore, in this Chapter, the developed analytical model will be used to model the heat addition process in a CSP electricity production application. The purpose of this analysis is to demonstrate the applicability of the developed model in analysing the characteristics of a storage integrated ultra-high temperature CSP plant.

8.1. Solar power plant model

A model is described in this Chapter for a CSP plant based on the novel receiver at the nominal operating conditions. Performance during transient phases or abnormal conditions are not covered by this analysis. Accordingly, comparison with reference plants will be demonstrated at their rated performances (i.e. at their nominal operating conditions). The CSP plant model is composed of four main stages as illustrated in Figure 8.1(a), while the energy flow through the CSP plant is demonstrated in Figure 8.1(b). The significance of each energy loss mechanism is represented by an arbitrary range of typical values of its corresponding component.

8.1.1. Field data

The input solar resource will be held fixed throughout the different cases using the same annual solar resource \( DNI = 2.012 \text{ MWhm}^{-2} \) used in the previous analyses of the solar receiver. Therefore, the daily average incident solar radiation \( G_{\text{helo}} \) on the heliostats is estimated as:

\[
G_{\text{helo}} = A_{\text{helo}} DNI / 365 \tag{130}
\]

where \( A_{\text{helo}} \) is effective surface area of all reflector heliostats in the field.
Three different plant capacities are considered in this study: 20 MW$_{\text{elec}}$, 40 MW$_{\text{elec}}$, and 100 MW$_{\text{elec}}$. Field data, such as heliostats surface areas and tower heights, are obtained based on existing CSP power plants based on the tower receiver technology; a review of current tower receiver plants was provided in section 2.4.1. A summary of the prescribed field data to the model is presented in Table 34. Please note that due to the lack of commercial power plants >20 MW$_{\text{elec}}$ based on the cavity receiver technology, data from plants based on external receivers were used and approximated accordingly. The optical efficiency ($\eta_{\text{o}}$) is fixed at 86% for the three plants.
Table 34. Prescribed field data at three different plant capacities.

<table>
<thead>
<tr>
<th>Electrical Power Rating (MW&lt;sub&gt;elec&lt;/sub&gt;)</th>
<th>Aperture Area of a Single Heliostat Module (m&lt;sup&gt;2&lt;/sup&gt;)</th>
<th>Number of Heliostats</th>
<th>Incident Solar Power on Heliostats (MW)</th>
<th>Tower Height (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>120</td>
<td>1255</td>
<td>150.60</td>
<td>165</td>
</tr>
<tr>
<td>40</td>
<td>140</td>
<td>2516</td>
<td>352.24</td>
<td>200</td>
</tr>
<tr>
<td>100</td>
<td>170</td>
<td>5180</td>
<td>880.60</td>
<td>250</td>
</tr>
</tbody>
</table>

8.1.2. Receiver design

The proposed receiver is represented here with the absorptive cavity configuration given its superior performance demonstrated in Chapter 7. The receiver design is scaled up based on the increased heliostats surface area, which determines the aperture diameter and HTF flow rate. Subsequently, cavity dimensions are determined based on the aperture diameter, while the input velocity of the HTF film through the 20 mm inlet slit thickness is determined based on the flow rate in accordance to the guidance explained in section 4.4. Table 35 displays the main design parameters of the three cavity receivers in this study and their corresponding receiver efficiencies (η<sub>rc</sub>).

Table 35. Calculated design parameters and energy efficiencies for the receiver at the three studied plant capacities.

<table>
<thead>
<tr>
<th>Electrical Power Rating (MW&lt;sub&gt;elec&lt;/sub&gt;)</th>
<th>Diameter of the Receiver Aperture (m)</th>
<th>Inclined Surface Dimensions (m)</th>
<th>Inlet Film Velocity (ms&lt;sup&gt;-1&lt;/sup&gt;)</th>
<th>Receiver Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>4.38</td>
<td>4.80 18.33</td>
<td>0.533</td>
<td>73.01%</td>
</tr>
<tr>
<td>40</td>
<td>6.70</td>
<td>7.20 27.50</td>
<td>0.829</td>
<td>72.96%</td>
</tr>
<tr>
<td>100</td>
<td>10.59</td>
<td>11.10 42.39</td>
<td>1.344</td>
<td>72.92%</td>
</tr>
</tbody>
</table>

8.1.3. Thermal energy storage

The TES here is modelled based on a system developed by Robinson (2017). Assuming negligible conductive losses, which is justified for this design (Robinson, 2018), the efficiency of the storage unit (η<sub>storage</sub>) is evaluated using:

\[
\eta_{storage} = 1 - \frac{P_t \times \left(86400 \frac{s}{day}\right)}{E_{storage}}
\]  

(131)
where $P_r$ is daily average radiative power loss from the storage unit and $E_{storage}$ is energy capacity of the storage unit. $P_r$ is calculated as follows:

$$P_r = \frac{\sigma A_{storage}(T_{high}^4 - T_{low}^4)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2} \left(\frac{d_1}{d_2}\right)^2}$$

where $A_{storage}$ is surface area of storage unit. $T_{high}$ and $T_{low}$ are the top and bottom temperatures for the storage cycle, respectively. The subscripts 1 and 2 here denote the inner and outer surfaces of the spherical storage unit, respectively. The temperature limits of the storage cycle are made equivalent to that of the solar circuit (i.e. $T_{high} = 1673$ K and $T_{low} = 800$ K for the base case of all plants).

The storage energy capacity is composed of three phases for the storage medium: sensible solid capacity, latent capacity, and sensible liquid capacity. Therefore, $E_{storage}$ is evaluated as follows:

$$E_{storage} = (\rho V)_{core}[c_{solid}(T_m - T_{low}) + \xi + c_p(T_{high} - T_m)]$$

where $c_{solid}$ and $c_p$ are the solid and liquid specific heat capacities of the storage medium, respectively. $\xi$ is specific latent heat of storage medium. $T_m$ is melting temperature of the storage medium. In the base case, aluminium is used as the storage medium for the reasons explained by Robinson (2017). The main characteristics of the storage system at each plant capacity are demonstrated in Table 36. It can also be noticed that the storage efficiency increases with the plant capacity due to the reduced surface area to volume ratio of the storage core.

Table 36. Evaluated design parameters and efficiencies for the thermal storage system at the three studied plant capacities.

<table>
<thead>
<tr>
<th>Electrical Power Rating (MW_{elec})</th>
<th>Storage Core Mass (Tonnes)</th>
<th>Storage Capacity (MWh) Buffer Available</th>
<th>Radiative Power Loss (MW)</th>
<th>Storage Hours at Rated Capacity</th>
<th>Storage Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>960</td>
<td>122.3</td>
<td>351.9</td>
<td>3.78</td>
<td>3.8h</td>
</tr>
<tr>
<td>40</td>
<td>2244</td>
<td>285.8</td>
<td>822.3</td>
<td>6.64</td>
<td>6.0h</td>
</tr>
<tr>
<td>100</td>
<td>5607</td>
<td>714.1</td>
<td>2054.5</td>
<td>12.18</td>
<td>6.5h</td>
</tr>
</tbody>
</table>
8.1.4. Energy conversion

The energy extraction cycle presented by Robinson (2017) is similar to the CCGT cycle demonstrated in Figure 8.2. The theoretical energy conversion efficiency can be predicted using the idealised Carnot cycle model:

\[
\eta_{\text{Carnot}} = 1 - \frac{T_{\text{low}}}{T_{\text{high}}}
\]  \hspace{1cm} (134)

Figure 8.2. Simplified layout of the thermal energy storage system integrated with the energy extraction (Robinson, 2017). The high temperature in this demonstrated system is 1800 K.

Carnot approach is an idealistic approach, which does not account for thermodynamic irreversibility of heat engines. Endo-reversible thermodynamics, which was introduced by the works of Chambadal (1957) and Novikov (1958), offers a more realistic approach for estimating the efficiency of a thermodynamic cycle by accounting for the exergy destruction during irreversible heat transfer. The endo-reversible efficiency of a Chambadal–Novikov engine is given by:

\[
\eta_{\text{Chambadal–Novikov}} = 1 - \sqrt{\frac{T_{\text{low}}}{T_{\text{high}}}}
\]  \hspace{1cm} (135)
According to Curzon and Ahlborn (1975), Equation (135) was presented to result in more closely matching efficiencies, than Carnot efficiencies, to the observed efficiencies of real heat engines. The studied real heat engines included West Thurrock coal-fired power plant in the UK ($T_{\text{high}} = 838 \text{ K}, T_{\text{low}} = 298 \text{ K}$), CANDU nuclear power plant in Canada ($T_{\text{high}} = 573 \text{ K}, T_{\text{low}} = 298 \text{ K}$), and Larderello geothermal power plant in Italy ($T_{\text{high}} = 523 \text{ K}, T_{\text{low}} = 353 \text{ K}$).

For CCGT, the actual cycle efficiencies are found to exceed their endo-reversible efficiencies by ~2% at ultra-high temperatures (Hada et al., 2012). Thus, the energy conversion efficiency at the given temperature scale is expressed as follows:

$$\eta_{102\%\text{endo-reversible}} = 1.02 \left[ 1 - \sqrt{\frac{T_{\text{low}}}{T_{\text{high}}}} \right]$$

(136)

Figure 8.3 displays the variations of Carnot and 102% endo-reversible efficiencies with the high temperature for the energy extraction from the storage system described in Figure 8.2. This plot exhibits the substantial conservative nature of the 102% endo-reversible approach compared to the reversible Carnot.

In Table 37, estimates from Equations (134) and (136) are compared with commercial CCGT efficiencies, which verify the endo-reversible expressions for modelling the energy conversion at ultra-high temperatures. Accordingly, the energy conversion ($\eta_{\text{conv}}$) here is modelled using the following conditional expression:

$$\eta_{\text{conv}} = \begin{cases} 
0.94 \left[ 1 - \sqrt{\frac{T_{\text{low}}}{T_{\text{high}}}} \right] & T_{\text{high}} < 1660 \text{ K} \\
1 - \sqrt{\frac{T_{\text{low}}}{T_{\text{high}}}} & 1660 \text{ K} \leq T_{\text{high}} < 1820 \text{ K} \\
1.02 \left[ 1 - \sqrt{\frac{T_{\text{low}}}{T_{\text{high}}}} \right] & T_{\text{high}} \geq 1820 \text{ K}
\end{cases}$$

(137)

8.1.5. Electrical parasitic losses

A fraction of the gross electrical output from the plant is typically used to power some equipment in the plant. The pumping load to circulate the liquid metal HTF between the thermal energy storage and receiver is typically the primary source for electrical power loss. Another major source for electric parasitic losses in CSP plants based on molten salts is electric heat tracing to prevent freezing of molten salts. In the case of
Figure 8.3. Variation of storage characteristics with temperature for a spherical aluminium storage core with a diameter of 24.3 m and low temperature of 300 K (Robinson, 2017).

Table 37. Comparison of evaluated energy conversion efficiencies using different models with commercial CCGT efficiencies at ultra-high temperatures. The low temperature is taken here as 298 K to match the reference low temperature.

<table>
<thead>
<tr>
<th>High Temperature (K)</th>
<th>Commercial CCGT (Mitsubishi, 2023)</th>
<th>Carnot Equation (134)</th>
<th>Chambadal–Novikov Equation (135)</th>
<th>102% Endo-reversible Equation (136)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1423</td>
<td>48.00%</td>
<td>79.06%</td>
<td>54.24%</td>
<td>55.32%</td>
</tr>
<tr>
<td>1523</td>
<td>52.00%</td>
<td>80.43%</td>
<td>55.77%</td>
<td>56.88%</td>
</tr>
<tr>
<td>1623</td>
<td>54.00%</td>
<td>81.64%</td>
<td>57.15%</td>
<td>58.29%</td>
</tr>
<tr>
<td>1673</td>
<td>57.00%</td>
<td>82.19%</td>
<td>57.80%</td>
<td>58.95%</td>
</tr>
<tr>
<td>1773</td>
<td>59.00%</td>
<td>83.19%</td>
<td>59.00%</td>
<td>60.18%</td>
</tr>
<tr>
<td>1873</td>
<td>61.50%</td>
<td>84.09%</td>
<td>60.11%</td>
<td>61.31%</td>
</tr>
<tr>
<td>1973</td>
<td>64.00%</td>
<td>84.90%</td>
<td>61.14%</td>
<td>62.36%</td>
</tr>
</tbody>
</table>

the liquid tin based CSP plant, the hot cavity fluid could possibly be utilised to minimise the anti-freezing parasitic loss from the plant. The total electrical loss \(W_{elec,loss}\) from the plant can be expressed as:

\[
W_{elec,loss} = W_{pump} + W_{parasitic}
\] (138)
where $W_{\text{pump}}$ is pumping load and $W_{\text{parasitic}}$ is secondary parasitic losses, which is assumed as 1% of the rated power. The pump load ($W_{\text{pump}}$) can be estimated from:

$$W_{\text{pump}} = \frac{W_{\text{static}} + W_{\text{circulation}}}{\eta_{\text{pump}}}$$

$$= \frac{\rho_{htf} g H_{\text{static}} A_{\text{pipe}} v_{htf} + f \frac{L}{d_{\text{pipe}}} \frac{v^2}{2g}}{\eta_{\text{pump}}}$$

(139)

where $H_{\text{static}}$ is static head, which is mainly the tower height plus minor head losses of the HTF circuit components. $A_{\text{pipe}}$ is cross-sectional area of the graphite feed pipes. $v_{htf}$ is velocity of the liquid metal HTF. $f$ is Moody friction factor (~0.0366 for a graphite pipe). $L$ is the total length of pipes. $d_{\text{pipe}}$ is internal diameter of the feed pipe. $\eta_{\text{pump}}$ is pump efficiency, which is taken here as 75% (Fritsch et al., 2015).

Pumping the liquid metal against the static head of the tower is the main component of the pumping load. Given that the density of molten tin is nearly four times the density of molten salts; the $W_{\text{static}}$ here is expected to be significantly greater than in conventional CSP power plants. However, this load may be only required during the transient initiation phase of the liquid metal circuit. In CSP plants integrated with direct storage, the outlet HTF from the receiver loses its kinetic energy, as it stalls at the hot reservoir; hence, the pump is required to continuously operate against the static head of the tower. In a plant with an indirect-storage, the same HTF mass circulates in a closed circuit; hence after initiating the mass flow rate in the circuit, the pumping load would then normalise at the frictional and minor head losses in the receiver and heat exchanger of the TES.

The estimated total electrical loss from each of the studied plant is presented in Table 38, which generally ranged between 1-7% of the gross electrical output. This range conforms to literature estimations for solar tower receiver plants based on molten salts, liquid sodium, and particle receivers (Gallo et al., 2015, Ramorakane and Dinter, 2016, Trainer, 2019, Zheng et al. 2020). Please note that these rough estimates may vary in practice based the actual pump efficiency, type of heat exchanger used at the TES system, and the employed anti-freezing measure.
Table 38. Estimated electrical losses in the three studied plants.

<table>
<thead>
<tr>
<th>Electrical Power Rating (MW\textsubscript{elec})</th>
<th>Internal Diameter of the HTF Circulation Pipes (m)</th>
<th>Feed Velocity of the HTF (ms\textsuperscript{-1})</th>
<th>Estimated Electrical Losses (MW\textsubscript{elec})</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>0.35</td>
<td>0.533</td>
<td>0.72</td>
</tr>
<tr>
<td>40</td>
<td>0.43</td>
<td>0.829</td>
<td>2.04</td>
</tr>
<tr>
<td>100</td>
<td>0.53</td>
<td>1.344</td>
<td>6.86</td>
</tr>
</tbody>
</table>

8.1.6. Overall plant efficiency

The performance metric used in the analysis of this Chapter will be the solar-to-electric efficiency ($\eta_{\text{sol-\text{elec}}}$), which is evaluated as follows:

$$\eta_{\text{sol-\text{elec}}} = \frac{\text{Net electrical output}}{\text{Incident solar radiation on the field}} = \frac{(\eta_0 \eta_{rc} \eta_{\text{storage}} \eta_{\text{conv}} A_{\text{hetio}} I) - W_{\text{pump}}}{A_{\text{hetio}} I}$$  \hspace{1cm} (140)

8.2. Power plant analysis

In Chapters 1, it was concluded that operating solar thermal at ultra-high temperatures would benefit the overall power generation system by enabling integration with an equivalent TES and advanced energy conversion cycles. In Chapter 2, the aim of the thesis was specified on developing a solar receiver, which could provide ultra-high temperatures for large-scale applications, while being compatible to a state-of-the-art TES technology. In this section, the TES integrated CSP power plant model described in section 8.1 is used to demonstrate the benefits of ultra-high temperature operation at large-scale applications.

8.2.1. Benefits of ultra-high temperature and large-scale operation

The conceptual significance of concentrating solar radiation to achieve feasible ultra-high receiver temperatures was presented in sections 2.1 and 7.1.2. However, increasing the receiver temperature in these analyses was shown to have a negative effect on the thermal efficiency of the receiver due to increased thermal losses. Similar effect of temperature on the storage efficiency was illustrated in Figure 8.3. The gains from increasing the temperature is only established in the energy conversion phase;
however, these gains may be superseded by the increased thermal losses with temperature in the solar receiver and TES systems. In Figure 8.4(a), the variation of the 20 MW\textsubscript{elec} power plant performance with temperature of the proposed receiver is displayed at four different levels of geometric concentration ratios. The plot displays the benefit of increasing the receiver temperature at each concentration level up to an optimum point beyond which the overall efficiency of the plant would drop with temperature. The main reason behind this behaviour is due to the exponentially increased radiative losses from the receiver and TES at ultra-high temperatures, which outweigh the gains in the energy conversion phase.

The previous profiles are iterated for the larger scales as shown in Figure 8.4(b) and Figure 8.4(c). When comparing the profiles across the three studied capacities, the overall plant efficiency is shown to directly correlate with the plant capacity, which is fundamentally attributed to the direct correlation between storage efficiency and capacity, as explained previously in Chapter 1. These results demonstrate the importance of maximising the scale of CSP applications. For reference, the overall efficiencies of some of the existing CSP plants, which are integrated with TES systems, are presented in Table 39. Current plants typically operate at temperatures <1000 K with optical concentration ratios lower than 1000 suns, which lead to overall efficiencies <20%. The data also demonstrate the higher efficiencies of plants based on cavity receivers than plants based on external receivers.
Figure 8.4. Influence of receiver temperature on CSP efficiency at different levels of concentration ratio and plant capacity. (a) 20 MW<sub>elec</sub>. (b) 40 MW<sub>elec</sub>. (c) 100 MW<sub>elec</sub>.

### 8.2.2. Optimum receiver temperature

In the theoretical analysis presented in section 2.1.1, the maximum attainable receiver temperatures, under a range of given concentration ratios, were presented. In this section, the optimum receiver temperature is presented for a range of concentration ratios. The optimum receiver temperature at a given concentration ratio was defined in Figure 8.4 as the peak point of the curve. The plot displayed in Figure 8.5 specifies the required concentration ratio and receiver temperature to maximise the overall efficiency of the plant. It also shows that the advantage of operating at higher temperatures is manifested at larger plant capacities.
Table 39. Comparison of existing TES integrated tower receiver systems per receiver design (NREL, 2022b, Hennecke et al., 2009, Graham-Cumming, 2009, ARENA, 2013).

<table>
<thead>
<tr>
<th>CSP Plant</th>
<th>Rated Capacity (MW&lt;sub&gt;elec&lt;/sub&gt;)</th>
<th>Geometrical Concentration Ratio</th>
<th>Receiver Temperature (K)</th>
<th>TES System (Storage Hours at Rated capacity)</th>
<th>Solar-to-Electrical Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Cavity Receiver Based</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Jülich</td>
<td>1.5</td>
<td>909</td>
<td>973</td>
<td>Ceramic heat sink (1.5h)</td>
<td>N/A</td>
</tr>
<tr>
<td>PS10</td>
<td>11</td>
<td>300</td>
<td>548</td>
<td>Two-tank system (1h)</td>
<td>15.53%</td>
</tr>
<tr>
<td>PS20</td>
<td>20</td>
<td>N/A</td>
<td>803</td>
<td>Superheated steam (2h)</td>
<td>15.84%</td>
</tr>
<tr>
<td>Khi</td>
<td>50</td>
<td>N/A</td>
<td>803</td>
<td></td>
<td>10.57%</td>
</tr>
<tr>
<td><strong>External Receiver Based</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lake Cargelligo</td>
<td>3</td>
<td>N/A</td>
<td>773</td>
<td>Core graphite (1h)</td>
<td>3.63%</td>
</tr>
<tr>
<td>Crescent Dunes</td>
<td>110</td>
<td>N/A</td>
<td>838.7</td>
<td>Two-tank system (10h)</td>
<td>5.97%</td>
</tr>
</tbody>
</table>

Figure 8.5. Variation of the optimum receiver temperature with concentration ratio at different concentrated solar power plant capacities.
Chapter 9. Conclusions and Future Work

Meeting the demanding energy targets set in the IPCC’s Paris Accords in 2015 necessitates finding innovative solutions to counter the obstacles restraining renewable technologies from becoming feasible alternatives to conventional fossil-fuel-based systems. Solar energy is renewable and abundant, which could supply the entire global energy consumption if harvested efficiently. Solar thermal energy can be used directly to supply thermal processes or converted into electricity in CSP plants. CSP-TES systems can potentially become more feasible and sustainable than PV-battery systems for power generation (Achkari and El Fadar, 2020, Jorgenson et al., 2016).

Operating CSP and TES at ultra-high temperatures enhances their power densities and compactness of design. Running CSP at temperatures >1300 K would enable their integration with high-efficiency power cycles, such as CCGT, without requiring supplementary fossil fuel combustion to compensate for the gap between the outlet HTF temperature from the solar receiver and the inlet turbine temperature of the CCGT cycle. Additionally, harvesting solar thermal energy at ultra-high temperatures can be utilised in various energy-intensive thermal processes, such as in hydrogen and cement production. Nevertheless, after reviewing the technological status of solar thermal and TES, it was evident that the former lags behind the latter in terms of temperature.

This thesis presents an original solar receiver designed to facilitate a reliable operation at ultra-high temperatures. The proposed receiver involves novel operational concepts and design features, including the use of a directly irradiated liquid metal HTF flowing down a corrugated back plate. A computational-based research methodology has been developed and used to evaluate the technical performance of the proposed receiver at different operational and design configurations.

9.1. Development of a novel solar receiver

In Chapter 2, concentrating the solar radiation is established as a critical factor for sustaining a feasible operation at ultra-high temperatures. After reviewing the prospective solar thermal technologies, the tower receiver emerged as the most promising technology to meet the aims of this thesis.
The main characteristics of an ultra-high temperature receiver were identified in Chapter 3 as follows:

1. Solar absorption must be performed within an apertured enclosure to entrap the concentrated solar radiation and minimise the thermal losses to the ambient;
2. Secondary optics must be used incorporated to compensate lost concentrations from large solar fields and to facilitate the required high concentration ratios to minimise thermal losses from the cavity;
3. Liquid metals are strong candidates as heat transfer fluids at ultra-high temperatures, given their superior thermo-physical properties and high boiling points compared to other fluids;
4. The conventional use of a solid absorber to receive the concentrated solar radiation must be abolished by allow the opaque HTF to directly receive the concentrated radiation to avoid the thermomechanical and chemical degradation of solid materials under high radiative fluxes.

A novel solar receiver was proposed in Chapter 4 based on the previous characteristics with liquid tin used as an optically exposed HTF. Tin is selected as the candidate HTF to exploit its wide liquid temperature range with comparable freezing temperatures to conventional solar salts, and its chemical compatibility with market available refractory containment materials, such as graphite, which is proven to alleviate its corrosiveness at ultra-high temperatures. The liquid tin HTF inserted into the cavity through a wide slit to form an opaque gravity-driven film over an inclined surface at the back of the cavity. The inclined back plate was corrugated with a sinusoidal profile to preserve the HTF film continuity, which is vital to protect the former from damage under highly concentrated solar radiation.

Secondary optics can be worthwhile for ultra-high temperature applications (Li et al., 2020b), which can significantly augment the optical concentration ratio by over four times and reduce spillage loss by about 60% (Li et al., 2019). Although the cavity design was found considerably immune from wind-induced convection, the aperture still required to be closed with a window to protect the liquid metal and its containment materials from oxidation and burning at ultra-high temperatures. The limitations and concerns regarding the use of quartz glass and fluidic seals in this application were
presented in Chapter 4, which led to proposing a transparent ceramic window fabricated from MgAl$_2$O$_4$ spinel to utilise its favourable thermal and mechanical properties, and maintenance of high optical transparency over a wide range of wavelengths within the solar radiation spectrum. A steady-state energy analysis on the window was presented in Chapters 5-7.

Two configurations were considered for the cavity walls: reflective and absorptive. In the reflective cavity, solar collection is only handled through direct absorption by the gravity-driven HTF film flow, while cavity walls are made reflective, by lining their interior surfaces with a reflective and thermal-resistant metal, to redirect the reflected beams from the film surface back to the film for further absorption. In the absorptive cavity, the role of the exposed film is to moderate the intensity of the concentrated solar radiation and diffusively reflect them towards the larger cavity walls, which are lined with a compatible absorptive coating to absorb the majority of the solar radiation and preheat the HTF before its exposed phase. The detailed design of the cavity walls and their insulations were presented in Chapter 4 and numerically tested in Chapter 7.

9.2. Modelling methodologies

In Chapter 5, a quasi-steady-state model was presented, which described the receiver performance using analytical and semi-empirical expressions. The model used various approximations to simplify the analysis, which were verified against a radiation-coupled CFD numerical solution described in Chapter 6. The CFD model combined the discrete ordinates radiation model (DOM) and the Volume of Fluid (VOF) multiphase model to simulate the volumetric absorption of solar radiation by the exposed HTF film flow. The model was tailored to accommodate the transient 3D features of the gravity-driven HTF film flow and buoyancy-driven cavity fluid flow as described in Chapter 6. Due to the large difference in timescales required to converge the two flows, the numerical model was developed in form of three sequential simulation sets, so that each flow was simulated in a separate transient simulation set before combining their pseudo-steady-state outcomes into a third steady-state simulation. The employed numerical sub-models were validated in Chapter 7 against experimental and numerical data provided from the literature.
9.3. **Main Findings**

A summary of the main findings on the design, performance, and application of the novel receiver is presented in this section.

9.3.1. **Energy performance of the proposed receiver**

1. The proposed receiver concept was shown successful to attain outlet liquid tin temperatures within <5% of the target;
2. The reflective configuration displayed poor thermal efficiencies (<40%), mainly due to radiative absorption by the reflective walls;
3. The absorptive configuration showed more promising efficiencies exceeding 70%, which are comparable to low-temperature cavity receivers;
4. The performance of the reflective cavity was significantly influenced by the optical properties of the liquid metal and cavity walls;
5. The performance of the absorptive cavity displayed a greater degree of insensitivity to the optical properties of the liquid metal and cavity walls.

9.3.2. **Mitigation of the main concerns about the proposed receiver concept**

1. Liquid metals can be highly corrosive at ultra-high temperature. Refractory materials, such as graphite and silicon carbide, were recently proposed and demonstrated in the literature as feasible containment materials for liquid tin;
2. Pumping liquid metals at ultra-high temperatures can be unfeasible given the low efficiency electromagnetic pumps. Recently, a mechanical pump made from Shapal was demonstrated to successfully pump liquid tin at 1673 K;
3. Liquid metals are susceptible to oxidation, there is an existing practical base of knowledge for handling liquid tin at ultra-high temperatures in the glass industry using the Pilkington process, including oxidation protection measures;
4. Exposure of solid absorptive materials to high solar fluxes can instigate thermomechanical and chemical degradation. The absorptive cavity walls are shown to be subjected to reduced fluxes, as the highly concentrated solar beams are moderated by absorption and diffuse reflection by the wavy liquid metal film flow;
5. The solar absorptance of liquid metals are naturally low for direct absorption. The absorptive cavity configuration showed minimal sensitivity to the liquid metal emissivity. Additionally, emissivity of liquid metals can be enhanced by colouration or addition of chemically compatible ceramic particles.

9.3.3. **Continuity of the liquid metal film**

1. The combination of flow inclination and surface corrugations were proven successful in preserving the continuity of the liquid metal HTF film flow;
2. The liquid metal film flow over the corrugated surface is still found to be susceptible to two forms of discontinuity. At low amplitude-to-wavelength ratios, film breaks down into thinning threads due to excessive acceleration. At high amplitude-to-wavelength ratios, primary flow is blocked by stagnant regions, which initiate flow separation zones. Visual representations of the two forms of film discontinuity were displayed in Chapter 7;
3. There is a range of amplitude-to-wavelength values at which the film flow can remain preserved. This range is correlated to the flow rate and physical properties of the liquid metal;
4. The amplitude-to-wavelength ratio of the corrugations were smaller in the reflective cavity given its slower HTF flow rate, as the exposed liquid metal required longer residence time to reach the target temperature. At the studied scale and inlet conditions, this resulted in a significantly greater, by 21.63%, heat transfer area for the film in the absorptive cavity, which indicates a higher power density than of the reflective cavity.

9.3.4. **Conceptual limitations of the reflective cavity receiver**

1. The poor performance of the reflective cavity is primarily attributed to the unavoidable absorbed fraction of energy by the reflective walls, which is found to be substantial even at a wall reflectance of 0.9;
2. Since the reflective cavity walls are required to be maintained at the lowest possible temperature to maintain the integrity and reflectance of its lining material, active cooling is found necessary for the reflective walls;
3. It is unlikely to use the liquid tin HTF as the coolant at such low temperatures, and small temperature differences, as it was demonstrated to result in
ineffective cooling for the walls, while risking dropping its temperature to below the freezing temperature in the process;

4. Using a different fluid from the HTF to cool the walls would complicate, if not prohibit, the energy recovery from the reflective walls. Nevertheless, the cavity fluid was proposed and demonstrated as an effective coolant for the reflective walls, which may be used to indirectly preheat the HTF or maintain its liquidity during the night-time phase;

5. The recommended insulation layers for the two cavity walls configurations were demonstrated not to exceed their maximum allowable temperatures;

6. The reflective cavity is shown to be significantly impacted by the optical properties of the internal surfaces, while the absorptive cavity displayed more insensitive performance to the optical properties;

7. Maximising the solar absorptance of the liquid metal HTF, which could be implemented through colouration or addition of ceramic particles, was shown to substantially improve the efficiency of the reflective cavity; however, it will not likely exceed the efficiency of the absorptive cavity;

8. The requirement to maintain a highly reflective (reflectance >0.95) walls at ultra-high temperature marks the central conceptual limitation of the reflective cavity configuration, as the candidate lining materials are likely to be metallic, while the reflectance of most metals are shown to deteriorate significantly with temperature;

9. The structure of the reflective cavity is found to require materials that are not abundant on Earth, such as silver or platinum. This violates the energy sustainability target established in Chapter 1.

9.3.5. **Transparent ceramic window**

1. The cavity aperture is closed with a window to protect the liquid metal and its containment materials from oxidation and burning at ultra-high temperatures;

2. Conventionally used quartz glass has shown thermomechanical limitations in the literature due to degradation by recrystallization induced by the thermal cyclic nature at temperatures >1073 K, while fluidic seals may not perfectly isolate the oxidant ambient from diffusing into the cavity fluid;
3. Ytrria doped magnesium aluminate spinel is proposed as a potentially feasible transparent ceramic window, which can withstand thermal stresses and fatigue at temperatures up to 2473 K;
4. The proposed ceramic window is characterised by its long-term resistance to chemical corrosion, high thermal shock resistance, and high fracture toughness, which can enable safe operation at a thickness of only 4 mm;
5. Concerns about the fabrication costs of spinel and anti-soiling of windows are considerably mitigated with the ongoing procedures and measured developed in the literature (see Chapter 4 for more details);
6. Although passive cooling of the ceramic window was found sufficient to maintain the window temperature below its maximum allowable temperature, active cooling of windows, possibly by the cavity fluid, is likely to be required in practice, particularly at higher concentration ratios.

9.3.6. Suggested cavity wall structures

1. The reflective cavity walls are composed of a single layer of microporous insulation, which was shown to provide a sufficient thermal insulation at temperatures up to 1273 K;
2. The absorptive cavity walls are composed of a microporous layer and an additional insulation layer made from zirconium oxide fibres to accommodate the temperature gradients >1273 K;
3. Silver is nominated as the lining material for the reflective cavity walls, given its unique high reflectance (>0.94) at temperatures up to its melting temperature (~1235 K);
4. For reflective walls at temperatures exceeding 1200 K, the option of preheating the HTF by cooling the walls may become feasible, hence, zirconium-platinum lining was suggested as a compatible reflective lining material to graphite; however, its reflectance may drop to 0.8 at temperatures >2000 K;
5. Iron–cobalt–chromite black spinel is nominated as the coating material for the absorptive cavity walls due to its chemical compatibility with graphite, thermal resistance, and high solar absorptance (>0.85) at ultra-high temperatures;
9.3.7. **Significance of the buoyancy-driven cavity fluid flow**

1. The buoyancy-driven flow of the cavity fluid was found to pose negligible convective and shearing effects on the heat transfer and continuity of the liquid metal HTF. Nevertheless, in case the cavity fluid is circulated inside the cavity at higher velocities to actively cool the walls and/or window, this might result in a non-negligible forced convection effect on the HTF;
2. Modelling the buoyancy-driven flow as an incompressible ideal gas is verified by the low Mach numbers of the cavity fluid;
3. The surface tension of highly dense liquid metals is shown to pose negligible effects on the gravity-driven HTF film flow; however, this must be checked by ensuring the Weber number is $>>1$.

9.3.8. **Verification of the analytical model**

1. The analytical model was verified to deliver energy breakdown results and receiver efficiency results with discrepancies $<10\%$ and $6\%$ from the results obtained from the computationally expensive numerical solution, respectively;
2. The only exception was the natural convection, which displayed greater discrepancies, as the used semi-empirical correlation underestimated the heat transfer coefficients by up to $<14.3\%$. Nevertheless, given the insignificance of natural convection ($<<1\%$ of input solar energy), these discrepancies did not impact the overall receiver energy results;
3. Various empirical correlations were tested but failed to generate close results to the numerically solution. This indicates that generic natural convection correlations developed for basic geometries and solid boundaries are not suitable here. Therefore, a natural convection correlation may need to be tailored to the proposed receiver design and HTF flow configuration;
4. The diffuse radiation assumption employed in the analytical mode was verified as an acceptable approximation, particularly for the absorptive cavity;
5. The assumption of evaluating all fluid properties at their mean flow temperatures, which was used in the analytical model, was verified with $<1\%$ and $<3\%$ disparities in the evaluated thermo-physical and optical properties, respectively. It was also shown that evaluating the optical properties at the
representative radiation temperature, instead of the mean flow temperature, would drop their disparities to <1%.

9.3.9. Why solar thermal should be large-scale and at ultra-high temperatures?

1. The benefits of operating the receiver at ultra-high temperatures and large capacities were manifested when incorporated in an application, such as the CSP plant demonstration presented in Chapter 8;
2. It was verified that maximising the receiver temperature must be coupled with the maximising the concentration ratio; otherwise, the gains from running the energy extraction cycle at higher temperatures can be outweighed by the exponentially increased radiative loss from the receiver;
3. There is an optimum receiver temperature, at given concentration ratio, beyond which the overall efficiency of the plant would decline if the receiver temperature increases. This optimum temperature was shown to increase at higher concentration ratios and larger plant capacities.

9.4. Recommendations for future work

In view of the results presented in this thesis, the proposed solar receiver, at its absorptive cavity wall configuration, emerged as a promising starting point for enabling solar thermal energy to be harnessed and transported at ultra-high temperatures. The absorptive cavity is then considered qualified for an experimental demonstration, which will likely be centred on the practicalities of the gravity-driven liquid metal film flow over a corrugated inclined surface in terms of film continuity, volumetric radiation absorption, reliability of the proposed containment materials, and thermal-structural testing of the proposed walls and window. A small-scale prototype of the cavity receiver with mercury as the HTF may be used for the demonstration to utilise mercury’s liquidity at room temperature with comparable optical ($\varepsilon_{Hg} \sim 0.1$) and thermo-physical properties ($\rho_{Hg} \sim 13,545 \text{ kgm}^{-3}$, $c_{p,Hg} \sim 0.139 \text{ Jkg}^{-1}\text{K}^{-1}$, $k_{Hg} \sim 8.69 \text{ Wm}^{-1}\text{K}^{-1}$, $\mu_{Hg} \sim 1.544\times10^{-6} \text{ kgm}^{-1}s^{-1}$) to the candidate liquid metals. A high-flux solar simulator, such as the 54 kW setup developed in the University of Adelaide (Dong et al., 2012), may then be used to demonstrate the volumetric absorption performance of the film. Furthermore, the University of Adelaide has developed a
novel solar simulator capable of producing equivalent radiations of up 30,000 suns (Alwahabi et al., 2016), which may be used to study the optical performance and structural integrity of the proposed ceramic window.

After the experimental demonstration, the next step is likely to be optimising the design of the proposed receiver for a selected number of applications. For example, the geometric parameters of the cavity receiver may be optimised to maximise the thermal efficiency using the analytical model described in Chapter 5. In Chapter 7, it was emphasised that the internal dimensions of the cavity must be guided by the resulted flux distribution to guarantee no solid surface would be exposed to the highly concentrated solar radiation. Moreover, exposed regions of the film, which are not subjected to high radiative fluxes should be covered to minimise the emissive losses from the heated HTF. Another potentially useful investigation is using the numerical model described in Chapter 6 in optimising of the corrugations’ amplitude-to-wavelength ratio to maximise the diffuse fraction of the inclined surface, while preserving the continuity of the gravity-driven film flow.

The author recommends performing a techno-economic-assessment to justify the design selections made in this study for the proposed receiver, such as the suggested transparent ceramic window and materials used to build the cavity wall structures. Since the capital cost is a major obstacle for large-scale solar thermal projects as discussed in Chapter 1, the payback period to recover the initial investment will need to be evaluated for an application incorporating the proposed receiver and be compared with a conventional system. Similar approach may be used in CSP applications by evaluating the net cumulative savings and levelised cost of electricity resulted from each system. In case of a comparison with a fossil fuel based system, the analysis should also encompass an environmental impact assessment of each system.

In Chapter 4, the application of an infrared-reflective coating at the internal surface of the ceramic window was suggested to internally reflect the re-radiated energy from the cavity. This approach was demonstrated by Röger et al. (2009) to reduce the receiver losses by 11% and window temperature by 180 K. An alternative approach may include modifying the crystalline structure of the ceramic window to become
spectrally selective. Further investigations and developments in these areas may be worthwhile to enhance the efficiency of the proposed receiver.

The future work may involve investigating different cooling techniques for the ceramic window and cavity walls. The present research laid out the theoretical base for this investigation and proposed the use of the cavity fluid as a coolant. However, detailed designs of the cooling mechanisms, and energy recovery, are still required. Similarly, the absorptive cavity wall heat exchanger is yet to be designed in details, which may benefit from the preliminary analysis provided in Chapter 7. Finally, the design of the liquid metal heat exchanger, which will connect the receiver to the TES system is yet to be designed and tested.

The receiver potential was demonstrated for a power production application in Chapter 8. However, this potential may also be extended to other ultra-high temperature applications, such as the thermal processes used in hydrogen and cement production systems. Currently, over 85% of globally produced hydrogen is still sourced from fossil fuels (Qureshi et al., 2022), while cement production is responsible for 6-10% of global CO$_2$ emissions due to combustion of fossil-based kiln fuels (Liu et al., 2020). The developed solar receiver is demonstrated at a comparable temperature to the ultra-high temperature processes in both applications; hence, it can present as a sustainable thermal reactor to replace the conventionally employed combustion reactors. A techno-economic-environmental analysis of such replacements can be worthwhile now given the provision of the ultra-high temperature solar thermal technology.

Potential future work may include upgrading the analysis models developed in this research. For instance, the evaluation of the convective loss inside the receiver requires the development of a customised correlation to the cavity geometry. The numerical model may be upgraded to model the cavity walls as non-isothermal boundaries to check the validity of the isothermal wall assumption, which was justified here the high thermal conductance of the used interior materials and their insensitive optical properties to temperature at ultra-high temperatures. Finally, the applicability of the developed numerical model in other similar applications, involving volumetric radiation by fluidic flows, may also be investigated.
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### Appendix A1. Thermo-physical Properties

Table A1.1. Thermo-physical properties of nominated storage materials to operate at ultra-high temperatures (Datas, 2021b). CTE here is coefficient of thermal expansion.

<table>
<thead>
<tr>
<th>Material</th>
<th>$T_{max}$ (°C)</th>
<th>$T_{loss}$ (°C)</th>
<th>$k$ (W/mK)</th>
<th>Specific Heat (kJ/kg.K)</th>
<th>$\rho$ (g/cm$^3$)</th>
<th>$\lambda$ (W/mK)</th>
<th>$\mu$ (mPa.s)</th>
<th>$\tau$ (KJ/kg)</th>
<th>$\rho$ (kg/m$^3$)</th>
<th>$\lambda$ (W/mK)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Graphite</td>
<td>3600</td>
<td>18</td>
<td>40</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
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<tr>
<td>Ceramic cement</td>
<td>3072</td>
<td>6.31</td>
<td>0.7</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
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<tr>
<td>MgO</td>
<td>3025</td>
<td>1.7</td>
<td>7.7</td>
<td>3.58</td>
<td>4.65</td>
<td>5.99</td>
<td>154</td>
<td>650</td>
<td>20</td>
<td>11</td>
</tr>
<tr>
<td>Al$_2$O$_3$</td>
<td>3072</td>
<td>1.2</td>
<td>6.6</td>
<td>3.97</td>
<td>5.27</td>
<td>5.88</td>
<td>96</td>
<td>750</td>
<td>0.4</td>
<td>2</td>
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<tr>
<td>SiO$_2$</td>
<td>1710</td>
<td>1.4</td>
<td>4.2</td>
<td>2.63</td>
<td>3.1</td>
<td>3.61</td>
<td>0.5</td>
<td>300</td>
<td>12</td>
<td>6</td>
</tr>
<tr>
<td>C$O$</td>
<td>2572</td>
<td>2.2</td>
<td>7</td>
<td>3.35</td>
<td>0.97</td>
<td>3.34</td>
<td>14.3</td>
<td>450</td>
<td>5.7</td>
<td>10</td>
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<tr>
<td>Granite</td>
<td>1240</td>
<td>1.3</td>
<td>2.8</td>
<td>2.64</td>
<td>0.82</td>
<td>2.17</td>
<td>2.4</td>
<td>200</td>
<td>10</td>
<td>2</td>
</tr>
<tr>
<td>Sandstone</td>
<td>1500</td>
<td>1.2</td>
<td>1.8</td>
<td>2.2</td>
<td>0.71</td>
<td>1.56</td>
<td>5.95</td>
<td>220</td>
<td>5.6</td>
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<td>SIC</td>
<td>2527</td>
<td>9.9</td>
<td>40</td>
<td>3.21</td>
<td>1.26</td>
<td>4.05</td>
<td>17.7</td>
<td>560</td>
<td>5</td>
<td>10</td>
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<tr>
<td>Fe (solid)</td>
<td>1538</td>
<td>5</td>
<td>28</td>
<td>7.66</td>
<td>0.72</td>
<td>5.64</td>
<td>12.6</td>
<td>790</td>
<td>0.15</td>
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<tr>
<td>Si (solid)</td>
<td>1141</td>
<td>11</td>
<td>26</td>
<td>2.33</td>
<td>0.98</td>
<td>2.28</td>
<td>7.66</td>
<td>4.38</td>
<td>330</td>
<td>2</td>
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<tr>
<td>Fe (liquid)</td>
<td>1538</td>
<td>6.2</td>
<td>40</td>
<td>7.86</td>
<td>0.82</td>
<td>6.48</td>
<td>16.1</td>
<td>40</td>
<td>900</td>
<td>0.15</td>
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<tr>
<td>Si (liquid)</td>
<td>1141</td>
<td>26</td>
<td>60</td>
<td>2.33</td>
<td>0.67</td>
<td>2.26</td>
<td>11.6</td>
<td>12</td>
<td>310</td>
<td>2</td>
</tr>
<tr>
<td>Al (liquid)</td>
<td>660</td>
<td>29</td>
<td>91</td>
<td>2.7</td>
<td>1.18</td>
<td>3.18</td>
<td>17</td>
<td>45</td>
<td>440</td>
<td>1.8</td>
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<tr>
<td>Cu (liquid)</td>
<td>1085</td>
<td>36</td>
<td>170</td>
<td>8.92</td>
<td>0.52</td>
<td>4.61</td>
<td>27.7</td>
<td>54</td>
<td>640</td>
<td>5</td>
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<tr>
<td>Chlорide salt</td>
<td>425</td>
<td>0.32</td>
<td>0.35</td>
<td>1.25</td>
<td>1.71</td>
<td>0.97</td>
<td>91</td>
<td>260</td>
<td>0.5</td>
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Table A1.2. Thermo-physical properties of nominated HTFs to operate at ultra-high temperatures (Datas, 2021b).

<table>
<thead>
<tr>
<th>Heat transfer fluid</th>
<th>$T_{max}$ (°C)</th>
<th>$T_{loss}$ (°C)</th>
<th>$k$ (W/mK)</th>
<th>Specific Heat (kJ/kg.K)</th>
<th>$\rho$ (g/cm$^3$)</th>
<th>$\lambda$ (W/mK)</th>
<th>$\mu$ (mPa.s)</th>
<th>$\tau$ (KJ/kg)</th>
<th>$\rho$ (kg/m$^3$)</th>
<th>$\lambda$ (W/mK)</th>
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</thead>
<tbody>
<tr>
<td>Aluminum (Al)</td>
<td>660</td>
<td>2450</td>
<td>18.4</td>
<td>0.9</td>
<td>0.92</td>
<td>1.75</td>
<td>42.36</td>
<td>0.010</td>
<td>2</td>
<td>5.905</td>
</tr>
<tr>
<td>Iron (Fe)</td>
<td>1528</td>
<td>2662</td>
<td>37.0</td>
<td>50.3</td>
<td>72.0</td>
<td>7.5</td>
<td>15.4</td>
<td>5.65</td>
<td>0.184</td>
<td>0.3</td>
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<tr>
<td>Silicon (Si)</td>
<td>1414</td>
<td>3372</td>
<td>6.0</td>
<td>28.2</td>
<td>25.0</td>
<td>0.79</td>
<td>12.5</td>
<td>24.50</td>
<td>0.017</td>
<td>0.3</td>
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<td>Gallium (Ga)</td>
<td>30</td>
<td>2405</td>
<td>65.0</td>
<td>26.1</td>
<td>549.2</td>
<td>0.59</td>
<td>11.6</td>
<td>31.78</td>
<td>0.003</td>
<td>0.5</td>
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<td>Lithium (Li)</td>
<td>180</td>
<td>1347</td>
<td>6.5</td>
<td>28.9</td>
<td>43.6</td>
<td>0.21</td>
<td>10.7</td>
<td>34.85</td>
<td>0.012</td>
<td>0.4</td>
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<tr>
<td>Tin (Sn)</td>
<td>232</td>
<td>2628</td>
<td>67.0</td>
<td>37.1</td>
<td>6452.2</td>
<td>0.98</td>
<td>9.7</td>
<td>23.14</td>
<td>0.007</td>
<td>0.6</td>
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<tr>
<td>Indium (In)</td>
<td>156</td>
<td>2080</td>
<td>62.0</td>
<td>20.1</td>
<td>6095.3</td>
<td>0.81</td>
<td>8.3</td>
<td>35.35</td>
<td>0.002</td>
<td>0.5</td>
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<tr>
<td>Lead (Pb)</td>
<td>327</td>
<td>1745</td>
<td>23.0</td>
<td>25.8</td>
<td>9047</td>
<td>1.05</td>
<td>9.3</td>
<td>18.84</td>
<td>0.006</td>
<td>0.2</td>
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<tr>
<td>Lead–bismuth (Pb-40%Bi)</td>
<td>125</td>
<td>1436</td>
<td>20.0</td>
<td>25.9</td>
<td>9113</td>
<td>0.89</td>
<td>4.9</td>
<td>17.17</td>
<td>0.005</td>
<td>0.7</td>
</tr>
<tr>
<td>Bismuth (Bi)</td>
<td>271</td>
<td>1352</td>
<td>19.0</td>
<td>27.2</td>
<td>10724</td>
<td>1.06</td>
<td>5.4</td>
<td>13.54</td>
<td>0.007</td>
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</table>

Table A1.3. Thermo-physical properties of selected liquid metals (Lorenzin and Abánades, 2017). $\lambda$ is thermal conductivity.

<table>
<thead>
<tr>
<th>HTF</th>
<th>$T_{max}$ (°C)</th>
<th>$T_{loss}$ (°C)</th>
<th>$C_p$ (KJ/kg.K)</th>
<th>$\lambda$ (W/mK)</th>
<th>$\rho$ (kg/m$^3$)</th>
<th>$\nu$ (mPa.s)</th>
<th>$\tau$ (KJ/kg)</th>
<th>$\rho$ (kg/m$^3$)</th>
<th>$\lambda$ (W/mK)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Melted Tin (Sn)</td>
<td>232</td>
<td>273</td>
<td>0.24</td>
<td>39.8</td>
<td>4330</td>
<td>1.01</td>
<td>1539.2</td>
<td>15.9</td>
<td></td>
</tr>
<tr>
<td>Gallium (Ga)</td>
<td>29.8</td>
<td>2403</td>
<td>5.75</td>
<td>59.5</td>
<td>5673</td>
<td>0.69</td>
<td>21273.7</td>
<td>252</td>
<td></td>
</tr>
<tr>
<td>Sodium (Na)</td>
<td>687</td>
<td>883</td>
<td>1.26</td>
<td>57.5</td>
<td>761</td>
<td>0.16</td>
<td>958.8</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>Lithium (Li)</td>
<td>180</td>
<td>1347</td>
<td>6.15</td>
<td>63.3</td>
<td>436</td>
<td>0.20</td>
<td>1013.7</td>
<td>13.8</td>
<td></td>
</tr>
<tr>
<td>Lead–bismuth (Pb-40%Bi)</td>
<td>125</td>
<td>1638</td>
<td>0.146</td>
<td>17.7</td>
<td>5710</td>
<td>1.33</td>
<td>1417.6</td>
<td>13</td>
<td></td>
</tr>
<tr>
<td>Gallium (Ga6%–20%Sn–13.5%Zr)</td>
<td>-59</td>
<td>&gt;1300</td>
<td>0.29</td>
<td>16.5</td>
<td>6440</td>
<td>2.4</td>
<td>1857.6</td>
<td>450</td>
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</tr>
</tbody>
</table>

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Table A1.4. Specific volume ($v$), internal energy ($u$), enthalpy ($h$), and entropy ($s$) of steam at various temperatures ($T$) – in degree Celsius – and two pressure ($P$) levels (Çengel, 2019).

Table A1.5. Variation of specific heat capacity of steam with temperature (Engineering-ToolBox, 2005).
Figure A1.1. Variation of thermo-physical properties of steam with temperature in degree Celsius (Haar et al., 1984). (a) Viscosity. (b) Thermal conductivity. (c) Prandtl number.
Figure A1.2. Variations of density and viscosity of FLiBe molten salt with temperature (Sohal et al., 2010).

Figure A1.3. Variations of surface tension and specific heat capacity of FLiBe molten salt with temperature (Sohal et al., 2010).
Table A1.6. Properties of air at 1 atm (Çengel, 2003). Please note that temperatures here are displayed in degree Celsius.

Table A1.7. Thermodynamic properties of air at various temperatures (T) pressure (P) levels (Shpilrain, 2011). \(v\) is specific volume \((10^3 \times m^3 \cdot kg^{-1})\), \(h\) is specific enthalpy \((kJ \cdot kg^{-1})\), and \(s\) is specific entropy \((kJ \cdot K^{-1} \cdot kg^{-1})\) of air.
Table A1.8. Specific heat capacity, kJ kg\(^{-1}\) K\(^{-1}\), of air at various temperatures \(T\) pressure \(P\) levels (Shpilrain, 2011).

<table>
<thead>
<tr>
<th>T,K</th>
<th>(p = 1), bar</th>
<th>(p = 10), bar</th>
<th>(p = 100), bar</th>
<th>(p = 1000), bar</th>
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</thead>
<tbody>
<tr>
<td>100</td>
<td>1.032</td>
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<td>1.007</td>
<td>1.021</td>
<td>1.158</td>
<td>1.303</td>
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<td>400</td>
<td>1.014</td>
<td>1.021</td>
<td>1.087</td>
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<td>1.055</td>
<td>1.080</td>
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<td>700</td>
<td>1.075</td>
<td>1.077</td>
<td>1.096</td>
<td>1.168</td>
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<td>1.100</td>
<td>1.114</td>
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<td>900</td>
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<td>1.159</td>
<td>1.160</td>
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<tr>
<td>1200</td>
<td>1.175</td>
<td>1.175</td>
<td>1.181</td>
<td>1.225</td>
</tr>
<tr>
<td>1300</td>
<td>1.189</td>
<td>1.189</td>
<td>1.194</td>
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</tr>
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</table>

Table A1.9. Thermal conductivity, \(10^{-3} \times \text{W m}^{-1} \text{K}^{-1}\), of air at various temperatures \(T\) pressure \(P\) levels (Shpilrain, 2011).

<table>
<thead>
<tr>
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<th>(p = 0.1), bar</th>
<th>(p = 1.0), bar</th>
<th>(p = 10), bar</th>
<th>(p = 100), bar</th>
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<tr>
<td>100</td>
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<td>100</td>
<td>100</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>200</td>
<td>165</td>
<td>144</td>
<td>137</td>
<td>135</td>
<td>134</td>
</tr>
<tr>
<td>300</td>
<td>812</td>
<td>813</td>
<td>486</td>
<td>304</td>
<td>238</td>
</tr>
<tr>
<td>400</td>
<td>529</td>
<td>417</td>
<td>620</td>
<td>701</td>
<td>489</td>
</tr>
<tr>
<td>500</td>
<td>1281</td>
<td>1424</td>
<td>718</td>
<td>604</td>
<td>663</td>
</tr>
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<td>6000</td>
<td>3191</td>
<td>4287</td>
<td>2272</td>
<td>1087</td>
<td>749</td>
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</table>

Table A1.10. Dynamic viscosity, \(10^{-7} \times \text{N s m}^{-2}\), of air at various temperatures \(T\) pressure \(P\) levels (Shpilrain, 2011).

<table>
<thead>
<tr>
<th>T,K</th>
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<th>(p = 10), bar</th>
<th>(p = 100), bar</th>
<th>(p = 1000), bar</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>71.1</td>
<td>837.8</td>
<td>1019</td>
<td>—</td>
</tr>
<tr>
<td>200</td>
<td>132.5</td>
<td>134.6</td>
<td>181.2</td>
<td>—</td>
</tr>
<tr>
<td>300</td>
<td>184.6</td>
<td>185.9</td>
<td>205.0</td>
<td>545.5</td>
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<tr>
<td>400</td>
<td>230.1</td>
<td>231.2</td>
<td>243.4</td>
<td>463.4</td>
</tr>
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<td>270.9</td>
<td>279.9</td>
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<td>306.4</td>
<td>313.6</td>
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<td>700</td>
<td>338.8</td>
<td>339.4</td>
<td>345.3</td>
<td>442.7</td>
</tr>
<tr>
<td>800</td>
<td>369.8</td>
<td>370.3</td>
<td>375.4</td>
<td>456.4</td>
</tr>
<tr>
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<td>398.5</td>
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<td>472.0</td>
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<td>424.8</td>
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<td>488.7</td>
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<tr>
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<td>449.3</td>
<td>452.8</td>
<td>505.8</td>
</tr>
<tr>
<td>1200</td>
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<td>473.2</td>
<td>476.5</td>
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<tr>
<td>1300</td>
<td>496.0</td>
<td>496.3</td>
<td>499.2</td>
<td>541.8</td>
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</table>
Table A1.11. Density, kg m\(^{-3}\), of CO\(_2\) at various temperatures and pressure levels (Vukalovich et al., 1963). Please note that temperatures and pressures here are displayed in degree Celsius and bar, respectively.

<table>
<thead>
<tr>
<th>Temp, °C</th>
<th>1</th>
<th>5</th>
<th>10</th>
<th>20</th>
<th>30</th>
<th>50</th>
<th>70</th>
<th>100</th>
</tr>
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<tbody>
<tr>
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<td>10.022</td>
<td>20.287</td>
<td>35.659</td>
<td>53.997</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>20</td>
<td>1.4544</td>
<td>8.3524</td>
<td>17.048</td>
<td>29.348</td>
<td>45.826</td>
<td>62.852</td>
<td>80.823</td>
<td>120.24</td>
</tr>
<tr>
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<td>45.826</td>
<td>62.852</td>
<td>80.823</td>
<td>120.24</td>
</tr>
<tr>
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<td>34.504</td>
<td>46.427</td>
<td>56.561</td>
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<tr>
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<td>10.175</td>
<td>20.464</td>
<td>31.086</td>
<td>41.335</td>
<td>51.987</td>
<td>85.592</td>
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<tr>
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<td>34.085</td>
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<td>25.599</td>
<td>31.855</td>
<td>44.006</td>
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<td>5.1019</td>
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<td>18.081</td>
<td>23.997</td>
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<td>40.633</td>
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</tbody>
</table>

Table A1.12. Specific heat capacity, kJ kg\(^{-1}\) K\(^{-1}\), of CO\(_2\) at various temperatures and pressure levels (Vukalovich et al., 1963). Please note that temperatures and pressures here are displayed in degree Celsius and bar, respectively.

<table>
<thead>
<tr>
<th>Temp, °C</th>
<th>1</th>
<th>5</th>
<th>10</th>
<th>20</th>
<th>30</th>
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<th>70</th>
<th>100</th>
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</thead>
<tbody>
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<td>1.0826</td>
<td>1.1061</td>
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<tr>
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<td>0.8782</td>
<td>0.9009</td>
<td>0.9461</td>
<td>0.9782</td>
<td>1.0122</td>
<td>1.0489</td>
<td>1.0826</td>
<td>1.1061</td>
</tr>
<tr>
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<td>0.9972</td>
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<td>1.0254</td>
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</tr>
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<td>1.0970</td>
<td>1.1168</td>
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<td>1.1402</td>
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<td>1.1165</td>
<td>1.1265</td>
<td>1.1326</td>
<td>1.1402</td>
<td>1.1402</td>
<td>1.1479</td>
<td>1.1479</td>
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<tr>
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<td>1.1471</td>
<td>1.1471</td>
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<td>1.1522</td>
<td>1.1479</td>
<td>1.1479</td>
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<tr>
<td>70</td>
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<td>1.1881</td>
<td>1.1917</td>
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<td>1.1881</td>
<td>1.1917</td>
<td>1.1953</td>
<td>1.2023</td>
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</tbody>
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Table A1.13. Thermal conductivity, 10-3 × W m-1 K-1, of CO2 at various temperatures ( )
pressure ( ) levels (Vesovic et al., 1990).

Table A1.14. Dynamic viscosity, 10-6 × N s m-2, of CO2 at various temperatures ( ) pressure
( ) levels (Vesovic et al., 1990).

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Figure A1.4. Variation of density of liquid tin with temperature (Assael et al., 2010).

Figure A1.5. Variation of viscosity of liquid tin with temperature (Assael et al., 2010).
Figure A1.6. Variation of specific heat capacity of liquid tin with temperature (Khvan et al., 2019).

Figure A1.7. Variations of thermo-physical properties of liquid tin with temperature. (a) Surface tension (Yuan et al., 2002). (b) Thermal conductivity (Giordanengo et al., 1999).

For solar dishes, the only foreseen viable option for large-scale power production application would be using a centralised TES in a micro-turbine based thermodynamic cycle similar to the hypothetical system presented in Figure A2.1. However, the economic feasibility of this configuration is yet to be established first, particularly with the commercial lack of confidence after a history of failed utility-scale projects due to excessive operation and maintenance costs (Salameh, 2014).

Figure A2.1. A hypothetical configuration for solar dishes in a large-scale combined power generation cycle (Karni, 2003).

A hypothetical solar furnace configuration (Figure A2.2) is proposed for integration with a liquid-metal-based ultra-high temperature TES system (Figure A2.3) developed by Robinson (2017). The liquid metal used in the TES storage core circulates in an optically exposed extrusion, which will be illuminated by concentrated solar radiation. This heat addition process to the storage core will replace, or work in conjunction with, the electric heating used to charge the storage core of the original TES design. Some design and operational aspects of this concept are already addressed in this thesis (in Chapter 3), such as the direct irradiation, pumping, and containment of the liquid metal HTF. On top of the conceptual limitations associated with conventional solar furnaces, the main challenges of this new configuration will be the insulation of the extrusion and effectiveness of heat transfer across the storage core.
Figure A2.2. A preliminary setup for a TES integrated solar furnace.

Figure A2.3. An ultra-high temperature TES nominated for integration (Robinson, 2017).
Two potential setups for integrating the liquid metal based ultra-high temperature thermal energy storage (UHTS) with the beam-down solar tower concept are illustrated in Figure A2.4.

Figure A2.4. Potential storage integration arrangements for the beam-down solar tower setup. UHTS is Ultra-High Temperature thermal energy Storage. CPC is Compound Parabolic Concentrator.
A ‘liquid metal fountain’ concept (Figure A2.5) was also considered to minimise the distance of free-falling and preserve the continuity of the exposed liquid metal flow inside the cavity receiver. While a fountain can be designed to deliver a 360° dome of liquid metal, allowing for use with surround heliostat fields. However, given the high density of liquid metals, the HTF flow rate is unlikely to be sufficient to maintain a continuous dome. Therefore, the concept was proposed for a polar-facing type of heliostat fields. The use of multi-aperture receiver is avoided here due to its ineffectiveness at this power scale and operational temperature (detailed explanation were provided in section 3.1). Nevertheless, even with a single flow, the small flow rate of liquid tin was found to break and expose the solid components to high solar irradiations.

Figure A2.5. Polar-facing ‘liquid metal fountain’ receiver concept integrated with an ultra-high temperature thermal energy storage (UHTS).
Appendix A3. Supplementary CFD Simulations

Wind tunnel test on open-aperture receiver

General-use correlations proposed in the literature for evaluating the wind-induced convective effects produce contradicting results or limited to certain geometries and operating conditions (Lee et al., 2019, Ma, 1993, Prakash et al., 2009, Sinha et al., 2019). Therefore, the significance of wind-induced convection on the proposed receiver design, with an open aperture, is tested in a virtual ‘wind tunnel’ using the CFD code *Ansys Fluent* (Fluent, 2016).

To run the wind tunnel simulations, the CAD model of the receiver was imported to *Ansys DesignModeler* and its solid volume was then subtracted from a 125,000 m$^3$ cube, creating cavities in the fluidic volume representing the solid boundaries of the receiver. The wind inlet was then applied horizontally and normal to the cube’s front face, as shown in Figure A3.1, with a prescribed temperature of 300 K. The magnitude of the wind velocity was varied from 0 ms$^{-1}$ up to 9 ms$^{-1}$ to study its convective effects. Wind direction is fixed at the facing direction to the receiver aperture to predict the wind-induced effect at its maximum component towards the aperture. Only the momentum and energy equations were activated, while the inclined surface, displayed in Figure A3.2, of the receiver was prescribed a temperature of 1673 K.

![Figure A3.1. Model design for the wind tunnel simulations.](image)
Figure A3.2. The inclined surface (highlighted in red), which will be supporting the HTF film flow, is prescribed at the target ultra-high temperature.

The wind flow dynamics along the external surfaces of, and inside, the receiver are displayed in Figure A3.3 at two wind speed levels: 9 m s$^{-1}$ and 3 m s$^{-1}$. From the presented streamlines plots, it can be observed that the wind flow decelerates and separates near the aperture. At both wind speeds, the streamlines were found to lose two thirds of their kinetic energy, allowing only the bottom region of cavity to be occupied by the external air flow as shown in Figure A3.3(c) and (f). It should be noted here that the bottom region will be host the outlet of the heated liquid metal HTF, hence, subjecting the gravity-driven film to flow disturbances and severe convective losses.
Figure A3.3. Visualised streamlines (coloured according to velocity magnitudes) of wind-driven air generated from the wind tunnel simulations. (a) Top view of the external streamlines in at a wind speed of 9 m s$^{-1}$. (b) Isometric view of the streamlines at a wind speed of 9 m s$^{-1}$. (c) Focused streamlines entering the receiver through the aperture at a wind speed of 9 m s$^{-1}$. (d) Isometric view of the streamlines at a wind speed of 3 m s$^{-1}$. (e) Separation of streamlines near the receiver aperture at a wind speed of 3 m s$^{-1}$. (f) Focused streamlines entering the receiver through the aperture at a wind speed of 3 m s$^{-1}$. 
The hot outlet was protected by a reflective cover at the bottom, as illustrated in Figure 4.18. In the case of an open aperture, the majority of the heat air was found to escape directly through the aperture. Enclosing the hot outlet by the cover helps restrain the air, or cavity fluid, from extracting heat from the outlet liquid metal flow. Sealing the outlet pit by creating a protective air/cavity fluid vortex at its entrance was also considered.

Figure 4.18. Sectional 3D view of the solid CAD model of the proposed cavity receiver. (a) The imported model to the wind tunnel simulations with labelled main features. (b) The improved outlet, from pit to a rectangular slit, created by adding a reflective cantilevered cover to protect the hot HTF outlet from radiative and convective thermal losses. It should be noted that this cover, being at the bottom, is not subjected directly to the concentrated solar beams.

Positioning the aperture at the bottom of the receiver helps entrapping the heated air at the upper part of the cavity, as shown in Figure A3.3, while creating a horizontal free shear layer at middle of the cavity, which isolates the stagnant upper hot zone from the cold wind-induced turbulence at the aperture level. This flow pattern was observed and justified by various studies of cavity receivers with open apertures (Xiao et al., 2012, Tan et al., 2009, Clausing, 1983). The wind magnitude determines the elevation of the protective free shear layer and, in turn, the volume of the hot zone. Nevertheless, The resulted wind-induced energy losses from the cavity were found
minimal (<0.5 MWth), even at a wind speed of 9 ms\(^{-1}\) as demonstrated in the flux report in Figure A3.4. This provides a vital insight about the immunity of this particular design against wind-induced convection, which might not be valid for other receiver designs.

Figure A3.4. Net heat transfer rate between inlet and outlet boundaries of the wind tunnel, designating the wind-induced convective loss from the cavity at wind speed of (a) 3 ms\(^{-1}\) and (b) 9 ms\(^{-1}\).

Figure A3.5. Velocity vectors from the wind tunnel tests, at a wind speed of 3 ms\(^{-1}\), without including the energy equation (no temperature effects) at the sectional (a) side and (b) top views.
Figure A3.6. Streamlines vectors from the wind tunnel tests, at a wind speed of 3 ms\(^{-1}\), with including the energy equation (temperature effects).

Other Simulations

Figure A3.7. Disintegration of a vertical flow of liquid tin, which is infused through the receiver ceiling, under gravitational acceleration. Red displays liquid tin; blue displays cavity fluid; other colours display disintegrating/thinning liquid tin.
Figure A3.8. Velocity vectors of the buoyancy-driven cavity fluid displaying the results from two different pressure interpolation schemes: (a) PRESTO and (b) body-forces weighted.

Figure A3.9. Velocity vectors of the buoyancy-driven cavity fluid displaying results from four different turbulence models: (a) standard $K - \varepsilon$ (Lauder, 1972), (b) Realisable $K - \varepsilon$ (Shih et al., 1995), (c) Wilcox’s $K - \omega$ (Wilcox, 1998), and (d) SST (Menter, 1994).
Figure A3.10. Temperature distribution across the wall sections (black lines are used to mark the borders between different materials) of the absorptive cavity receiver cooled by molten tin at: (a) 800 K (average interior wall temperature is 1138.35 K), (b) 1124.34 K (average interior wall temperature is 1460.99 K), and (c) 1448.68 K (average interior wall temperature is 1783.93 K). The preheat pipes are distanced 1 m apart.
Appendix A4. Solution Scripts

```matlab
% This function calculates view factors between two planar surfaces.
% The function receives 3 parameters, which are the coordinates of
% both figures and the desired number of significant digits.
% 
% viewfactor(factor1,factor2,area1,area2)=viewfactor(coord1,coord2,ndigits)
% 
% coord1 and coord2 are the coordinates of the vertices that
% meet the outline of figures 1 and 2. It must be entered this way:
% coord1=[[x1,y1,z1];[x2,y2,z2];[...,...,...];[xn,yn,zn]]
% 
% ndigits is the desired number of significant digits.
% The function displays more digits but only `ndigits`
% are significant.

function [facteur1,facteur2,aire1,aire2] =functionviewfactor(figure1,figure2,niveau)

% NOTE: THIS .m FILES HAS BEEN CODED IN FRENCH, THE COMMENTS HAS BEEN
% TRANSLATED.

% We want a long result
format long;

% How many vertices on each figures?
 nbpoint1,nbcoord1 = size(figure1);
 nbpoint2,nbcoord2 = size(figure2);

% We add the first point as the last point to close the outline of the figure
figure1(1,:)=figure1(1,:);
figure2(1,:)=figure2(1,:);

% Global variables: pt1 to pt4 are the points that define the segment to integrate, normal1 and normal2 are the unit normal vector.

    global pt1;
    global pt2;
    global pt3;
    global pt4;
    global normal1;
    global normal2;
    global normal;

% Unit normal vector of figure 1
    cote1=figure1(1,:)-figure1(4,:);
    cote2=figure1(4,:)-figure1(1,:);
    normal1=cross(cote1,cote2)/(norme(cross(cote1,cote2))));

% Unit normal vector of figure 2
    cote1=figure2(1,:)-figure2(4,:);
    cote2=figure2(4,:)-figure2(1,:);
    normal2=cross(cote1,cote2)/(norme(cross(cote1,cote2))));

% Preliminary calculation of area to determine which precision to use in calculation to respect the number of significant digits
airetemp=0;
normal=normal1;
for i=1:nbpoint1
    pt1=figure1(i,:);
    pt2=figure1(i+1,:);
```
%Integration to calculate the area of figure 1
aire1temp = aire1temp + quadl('functionviewfactorarea', 0, 1, 0, 1);
end
aire1temp = abs(aire1temp);
prec = 10^(-niveau) / (npoint1 * npoint2 / (2 * pi * aire1temp) + npoint1 / aire1temp);

% The calculation of area of figure 1 with the right precision.
 aire1 = 0;
normale = normal1;
 for i = 1:npoint1
   pt1 = figure1(i, :);
   pt2 = figure1(i+1, :);
   aire1 = aire1 + quadl('functionviewfactorarea', 0, 1, 1, prec);
 end
 aire1 = abs(aire1);

% The calculation of area of figure 1 with the right precision.
 aire2 = 0;
normale = normal2;
 for i = 1:npoint2
   pt1 = figure2(i, :);
   pt2 = figure2(i+1, :);
 end
 aire2 = abs(aire2);

% Integration to calculate the area of figure 2
 aire2 = aire2 + quadl('functionviewfactorarea', 0, 1, 1, prec);
 end
 aire2 = abs(aire2);

% Double integral that we need to calculate the view factor
 sommeintegrale = 0;
 for i = 1:npoint1
   for j = 1:npoint2
     pt1 = figure1(i, :);
     pt2 = figure1(i+1, :);
     pt3 = figure2(j, :);
     pt4 = figure2(j+1, :);
     sommeintegrale = sommeintegrale + dblquad('functionintegral', 0, 1, 0, 1, prec, 'quadl');
   end
 end

% Calculation of the view factors
 facteur1 = abs(sommeintegrale) / (2 * pi * aire1);
 facteur2 = abs(sommeintegrale) / (2 * pi * aire2);

% Function to integrate the view factor
 function sortie = functionintegral(x, y)

% Global variables: pt1 to pt6 are the points that define the segment to integrate, normale1 and normale2 are the unit normal vector.
global pt1;  
global pt2;  
global pt3;  
global pt4;  
global normale1;  
global normale2;
Figure A4.1. MATLAB functions used to build the algorithm to calculate the view factors of internal surfaces of the cavity receiver. The functions were prepared by Lauzier (2022).

Figure A4.2. User-defined C functions used to define the Gaussian distribution of intensity on the aperture boundary.
Figure A4.3. A sample of the bash Unix shell code used to initiate Ansys Fluent on a Linux based High Performance Computer.
define/models/viscous/rem-bil-based yes
35 ;
36 ; Other Tested Turbulence models (All are commands are
37 ; disabled with ";")
38 ;
39 rSST model
40 rdefine/models/viscous/kw-set yes
41 ;
42 k-kl-w model
43 rdefine/models/viscous/k-kl-w model yes
44 ;
45 k-kl RSL model
46 rdefine/models/viscous/kw-bil yes
47 rdefine/models/viscous/kw-low-re-correction yes
48 rdefine/models/viscous/curvature-correction yes
49 rdefine/models/viscous/turbulence-expert/turbulance-damping
50 ; yes 10
51 ;
52 rStandard k-epsilon model
53 rdefine/models/viscous/kw-standard yes
54 ; Enabling the k-epsilon realizable model
55 rdefine/models/viscous/k-realizable yes
56 ;
57 rk-epsilon RNG model
58 rdefine/models/viscous/kw-rng yes
59 rdefine/models/viscous/rng-differential-visc yes
60 ; Other k-epsilon options
61 rdefine/models/viscous/near-wall-treatment/enhanced-wall
62 ; treatment yes
63 rdefine/models/viscous/near-wall-treatment/wf-thermal-effects
64 ; yes
65 ;
66 rDefine the inlet boundary conditions
67 rdefine/bc/velocity-inlet/inlet mixture no no yes no 0.5329
68 ; no no no 0 1448.68 no no yes 20 10 yes yes 0
69 ;
70 ; Define a variable absorption coefficient for the liquid
71 ; metal HTF
72 ;
73 define/materials/change tinc> tinc> no no no no no no no
74 ; yes piecewise-linear 20 100 7.799019 200 10.239 200 14
75 . . . 3.142797 400 18.514617 500 22.56434 600 17.4365 700 18.5146
76 900 18.25296 900 20.078 1000 23.97034 1100 21.66993 1200 22
77 . . . 4.76658 1300 23.281 1400 24.11289 1500 24.9425726 1600 25
78 . . . 7.8017 1700 26.6258 1800 27.4097 1900 28.342 2000 29.212765
79 no no no yes 2.16 no
80 ;
81 ; Define the other materials (cavity fluid has no buoyancy
82 ; effects here)
83 define/materials/change spinel spinel no no yes constant 0
84 ; yes constant 0.00001 yes 1.7214
85 ;
86 define/materials/change cavity-liquid cavity-liquid yes constant 0.54295 yes constant 1040 yes constant 0.00001 yes constant 3.6337502052639e-05 no no no no no no no
87 ;
88 ; Define phase 2 as the HTF liquid metal
89 rdefine/phases/phase-domain/phase-2 htf no
90 ;
91 ; Define the interaction and surface tension between the two
92 ; fluids
93 define/phases/interaction-domain 0 yes yes yes
94 yes yes yes yes yes yes yes yes yes yes yes yes yes yes yes
95 ;
96 ; defining user-defined material database
97 define/materials/database/database-type user-defined
98 /export/case/eddie/eng/groups/HITI_Data/Task/k/ANSYS
99 /reflective/materials
100 ;
101 rREAD data file
102 rfd abs.dat.gz
103 ;
104 ; I0 angular discretisation
105 rdefine/models/radiation/discrete-ordinates yes 5 5 5 5
106 ;
107 ; setting the solution controls
108 rSOLVE/set/under-relaxation pressure 0.75
109 rSOLVE/set/under-relaxation mom 0.4
110 rSOLVE/set/n-v-controls 200 0.75 0.75

343
344
Figure A4.4. A sample of input commands to the Text User Interface version of Ansys Fluent on the High Performance Computer.
Appendix A5. Published Article


Tarek I. Abdelmassih, Zhao Tian, Adam Robinson

Abstract
Solar thermal energy has the theoretical potential to deliver heat at ultra-high temperatures (>1300 K), which can enable integration with state-of-the-art thermal energy storage systems and unlock new applications, including advanced power cycles and thermal processes. Liquid metals are prospective heat transfer fluids for such systems, given their favourable thermo-physical properties, while their aggressive corrosion is shown to be mitigated using compatible refractory containment materials. The conventional approach of collecting concentrated solar energy typically involves intermediate solid absorbers, in form of tubes or porous structures, which are prone to thermomechanical and chemical failure under high solar radiation. This paper investigates the use of directly irradiated liquid metal (bio) film, operating between 1000 and 1673 K, in two possible cavity configurations: A ‘reflective cavity’ and an ‘absorptive cavity’. The former employs cavity walls as internal reflectors to entrap radiation by secondary reflections until directly absorbed by the liquid metal. In the latter, the directly irradiated film is used to moderate the initial sheet of concentrated solar radiation before diffusively reflecting them to the absorptive cavity walls, which perform in a radiative heat exchanger used to preheat the liquid metal. The concept performance is evaluated using an approximate quasi-steady-state energy model of the receiver. The reflective cavity performance is found strongly dependent on the optical properties of its internal surfaces, which rendered in poor efficiencies (~48%) without special treatments. The absorptive cavity demonstrated higher efficiencies (~70%) with greater insensitivity to the optical properties, hence, promising its consideration in future developments of this concept.

1. Introduction
Solar thermal energy can be used in various mechanical and thermomechanical applications, including driving heat engines in concentrated Solar Power (CSP) plants. In principle, the heat engine efficiency can be improved by increasing the hot source temperature, which is subjected to thermal and material constraints (Gogbit, 2019). Solar-only CSP systems are currently limited to temperatures <1000 K due to chemical limitations of molten salts used as the Heat Transfer Fluid (HTF) (Ivan Khan et al., 2020). Running CSP at an Ultra-High Temperature (UHT) (>1300 K) would unlock the use of advanced power cycles, which can improve the solar-to-electricity efficiency by up to 50% (Keblishofer et al., 2009; Stein and Beck, 2017). Alternatively, the sourced heat could be exploited in other UHT thermal applications, such as glass melting processes (Ahmad et al., 2017) or thermochemical water splitting for hydrogen production (Mulsch et al., 2016). Thermal energy storage has been demonstrated viable at UHT, which promises high energy densities and roundtrip efficiencies (Amy et al., 2019; Datan, 2021; Robinson, 2017). Therefore, the missing link to a dispatchable UHT storage-integrated CSP is the provision of a UHT solar collector technology.

The proposed solution is investigated in three parts. This paper (Part 1) describes a novel UHT solar receiver based on a Liquid Metal (LM) HTF and investigates its performance using an analytical energy model. The transient fluid dynamics and radiation aspects of the concept will be addressed in Parts 2 and 3, respectively.

2. Central concepts for ultra-high temperature solar receivers
The fundamental conceptual elements of the proposed UHT solar receiver are outlined in this section, which serve as the basis for the receiver design described in the next section.
2.1. Solar concentration

Given the ultra-high surface temperature of the Sun (5778 K), sustaining solar receivers at UHT is permitted by the second law of thermodynamics. Concentrating solar radiation is fundamental to sustaining an efficient radiative transfer at UHT (Ho and Iveson, 2014). Ideally, a concentration of 2000 suns is sufficient to maintain a spot at 2500 K (Fletcher et al., 2001); however, greater concentrations are required in practical applications to compensate for the incurred optical and thermal losses (Riddin et al., 2017; Sheard, 2002). While spillage losses may increase at higher concentrations, Li et al. (2019) found that the incorporation of a 3D compound parabolic concentrator (CPC) can minimize the spillage loss by > 90%.

2.2. Cavity receiver

External receivers incur substantial thermal losses and are not recommended at temperatures > 1000 K (Li et al., 2016). The UHT alternative is cavity receivers, which enclose and isolate the absorber from ambient effects. However, this benefit comes at the price of aperture-restricted contact with heliostats, and increased construction and O&M costs.

2.3. Liquid metal

LMs have been recommended in various studies as HTFs in CSP systems (Prinz et al., 2014; Pietro et al., 2015; Hesami et al., 2017; Law and Abidin, 2016; Pietro and Wurzel, 2013; Pietro et al., 2013). The replacement of the conventional ‘solar salts’ (nitrites) with LMs was estimated to improve the receiver efficiency by 20% at 2000 suns and 1173 K (Pietro et al., 2013) and reduce the levelised cost of producing electricity by up to 15% (Singer et al., 2010). In addition to their high boiling points and thermal conductivities, LMs offer a single-phase heat transfer process over a wide range of temperatures (Table 1).

Table 1

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Density (kg/m³)</th>
<th>Specific heat capacity (J/kg·K)</th>
<th>Liquid phase (K)</th>
<th>Thermal conductivity (W/m·K)</th>
<th>Dynamic viscosity (mPa·s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Alumina</td>
<td>2100</td>
<td>1050</td>
<td>925.2743</td>
<td>257</td>
<td>0.750</td>
</tr>
<tr>
<td>Tl</td>
<td>6626</td>
<td>355</td>
<td>505.3275</td>
<td>66.8</td>
<td>0.005</td>
</tr>
<tr>
<td>Lead-</td>
<td>8710</td>
<td>146</td>
<td>307.1041</td>
<td>17.7</td>
<td>1.33</td>
</tr>
<tr>
<td>bismuth</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sulfur</td>
<td>761</td>
<td>1260</td>
<td>371.1156</td>
<td>63.3</td>
<td>0.15</td>
</tr>
<tr>
<td>Solar oil</td>
<td>±1100</td>
<td>±1200</td>
<td>±559.030</td>
<td>±59</td>
<td>±1.69</td>
</tr>
<tr>
<td>(at 300 K)</td>
<td>2100</td>
<td>1520</td>
<td>0.077</td>
<td>0.047</td>
<td></td>
</tr>
</tbody>
</table>

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Heavy metals, such as tin, are characterized by their high boiling points and chemical stability at UHT. Their high volumetric heat capacities and densities also facilitate compact receiver designs. Their low specific heat capacities could be compensated by their wide liquid ranges. However, they are disadvantaged by their increased corrosiveness at UHT (Lorenzin and Abainiades, 2016; Pace and Wetzel, 2012). Zhang et al. (2013) confirmed that commercially available tubes made from graphite, silicon carbide and mullite could provide effective containment for molten tin at 1623 K. Tin is characterized by its low vapour pressure (0.0094 Pa) compared to other LMs at 1300 K, such aluminium (16.584 Pa), lead (129.81 Pa), and bismuth (1062.9 Pa) (Hildebrand, 1918). Therefore, vaporization losses from tin is projected to be substantially lower than other LMs. Molten tin was suggested by Zheng and Ye (2010) as HTF in a hypothetical solar tower system used for hydrogen production.

Pumping LMs at UHT can be a concern. Although large electromagnetic pumps might be favoured over mechanical pumps in terms of maintainability and durability at UHT, their efficiencies are significantly lower for pumping heavy metals (Fritsch et al., 2015). Continuous pumping of molten tin at 1673 K for 72 h was demonstrated by Any et al. (2017), which displayed minimal failure to the containment components.

2.4. Direct Irradiation

The absorption of concentrated solar radiation is conventionally handled by a solid absorber, which then transmits the heat to the HTF by convection and conduction. For a transparent HTF, air or water, an intermediate opaque absorber is required to absorb the radiation despite adding parasitic resistance to heat flow. Directly irradiated heat exchangers and tubular receivers cannot deliver outlet HTF temperatures > 1100 K due to the risk of developing thermal stresses, leading to material degradation, under high solar fluxes (Chapuis et al., 2012; Ho, 2016).

Particle receivers use directly illuminated ceramic particles as heat transfer and storage mediums at temperatures < 1273 K (Düko et al., 2018; Ho, 2016; Wu et al., 2014). However, the solar-weighted absorbance of particles exposed to air for 24 h was found to deteriorate when temperature increased from 973 K to 1273 K (Ciesiel et al., 2014). This cyclic decay has raised concerns over the durability of particle receivers at UHT. Direct absorption by molten salts was also considered in designs of external and cavity receivers; however, performance was limited by wind-induced disruptions (Böhm, 1997; Ho and Iversen, 2014). Optically exposed mercury was suggested by Timmusk, 1996 as an HTF and reflector in a dish receiver.

2.5. Direct absorption by the liquid metal

The low emittance/absorbance of LMs might not prohibit them as solar absorbers. The influence of cavity emittance on the receiver efficiency was found minimal (c.5%) (DiAngelis et al., 2016; Pang et al., 2016). Additionally, LM emmittancies increase with temperature, which may exceed 0.2 for molten tin at UHT (Giesensie, 1969; Jack et al., 1969). The solar absorbance can be furtherly enhanced by collocation or addition of absorptive ceramic particles (Ho et al., 2018; Plehan et al., 2013). The emissivities of different LMs are presented in Table 2.

In view of the previous discussion, the proposed UHT receivers will feature direct absorption by molten tin in a cavity receiver operating under high solar concentration. In the following sections, description of the receiver design is presented, followed by an energy analysis.

3. Design description

The proposed receiver is designed for a hypothetical solar tower system with specifications presented in Table 3. The field is estimated to deliver 101.54 MW at 9750 suns to the receiver. This concentration level

<table>
<thead>
<tr>
<th>Liquid Metal</th>
<th>Wavelength (nm)</th>
<th>Temperature (K)</th>
<th>Emittance</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminium</td>
<td>300</td>
<td>1173</td>
<td>0.070-0.100</td>
<td>(Gold et al., 2012)</td>
</tr>
<tr>
<td>Copper</td>
<td>632.5</td>
<td>1000</td>
<td>0.040</td>
<td>(Gleesman and Sundman, 1993)</td>
</tr>
<tr>
<td>Gold</td>
<td>632.5</td>
<td>1400</td>
<td>0.070</td>
<td>(Clegg et al., 1994)</td>
</tr>
<tr>
<td>Iron</td>
<td>630.0</td>
<td>2022</td>
<td>0.075</td>
<td>(Ekelund and Sundman, 1994)</td>
</tr>
<tr>
<td>Nickel</td>
<td>514.5</td>
<td>1000</td>
<td>0.410</td>
<td>(Gleesman and Sundman, 1993)</td>
</tr>
<tr>
<td>Palladium</td>
<td>514.5</td>
<td>1950</td>
<td>0.030</td>
<td>(Clegg et al., 1994)</td>
</tr>
<tr>
<td>Platinum</td>
<td>514.5</td>
<td>2250</td>
<td>0.050</td>
<td>(Clegg et al., 1994)</td>
</tr>
<tr>
<td>Silver</td>
<td>514.5</td>
<td>1500</td>
<td>0.050</td>
<td>(Clegg et al., 1994)</td>
</tr>
<tr>
<td>Tin</td>
<td>514.5</td>
<td>1400</td>
<td>0.200</td>
<td>(Clegg et al., 1994)</td>
</tr>
<tr>
<td>Zirconium</td>
<td>632.5</td>
<td>2125</td>
<td>0.300</td>
<td>(Clegg et al., 1994)</td>
</tr>
</tbody>
</table>

Table 3: Spectral emissivities of different liquid metals. The emittance values here are average values evaluated at the stated temperatures.

<table>
<thead>
<tr>
<th>Energy Source</th>
<th>Total Aperture (m²)</th>
<th>Number of Heliostats</th>
<th>Geometric Concentration Ratio</th>
<th>Overall Optical Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>Annual Solar Resource (L/M)</td>
<td>1250</td>
<td>10,000</td>
<td>10,000</td>
<td>67.9%</td>
</tr>
</tbody>
</table>

Table 3: Energy and geometrical specifications of the hypothetical solar field used as input conditions to the receiver model.

is equivalent to medium-power lasers used in etching semiconductor materials (Nathan, 1993). Yet, this concentration level has not been demonstrated in commercial CSP, which are still limited to < 1000 suns, mainly due to material constraints of conventional tubular receivers when operating at high solar fluxes (Imsin et al., 2020). Solar tower systems can deliver up to 5000 suns with a room for further concentration, theoretically up to 23,000 suns for a dish angle of 45° (Stoia and Pájuelo, 2001), by incorporating non-imaging secondary concentrators, such as CPCs (Becker and Vanh-Hull, 1991; Li et al., 2016; Schneider, 2006). A 3D CPC was demonstrated by Li et al. (2019) to increase the flux concentration from a multi-source simulator by 4.13x at 3000 suns. The design, cooling, and manufacturing of the CPC are not covered in this paper; however, it is projected to be a metalized structure with a similar water-cooling system as described by Li et al. (2019). The prospect of using the cavity fluid CPC coolant is yet to be investigated. While this study covers only the receiver technology, future work may involve optimizing the heliostats layout for the proposed receiver, which may benefit from (Pizzi et al., 2011) and legal (2012) in accounting for the constrained CPC acceptance.

The proposed receivers here supplies heat to a circulating molten tin, as demonstrated in Fig. 1, to raise its temperature to 1673 K, which would enable integration with a CO2 cycle. The low temperature was set at 800 K based on an LM cycle in a literature UHT thermal energy storage (Robinson, 2017), which coherently provides a safe tolerance above the melting temperature of tin. anti-freezing measures employed for LMs in nuclear and CSP plants (Deeg et al., 2011; Frigo et al., 2019; Kazan et al., 2011) can be utilized to prevent HTF solidification during the diurnal cycle.

The molten tin circuit can be constructed using mechanical and sealing components developed by Any et al. (2017) and containment materials recommended by Zhang et al. (2018). Although the manufacturability of curved geometries with brittle ceramics could be challenging, it may become viable with the advancement of large-scale rapid prototyping and other fabrication techniques. Fabrication of curved and complex ceramic structures was demonstrated using additive manufacturing (Klosterman et al., 1999; Lakhdar et al., 2021;
Mohammadi et al., 2022), which are currently being used in manufacturing graphite enclosures and components in marine and automotive applications (Gupta AM, 2022). The proposed receiver can also benefit from the existing base of practical experience with handling molten tin at UHT in the ‘Pilkington’ process, where an inert atmosphere of N₂ and H₂ is used to protect against oxidation (Francis, 2016; Pilkington, 1969). This measure would also protect the containment materials from oxidation and burning at UHT.

The proposed receiver module is displayed in Fig. 2. The coiled cavity shape was chosen, as it offers thermal (Lasohemipathy et al., 2020) and optical (Azerifi et al., 2021) advantages over cylindrical cavities. The aperture plane is tilted 30° downward to compensate for the restricted CPC acceptance angle (Erichschmidt et al., 2012; Vani-Hall, 2021). For un-windowed apertures, this tilting helps minimising the wind-induced convection and disruption of balistic seals (Pesch et al., 2014). The aperture diameter was determined based on the required flux following the guidance from Steinfield and Schueller (1992).

3.1. Inclined film flow

A ‘free-fall’ flow configuration was initially considered to minimise the receiver size. This configuration is employed in particle receivers and was found inefficient due to the uncontrolled gravitational acceleration, which minimizes the residence time required for heat transfer (He et al., 2019). This limitation is likely more severe in the cases of dense and less absorptive LM. Consequently, the HTF here is inserted as a film flowing over an inclined surface to reduce the gravitational component and enable control over the flow. Flow inclination reduces the flow factor with the aperture; hence, minimizes the re-radiated losses from the film. However, increasing the inclination angle (θ) expands the cavity depth, and would require heliostat relocation further away from the tower, as illustrated in Fig. 3, resulting in reduced land utilisation and increased spillage. Therefore, a supplementary hydraulic control measure is necessary to maintain the film continuity at a reasonable flow inclination (θ = 30°). In Part 2, surface corrugations are demonstrated effective in preserving the film continuity.

3.2. Transparent ceramic window

For receivers enclosing directly irradiated fluids/particles, it is recommended to seal the aperture with transparent windows to prevent leakage and oxidation (Romero and González-Aguilar, 2016). While fluidic seals can be an optically efficient alternative to glazed windows, their contemporary technologies are still underdeveloped to effectively seal the aperture against oxygen diffusion and transient wind conditions (Alpinvistazoomgh, 2009; Tais et al., 2009). Windows can have the advantage of being spectrally selective to facilitate a ‘one-way’ passage for solar radiation by maximising transmittance at the peak solar wavelength and maximising reflectance across the infrared range where blackbody emissions peak at the cavity temperature (Maing et al., 2010; Rusean and Steinfield, 2015). Concerns over window cleaning can be mitigated through the ongoing development of transparent anti-soiling coatings for solar applications (Dahlichi et al., 2022; Huang et al., 2021; Bai et al., 2019; Quan and Zhang, 2017; Zhang et al., 2019), which were proven feasible for solar thermal systems (Lorenz et al., 2014). Windows exposed to UHT liquid metals in closed environments may be subjected to the corrosive and opaque metallic vapours, which may be alleviated by running fluidic curtains on their interior surfaces. Glass windows are often made thicker than optically desired to withstand the thermal stresses at UHT (Zambelli and Good, 2019; Becker et al., 2013). For a pressurised cavity receiver at 1973 K under 10,000 bars, a quartz glass window thickness of 10-20 mm was required (Kami et al., 1995). Glass windows are limited to diameters < 1 m (Avila-Main, 2011; Hinkel et al., 2019; Romero and González-Aguilar, 2014) and not recommended for temperatures > 1703 K (Röper et al., 2006). Transparent ceramics offer a promising glazing alternative, which avoids the structural limitations of quartz glass at UHT, with prospective large-scale manufacturability (Goldman et al., 2017; Sepulveda et al., 2013; Kog et al., 2015), which promoted them in solar thermal (Erickson and Gavilan, 2014) and photocatalytic (Liu et al., 2011) applications. Magnesium-aluminate-spinel (MgAl₂O₄), doped with 2 wt% yttrium(III) oxide (Y₂O₃), is selected here for the 10-mm thick window to exploit its high strength (~4x of quartz glass), high fracture toughness (~7x of quartz glass), and long-term resistance against creep and chemical corrosion at temperatures up to 2400 K (Głochołajski et al., 2015; Liu et al., 2019), which qualified them as the prime candidate window material for high-energy laser applications subjected to hostile environments (Quadri et al., 2019; Sanghera et al., 2011). The optical transmittance of a 10 mm thick spinel varies between 0.8 and 0.95 across the 0.3-4.3 μm spectral band (Sepulveda et al., 2013), which is a wider transmittance range compared to other glazing materials (Destiolo et al., 2009; Harris et al., 2013; Sanghera et al., 2011; Sanghera et al., 2013). Quadri et al., 2019 demonstrated that spinel transmittance could be enhanced up to 0.923 by using anti-reflective surface structures. Furthermore, the absorbance of spinel at 2000 K was found minimal (transmission average ~ 0.00262) at wavelengths < 3 μm (Fennell et al., 1962; Harris, 1999; Solé et al., 2021; Sanghera et al., 2011; Savva et al., 1998), which helps minimise the window cooling requirement. Minimising the absorptivity/ emissivity at wavelengths > 3 μm via application of infrared-reflective coatings is demonstrated viable and effective for windows, including a spinline window (Röper et al., 2009; Sanghera et al., 2011). The main disadvantage of using spinel is the high manufacturing costs, which could be minimised substantially by employing alternative fabrication techniques, such as processes proposed by Sanghera et al. (2015) and Villalobos et al. (2012).

3.3. Cavity wall configurations

Two cavity wall configurations are investigated to determine whether cavity walls should participate in solar absorption or remain reflective.

The first configuration is the ‘reflective cavity’ illustrated in Fig. 4(a). The HTF here is the sole absorber, while cavity walls interior are lined with a highly reflective metallic coating to sustain the UHT operation. The rationale is that the high reflectance, combined with the large depth-to-aperture ratio of the cavity, will entrap most of the original solar input inside the cavity through internal reflections before being

---

1. Yb-doped MgAl₂O₄ maintains a fracture toughness ~1.6 MPa.m^0.5 at 2073 K (Conn et al., 2013), which is more than double the fracture toughness of quartz glass at the room temperature (Gross et al., 2020).
predominantly absorbed by the HTF. However, the requirement for highly reflective cavity walls could be a practical limitation due to reduced reflectance of metals at high temperatures (Garavan et al., 1997; Uijbergen, 1972). Silver is the primary lining candidate, as it maintains specular reflectance > 60% up to its melting point (1234 K) (Uijbergen, 1972). For higher wall temperatures up to 2443 K, similar wall structure to the absorptive cavity may be used with a compatible reflective lining, such as zircaloy-platinum (Abrey and Geagea, 1991).

The second configuration is the "absorptive cavity" illustrated in Fig. 4(b). The design comprises a tubular solar absorber attached to cavity walls to preheat the HTF before its direct solid absorber phase. Preheating, instead of post-heating, is used here to minimize the temperature of the solid absorber. The premise here is to use the exposed LM to attenuate the highly concentrated solar input by absorption and diffuse reflection. The weakened and diffusely reflected beams from the wavy film surface will then strike a larger absorbing area (cavity walls), hence, alleviating the material degradation concerns discussed in section 1.4 – this supposition will be verified in Part 3 by generating the incident radiation contours on internal walls using a radiation-coupled numerical solution. Furthermore, if hot spots are developed at the absorptive walls, their exothermic radiation is less likely to escape from the cavity, given their small view factors with the aperture.

Cavity walls are isolated by microporous insulation, which combines superior thermal properties with compactness at temperatures up to 1273 K (Unifax, 2018). The absorptive cavity walls contain an additional, graphite-compatible, buffer insulation layer of zircaloy.
oxide fibres to cover the temperature gradient down to 1273 K. Graphite coated with iron-cobalt-chromium spinel (Wang et al., 2021) can be used to build up the tubular absorber/preheater as shown in Fig. 5. Since the insulated cavity walls are subjected to trivial thermal loss to ambient, the absorbed energy by the walls are expected to require active cooling to protect the lining material from melting. The exposure of reflective walls to the metallic vapours at UHT may also deteriorate its performance, which requires further investigation before progressing with the reflective cavity configuration.

LMs are not recommended as efficient coolants for the reflective walls, given their high freezing points and low specific heat capacities. Therefore, a secondary fluid is likely to be used as a coolant for the reflective walls, which adds further complexity and inefficiencies. The cavity fluid can be a viable coolant (specific heat capacity > 7% of liquid tin); however, direct recovery of its energy could be challenging. Circulating the cavity fluid from walls, and potentially the window and CPC, to the cavity would enable active control over the protective atmosphere inside the cavity. Contrary to the absorptive configuration, the coolant tubes/ducts will be embedded within the reflective walls, as shown in Fig. 5, to sustain their specularity. The effectiveness of different cooling mechanisms at each configuration is yet to be investigated.

4. Energy model

An approximate quasi-steady-state energy model of the receiver is described and solved using the sequentially non-iterative approach to evaluate the receiver efficiency at each configuration. Model verification against a numerical solution will be presented in Part 3.

4.1 Assumptions

All internal cavity re-radiations are assumed diffuse to enable expressing the radiative transfer in terms of area ratios and view factors, which can be solved analytically. The diffuse assumption is particularly justified for rough graphite surfaces (Wang et al., 2014) and will be verified for the HTF film in Part 3 based on its surface waviness. Modelling the reflective walls as diffuse surfaces will be demonstrated in Part 3 as the primary source for the discrepancy of results against a reference numerical solution, due to underestimating the radiative losses from the reflective cavity; nevertheless, the maximum discrepancy was 5.7%. Therefore, the model is still considered a useful design tool.
given its low computational requirements.

Window spectral selectivity is not considered in this analysis. Therefore, the model employs the grey radiation assumption, which is justified for the spinel window, given its stable transmittance at relevant spectral wavelengths as discussed in section 2.2. The solar source is treated as a blackbody thermal reservoir at 5200 K (Gere and Pye, 2013). Accordingly, the optical properties are evaluated at the solar peak wavelength ($\lambda_{max}$), which is calculated using Wein's displacement law:

$$\lambda_{max} = \frac{2988}{T_{solar}}$$

(1)

where $T_{solar}$ is the solar blackbody temperature.

Cavity walls are assumed adiabatic, as conduction is typically negligible compared to other heat transfer mechanisms at UHT (Heiber and Vant-Hull, 1994; Li et al., 2019). The conduction loss through the cavity walls may become significant when their effective thermal conductivity exceeds 0.1 W m$^{-1}$ K$^{-1}$ (DeAngelis et al., 2010). Here, the evaluated values are 0.031 W m$^{-1}$ K$^{-1}$ and 0.092 W m$^{-1}$ K$^{-1}$ for the reflective and absorptive cavity walls, respectively.

Cavity walls are assumed isothermal at 500 K and 1623.6 K for the reflective and absorptive configurations, respectively. The temperature of reflective walls should be kept as low as possible to maximize reflectance and minimize emissive losses. For the absorptive walls, their temperature should be higher than the desired HTF temperature after the preheat phase to account for the effectiveness of the wall heat exchanger (taken here as 90%). According to Rie et al. (1995), an isothermal cavity at 1673 K with at least two predefined temperature partitions – the cavity here has three temperature-defined partitions: HTF film, cavity walls, and window – is expected to underestimate the receiver efficiency by < 6%.

This underestimation inaccuracy tends to increase at temperatures closer to the equilibrium temperature (3033.1 K at 6750 K), hence, it is projected to be minimal for the < 1000 K reflective cavity. Furthermore, this inaccuracy is easy to be small here, given the use of highly conductive wall materials (silver and graphite), which are also characterized by their high reflective efficiencies, emissivities to temperatures (Thorn and Simpson, 1953; Uijtewaal, 1972).

There is no available analytical steady-state solution for unstable liquid flow over corrugated surfaces (Tsubakio et al., 2013), so Nusselt's solution (Nusselt, 1916) would overestimate the film thickness resulting in inaccurate velocity profiles when applied to wavy and turbulent films (Moran et al., 2002; Tsubakio et al., 2013; Vila, 1963). Therefore, the HTF film is modeled as an opaque and continuous porous material with surface optical properties. Turbulence and contiguity of the film flow will be numerically investigated in Part 2, where the geometrical properties of the corrugations will be demonstrated as a key factor in preserving the continuity. HTF thermo-physical and optical properties are evaluated at the mean flow temperature (1236.5 K), since variations of most tin properties at relevant temperatures are small (<5%) (Auedel et al., 2014; Elhassan et al., 2019). While uncertainties in the dynamic viscosity and thermal conductivity of molten tin from the mean temperature are more significant (<30%), these variations exhibited linear trends between 800 K and 1673 K. Hence, a property value at the mean temperature is still justified (Auedel et al., 2014; Auedel et al., 2017; Giardengo et al., 1996). Similarly, the emissivity of molten tin is taken as 0.2588 (Greenstein, 1976). Wall emissivities were approximated as 0.1 for metallic reflective walls (Omanachana and Vemula, 1977) and 0.8 for graphite absorptive walls (Thorn and Simpson, 1953). The cavity fluid is modeled as an inert mixture (90 wt% N$_2$ and 10 wt% H$_2$) gas.

4.2. Model description

The maximum receiver efficiency ($\eta_{max}$) is expressed by Steinfield (2003) in term of efficiency ($\eta$) and average incident flux on the field ($\gamma_{in}$):

$$\eta_{max} = 1 - (\sigma T^4 / CR_{Earth})$$

(2)

where $\sigma$ is Stefan-Boltzmann constant and CR is the geometric concentration ratio. $\eta_{max}$ is 95.6% at $T = 1673$ K for a CR of 10,000 and $\gamma_{in}$ of 1600 W m$^{-2}$. However, the actual efficiency ($\eta_{act}$) would be lower, typically ranges between 40 and 50% (Le Roux et al., 2014; Matsu et al., 2022), due to additional energy losses. $\eta_{act}$ is defined here as:

$$\eta_{act} = \frac{Q_{out}}{Q_{in}} = \frac{Q_{out}}{Q_{in} / (\rho c_{p} \gamma_{in})}$$

(3)

where $Q_{out}$ is the net (useful) rate of energy absorption/collector, $\rho$ is the aperture area, $\rho c_{p}$ and $\gamma_{in}$ are the concentrated solar power and flux impinging on the external surface of the aperture window, respectively. $P_{col}$ is given by:

$$P_{col} = \frac{\pi}{4} \lambda_{in} (\pi r^2) \int_{\Omega_{in}} f_{in} (\Omega_{in}) \sin \Omega_{in} d\Omega_{in}$$

(4)

where $\lambda_{in}$ is the solid angle, $\Omega_{in} (\Gamma, \Phi)$ is the incident solar radiation on an elemental solid angle area with a position vector $\Gamma$ and normal $\Phi$, and $\eta_{in}$ the overall optical efficiency. The optical efficiencies of the solar-field and CPC are taken as 75% (Rinaldo et al., 2014) and 90% (Li et al., 2019), respectively, which result in $\eta_{in}$ of 0.75. Therefore, the optical concentration ratio (CR$_{opt}$) and $P_{col}$ can be expressed as:

$$CR_{opt} = \frac{\eta_{in} \rho c_{p} \gamma_{in}}{\rho c_{p} \gamma_{in}} = CR_{Earth}$$

(5)

$$P_{col} = CR_{opt} \pi \lambda_{in} \int_{\Omega_{in}} f_{in} (\Omega_{in}) \sin \Omega_{in} d\Omega_{in} = CR_{Earth} \pi \lambda_{in}$$

(6)

$P_{col}$ is attenuated by the window before entering emissive ($Q_{es}$), reflective ($Q_{ref}$), and natural convective ($Q_{con}$) losses inside the cavity. The unrecovered energy absorbed by the walls is also accounted as a loss. Thus, $Q_{es}$ can be expressed as:

$$Q_{es} = \nu_{w} \gamma_{in} - Q_{ref} - Q_{con} = Q_{col} - Q_{es} = Q_{es} + \nu_{w} \gamma_{in}$$

(7)

where $\nu_{w}$ is the window transmittance (taken as 0.06). $\gamma_{es}$ is the window internal absorption, $Q_{col}$ the HTF film absorption, $Q_{es}$ the wall absorption, and $\nu_{w}$ the recovered energy fraction from walls, which is zero for the reflective cavity and taken as 0.9 for the absorptive preheater based on efficiencies of existing LM nodular heat exchangers (Gandhi, 1963; Schied and Groep, 1981). In theory, $\nu_{w}$ could be > 0 in the reflective cavity if energy extracted by the coolant is recovered; however, its temperature is unlikely to be sufficiently higher than the HTF inlet to facilitate an effective heat transfer.

Considering a continuous HTF film, its mass flow rate ($\dot{m}$) and film thickness at location $x$ ($t_{film}$) can be estimated from:

$$\dot{m} = Q_{es} / (c_{p} \Delta T)$$

(8)

$$t_{film} = \dot{m} / (\rho c_{p} \gamma_{in})$$

(9)

where $\Delta T$ is temperature difference between HTF inlet and outlet, $c_{p}$ the specific heat capacity and density of the HTF, respectively.
4.2.1. Emission

Radiative losses are composed of emissive and reflective components escaping from the aperture through the aperture. To evaluate the emissive loss, the internal cavity area ($A_{in}$) is discretized into a finite number ($n$) of planar surfaces, where each surface is labelled with an index $i$ and has a defined surface area ($A_i$), temperature ($T_i$), emissivity ($e_i$), and a view factor towards the aperture ($F_{Ai}$). Therefore, by utilizing the grey and diffuse assumptions, the summation and reciprocity rules can be used to evaluate the net emissive transfer at the aperture from other surfaces of the enclosure as (Incropera et al., 2017):

\[
\dot{Q}_e = e_{in} \varepsilon_{in} \sum_{i=1}^{n} \frac{T_i^4 - T_e^4}{1 - \varepsilon_{Ai}(1 - \varepsilon_{in})} F_{Ai}
\]  

(1)

where $T_e$ is ambient temperature. By substituting with the mean HTF and internal wall temperatures and emissivities, equation (10) becomes:

\[
\dot{Q}_e = e_{in} \varepsilon_{in} \sum_{i=1}^{n} \frac{T_{Ai}^4 - T_e^4}{1 - \varepsilon_{Ai}(1 - \varepsilon_{in})} F_{Ai}
\]  

(11)

where $t$ and $w$ subscripts denote the HTF film and cavity walls, respectively. The view factors are computed using a MATLAB code by Launier (2004) based on the contour double integral formulation, which is verified for complex geometries (Francisco et al., 2014), with vertices coordinates imported from the CAD model.

4.2.2. Reflection

Reflective loss transmitted through the window can be expressed in terms of geometrical and optical parameters as (Duffie et al., 1988; Zou et al., 2017):

\[
\dot{Q}_{ref} = \frac{1}{1 - (1 - e_{in})(1 - A_{w}/A_{in})} e_{ref} F_{ref}
\]  

(12)

where $e_{ref}$ is an effective cavity emissivity approximated by area-weighted averaging of HTF ($e_{in}$) and cavity wall lining ($e_w$) emissivities:

\[
e_{ref} = (1/A_{in}) \sum_{i=1}^{n} e_i A_{ref,i} + e_w \sum_{i=1}^{n} (A_{ref,i})
\]  

(13)

The transmitted solar input is not diffuse; hence, the first absorption instance by the HTF film ($Q_{in,HTF}$) and cavity walls ($Q_{in,wall}$) are evaluated as:

\[
Q_{in,HTF} = (1-x)\varepsilon_{in} e_{in} F_{ref} P_{in}
\]  

(14)

\[
Q_{in,wall} = x e_{in} e_{w} F_{ref} P_{in}
\]  

(15)

where $x$ is the fraction of transmitted solar radiation striking the walls instead of the HTF film. This fraction depends on the refractive index of the window, distribution and directional vector of the incident solar beams on the window, and cavity width. $x$ can be evaluated using the discrete ordinates model, as demonstrated in Part 3.

Secondary (internal) reflections of the solar input is assumed diffuse, defined here as internal reflections occurring after the solar input strikes any internal surface once, are assumed diffuse. The total absorption from the secondary reflections ($Q_{in,2nd}$) is evaluated by subtracting the initial absorption instances from the total absorption:

\[
Q_{in,2nd} = Q_{in} - (Q_{in,HTF} + Q_{in,wall})
\]  

(16)

\[
= \dot{Q}_{in,HTF} + \dot{Q}_{in,wall}
\]  

(17)

where $\varepsilon_{in}$ is window emissivity. Given the small $Q_{in,2nd}$ quantities (0.9-2.2% of input power), it is assumed to be completely absorbed by the internal surfaces.

4.2.3. Absorption

Utilizing the diffuse re-radiation assumption, $Q_{in,2nd}$ and $Q_{in,wall}$ are distributed on the HTF and walls weighted by each's surface area and emissivity as:

\[
Q_{in,HTF} = \dot{Q}_{in,HTF} + \frac{\varepsilon_{in} e_{in} A_{HTF}}{1} + \frac{\varepsilon_{wall} e_{wall} A_{wall}}{1} Q_{in,wall}
\]  

(18)

\[
Q_{in,wall} = \dot{Q}_{in,wall} + \frac{\varepsilon_{wall} e_{wall} A_{wall}}{1} + \frac{\varepsilon_{in} e_{in} A_{HTF}}{1} Q_{in,HTF}
\]  

(19)

The total window absorption ($Q_{in}$) is evaluated using:

\[
Q_{in} = \varepsilon_{in} e_{in} F_{ref} P_{in} + \varepsilon_{wall} e_{wall} A_{wall} Q_{in,wall}
\]  

(20)

4.2.4. Convection

For an insulated and windowed cavity, wind-induced convection is considered an indirect loss mechanism, which degrades the energy lost through window aborption. Therefore, the wind-induced convection loss is implicit within the $Q_{in,wall}$ and $Q_{in,HTF}$ terms in Equation (7), while the natural convection loss is represented explicitly using the $Q_{n}$ term.

The convective heat transfer coefficient ($h$) of any domain surface is defined as:

\[h = \left(\frac{k_o}{\delta}ight)^{\frac{1}{2}}
\]  

(21)

where $k_o$ is the thermal conductivity of fluid, $\delta$ the characteristic length, and $Nu$ the Nusselt number, which is related to Rayleigh number ($Ra$) via:

\[Nu = c(Ra)^{\frac{1}{2}}
\]  

(22)

where $c$ and $\alpha$ are constants describing the geometry/orientation of the absorbing surface(s) and flow regime, respectively. Constants for different shapes, orientations, flow regimes, and fluids are available in literature (Aydin and Gessoun, 2001; Berkoo and Yant-Hull, 1991; Bennett, 1974; Tsui and Nagan, 1990). Interaction effects between surface radiation and turbulent natural convection are negligible (variables in $Nu$ values < 1 %) for rectangular enclosures with aspect ratios $>3$ (Utechman et al., 2001); hence, $Nu$ is acquired from a decoupled correlation. Constants for rectangular enclosures with inclined surface ($c$) are provided by Cantor (1976). For open aperture cavities, (Gonzales et al., 2015) recommends Clasing (1963). Different literature correlations will be compared against a CFD solution to examine their validity in Part 3.

The calculated $Ra$ values were $256 \times 10^3$ and $1.24 \times 10^6$ for the reflective and absorptive cavities, respectively. Therefore, for a cavity fluid with a Prandtl number $-5.76$, the buoyancy-driven flow is considered fully turbulent in both configurations (Krisnanorti, 1973). Please note that no empirical correlation or analytical expression is available to model the interfacial heating effects on natural convection.

In Part 2, a CFD solution is used to demonstrate the insignificane of such viscous effects on the bulk flow.

The convective loss per configuration is based on the temperature(s) and convective heat transfer coefficient(s) of its respective absorbing surfaces.
\[ Q_{\text{effective}} = A_{\text{eff}} h_{\text{a}} (T_{\text{a}} - T_{\text{en}}) \]  
(23)

\[ Q_{\text{absorpt}} = A_{\text{a}} h_{\text{a}} (T_{\text{a}} - T_{\text{en}}) + A_{\text{a}} h_{\text{e}} (T_{\text{e}} - T_{\text{en}}) \]  
(24)

where \( T_{\text{a}} \) is the effective cavity temperature calculated as:

\[ T_{\text{en}} = \left( \frac{1}{A_{\text{a}}} \right) (T_{\text{a}} A_{\text{a}} + T_{\text{e}} A_{\text{e}} + T_{\text{en}} A_{\text{en}}) \]  
(25)

where \( T_{\text{en}} \) is the window temperature.

5. Results and discussion

The energy performance of the proposed receiver is presented to investigate its feasibility and study the influence of selected parameters on its energy efficiency.

5.1. Energy flow

The rate of energy flow in each cavity configuration is presented in Fig. 6. The overall efficiency of the absorptive cavity is 74% compared to only 31.9% for the reflective cavity. While the latter was successful in entrapping \( > 80 \% \) of transmitted solar radiation into the cavity, more than half of this energy was absorbed by the walls despite their high reflectance. This is a result of the large walls-to-wall area ratio (~4.66), as 52% of all secondary reflections were absorbed by the walls. The absorptive cavity captured \( > 90 \% \) of the transmitted radiation with \( < 8 \% \) unrecovered energy from the walls. While the emissive loss was 4X greater in the absorptive cavity due to its higher temperature and effective emissivity, the total re-entrant losses from its aperture were \( 60 \% \) lower than in the reflective cavity due to the latter’s substantial reflective loss.

The significant wall absorption marks a vital conceptual concern for the reflective cavity, as walls necessitate active cooling to discharge over 39% of the solar input. A full recovery of this energy may boost the efficiency up to 70.3%, which would still be less efficient than its absorptive counterpart. In practice, a full, or even sizeable, recovery from the reflective walls is unlikely, as discussed in section 2.3. Nevertheless, instead of direct energy recovery during nominal operation, heat collected by the coolant could be utilized in preheating the HTF above its melting point. Assuming 80% efficient heat transfer from the reflective walls to cavity fluid for 6 h (sun hours) and 80% efficient heat transfer to the stored liquid at \( 38 \) h (night-time hours), the resultant energy can cover temperature drops up to 225 K at the studied scale. This limitation could be minimised by maximising the wall reflectance and the HTF emissivity. The influences of these two optical properties are addressed in sections 4.4 and 4.5.

5.2. Concentration ratio

Solar concentration is essential at UHT to minimise the energy losses from the aperture. However, beyond an optimal concentration ratio, further concentration might not save sufficient energy to outweigh the increased spillage and CPC losses. To model this effect, \( q_{\text{c}} \) was linearly varied from 75% to 55% to correspond to an increasing \( C_{\text{R}} \) from 1 sun to 20,000 suns, respectively. As indicated in Fig. 7, concentration ratios \( > 1000 \) suns are more justifiable at UHT for an absorptive cavity than a

Fig. 6. Rate of energy flow throughout the (a) reflective and (b) absorptive cavities. Colour convention: red marks losses; green marks energy collection; orange marks input. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)
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reflective cavity.

5.3. Natural convection

Natural convection was found to pose an opposite effect per cavity configuration (Table 4). The cold reflective walls resulted in a lower cavity temperature than the mean HTF film temperature; hence, the resultant buoyancy-driven effect inside the cavity is the extraction of heat from the HTF film. In the absorptive cavity, the temperatures of the absorbing surfaces, HTF film and walls, are lower than the cavity temperature; therefore, the resultant buoyancy-driven effect is conveying heat from the window to the absorbing surfaces. The latter mechanism is favourable, as it passively provides heat recovery from, and cooling to, the window. Nevertheless, the convective magnitude of this mechanism is minor in the absorptive cavity due to the small temperature difference across its domain.

The influence of \( h_{\text{aw}} \) on the overall receiver efficiency is demonstrated in Fig. 5. A typical \( h_{\text{aw}} \) value in a buoyancy-dominated cavity ranges between 1 and 15 W m\(^{-2}\) K\(^{-1}\) (DeAngelis et al., 2011), which may become slightly higher when accounting for the interfacial shearing effects. If viscous effects are ignored, \( h_{\text{aw}} \) would depend on the temperature difference across the domain \( (T_{\text{ah}} - T_{\text{aw}}) \). Accordingly, natural convection posed a more significant impact on the absorptive cavity efficiency at higher temperatures, as shown in Fig. 5(a). Since the favourable convective mechanism in the absorptive cavity occurs only when the cavity is at a lower temperature than the window, this effect, resembled by the positive slope in Fig. 5(b), diminishes at higher receiver temperatures before becoming a less mechanism at temperatures \( > T_{\text{aw}} \). Overall, natural convection is shown to pose minimal influence on the efficiency of a waneden cavity receiver.

5.4. Window temperature

The quasi-steady-state window temperature is estimated by equating the total window absorption to the emissive and convective losses from the window using the parameters displayed in Table 5. In Part 3, the volumetric optical properties of spinline and the discrete ordinates method will be used to verify these results.

Table 4: Calculated temperatures and convection heat transfer coefficients.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Reflective</th>
<th>Absorptive</th>
</tr>
</thead>
<tbody>
<tr>
<td>( T_{\text{aw}}(K) )</td>
<td>1224.5</td>
<td>1641.9</td>
</tr>
<tr>
<td>( T_{\text{ah}}(K) )</td>
<td>2090.00</td>
<td>2152.05</td>
</tr>
<tr>
<td>( T_{\text{aw}}(K) )</td>
<td>916.4</td>
<td>1037.8</td>
</tr>
<tr>
<td>( h_{\text{aw}}(\text{W m}^{-2}\text{K}^{-1}) )</td>
<td>1.1272</td>
<td>0.0105</td>
</tr>
</tbody>
</table>

*The negative sign indicates an opposite heat transfer (energy gain by the absorbing surface due to \( T_{\text{aw}} > T_{\text{aw}} \).*

The evaluated temperatures of the passive-cooled window are below the maximum allowable temperature specified in section 2.3 as 2400 K. Nevertheless, the window is likely to necessitate active cooling to factor in the transient heating/cooling. For zero allowable strain, the thermal stress \( (\sigma_{\text{th}}) \) can be estimated using:

\[
\sigma_{\text{th}} = \frac{E \Delta T_{\text{aw}}}{\text{thickness}}
\]

where \( E \) is Young’s Modulus (232 GPa) and \( \Delta T_{\text{aw}} \) is the thermal expansion coefficient \( \left(5.9 \times 10^{-6} \text{K}^{-1}\right) \). For spinline’s fracture strength of 350 MPa (Sanphra et al., 2011), the active cooling should keep the transient \( \Delta T_{\text{aw}} < 200 \text{ K} \). The utilisation of cavity fluid as window coolant is yet to be investigated.

5.5. Liquid metal emissivity

The HTF emissivity was varied to determine its influence on the receiver efficiency (Fig. 9). Results are compared with data from DeAngelis et al. (2018) of a tubular LM receiver at 1623 K. Overall, the HTF emissivity was found to influence the performance of the absorptive cavity more than the absorptive cavity. The absorptive receiver exhibited comparable efficiencies to the reference receiver. Increasing the HTF emissivity in the absorptive cavity from 0.2289 to 0.85 can substantially reduce energy wasted through cavity walls by 73 %, while improving the solar extractament by 9 %, as illustrated in Fig. 10. Still, the collected energy remains lower than the absorptive cavity without a modified HTF emissivity.

5.6. Wall reflectance

The wall reflectance \( (1 - \eta_{\text{aw}}) \) has an opposite effect on the performance of each cavity configuration, as demonstrated in Fig. 11. Regardless of the configuration, this effect is minimal at wall reflectance < 0.6 but becomes dramatically significant at higher reflectance. This highlights the importance of maximising wall reflectance in the absorptive cavity, whereas wall optical properties are less critical for the absorptive cavity. The reflectivity of silver, at 0.69 µm, drops from 0.972 to 0.939 when temperature increases from 900 K to 1234 K (Kishkara, 1972), which, from Fig. 11, corresponds to decreased reflective receiver efficiency of ~ 8 %. However, this import can be considerably higher for other lowing metals (Kishkara, 1972), which emphasise the importance of preserving the reflective walls at the lowest possible temperature.

6. Conclusion

A new UT window receiver concept was proposed featuring an optically exposed LM HTF to mitigate previous concerns about subjecting solid absorbers to concentrated irradiation. The concept is theoretically investigated for heating liquid tin from 900 K to 1673 K at two possible cavity configurations, using an approximate quasi-steady-state energy model. The presented design and modelling methodology may be
Fig. 8. Effect of convection heat transfer coefficient ($h_{con}$) on the receiver efficiency at different HTF outlet temperatures in the (a) reflective and (b) absorptive cavities. $h_{con}$ is presented here as an absolute quantity, which explains the trends with positive slopes, where heat is transferred to, rather than from, the absorbing surface.

Table 5
Evaluated parameters used to estimate the quasi-steady-state window temperature.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Reflective</th>
<th>Absorptive</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{in}(K)$</td>
<td>300</td>
<td>300</td>
</tr>
<tr>
<td>$T_{out}(K)$</td>
<td>910.4</td>
<td>1628.7</td>
</tr>
<tr>
<td>Wind speed (m/s$^{-1}$)</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>$h_{conv}(W/m^2\cdot K)$</td>
<td>64000</td>
<td>64000</td>
</tr>
<tr>
<td>$h_{conv,rel}(W/m^2\cdot K)$</td>
<td>2726</td>
<td>10442</td>
</tr>
<tr>
<td>Total window absorption (kW)</td>
<td>309.64</td>
<td>309.74</td>
</tr>
<tr>
<td>Total convective loss from window (kW)</td>
<td>229.33</td>
<td>197.49</td>
</tr>
<tr>
<td>Total emissive loss from window (kW)</td>
<td>81.31</td>
<td>66.25</td>
</tr>
<tr>
<td>$T_{rel,0D}$</td>
<td>3099.08</td>
<td>2102.05</td>
</tr>
</tbody>
</table>

generalized for broad receiver analysis.

The main findings are summarized as follows:

1. Using a LM as the sole absorber in a reflective cavity would result in poor efficiencies (<40%) attributed to the substantial wall absorption and reflective loss. Higher efficiencies (~70%) are attainable by employing the LM film as an internal reflector/moderator to the concentrated solar input, while the majority of absorption is handled by the absorptive cavity via radiative wall absorber.

2. Reflective cavity efficiency is found to be sensitive to the internal optical properties than the absorptive cavity. While some measures were proposed to alleviate this limitation, it would still be less efficient than an equivalent absorptive cavity.

3. Reflective cavity walls require lining with a highly reflective and thermally resistant material, which may be unachievable, as metallic

Fig. 9. Effect of HTF emissivity ($\varepsilon_{HTF}$) on the receiver efficiency in the (a) reflective and (b) absorptive cavities. The emissivity values of [Djelalic et al. (2015)] data are based on the emissivity of the absorber tubes instead of optically exposed HTF.

Fig. 10. Energy flow diagram through the reflective cavity with an improved HTF emissivity of 0.05. Colour convention: red marks losses routes; green marks collection routes. Colour convention: red marks losses; green marks energy collection; orange marks input. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)
Fig. 11. Effect of cavity wall reflectance on the receiver efficiency at both cavity configurations.

reflectance deteriorate with temperature. Protecting reflectance at UHT, and under exposure to corrosive metal vapours, necessitates substantial cooling. While using the LMT HTF as a coolant is found unbeatable, the cavity fluid is projected to be an effective alternative; yet, energy recovery is unlikely to be as efficient as in the absorptive cavity.

4. Sealing the cavity aperture is required to protect against oxidizing ambient effects. A transparent ceramic window was proposed to exploit its superior thermo-mechanical and optical properties at UHT compared to conventional glazing materials. Preliminary qualitative and quantitative (thermal analysis) assessments demonstrated its potential and challenges, including requirement for active cooling. Further investigations are still required to determine its practical feasibility at large scales. While currently undeveloped for this application, fluoride seal may eventually become an optically efficient alternative to windows.

5. At UHT, increasing concentration ratios beyond 1000 suns is particularly advantageous for absorptive cavity receivers.

6. Natural convection is unlikely to pose any significant influence on the energy performance of the windward receiver.

Some practical and operational challenges were emphasised and conceptually addressed, including manufacturing of the components and energy recovery from the walls. The feasibility of the suggested measures are yet to be practically investigated. In Parts 2 and 3, a transient radiation-CFD coupled analysis will be described to verify the results and appraise the employed assumptions of the presented analytical model. The promising performance of the absorptive cavity prompts performing a techno-economic and environmental analysis to justify, or improve, the suggested design selections made in this paper for a particular UHT application, such as a thermochemical process.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have influenced the work reported in this paper. 

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References


