Thermohydraulic analysis of single phase heat transfer fluids in CSP solar receivers

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Abstract

Theoretical modelling techniques are used to compare the thermohydraulic performance and thermal storage characteristics of molten salt, liquid sodium, and lead-bismuth in a CSP solar receiver concept. For molten salt, the performance of a number of heat transfer augmentation techniques are also studied. Sodium and lead-bismuth both yield excellent receiver thermal efficiency (max $\sim$92%), when compared to molten salt (max $\sim$ 90%), due to high thermal conductivity values that lead to large heat transfer coefficients. A high pressure drop penalty for lead-bismuth largely offsets its thermal performance gain over molten salt, however sodium retains its advantage as a receiver working fluid with a low pumping parasitic. The implementation of heat transfer enhancement techniques can significantly improve the performance of a molten salt receiver when compared to smooth tube designs. The low specific heat capacity and high unit cost of lead-bismuth is prohibitive towards its use as a storage medium in storage-integrated plant designs, resulting in very high LCOE values. Sodium is the most economically feasible fluid for systems with low storage (< 3 hour), however the low per-unit cost and high specific heat capacity of molten salt means that this is the most effective working fluid in systems with larger storage requirements.

Keywords: Concentrated solar power, solar receiver, molten salt, liquid metals, thermohydraulic performance, thermal storage

1. Introduction

Concentrated solar power (CSP) is one of the fastest growing renewable energy technologies [1]. CSP has the potential to contribute towards a significant proportion of commercial electricity generation in years to come, with a 12% share in global electricity capacity forecast for 2050 [2]. A key advantage of CSP over other commercial renewable energy technologies lies in its ability to store thermal energy. Thermal energy storage (TES) allows for generation of electricity at times of little or no solar exposure, adding to the value and flexibility of CSP [3]. A critical challenge with CSP lies in its current high levelized cost of electricity (LCOE) relative to other commercial generation technologies, hindering its competitiveness in the energy market. Research and development into CSP is focussed on...
making the technology attractive to electricity producers and policy makers, through performance improvements and capital cost reductions [4].

CSP technologies are categorised by the manner in which sunlight is concentrated onto the solar receiver; parabolic trough, parabolic dish, linear Fresnel and power tower. Parabolic trough plants currently make up the majority of CSP systems in operation [5], however there are significant potential improvements to power tower systems that can ensure its future as the dominant CSP technology [4]. High fluid outlet temperature (> 1000°C) and concentration ratio (> 10^3 suns) capabilities of power tower systems make it a promising technology in which to increase CSP
penetration in the market going forward [6]. A schematic of a CSP tower system is shown in Figure 1.

The power tower system uses a large number of computer controlled heliostats to track and reflect incident sunlight onto the target receiver, fixed atop a tower structure. The role of the receiver is to convert concentrated solar energy from the heliostat field into workable thermal energy in a heat transfer fluid (HTF). The high temperature HTF can either pass to the turbine directly (as is the case for many water/steam HTF systems) or indirectly via a thermal storage tank where it is later used to generate electricity via the power block (steam generator, turbine, condenser, electrical generator). The success of a CSP plant is dependent on the performance of each individual subsystem, with the overall plant efficiency described by the following [7]:

$$\eta_{\text{plant}} = \eta_{\text{optical}} \cdot \eta_{\text{receiver}} \cdot \eta_{\text{transport}} \cdot \eta_{\text{storage}} \cdot \eta_{\text{power}}$$

(1)

Receiver thermohydraulic performance is of critical importance as the component effectively acts as the link between the solar collector and the electricity generator. There are numerous power tower receiver designs and concepts using either a solid, liquid, or gaseous HTF [6], however liquid tubular receivers represent the most popular configuration, largely due to its relative simplicity in design and operation and similarity with traditional heat exchanger technology [8]. Tubular receivers have been in use since the early stages of CSP power tower development, such as in the pioneering 10 MW<sub>e</sub> plants at Solar One and Solar Two in the 1980’s and 1990’s [9]. The receiver is constructed of banks of tubes that carry a HTF, with the HTF temperature raised as the tubes intercept an envelope of concentrated sunlight from the heliostat field. The receiver is classified as being external or internal in its configuration. Tubes on an external receiver are typically formed in a quasi-billboard shape, and can be used as a flat panel billboard receiver for equator facing heliostat fields, or multiple billboard panels can be arranged to approximate a cylindrical shape for heliostat fields that surround a centrally located tower/receiver [10]. The tubes in an internal configuration are placed in a box-like cavity structure that allows concentrated sunlight through an aperture. Internal cavity receivers offer greater protection for the receiver tubes to environmental conditions than external receivers, affecting lower heat losses, however the limited view factor offered by the cavity aperture means that internal cavity receivers may suffer greater optical losses through spillage.

The selection of an appropriate receiver HTF is one of the most important considerations made at the design stage.
Various different HTF such as water/steam, molten salts, and liquid metals have been tested and operated in tubular receivers since the 1980’s. Water/steam is a popular HTF in tower systems, where the receiver effectively acts as the steam generator for the power cycle. The direct nature of steam generation in the receiver reduces the complexity of the plant, and may lead to an improved overall plant efficiency [11]. Water/steam systems are used in several CSP tower projects throughout the world, such as at the world’s largest CSP plant - the 377 MW$_e$ Ivanpah facility [5]. Water/steam systems have limited short term energy storage that can be used as a buffer between cloud transients through the steam accumulation technique [7], such as at the PS10 and PS20 facilities in Seville, Spain, where 1 hour TES is possible [5].

A single phase receiver working fluid may be integrated into a relatively straightforward sensible heat storage system, either as the storage medium itself in a two-tank direct system, or in conjunction with a more appropriate storage medium in an indirect system. From a dispatchability standpoint, molten salt represents a very attractive HTF for long term thermal storage due to its large specific and volumetric heat capacity [12]. The value adding nature of dispatchable electricity is a key to the attractiveness of CSP, with the Solar Two project demonstrating commercial scale tower system with 3 hours of molten salt storage in the 1990’s [9]. The 19.9 MW$_e$ Gemasolar plant in southern Spain took the molten salt storage concept further with 15 hours of storage, allowing for around the clock electricity production [1]. The 110 MW$_e$ Crescent Dunes Solar Energy plant in NV, USA is currently the largest tower plant in the world to employ molten salt receiver technology, with 10 hours storage capacity [13]. The heat transfer performance of molten salt may be improved with heat transfer enhancement techniques. Yang et al. [14] reported a Nusselt ($N_u$) number enhancement of three when comparing molten salt heat transfer in a spirally grooved tube to that of a smooth tube. There are numerous heat transfer enhancement techniques that can be extended towards molten salt flows that are discussed further in this paper.

Liquid metals such as sodium (Na) and lead-bismuth eutectic (Pb-Bi) have excellent heat transfer characteristics that could make them suitable candidates for use in solar thermal receivers for future projects [15]. Liquid sodium was trialled in the 1970’s and 1980’s as a receiver working fluid and storage medium at the IEA-SSPS facility in Almeria, Spain [16]. Sodium cooled receivers tested at the IEA-SSPS facility could operate to > 2.5 MW/m$^2$ [17], significantly greater than the heat flux capabilities of molten salt (~ 0.8 MW/m$^2$) and water/steam systems (~ 0.6 MW/m$^2$) [10]. More recently, the Vast Solar facility, under construction in NSW, Australia, pioneered the use liquid sodium as the HTF with 3 hours of TES [18]. The relative lack of development of sodium as a HTF in modern CSP plants may be attributed to inherent operational hazards. As an alkali metal, liquid sodium is flammable in air and explosive in water, these volatile properties lead to a sodium spray fire at the SSPS facility in 1986, shutting down sodium testing at the facility thereafter [15]. There are currently no solar receivers in operation using lead-bismuth eutectic. However, it is relatively less hazardous working fluid than sodium, and its excellent heat transfer characteristics and operational experience from the nuclear industry means it may become a useful receiver HTF in future projects [19]. Tube wall to fluid heat transfer coefficients associated with liquid sodium can be an order of magnitude greater than molten salts, due to possessing a thermal conductivity that is approximately two orders of magnitude higher [20]. The advantageous
heat transfer properties of liquid metals as candidate receiver HTF was highlighted in a study by Boerema [12]. It was
found that liquid sodium receivers could be sized significantly smaller, and achieve a thermal efficiency increase over
receivers using Hitec molten salt due to higher solar concentration ratios attainable.

Optimizing the receiver design in order to deliver maximum thermohydraulic performance is a key aspect of CSP
research and development [4]. This can be achieved by using an appropriate HTF, affecting lower thermal losses
by reducing the heat transfer area and tube diameter, and utilising a large temperature difference across the receiver
inlet-outlet for greater thermodynamic performance. Although there may be improved heat transfer performance
associated with liquid metals in comparison to molten salt, TES capabilities and the possibility of employing heat
transfer enhancement techniques to molten salt mixtures may act to equalize the potential of each fluid as a suitable
receiver HTF.

The focus of this study is to investigate the suitability of a number of different single phase heat transfer fluids
in a receiver concept. Theoretical heat transfer models are used to predict thermal and hydraulic performance of
a commercial molten salt mixture known as solar salt, liquid sodium, and lead-bismuth eutectic in a smooth tube.
A number of different heat transfer enhancement mechanisms are investigated for the molten salt fluid. A range of
receiver geometries are investigated for the various HTF under a fixed thermal rating, allowing for an exploration of
the pertinent performance characteristics. There are examples in literature where authors have presented and discussed
significant properties of different HTF pertinent to solar receiver operation, however this paper addresses the issue by
offering a discussion on the various fluids as well as directly comparing them through receiver modelling under typical
operating conditions, thus allowing for the identification of various benefits and shortcomings of each fluid in a typical
CSP design.

2. HTF properties

A variety of molten salt mixtures are commercially available, such as solar salt, Hitec, and Hitec XL [21], however
this study is particularly concerned with the solar salt mixture, due to its popularity in a number of CSP projects and
higher thermal stability limit. Sodium and lead-bismuth eutectic are the liquid metals investigated in the current work.

- Solar salt is composed of sodium nitrate and potassium nitrate (60 wt % NaNO₃ + 40 wt % KNO₃), with a
  liquidus temperature range of 260 – 621°C, and unit cost of US $0.5/kg

- Liquid sodium (100 wt % Na) has a liquidus temperature range of 98 – 881°C, and unit cost of US $2/kg

- Lead-bismuth eutectic (44.5 wt % Pb + 55.5 wt % Bi) has a liquidus temperature range of 125 – 1670°C, and
  unit cost of US $13/kg

The unit cost for each fluid is given by Pacio & Wetzel [22]. Some desirable characteristics of HTF for solar
thermal receivers include;
High heat transfer coefficients ($h$) so as to minimise the temperature drop between the heat transfer surface and fluid, instigating higher thermal efficiencies and reduced thermomechanical stresses [12]. A high HTF conductivity ($k$), such as that associated with liquid metals, results in greater heat transfer coefficients. Heat transfer enhancement techniques may be used to augment the heat transfer performance of molten salt mixtures in tubes [14].

A broad liquidus temperature range allows greater scope for manipulation of sensible thermal energy storage. Low melting temperatures reduces the risk of material freezing in the system, thereby avoiding a complex remelting procedure [7]. Thermal stability at elevated temperatures allows for a high HTF outlet temperature for more efficient thermodynamic power cycles [6].

A high specific heat capacity ($C_p$) results in more cost-effective sensible heat storage solutions. A greater volumetric heat capacity ($\rho \cdot C_p$) means lower fluid volumes are required to store a particular quantity of thermal energy.

Low raw material cost and high availability will help reduce costs associated with the plant, maximising overall performance and cost-effectiveness for significant thermal energy storage.

Low density ($\rho$) results in a lower receiver pressure drop ($\Delta P$) and therefore lower pumping requirement, a parasitic load that reduces the net power output of the plant.

Ease of handling and environmental impact must be considered at the design stage. Certain HTF may be volatile and require special handling considerations, incurring greater system complexity and cost.

Low vapour pressure for operation at high temperature and for use as a TES medium. Liquid sodium, lead-bismuth, and solar salt all have low vapour pressures.

The HTF should be compatible with receiver tubes and headers, process piping, and containment material in terms of corrosion. This is particularly important if the receiver is required to have a long service life.

Material freezing is a particular concern with molten salt mixtures due to high melting temperatures (> 200°C). Molten salts experience a volumetric expansion upon melting (4.6% for the solar salt composition [23]), therefore allowing the material to freeze in tubes and then subjecting components to a subsequent thawing procedure poses a potential risk to mechanical damage through plastic deformation [9]. Great care must be taken in receiver and power plant control during abnormal operational scenarios (component/instrumentation failure) and transient events (cloud passages). Efficient thermal insulation and heat tracing is required to keep the fluid molten in plant pipework during off-operation hours, and the high melting temperatures associated with molten salts will effectively decrease the net power output due to an increase in the heat trace parasitic. Freezing concerns are lessened with liquid metals, due to significantly lower melting temperatures.
The compatibility of the receiver working fluid with materials in contact is critical. The heat transfer fluid acts as the electrolyte in the corrosion process, attacking the containment material (receiver tubes/headers, process piping, storage tanks) [11]. Significant levels of corrosion may inflict costly mechanical damage to the receiver, through corrosion accelerating the fatigue process when they act simultaneously, and through stress corrosion cracking. The corrosive nature of solar salt operating at high temperatures is widely reported for a variety of different tube materials. In terms of commonly used receiver tube materials, a corrosion rate of $6 - 15 \mu m/year$ has been reported for 304 and 316 Stainless Steels at 570°C [24], however it has a slow corrosion rate with Incoloy 800 up to temperatures of 595°C [25]. Sodium exhibits the greatest compatibility with stainless steels with nickel based alloys of the HTF investigated [22]. A very large corrosion rate of $> 250 \mu m/year$ for lead-bismuth attacking austenitic stainless steels and nickel based alloys has been reported [11], largely due to the high solubility of the alloying elements (Fe, Cr, Ni). Corrosion may be mitigated somewhat in lead-bismuth loops by controlling the oxygen content and by introducing inhibitors in the form of protective zirconia films on the surface [19]. A highly corrosive working fluid will require a robust receiver design to sustain a long operational life. This would incur a greater material cost and the additional thermal resistance associated with wall thickness would lead to reduced thermal performance and increased thermomechanical stresses. A receiver carrying a highly corrosive fluid may require a number of replacements over the life of the plant, which is detrimental to plant economics.

Graphs of fluid density, specific heat capacity, thermal conductivity, dynamic viscosity, and Prandtl number ($Pr = \frac{C_p\mu/k}{\nu}$) are shown in Figures 2 - 7 for the fluids investigated. Correlations used to calculate the temperature dependent fluid properties are taken from Zavoico [26] for solar salt, Fink & Leibowitz [27] for liquid sodium, and Sobolev [28] for lead-bismuth eutectic.

Liquid sodium has the lowest density of the three fluids, below $10^3 \text{ kg/m}^3$ across its liquidus temperature. Lead-bismuth has by far the greatest density, at $\sim 10^4 \text{ kg/m}^3$ across its temperature range, $\sim 5$ times more dense than molten salt, and an order of magnitude more dense than liquid sodium. A high density will act to increase the HTF
Reynolds number inside the tube, enhancing convective heat transfer in the turbulent regime and improving thermal performance. A high density working fluid may require a significant pumping effort to overcome tube pressure drop, reducing the net power output from the plant.

Liquid sodium has a slightly lower specific heat capacity \((C_p)\) than molten salt, however lead-bismuth is an order of magnitude smaller than both sodium and molten salt. A comparison of volumetric heat capacity is useful for the investigation of the thermal storage merits of each fluid, and is shown in Figure 4. The large specific heat capacity of the molten salt relative to the liquid metals means that a greater amount of thermal energy can be stored for the same volume of fluid. It is interesting to note that lead-bismuth outperforms liquid sodium in volumetric heat capacity terms, despite having a lower specific heat capacity (Figure 3). The low specific heat capacity of lead-bismuth is counterbalanced by a significantly higher density (Figure 2). This means that lower volumes of lead-bismuth are required to store thermal energy than sodium. A comparative cost breakdown of the fluids is presented in the results section for the TES analysis, providing further insight into the merits of each fluid as a storage medium.

![Figure 3: Specific heat capacity \((C_p)\) as a function of temperature \((T)\)](image)

![Figure 4: Volumetric heat capacity as a function of temperature \((T)\)](image)
The liquid metal candidates exhibit lower dynamic viscosities across the operational temperature range than molten salt (Figure 6), this is desirable in order to affect turbulent flow inside the receiver tubes, maximising thermal performance and minimising friction factor and pressure drop as a result. At higher temperatures, lead-bismuth has a slightly higher viscosity than molten salt. Liquid metals are characterised by high thermal conductivities (Figure 5), and low specific heat capacities (Figure 3), which results in very low Pr values \((Pr << 1)\) (Figure 7). The thermal conductivity of both liquid metals can be orders of magnitude greater than molten salt. These high conductivity values of liquid metals will yield significantly higher heat transfer coefficients in the receiver tubes when compared to molten salt, resulting in lower tube temperatures, leading to greater thermal efficiencies and reduced thermomechanical stresses.

Liquid metals behave differently than more conventional, higher \(Pr\) fluids, in terms of heat transfer. The thermal boundary layer \((\delta_{th})\) is much thicker than the viscous boundary layer \((\delta_u)\), meaning that the rate of heat diffused from the wall to fluid core is much larger than that of momentum diffusion. The temperature and velocity profiles for varying \(Pr\) fluids is shown in Figure 8, for two-dimensional flow over a plate at constant wall temperature [29].
\[ Pr \ll 1, \ \delta_{th} > \delta_u \]

\[ Pr \sim 1, \ \delta_{th} \sim \delta_u \]

\[ Pr \gg 1, \ \delta_{th} < \delta_u \]

For laminar fluid flows, conventional forced convection heat transfer correlations for constant wall temperature and heat flux can be applied to liquid metals in the same manner as higher \( Pr \) fluids as the velocity profile is non-
consequential, however this is not the case in the turbulent regime ($Re > 10^4$) [29]. The large thermal conductivity of liquid metals means that molecular conduction is the dominant transport mechanism of thermal energy, even into the core flow region [30]. This differs from higher $Pr$ fluids, where thermal energy transport by eddy currents has a large influence on forced convective heat transfer. The discrepancy in the transport of thermal energy and momentum means that forced convective heat transfer correlations for turbulent flow derived for more conventional higher $Pr$ fluids such as air, water, molten salts, and oils, are not compatible with liquid metals.

3. Thermohydraulic model

The thermohydraulic performance of each HTF for a receiver application is perhaps most appropriately studied by simulating their use under typical operating conditions. Theoretical thermal and hydraulic models are used to investigate the optimum size of a single pass billboard receiver design for each HTF, thus allowing for a straight comparison between the fluids. A 5 MWth rating is used as a starting point in the analysis, with the limits of the HTF explored by varying receiver geometry, while respecting the film temperature limit. The geometry of the quasi-flat panel is varied by adjusting the number of tubes on the receiver, and tube length, thus allowing for variations in both heat transfer area and aspect ratio (AR, active heat transfer area length divided by width). Such a receiver panel may be used as a standalone billboard receiver for an equator facing heliostat field, or multiple panels may be used to approximate a cylindrical receiver on the tower for a surrounding field. A constant tube geometry of 0.008 m outer diameter and 0.001 m wall thickness is maintained for simplicity. The HTF mass flow rate is adjusted in order to deliver a desired outlet temperature of 565°C from a set inlet temperature of 290°C, under the various modelling inputs and boundary conditions. These temperatures are typical of a solar receiver operation, corresponding to those used on the Solar Two molten salt receiver [26]. The 290°C inlet temperature allows a buffer against HTF freezing, with the molten salt being the most critical of the fluids investigated, while the 565°C fluid outlet temperature corresponds to steam inlet temperatures for a conventional steam turbine [31]. Simulating under these conditions serves to demonstrate the performance of each candidate fluid in a typical CSP configuration.

The tubes are constructed of Alloy 800H material, a Ni-Fe-Cr alloy that is suitable for high temperature receiver construction due to its corrosion resistance, and excellent creep-rupture and fatigue strength [32]. Alloy 800H is covered by the ASME Boiler & Pressure Vessel Code: Section III - Subsection NH [33], which stipulates design guidelines for high temperature components in service. Material creep and fatigue data is supplied up to temperatures of 1033 K, meaning confidence in mechanical reliability predictions is diminished at higher temperatures. Therefore, the maximum receiver surface temperature must remain within this upper limit.

For simplicity, a uniform heat flux boundary condition is assumed across the receiver panel. This is essentially an assumption of an ideal heliostat aiming strategy, which aims to homogenise the heat flux distribution on the receiver [34]. Due to the heat flux uniformity, heat transfer modelling need only be performed on a single tube and the results assumed similar across all receiver tubes. The tube is discretised into a number of finite elements in the axial ($z_{el}$)
and circumferential directions (θel), with one-dimensional heat conduction assumed across the wall thickness. The
discretisation of the tube into elements allows for the calculation of tube temperatures in three dimensions (r, θ, z),
thus allowing for accurate predictions of thermal energy losses. A description of tube discretisation for the modelling
procedure is given in Figure 9.

![Figure 9: Schematic of receiver tube discretisation into axial and circumferential element](image)

The tubes on a billboard receiver are closely aligned so that only one half of the tube is exposed to the incident
heat flux (Q^\text{so})). Indeed, the curvature of the receiver tubes means that the incident heat flux is also non-uniform on
the side exposed to concentrated solar energy. The incident heat flux therefore varies around the tube circumference,
and is described by a cosine function;

\[
Q^\text{so, zo, \theta, j=0} = \begin{cases} 
Q^\text{zo} \cos(\theta) & 0 \leq \theta \leq \pi/2 \\
0 & \pi/2 \leq \theta \leq 3\pi/2 \\
Q^\text{zo} \cos(\theta) & 3\pi/2 \leq \theta \leq 2\pi 
\end{cases} 
\]

(2)

Where θ = 0 is at the tube crown on the side receptive to concentrated sunlight. The model conducts an energy
balance between the heat absorbed by the fluid and heat lost to the environment. Heat transfer calculations are carried
out on each tube axial element in an iterative procedure described below:

1. As an initial iteration step (j = 1), the incident power falling on the tube axial element (Q^\text{net, zo, j}) is assumed to
be fully absorbed by the HTF ($Q_{f,zel,j} = Q_{net,zel,j}$).

2. Bulk fluid temperatures ($T_{f,zel,j}$) are calculated by assuming steady flow conditions along the length of the tube 
($Q_{f,zel,j} = \dot{m}_{f,tube,j} C_{p,f,zel,j} (T_{out,zel,j} - T_{in,zel,j})$).

3. Tube wall temperatures are computed in the axial and circumferential directions, on the inside and outside of the 
tube wall ($T_{si,zel,θel,j}, T_{so,zel,θel,j}$). This allows for the calculation of thermal energy losses through convection, 
radiation, and reflection at different positions on the tube ($Q_{error,zel,j} = Q_{f,net,zel,j} - (Q_{f,net,zel,j} + Q_{l,conv,zel,j} + Q_{l,rad,zel,j} + Q_{l,ref,zel,j})$).

4. For each axial element ($z_{el}$), the error that exists between the sum of the thermal energy absorbed by the fluid 
plus heat losses, against the power input, is used as the cost function in which to modify inputs into the next 
iteration ($Q_{error,zel,j} = Q_{net,zel,j} - (Q_{f,net,zel,j} + Q_{l,conv,zel,j})$).

5. For following iterations, the modelling error ($\pm$) is added to the previous assumption of heat transferred to the fluid ($Q_{f,net,j+1} = Q_{f,net,j} + Q_{error,net,j}$), and steps 2 through 5 are repeated.

6. After a series of model iterations, the sum of the error on all tube elements converges towards an acceptable 
value ($\sum_{j=1}^{N_{zel}} |Q_{error,zel,j}| < 1 W$), confirming an energy balance on the receiver.

A flow diagram describing the energy balancing procedure used in the thermal model is shown in Figure 10.

For each iteration of the model ($j$), the mass flow rate through the tube ($\dot{m}_{f,tube,j}$) must be iterated in order to deliver the desired outlet temperature through a separate iterative procedure. The fluid temperature development in
the axial direction is calculated using the following equation:

$$T_{out, z_{el}} = T_{in, z_{el}} + \frac{Q_{f, z_{el}}}{m_f, tube C_{p, f, z_{el}}}$$  \hspace{1cm} (3)

The outlet temperature of one axial element forms the inlet temperature to the next element in the flow direction ($T_{in, z_{el}} = T_{out, z_{el-1}}$). The bulk fluid temperature is used to define fluid properties for the model ($\rho, C_p, \mu, k$), as the fluid temperature increases along the length of the tube:

$$T_{f, z_{el}} = \frac{(T_{in, z_{el}} + T_{out, z_{el}})}{2}$$  \hspace{1cm} (4)

As the initial model iteration assumes a full transfer of incident heat to the fluid (Step 1), the heat flux at the wall-fluid interface will be equal to that falling on the outer surface ($Q_{s_i, z_{el}, \theta_{el}, j=1} = Q_{s_o, z_{el}, \theta_{el}}$). However, the heat flux profile will deviate for subsequent iterations ($Q_{s_i, z_{el}, \theta_{el}, j>1} \neq Q_{s_o, z_{el}, \theta_{el}}$), as it is adjusted according to local thermal losses on the tube (Step 3), which are non-uniform around the circumference due to variations in local wall temperature and view factor. The wall temperatures on the tube are used to calculate thermal energy losses, and these are established from the bulk fluid temperature ($T_{f, z_{el}}$) using the heat transfer coefficient ($h_{f, z_{el}}$) and local heat flux ($Q_{s_i, z_{el}, \theta_{el}}$). The heat transfer coefficient is initially found through the Nusselt number ($Nu_{f, z_{el}}$):

$$h_{f, z_{el}} = \frac{Nu_{f, z_{el}} k_{f, z_{el}}}{D_i}$$  \hspace{1cm} (5)

Where,

$$Nu_{f, z_{el}} = 6.3 + 0.0167 Re_{f, z_{el}}^{0.85} Pr_{f, z_{el}}^{0.93}$$  \hspace{1cm} (6)

is from Sleicher & Rouse [35] for turbulent liquid metal flows, ($0.004 \leq Pr \leq 0.1, 10^4 \leq Re \leq 10^6$). For the molten salt fluid, the Gnielinski [36] correlation is applied instead ($0.5 \leq Pr \leq 2 \times 10^3, 3 \times 10^3 \leq Re \leq 5 \times 10^6$):

$$Nu_{f, z_{el}} = \frac{(f_{el}/8)(Re_{f, z_{el}} - 1000) Pr_{f, z_{el}}}{1 + 12.7 (f_{el}/8)^{1/2} (Pr_{f, z_{el}}^{2/3} - 1)}$$  \hspace{1cm} (7)

The heat transfer coefficient is used to calculate the inner wall surface temperatures from the following:

$$T_{s_i, z_{el}, \theta_{el}} = T_{f, z_{el}} + \frac{Q_{s_i, z_{el}, \theta_{el}}}{h_{f, z_{el}}}$$  \hspace{1cm} (8)

The inner wall temperatures are used to indicate the maximum film temperature of the HTF inside the tube, a limiting factor in receiver design as thermal stability becomes problematic near the point of phase change. Outer wall temperatures are then calculated using the inner wall temperatures, and assuming one dimensional conduction across.
the tube wall;

\[ T_{so, zel, \theta e} = T_{si, zel, \theta e} + \frac{Q_{si, zel, \theta e}}{k_{i, zel, \theta e}} \left( r_o \left( \ln \frac{r_o}{r_i} \right) \right) \] (9)

Ambient conditions of a 4 m/s wind speed and 298 K air temperature are assumed in the analysis for the calculation of convective and radiative heat losses, corresponding to typical operational conditions of a CSP plant. The total convective loss from the receiver \( (Q_{l, \text{conv}, \text{rcv}}) \) is calculated by assuming the billboard panel as a vertical flat plate profile. The average surface temperature of the receiver \( (\bar{T}_{so}) \) is used to calculate free, mixed, or forced convection losses using \( Nu_{\infty} \) correlations from Siebers [37] and Siebers & Kraabel [38].

\[ Q_{l, \text{conv}, \text{rcv}} = h_{\infty} A_{\text{rcv}} (\bar{T}_{so, \text{rcv}} - T_\infty) \] (10)

Radiative losses \( (Q_{l, \text{rad}, \text{rcv}}) \) are dependent on the receiver surface temperature, emissivity and view factor between the receiver tubes and environment. The emissivity varies between \( \sim 0.8 - 0.9 \) as a function of tube surface temperature, and is calculated for Pyromark high temperature black paint [39]. The view factor \( (F_{\text{view}}) \) is established using the crossed strings method [40]. The radiative heat loss is found using the following equation;

\[ Q_{l, \text{rad}, \text{rcv}} = \sum_{\text{tube}} \left( \sum_{zel} \sum_{\theta e} N_{\text{tube}} \frac{\varepsilon \sigma}{A_{\theta e}} \left( T_{4, so, zel, \theta e} - T_\infty \right) \right) \] (11)

The Stefan-Boltzmann constant \( (\sigma) \) is \( 5.67 \times 10^{-8} \text{W/m}^2\text{K}^4 \). Reflective losses \( (Q_{l, \text{ref}, \text{rcv}}) \) are a function of the tube surface coating used to maximise solar absorptance, and minimise thermal emittance of the heat transfer surface. A maximum absorptivity of \( a = 0.95 \) for the Pyromark surface coating is assumed for the crown \( (\theta = 0) \), with the absorptivity then varying as a function of the irradiance incidence angle according to equations given by Ho et al.[39]. The reflective losses are calculated from the following;

\[ Q_{l, \text{ref}, \text{rcv}} = \sum_{\text{tube}} \left( \sum_{zel} \sum_{\theta e} N_{\text{tube}} Q_{so, zel, \theta e} (1 - a_{\text{zelo}, \theta e}) \right) \] (12)

The total thermal energy lost to the environment is used to establish receiver thermal efficiency, as per the following;

\[ \eta_{th, \text{rcv}} = \frac{(Q_{\text{net}, \text{rcv}} - Q_{l, \text{conv}, \text{rcv}} - Q_{l, \text{rad}, \text{rcv}} - Q_{l, \text{ref}, \text{rcv}})}{Q_{\text{net}, \text{rcv}}} \] (13)

An increase in pressure drop will result in an increase in the necessary pumping power. The pumping power required to shuttle the HTF through the system presents a significant parasitic load on the CSP plant [4], reducing the overall system efficiency. A hydraulic analysis is therefore used to evaluate the HTF pressure drop in the receiver for the various HTF and configurations investigated in the study. The pressure drop in the receiver is first established by
summing the pressure drop across each axial element in series in a single tube, as the tubes form a parallel flow path for the working fluid;

\[ \Delta P_{\text{rcv}} = \sum_{z_{el}=1}^{N_{zel}} f_{z_{el}} \left( \frac{L_{z_{el}}}{D_i} \right) \left( \frac{\rho f_{z_{el}} U_{f,z_{el}}^2}{2} \right) \]  

(14)

The friction factor is evaluated by using the Petukhov [41] equation for fully developed turbulent flow in smooth tubes \( 3 \times 10^3 \leq Re \leq 5 \times 10^6 \):

\[ f_{z_{el}} = \left( 0.79 \ln \left( Re_{f,z_{el}} \right) - 1.64 \right)^{-2} \]  

(15)

The necessary pumping power required to overcome the pressure drop and pump the HTF through the receiver is calculated with an assumed pump efficiency of \( \eta_{\text{pump}} = 0.8 \);

\[ W_{\text{rcv}} = \frac{\dot{m}_{f,\text{rcv}} \Delta P_{\text{rcv}}}{\eta_{\text{pump}} \rho_f} \]  

(16)

The thermal energy storage (TES) potential of each HTF is compared by simulating their use as an energy storage medium in a sensible two-tank direct system. The two-tank sensible system is one of the more straightforward TES concepts, and has been implemented successfully in a number of different commercial CSP facilities [5]. The HTF in a two-tank direct system is cycled between a hot tank and a cold tank, with heat from the HTF used to generate superheated steam for the power cycle from a feed-water supply in a steam generator. The ability to utilise the HTF as both a receiver working fluid and storage medium in a direct system is advantageous over indirect systems that require separate working fluids with additional costs and complexity of more storage units and heat exchangers needed to transfer thermal energy between the fluids. In the direct system, the temperature differential between the hot and cold storage tanks is used to generate steam in a heat exchanger, with the stored energy potential calculated using the following equation;

\[ \Delta Q_{h\rightarrow c} = m_f \int_{T_{f,h}}^{T_{f,c}} C_{p,f}(T_f) \, dT_f \]  

(17)

Where \( m_f \) is the HTF mass, \( T_{f,h} \) and \( T_{f,c} \) are the fluid temperatures in the hot and cold tanks respectively, and \( C_{p,f} \) is the specific heat capacity at the mean temperature \( (T_f) \). The thermal energy collected by the HTF may also be calculated by simply multiplying the net power input by the thermal efficiency \( (\Delta Q_{h\rightarrow c} = Q_{\text{net,rcv}} \cdot \eta_{\text{th,rcv}}) \). By investigating the thermal energy collected by the system during a period of operation, the potential of each HTF as a candidate TES medium can be ascertained in terms of required fluid volumes and cost.
4. Heat transfer enhancement techniques

Enhancing the heat transfer in tube flows has received widespread interest as it has a broad range of applications such as in refrigeration, heat exchangers, and HVAC [42]. Convection heat transfer can be improved by as much as 400% in a tube that is complimented with heat transfer augmentation techniques [40]. In the present study, various heat transfer enhancement techniques are investigated using the thermohydraulic model in order to evaluate their suitability for molten salt HTF in receiver concepts.

Heat transfer enhancement mechanisms are categorised as being either active or passive. With an active device, heat transfer augmentation is activated through an external power source, while passive techniques are simply a geometrical alteration to the tube passage that disturbs the flow regime. A number of passive techniques are studied herein, as these are more popular due to reduced complexity and expense [43].

The eddy transport of thermal energy plays a large role in turbulent forced convection heat transfer for molten salts (Pr > 1), as described in Figure 8. Liquid metals are largely unaffected by the eddy transport of heat well into the turbulent regime, indeed, thermal diffusion through eddy diffusivity only exceeds the contribution of molecular diffusion at $Re > 60000$ for lead-bismuth eutectic ($Pr = 0.025$) and $Re > 214000$ for sodium ($Pr = 0.007$) [44]. Convective heat transfer enhancement mechanisms have received little interest for lower Pr fluids, as the resultant increase in pressure drop will outweigh the gain in thermal performance, therefore heat transfer enhancement mechanisms are only applied to the molten salt HTF in this study.

Experimental studies into the effect of heat transfer enhancement in tube flows typically investigate both heat transfer and pressure drop effects. The introduction of passive devices may result in enhanced heat transfer in the tube, but a reduced hydraulic diameter, and increased flow path and surface roughness may increase the friction factor, which increases pressure drop (as per Equation 14). It is common practice in literature to present experimentally derived correlations for passive devices as a ratio between the smooth and enhanced tube, and some of these correlations are used in the analysis.

4.1. Surface roughness

The simplicity and effectiveness of surface roughness techniques makes them one of the most popular heat transfer enhancement mechanisms [45]. Surface roughness techniques act to disturb the laminar sub-layer adjacent to the wall, as this is a region of significant thermal resistance. These mechanisms typically take the form of ribs that are machined at the internal wall of the tube. The ribbed profiles may be formed transversely, or along a helix angle in order to induce swirl flow that further enhances convective heat transfer [46]. Tube pressure drop increases as a consequence of the increased flow disturbance. The profile of the rib ($n$, number of sharp corners facing the flow), extension of the rib into the tube core ($e$), pitch ($p$), helix angle ($\alpha$), and contact angle ($\beta$) all have an influence on the heat transfer and pressure drop characteristics. An illustration of these ribbed tubes are shown in Figure 11.

Ravigururjan & Bergles [45] has statistically collated a large number of data sets of surface roughness techniques from various authors, and generated correlations for the augmented tube Nusselt number $\text{Nu}_a$ and augmented tube
friction factor $f_a$. The correlations cover a broad range of $Re$ and $Pr$ which incorporate the conditions of molten salt flow in the receiver ($5 \times 10^3 \leq Re \leq 2.5 \times 10^5$, $0.66 \leq Pr \leq 37.6$). $Nu_a$ is found using the following:

$$
Nu_a/Nu_s = [(1 + [2.64Re^{0.036}(e/D_i)^{0.212} \,(p/D_i)^{-0.21} \,(\alpha/90)^{0.29} \,(Pr)^{0.024}]^{1/7})]
$$

Figure 11: Description of ribbed tube profiles

The above augmented-versus-smooth tube ratios have been developed based on the following smooth tube correlations;

$$
Nu_s = \left(\frac{f_s}{2}\right) RePr^{1/2} \left(\frac{Pr^{2/3}}{12.7} - 1\right) \quad (20)
$$

$$
f_s = (1.58lnRe - 3.28)^{-2} \quad (21)
$$

The maximum inner diameter ($D_i$) is used as the characteristic length to define $Re$ and the various different geometrical parameters. The above equations can be used to evaluate tube geometries with $0.01 \leq e/D_i \leq 0.2$, $0.1 \leq p/D_i \leq 7.0$, $0.3 \leq \alpha/90 \leq 1.0$, and $0 \leq \beta \leq 90$. These equations are applicable to wire coil inserts. The primary difference between the wire coil and ribs discussed above is that the wire coil can be retrofitted a smooth tube, rather than machined into the wall at the initial manufacturing process. Similar to surface roughness elements, wire coils disrupt the boundary layer at the wall-fluid interface, and generate secondary swirl flows. A schematic of a wire coil insert is shown in Figure 12.
4.2. Twisted tape inserts

Twisted tape inserts are used to augment heat transfer in tube flows primarily by inducing secondary flow structures, creating a larger flow path for the fluid [46]. The tape causes higher fluid velocities in the tube passage due to the partition that it creates. If good contact is maintained between the tape insert and tube wall, the tape may behave similarly to a fin and further augment heat transfer [47]. The increased flow path for the fluid and reduced hydraulic diameter of the flow passage results in an increased pressure drop. A schematic of the twisted tape is shown in Figure 13.

Smithberg & Landis [48] present equations for the Nusselt number and friction factor through the implementation of a twisted tape, covering a broad range of flow conditions ($2 \times 10^3 \leq Re \leq 10^5$, $0.7 \leq Pr \leq 10$);

\[
Nu_D = \frac{Re Pr}{1 + \frac{709}{Re f_a} \left( \frac{D_i}{D_H} \right) \left( \frac{H}{D_H} \right) Pr^{0.731}} \left[ 0.023 \left( \frac{D_i}{D_H} \right) Re^{-0.2} Pr^{-0.2} \left( 1 + \frac{0.0219}{(Pr)^{1.1}} \right) \right]^2
\]

(22)

\[
f_a = 4 \left[ 0.046 + 2.1 \left( \frac{p}{D_i} - 0.5 \right)^{1.2} \right] Re^{-0.2} \left[ 1 + 1.7 \left( \frac{p}{D_i} \right)^{-1.2} \right]
\]

(23)

These equations are applicable for a tape pitch of $3.62 \leq p/D_H \leq 22.0$. The hydraulic diameter ($D_H$) is calculated
using the minimum free flow area and wetted perimeter \((4A_{\text{free}}/L_{\text{per}})\). Equation 22 applies to a loosely fitting tape, additional terms apply for fin effectiveness when the tape is in close contact with the wall, and can be found in Smithberg & Landis [48].

4.3. Designs investigated

The appropriate device is one which maximises heat transfer with a minimal increase in pressure drop. A commonly used figure of merit for heat transfer enhancement devices is the performance evaluation criteria (PEC) [49]:

\[
PEC = \frac{\left( \frac{Nu_a}{Nu_s} \right)}{\left( \frac{f_a}{f_s} \right)^{1/3}}
\]

A parametric study of a number of different device geometries has been conducted in order to identify appropriate designs over the relevant range of \(Re\) and \(Pr\) \((5 \times 10^3 \leq Re \leq 3 \times 10^4, 2 \leq Pr \leq 10)\). Configurations possessing the maximum PEC and maximum \(Nu_a\) are put forward for simulation in the receiver model, listed in Table 1.

Table 1: Heat transfer enhancement devices simulated with solar salt (based on performance averaged over \(5000 \leq Re \leq 25000, 2 \leq Pr \leq 10\))

<table>
<thead>
<tr>
<th>Case</th>
<th>Device</th>
<th>Profile</th>
<th>(n)</th>
<th>(e/D_i)</th>
<th>(\alpha/90)</th>
<th>(p/D_{iH})</th>
<th>(H/D_i)</th>
<th>Performance indicator</th>
</tr>
</thead>
<tbody>
<tr>
<td>i</td>
<td>ribbed surface</td>
<td>□</td>
<td>2</td>
<td>0.15</td>
<td>0.3</td>
<td>0.5</td>
<td>90°</td>
<td>maximum PEC</td>
</tr>
<tr>
<td>ii</td>
<td>ribbed surface</td>
<td>□</td>
<td>2</td>
<td>0.2</td>
<td>0.9</td>
<td>0.5</td>
<td>90°</td>
<td>maximum (Nu_a/Nu_s)</td>
</tr>
<tr>
<td>iii</td>
<td>wire coil</td>
<td>○</td>
<td>∞</td>
<td>0.13</td>
<td>0.3</td>
<td>0.5</td>
<td>90°</td>
<td>maximum PEC</td>
</tr>
<tr>
<td>iv</td>
<td>wire coil</td>
<td>○</td>
<td>∞</td>
<td>0.2</td>
<td>1</td>
<td>0.5</td>
<td>90°</td>
<td>maximum (Nu_a/Nu_s)</td>
</tr>
<tr>
<td>v</td>
<td>twisted tape</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>4</td>
<td>0.25 maximum PEC &amp; (Nu_a/Nu_s)</td>
</tr>
</tbody>
</table>

5. Results & Discussion

Results are initially presented for the three HTF in smooth tubes, with augmented molten salt tubes discussed towards the end of this section. The results are presented in terms of receiver geometry, allowing for the identification of appropriate designs for the prescribed receiver thermal rating, and is convenient for comparing the performance of the HTF. Receiver thermal efficiency \((\eta_{th})\) and maximum outer surface temperature \((T_{so})\) at \(z/L_{\text{tube}} = 1\) is shown in Figure 14.

Thermal efficiency increases with a decrease in heat transfer area due to reduced convective and radiative losses (Equations 10 & 11). As mentioned in Section 3, any receiver design that breaches design point limits for film temperature or maximum surface temperature are excluded from the results section. It is clear from the results that the molten salt is quite limited in terms of the design options, however the liquid metals are much more flexible in this regard. Very compact receiver designs are possible with the liquid metals, with heat fluxes \(> 2.5\) MW/m\(^2\) possible regardless of the AR (length/width). Molten salt can be used in receivers up to heat fluxes of \(\sim 0.8\) MW/m\(^2\). A large AR facilitates more turbulent flow, as a greater portion of the total receiver mass flow rate is transported by fewer tubes, resulting in a greater fluid velocity and \(Re_f\). Molten salt is only suited to a design with a large AR (small
number of long tubes), such as those used to construct cylindrical receiver designs, while liquid metals may be used across a large design space, from large designs for a surrounding heliostat field down to very small sizes for modular CSP plants with equator facing fields. As small a receiver design as possible is preferable in any case as it saves on structural material costs in the tower due to reduced dead-weight and susceptibility to wind loading [12], saves on tube material costs, and aides in thermal efficiency and pressure drop. When equivalent receiver configurations are studied, liquid sodium is seen to possess the greatest thermal efficiency, followed by lead-bismuth, and finally molten salt. The greater flexibility of liquid metals means that thermal efficiencies of > 92% are possible, while molten salt is at a maximum of ~ 90% in a smooth tube. The most thermally efficient designs of each fluid are: sodium with $\eta_{th} = 92.34\%$ ($A_{rcv} = 2$ m$^2$, AR = 15), lead-bismuth with $\eta_{th} = 92.25\%$ ($A_{rcv} = 2$ m$^2$, AR = 15), and molten salt with $\eta_{th} = 90.16\%$ ($A_{rcv} = 6.5$ m$^2$, AR = 15). AR has a minimal affect on the thermal efficiency of the liquid sodium designs, with a $0.1 - 0.2\%$ efficiency improvement seen in the lead-bismuth designs for an increase in AR.

Figure 14: Thermal efficiency ($\eta_{th}$) and maximum surface temperature ($T_{so}$) of various receiver designs using (a) sodium, (b) lead-bismuth, and (c) solar salt.
Thermal efficiency is heavily influenced by surface temperature, which drives convective and radiative losses, with the temperature increasing for a decrease in heat transfer area due to an increase in heat flux concentration. The sodium designs have the lowest tube material temperatures, followed by lead-bismuth, then molten salt, which is reflected by the thermal efficiency results. As the same tube material properties are modelled for all three fluids, the magnitude of the outer surface temperature under similar heat flux levels and tube thermal conductivity is influenced by the inside surface temperature (Equation 8). The inner surface temperature is in turn driven by the heat transfer coefficient (Equation 8), and both are shown in Figure 15.

![Figure 15](image)

Figure 15: Maximum film temperature ($T_{si}$) and maximum heat transfer coefficient ($h_f$) of various receiver designs using (a) sodium, (b) lead-bismuth, and (c) solar salt.

The large thermal conductivity of sodium (Figure 5) is what results in very large heat transfer coefficients, shown in Figure 15 to be 2 – 4 times larger than lead-bismuth, and an order of magnitude larger than molten salt. The large heat transfer coefficients of the liquid metals facilitates a small wall-to-fluid $\Delta T$, as per Equation 8, and means that larger solar concentration ratios and thermal efficiencies are possible when coupled with a film temperature limit. The relatively narrow operational temperature range of molten salt means that the film temperature (894 K) is
A primary limitation in receiver design, while the liquid metal designs are generally limited by the tube material for the conventional inlet-outlet $\Delta T$ used here. Thermomechanical stresses on receiver tubes are influenced by material temperatures and the magnitude of radial and circumferential temperature gradients [50], which occur due to the non-uniform thermal boundary conditions on the sun-ward side (Equation 2). The radial temperature gradient is caused by the tube wall, an unavoidable thermal resistance that is necessary in the solar-to-thermal energy conversion process, and is influenced by heat flux, tube geometry, and material properties (Equation 9). The circumferential temperature gradient is heavily influenced by the HTF, as a larger heat transfer coefficient will result in a smaller temperature gradient around the circumference. The excellent convective heat transfer performance of sodium means that tube temperatures and thermomechanical stresses will be lower than a similarly sized receiver using either lead-bismuth and solar salt under the same heating conditions, resulting in greater mechanical reliability due to reduced creep-fatigue damage. The highly corrosive nature of lead-bismuth must be considered at the design stage, as it would act to accelerate mechanical damage by attacking the alloying elements in the tube material. Tube materials that are compatible with liquid metals, and possess excellent creep-fatigue strength under high temperatures and stresses may be used to exploit greater heat fluxes and fluid temperatures that facilitate more efficient thermodynamic cycles. The $N_{uf}$ for both liquid metals [35] and molten salt [36] are dependent on $Re_f$ and $Pr_f$ (Equations 6 & 7), with maximum $Re_f$ and velocity ($U_f$) shown in Figure 16.

The relatively low specific heat capacity of liquid metals (Figures 2 & 3) results in large fluid velocities through the receiver tubes as a large mass flow rate is required to achieve the desired outlet temperature (Equation 3). The large velocity, coupled with a small viscosity (Figure 6), results in very large $Re_f$, enhancing convective heat transfer through eddy conductivity, despite the fact that heat transfer to liquid metals is dominated by molecular conduction at the lower end of the turbulent $Re_f$ range [44]. The molten salt has a high $Pr_f$ due to a low thermal conductivity, therefore heat transfer is mainly facilitated in the turbulent regime by eddy conductivity, which increases with an increase in $Re_f$. The low $Re_f$ and low thermal conductivity of solar salt results in smaller heat transfer coefficients than liquid metals. The lower heat transfer coefficient results in a much larger wall-to-fluid $\Delta T$, and reduced overall thermal performance.

For all HTF, the pressure drop increases with a decrease in heat transfer area (Figure 17), signalling the effect that an increased fluid velocity has on hydraulic operation, as opposed to a decrease in flow path. For the same area, the pressure drop increases with an increase in AR, due to the twofold effect of a longer tube length and higher fluid velocity. The low thermal losses of a receiver with a small area and high AR means that the working fluid is travelling through long tubes at a very high velocity in order to deliver the desired outlet temperature, with the result being a significant pumping power requirement (Figure 17). The large fluid density (Figure 2) and velocity of lead-bismuth means that it has a far greater pressure drop penalty than the other HTF, with a pumping power requirement 3-4 times larger than that of sodium and more than an order of magnitude larger than molten salt. The high specific heat capacity of molten salt results in a lower fluid mass flow rate through the receiver, resulting in lower fluid velocities and reduced pressure drop.
The impact that pressure drop has on receiver performance is described in Figure 18 for the net power output, which is the product of thermal energy available to the receiver at design point (5 MW(th)) and thermal efficiency (\(\eta_{th}\)), minus the pumping parasitic (\(\dot{W}_{rcv}\)). For the most effective designs of each fluid, sodium has a 4.62 MW(th) net output (\(A_{rcv} = 2 \text{ m}^2, \ AR = 1\)), lead-bismuth has a 4.60 MW(th) net output (\(A_{rcv} = 2 \text{ m}^2, \ AR = 2\)), and molten salt has a 4.50 MW(th) net output (\(A_{rcv} = 6.5 \text{ m}^2, \ AR = 15\)). The best performing receiver designs are therefore different in terms of net power output when compared to thermal efficiency (Figure 14) for both liquid metals, with no difference to molten salt. For all fluids, a slight thermal efficiency increase for larger AR will be negated by an increased pumping power requirement. At the most disadvantaged lead-bismuth design (\(\Delta P\)), a pumping requirement of nearly 70 kW amounts to 1.4% of the total power input to the receiver. This parasitic loss is far greater than the 0.1 − 0.2% thermal performance gain that is offered by the larger AR receiver design over a smaller AR. The relatively small pressure drop penalty of molten salt means that the most thermally efficient molten salt design has a slightly higher net power output.
than a similarly sized lead-bismuth design ($A_{rcv} = 6.5$ m$^2$, AR = 15), however lead-bismuth generally outperforms molten salt in this capacity across other designs. The excellent thermal performance of sodium results in a higher net power output than both lead-bismuth and molten salt across all designs, despite having a higher pumping power requirement than molten salt.

The HTF volume and associated cost required to store 1 MWh (3600 MJ) of thermal energy is shown in Table 5 for a temperature increase from 290 – 565°C. Sodium requires the largest storage volume across the same temperature range, followed by lead-bismuth and then molten salt. A large storage volume may incur significant capital expenditure for the CSP plant due to larger sized storage tanks, with additional capital and operational costs accrued through a larger insulation requirement and heat trace protection. The large unit cost and low specific heat capacity of lead-bismuth means that it is a significantly more expensive storage medium than the other HTF. Cost effective TES is critical if CSP is to supply intermediate and base-load power [51], therefore the low cost and high specific heat

Figure 17: Receiver pressure drop ($\Delta P$) and corresponding pumping power requirement ($\dot{W}$) of various receiver designs using (a) sodium, (b) lead-bismuth, and (c) solar salt.
capacity of molten salt means that it has a significant advantage over liquid metals for direct TES. Lead-bismuth is clearly a very poor storage medium, as the large cost would negatively affect the plant LCOE, reducing competitiveness in the energy market. Lead-bismuth may be used in an in-direct TES system with a more cost effective storage medium such as molten salt, however additional costs and complexity would be incurred with such a system due to supplementary heat exchangers and storage tanks [7]. Despite being more expensive than molten salt, sodium appears to have a manageable TES cost, especially when compared to lead-bismuth. Molten salt is evidently a highly effective storage medium, however it is limited to fluid temperatures of $\sim 600^\circ$C. The broad operational temperature range of liquid metals may be exploited by raising the fluid outlet temperature in order to increase the TES potential. The high boiling temperatures of lead-bismuth (1670°C) and sodium (881°C) means that there is large scope for manipulation of fluid outlet temperatures for TES, however such high temperatures would require a very robust receiver design with an appropriate tube material.

Figure 18: Net power output and LCOE (shown for $CF = 0.3$) of various receiver designs using (a) sodium, (b) lead-bismuth, and (c) solar salt
A simplified LCOE study has been conducted in order to provide a final comparison of the candidate fluids in terms of plant economics for a typical CSP system with integrated TES. The various parameters investigated are listed in Table 3. The basis for the LCOE analysis is a reference 10 MW<sub>e</sub> plant located in California, USA (DNI = 2700 kWh/m<sup>2</sup>/yr). Capacity factors (CF) in the range 0.2 \( \leq \text{CF} \leq 0.5 \) are investigated, thus allowing for the evaluation of various plant configurations with different TES requirements. The solar multiple (SM) and TES capacity required will vary depending on the plant CF rating, and values for these are taken from estimates given by Falcone [10] for a system in Barstow, California. A 38% Rankine cycle efficiency is assumed, with a further 5% thermal energy loss attributed to TES and system piping [52]. The plant thermal rating and field size varies depending on the thermal-electric conversion efficiency, SM, and optical efficiency. The LCOE valuation has units of US $/kWh, and is calculated using the following equation [53]:

\[
\text{LCOE} = \frac{I_0 + \sum_{yr=1}^{N_{yr}} \frac{O&M_{yr}}{(1+r_d)^{yr}}}{Q_{yr, e}}
\]  

(25)

Where \( I_0 \) is the capital expenditure (CAPEX) of the full plant at \( yr = 0 \). O&M<sub>yr</sub> is the operation and maintenance costs of the plant in \( yr = n \), assumed as 65 $/kW<sub>e</sub>-yr [4]. \( Q_{yr, e} \) is the electricity produced in \( yr = n \), in units of kWh, calculated using receiver efficiency at design point, annual solar resource, annual average field optical efficiency of 56.5% for an equator facing configuration [54], plant availability of 90%, with the subtraction of pumping losses. \( r_d \) is the plant discount rate applied throughout the plant lifetime, assumed here as 6% [55]. LCOE is calculated for the full life of the plant from \( yr = 1 \) to the end of life at \( N_{yr} = 30 \). The CAPEX of the various plant components is established by referencing the heliostat field cost (conservative estimate of 200 $/m<sup>2</sup>[4]) and working backwards using a relative component cost breakdown given by Pitz-Paal [52] for a typical CSP tower plant, where the heliostat field is assigned 36% of the overall CAPEX. The cost of TES (if applicable) is then added to the overall plant cost. Despite variations
in size, the analysis assumes that all receiver designs share the same CAPEX and maintain mechanical integrity over
the plant lifetime, therefore no replacements are required. LCOE is plotted in Figure 18 for CF=0.3, and in Table 4
for the receiver configuration possessing the lowest LCOE valuation for each fluid at each CF.

Table 4: Lowest LCOE values of plant configurations described in Table 3

<table>
<thead>
<tr>
<th>CF</th>
<th>Na ($/kWh)</th>
<th>Pb-Bi ($/kWh)</th>
<th>Salt ($/kWh)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.20</td>
<td>0.1245</td>
<td>0.1255</td>
<td>0.1283</td>
</tr>
<tr>
<td>0.25</td>
<td>0.1204</td>
<td>0.2989</td>
<td>0.1214</td>
</tr>
<tr>
<td>0.30</td>
<td>0.1173</td>
<td>0.4113</td>
<td>0.1171</td>
</tr>
<tr>
<td>0.35</td>
<td>0.1156</td>
<td>0.4928</td>
<td>0.1136</td>
</tr>
<tr>
<td>0.40</td>
<td>0.1141</td>
<td>0.5541</td>
<td>0.1112</td>
</tr>
<tr>
<td>0.45</td>
<td>0.1129</td>
<td>0.6016</td>
<td>0.1093</td>
</tr>
<tr>
<td>0.50</td>
<td>0.1120</td>
<td>0.6398</td>
<td>0.1078</td>
</tr>
</tbody>
</table>

The trend shown in Figure 18 is consistent across all CF configurations in terms of LCOE, with the best performing sodium design a 2 m$^2$ area with AR=1, lead-bismuth is a 2.5 m$^2$ area with AR=1, and molten salt is a 6.5 m$^2$
area with AR=15. These correspond to designs with the highest net power output, which best balance thermal and
hydraulic performance. At CF=0.2, sodium and lead-bismuth have a lower LCOE than molten salt, as zero TES is
required (Table 3), meaning the relative fluid costs are negated (fluid costs in the receiver system alone are considered
negligible), and LCOE is a function of net power output alone. For lead-bismuth, CF=0.2 results in its lowest LCOE
across all CF investigated at 0.1255 $/kWh. As CF is increased and direct TES is integrated, the high $/\text{MW}_h$ cost of
lead-bismuth results in very large LCOE relative to sodium and molten salt, as fluid cost drives $I_0$ in Equation 25. The
broad temperature range of lead-bismuth may be beneficial for future CSP plants with receivers operating at very high
temperatures, however such high LCOE valuations significantly hinders its competitiveness in more conventional
plant configurations with direct TES. The thermohydraulic performance of sodium as a receiver HTF results in an
LCOE that is lower than molten salt for CF < 0.3, meaning that the high thermal efficiency and manageable pressure
drop of sodium receivers outweigh the additional fluid cost when compared to molten salt where a low TES capacity
is required < 3 hour. Molten salt has a lower LCOE than sodium for CF > 0.3, as the lower $/\text{MW}_h$ cost of molten salt
becomes an increasingly influential in $I_0$ in Equation 25. LCOE decreases for both sodium and salt as the CF, and by
extension TES requirement, is increased. The lowest sodium LCOE is 0.1120 $/kWh for CF=0.5, and for molten salt
is 0.1120 $/kWh for CF=0.5, however a large CF require a very large $I_0$ and larger land area for the collector system.
Sodium is highly competitive with molten salt for a system with direct TES across all CF, despite a significantly higher
$/\text{MW}_h$ cost and lower specific heat capacity. The broad operational temperature range of sodium may be exploited
beyond the inlet-outlet temperatures investigated here, resulting in greater stored energy potential (Equation 17). The
LCOE results indicate that the thermohydraulic performance of sodium is advantageous for systems with low TES
requirements, while the benefits of a high specific heat capacity and low $/\text{MW}_h$ cost through the use of molten salt
are realised for systems with a large TES requirement.

A number of heat transfer enhancement geometries (Table 1) are investigated for the most thermally efficient
molten salt design (smooth tube), which $A_{CV} = 6.5 \text{ m}^2$ and AR=15, with results presented in Figure 19 along with those of sodium and lead-bismuth in the same configuration. The designs investigated in the present study are the most effective in terms of $Nu$ enhancement and PEC for a broad range of geometrical parameters and relevant flow conditions. Surface roughness techniques (Cases i-iv), are found to be more effective at enhancing thermal performance than the twisted tape (Case v). Twisted tapes are more effective in the laminar flow regime than in turbulent flow, as it disrupts the core flow region rather than the viscous sub-layer at the wall [56]. The $Re_f$ regime of molten salt receivers at design point conditions is in the turbulent regime, and as a result, surface roughness techniques are more effective as they disrupt the viscous sub-layer adjacent to the wall. A heat transfer enhancement of $200 – 320\%$ is possible in molten salt receivers using either a ribbed surface or wire coil insert, bringing thermal efficiency of molten salt closer to that of the liquid metals. The PEC of the twisted tape is similar to that of a smooth tube, due to the large increase in friction factor ($f_a/f_s = 5$) that accompanies a moderate increase in $Nu_f (Nu_f/Nu_s = 1.7)$. Cases ii and iv have the maximum heat transfer performance ($\sim 320\%$), although they yield a friction factor that is an order of magnitude greater than the smooth tube. The increase in necessary pumping power largely offsets the gain in thermal performance, and this is indicated by the PEC valuation. The most effective device is Case iii, which is a wire coil insert that best combines heat transfer enhancement ($Nu_f/Nu_s = 2.07$) with a minimal friction factor increase ($f_a/f_s = 1.59$), thus delivering the maximum power output with a large PEC valuation (maximum PEC=1.77). The wire coil insert is also advantageous over machined ribs from a manufacturing standpoint, as they can be retrofitted.

Figure 19: Heat transfer enhancement device results summary ($A_{CV} = 6.5 \text{ m}^2$, AR=15). (a) thermal efficiency ($\eta_{th}$), (b) maximum outer surface temperature ($T_{sw}$), (c) maximum film temperature ($T_f$), (d) maximum Nusselt number ($Nu_f$), (e) maximum heat transfer coefficient ($h_f$), (f) average friction factor ($f_a$), (g) tube pressure drop ($\Delta P$), (h) pumping power requirement ($W$), (i) maximum $Nu_f/Nu_s$, (j) maximum $f_a/f_s$, (k) maximum PEC, and (l) net power output ($Q_{net}$).
to a smooth tube rather than being machined into the wall in a costly manufacturing process. An augmented tube not only improves thermal efficiency, but allows for a greater heat flux concentration by lowering the wall-to-fluid $\Delta T$ by enhancing the convective heat transfer coefficient, thus permitting smaller and more compact receiver designs. This enables molten salt receivers to be used in smaller and more modular plant layouts, broadening its appeal and bringing it closer to liquid metals in terms of flexibility. This analysis is concerned with receiver performance at design point conditions, however an application of any heat transfer enhancement device should be studied in greater detail with consideration given towards supplementary costs and the thermohydraulic performance across changeable conditions.

6. Conclusions

The thermohydraulic performance of a number of single phase receiver heat transfer fluids has been investigated in a typical CSP power tower configuration using numerical modelling techniques. Sodium and lead-bismuth have excellent heat transfer performance when compared to molten salt, allowing for greater heat flux capabilities and permitting very small and compact receiver designs. The most thermally efficient sodium design has an efficiency of $\eta_{th} = 92.34\%$ ($A_{\text{rcv}} = 2\, \text{m}^2$), lead bismuth with $\eta_{th} = 92.25\%$ ($A_{\text{rcv}} = 2.5\, \text{m}^2$), and molten salt with $\eta_{th} = 90.16\%$ ($A_{\text{rcv}} = 6.5\, \text{m}^2$), all at $\text{AR}=15$. A large AR is critical to the design of molten salt receivers, however the broad temperature range of liquid metals permits usage across a broad range of receiver designs. The low mass flow rate of molten salt results in a low pressure drop and pumping power requirement, worst being $W_{\text{rcv}} = 2\, \text{kW}$ at the most thermally efficient design. Sodium and lead-bismuth have a manageable pumping power parasitic at low AR, however the slight thermal performance gain at high AR is negated by a significant pumping power requirement, particularly for lead-bismuth (worst at $W_{\text{rcv}} = 15.8\, \text{kW}$ for sodium, $W_{\text{rcv}} = 70.7\, \text{kW}$ for lead-bismuth), meaning the most effective liquid metal designs shift to a $2\, \text{m}^2$, AR=1 design for sodium, and a $2\, \text{m}^2$, AR=2 design for lead-bismuth. The low specific heat capacity and current high unit cost of lead-bismuth renders it as a poor choice as HTF in contemporary CSP designs where direct TES is required, however the benefits of this fluid may be realised at higher operating temperatures in future projects through the use of compatible containment materials, and with indirect TES. Molten salt is an excellent storage medium, with a low fluid volume requirement and cost proving significant. Sodium is less effective as molten salt as a storage medium, however it has a significantly lower cost penalty than lead-bismuth, and it’s excellent thermal performance allows it to remain competitive across all CSP plant configurations investigated. The LCOE analysis indicates that the superior thermohydraulic performance of sodium overcomes the storage potential deficit to molten salt at low plant capacity factors ($\text{CF}<0.3$) where there is a low TES requirement ($<3\, \text{hour}$), however molten salt becomes a more economically viable option where a larger storage capacity is required ($\text{CF}>0.3$), and fluid volume and cost has an increasing influence on CAPEX. The heat flux capabilities and thermal performance of molten salt receiver tubes can be improved significantly with the implementation of heat transfer enhancement devices. Surface roughness techniques and wire coil inserts prove to be more effective in the $Re_f$ range for molten salt receiver applications than twisted tapes. A heat transfer enhancement device can be configured for
molten salt receiver tubes so as to augment heat transfer significantly without suffering a large increase in pressure drop. This analysis serves to highlight the merits of a number of candidate single phase heat transfer fluids in a liquid tubular receiver. As CSP seeks to have a greater influence on the renewable energy mix throughout the world, the appropriate receiver HTF will be selected based on the capital/operational costs and performance of the plant, which is heavily influenced on the thermohydraulic performance of the HTF and TES configuration.

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References


• Performance of molten salt, sodium, and lead-bismuth in a solar receiver is studied

• Sodium outperforms lead-bismuth and molten salt across all designs investigated

• Molten salt performance can be improved significantly through enhancement techniques

• Lead-bismuth is a poor storage medium, not well suited to systems with direct storage

• Sodium designs have a lower LCOE than molten salt for low storage (<3 hour)