An Investigation of Porous Materials for the Capture of Concentrated Solar Energy

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Abstract

Fossil fuel usage is resulting in climate change. There is a need to switch to renewable energies, but existing technologies lack the efficiency for widespread adoption. Concentrating solar energy to a receiver using a parabolic reflector is an efficient method of converting sunlight into thermal energy at a high efficiency. Current receivers suffer efficiency challenges due to significant re-radiation losses as they reach a high temperature at the front surface. This project is focused on the creation of a computational model to simulate the radiation heat transfer in porous geometries, which can be used to optimize the geometric properties of the concentrated solar receiver and improve the efficiency. Two different computational fluid dynamics (CFD) software were considered, and their capabilities were assessed. A parametric study was conducted that involved changing the input radiation flux, porous material, porosity, pore size, and porosity gradient of the block. Results showed that of the geometries and materials tested, a graphite block with 70% porosity had the highest output flux from the system.

Keywords: Concentrated Solar Power, Solar Energy, Radiative Heat Transfer, Thermal Receiver, Computational Fluid Dynamics, Porous Media, Variable Pore Structure
Summary for Lay Audience

Fossil fuel usage is resulting in climate change due to the emission of greenhouse gases. There is a need to switch to renewable energies, but existing technologies lack the efficiency for widescale adoption. Solar energy is the most abundant source of renewable energy; however, commercially available solar energy technologies typically have low efficiencies, making them less competitive compared to conventional fossil-based systems. Concentrated solar power (CSP) is promising approach to convert sunlight into thermal energy at very high efficiency. Among CSP systems, the parabolic-dish CSP system is considered the most efficient. In this system, a parabolic dish directs sunlight to a single focal point where it is concentrated and converted into thermal energy through absorption into a thermal receiver. A challenge with current designs is that the surface of the thermal receiver reaches a high temperature and does not efficiently transfer the heat due to radiation losses. Porous media can reduce these losses because they have a large internal surface area which allows for absorption of radiation at many internal surfaces. However, the radiation exchange process in the porous medium requires better understanding, which is crucial to design high-efficiency concentrated solar receivers.

This project is focused on the creation of a computational model which can be used to design a solar thermal receiver that reduces the front surface temperature and improves the heat transfer within the receiver. To create this computational model, a simple experimental study was conducted, and computationally replicated. Then, the initial stages of a parametric study were conducted with more complex porous models. This study involved changing multiple parameters and measuring the heat output from the system. The parameters that changed were the value of incoming radiation, the porous material, the porosity of the block, the pore diameter of the block, and the porosity gradient of the block. Results from this study showed that of the geometries tested, a graphite block with 70% porosity had the highest output heat flux from the system, however the author notes that more geometries need to be tested to further optimize the receiver’s design.
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Chapter 1

1 Background and Literature Review

The use of fossil fuels to produce electricity is contributing to global warming, a major threat to the survival of humans. Almost 73% of global electricity production still comes from burning fossil fuels, as seen in Figure 1-1 [1].

![Figure 1-1 Estimated share of electricity power consumption in 2018](image)

It is predicted that the world’s energy usage will increase by 50% in 2050 [2]. If the world is to meet this electricity demand and stay below global warming of 1.5°C as suggested by the International Panel on Climate Change (IPCC) [3], we need to continue reducing our reliance on fossil fuels as an energy source. Even from a practical standpoint, since fossil fuels are a finite resource, their utility is limited, and renewable energy sources will be required to supply the world’s energy. Many countries have committed to net-zero emissions by 2050 [4], and want to achieve this by increasing renewable energy capacity. To achieve net zero by 2050, it is projected that CO2 emissions in the power sector will need to fall by 60% by 2030, and renewables will need to produce 60% of the electricity by 2030 [5]. Figure 1-2 shows the likely scenario of what the mix of energy sources will be in the coming decades.
As seen, renewable energy will make up most of the energy produced. Renewable energies have seen a big increase in recent years, being the only fuel source that increased in demand during the 2020 pandemic [7], but this growth will need to accelerate if we want to tackle climate change and account for the depletion of fossil fuel resources.

1.1 Background

1.1.1 Renewable Energy

There are many different types of renewable energy. Wind energy converts air currents into usable electricity using wind turbine technologies. Bioenergy involves burning biological materials to create heat, which is then used in a power cycle to produce electricity. Geothermal energy uses heat from the earth as input in steam power cycles to also produce electricity. Hydropower utilizes dams and flowing water to turn turbines for electricity production. All these sources combined will help achieve net zero; however, direct solar radiation is the most promising and abundant form of energy for our use in energy conversion and production [1].

Thermal radiation is energy emitted through electromagnetic radiation from a surface. It does not require any medium to carry it. Solar radiation is emitted throughout the electromagnetic spectrum, but almost 99% of solar radiation energy lies within 0.15 to 0.4µm range. Approximately 1% of solar radiative energy lies within gamma rays, x-rays, and radio waves.
43% of energy is within the visible spectrum, 49% is within the near infrared spectrum, and 7% is in the ultraviolet range [8]. Figure 1-3 shows a distribution of incident solar radiation on Earth.

![Spectrum of Solar Radiation (Earth)](image)

Figure 1-3 Spectrum of solar radiation incident on Earth [9]

The sun provides the earth with $1.2 \times 10^5$ terawatts (TW) of energy continuously; much higher than global energy demand, which is approximately 13 TW [10]. However, this energy needs to be harnessed and converted to electricity. The most popular technology is photovoltaics (PV) which have seen a huge increase in popularity in recent years [1]. Photovoltaics harness energy from the sun and convert it directly to electricity. Photovoltaic cells usually only use the visual spectrum to generate electricity, as only radiation between the wavelengths of 380nm-750nm can create enough energy to generate an electrical current [11]. The remainder of the energy absorbed by solar cells is converted into waste heat. Since photovoltaic cells need to be exposed to direct sunlight to harness electricity, the surface can get extremely hot. Current PV technology can reach efficiencies around 40%, and this efficiency decreases the hotter it gets [12]. Additionally, PV panels degrade over time due to extreme heat and direct sunlight exposure [13]. PV panels can last 25-40 years [1], but there are currently no large-scale initiatives to effectively recycle them. Hazardous metals (Pb, Cr, Cd, Ni) and toxic substances in PV panels can be released into the environment if not recycled properly [14]. Another problem with PV is that, like most forms of renewable energy, the supply is incremental and thus storage in batteries is required. There have been major improvements to battery technologies in recent years, but challenges remain in terms of the availability of precious metals required for the batteries, the amount of energy they can provide, and the short lifespan of the batteries [15].
1.1.2 Concentrated Solar Power

Another approach to harnessing the sun’s energy is to capture it as Concentrated Solar Power (CSP). CSP converts all solar radiation into heat, which makes for a more efficient use of solar energy. It does this by capturing energy from the sun in the form of (high temperature) heat, which can then be used directly for space/water heating, or in a power cycle to convert it to electricity. By this approach, a series of mirrors are used to direct sunlight to a single focal point which reaches an extremely high temperature (upwards of 1000°C [16]). The heat is concentrated to the receiver, then transferred to a working fluid (such as air, water, or oil). This working fluid removes heat from the receiver and can be used to create steam to turn turbines, or else stored for later use. Figure 1-4 shows an overview of a CSP power plant.

![Schematic of a CSP power plant](image)

Figure 1-4 Schematic of a CSP power plant illustrating the process of sunlight conversion into electricity. Red indicates warm heat transfer fluid; blue indicates cold heat transfer fluid [17]

Since solar energy supply is intermittent, solar energy collected by CSP approaches require energy storage so that it can be used continuously. Thermal storage materials fit into three categories: sensible, latent, and chemical, and these categories describe the way heat is stored. Materials that are considered “sensible” are solids such as sand-rock minerals, reinforced concrete, bricks, or liquids including molten salt, or synthetic oils. Molten salts have been regarded as the best storage material owing to their stability, low vapour pressure, high conductivity, and non-toxicity [18]. Latent heat storage uses material such as phase change materials (PCMs) which store and release heat during phase changes (i.e., melting or solidification). They have the highest storage density, but high cost and low conductivity, a physical property which makes it challenging to add and remove heat energy rapidly. Research is
ongoing to incorporate high conductivity materials within PCMs to increase conductivity [19], [20]. Chemical heat storage involves the absorption or release of large amounts of thermal energy when chemical bonds are formed or broken [18]. There has not been extensive research into this type of storage owing to the complicated reactors needed for the specific chemical reactions.

There are many different types of solar collectors. Non-concentrating collectors, such as flat-plate collectors, have the same intercepting area as absorbing area. The collectors consist of an absorbing plate, insulation layers, and tubes filled with heat transfer fluid to remove heat from the surface [18]. These types of collectors are usually used for heating and cooling purposes in buildings, and there has been research conducted into combining them with solar PV panels to increase efficiency [21].

Concentrating collectors use sun tracking and mirrors to direct sunlight to a smaller receiving area that reaches a high temperature. There are many different styles of concentrating collectors, as seen in Figure 1-5. Parabolic trough collectors are currently the most popular. Curved mirrors direct sunlight to a central metal receiver tube that is coated in glass to reduce radiation losses. They usually have a concentration ratio of approximately 40. These types of collectors track the sun in 2D as opposed to 3D, which results in better tracking accuracy because 2D systems are less complex to orient, to achieve the optimal angle of concentration from the sun [18]. Linear Fresnel reflectors use flat or slightly curved mirrors to reflect sunlight to a receiver tube fixed above the mirrors. A small parabolic mirror is sometimes added on the receiver to further focus the sunlight [22]. Heliostat Field Collectors or Central Receiver Collectors have an array of sun-tracking mirrors (heliostats) to direct sunlight to one central receiver which has a steam generator to absorb thermal energy and create high temperature, high-pressure steam that produces electricity [18]. Parabolic dish collectors use arrays of parabolic mirrors to direct sunlight to a central point. A heat transfer fluid within the receiver is heated to a desirable temperature and pressure to be used in a Stirling (i.e., closed, reversible cycle) [23] or Brayton engine (i.e., constant pressure, heat engine) [24]. The collectors track the sun continuously on two axes, and can have concentration ratios up to 2000, making them the most efficient type of concentrator [25]. Existing prototypes dish-engine systems range from 7-25kW [18].
1.1.3 Thermal Receivers

Thermal receivers absorb heat from concentrated sunlight and transfer it to a heat transfer fluid. The heat transfer fluid is used to produce energy immediately by creating steam that is fed into a turbine, or else transferred to a thermal storage unit for later use. There are a few key technical challenges when it comes to designing thermal receivers as identified by Ho and Iverson [27]. The material of the receiver and the heat transfer fluid must be able to optimize solar irradiance absorption while minimizing heat loss. They also must be able to have high reliability and mechanical stability at high temperatures for thousands of cycles. Thermal receivers can also involve direct or indirect heating of the power cycle fluid. The advantages of direct heating include reduced exergetic losses due to intermediate heat exchange (i.e., between receiver and fluid). The advantage of indirect heating includes the ability to store excess heat for energy production during non-solar hours. Ho and Iverson [27] show that a high concentration ratio on the receiver and reduced radiative view factor are critical to maintain high thermal efficiencies at temperatures above 650°C. Reducing convective loss is less significant but can yield increases in thermal efficiencies at high temperatures. Increasing solar absorptance, or decreasing thermal emittance also increases thermal efficiency.

There are many different types of solar receivers. Liquid receivers can be in the form of tubular liquid receivers or falling film receivers. Tubular liquid receivers have an array of panels containing thin-walled tubes that move the working fluid in multiple passes through the incident sunlight.
sunlight [27]. This design is employed in central receiver plants. Typical temperatures are around 600ºC, with efficiencies as high as 90% [28]. The working fluid is the limiting factor for this receiver type. Molten nitrate salt is typically used as a working fluid; however, studies have been done to analyze liquid metals (such as sodium [29] or fluoride salt [30]) for their ability to reduce front-to-back thermal stresses owing to their increased conductivity.

Falling film receivers have a gravity driven fluid that flows down an inclined wall. The fluid may be irradiated directly or heated through a wall, as seen in Figure 1-6. Direct exposure reduces thermal resistance by using blackened molten nitrate salts with suspended submicron particles, reaching efficiencies as high as 90% [31]. The major concern with the direct exposure receiver is the influence of wind, and fluid contaminant. An internal falling fluid receiver is heated by convection from the surrounding walls which are heated by incoming radiation. These types of receivers have estimated efficiencies as high as 94.5% [27], [32]. Although promising, falling film receivers are still in a theoretical or experimental phase.

Other receivers have storage capabilities within solid particles. Falling solid particle receivers use sand-like ceramic particles that fall through a cavity receiver that is directly irradiated with sunlight. Small particle air receivers use submicron carbon particles suspended in air heated by concentrated sunlight in a pressurized cavity receiver utilized in high-pressure Brayton cycles. More details of these and other receivers can be found in [27].
Volumetric receivers have higher fluxes compared to tube receivers owing to their 3D configuration. A comparison of the two receivers can be seen in Figure 1-7. The operating principle of a volumetric receiver is straightforward. The receiver is comprised of interlocking structures of knit wire, foam, or foil made of metal or ceramic with a porosity such that radiation is absorbed into the depth of the structure. Concentrated solar radiation heats the material and a working fluid passes through, heated by forced convection, transforming solar radiation into thermal energy. The heat is then transferred to a working fluid, such as synthetic oil or water, through pipes that run along the back of the receiver. The volumetric effect results in the front surface of the receiver having a lower temperature than the outlet, however this effect is challenging to achieve in practice [16]. The advantages of volumetric receivers over other types include the availability of the working fluid (air), they are non-toxic, they have 3-5 hours of thermal storage, and the volumetric effect reduces thermal radiation losses resulting in higher efficiency thermodynamic cycles [16]. Volumetric receivers have not yet been commercially developed, but a 1.5MW demonstration plant has been in operation since 2009 [33]. Volumetric air receivers use porous structures (such as honeycombs or porous ceramics). Air flows through the structure and is heated to 800-1000°C for metals, up to 1200°C for ceramics, and up to 1500°C for SiC [16].

![Figure 1-7 Comparison of tubular and volumetric solar receivers. The tubular receiver (left) heats the material and then to the heat transfer fluid resulting in a lower heat transfer rate. The volumetric receiver (right) heats the material and gas simultaneously resulting in a higher absorbed irradiance. [16]](image)
1.1.4 Porous Materials

Because of current receiver geometries, significant heat is absorbed right at the surface of the receiver or else reflected. Since the geometry of the porous material limits heat transfer from the front surface, there is insufficient heat removal from the front surface which decreases the amount of heat transferred through the receiver to the working fluid. As a result, the front surface reaches a very high temperature relative to the rest of the receiver. This results in significant heat lost due to re-radiation because the high front surface temperature radiates heat to surroundings instead of through the receiver. If the heat is not being transferred from the front surface of the receiver, the temperature of the front surface increases and more radiation losses occur. This problem is amplified at high temperatures [27].

Heat transfer processes always result in some heat loss to the surroundings. The heat transfer process from sunlight to thermal storage comprises of two exchange processes. First is the absorption of solar radiation in the receiver and transfer of heat to the heat transfer fluid. Second is the transfer of heat from the heat transfer fluid to the storage medium. If there was a theoretical maximum efficiency during each heat exchange process of 80%, and with 2 heat transfer processes the overall maximum efficiency for this type of system drops to around 64% [34], although CSP plant efficiencies are much lower (~30%) [35].

Volumetric solar receivers have the potential to increase efficiencies owing to their porous structures. Porous media are classified by a solid matrix of heterogenous material. A fluid occupies the void spaces within the matrix. Porous mediums have been used in many heat transfer applications because of their high surface area-volume ratio. They are widely employed in convection cooling, as a conductive material carries heat to a porous medium through which cooling fluid is being passed [36],[37]. Porous mediums have gained popularity for radiation heat capture and are a viable option to improve the efficiency of a CSP system. Because of their high surface area and void spaces, radiation can penetrate a porous medium where it can be effectively absorbed in the volume of the structure and transferred through conduction while minimizing re-radiation as it is difficult for radiation to escape to surroundings.
Despite their advantages, porous media receivers still encounter the problem of high front surface temperatures leading to re-radiation [38]. One theory to address this is a decreasing variable porous media receiver that has larger openings at the front allowing for more radiation to penetrate deeper into the receiver. The porosity would gradually decrease the deeper into the receiver the radiation goes. An optimal pore geometry would trap sunrays, which would undergo internal reflections before being fully absorbed into the porous medium (similar to a black body approximation as seen in Figure 1-8). This would lead to negligible re-radiation losses and more efficient systems overall. Studies that have investigated porous materials as CSP receivers will be reviewed in the next section.

Figure 1-8 An approximation of a black body. A small opening allows radiation to enter and be reflected multiple times before it is fully absorbed in the medium. A similar phenomenon could be achieved in a porous medium with multiple internal reflections occurring in the void spaces before being absorbed into the porous material. [39]

1.2 Literature Review

The computational model was developed using CFD software through ANSYS. The different radiation models that are available will first be reviewed. This will be followed by a review of the existing studies into improving the efficiency of volumetric receivers. The last two sections include a review of the existing research into porous materials, specifically their application as receivers for CSP.

1.2.1 Radiation Modelling in CFD

Many different models exist for modelling radiation in CFD codes. This section will explore the different options and their relative advantages and disadvantages.
The **Radiative Exchange Model (REM)** can model radiative heat transfer between enclosed surfaces with non-participating media. This method splits surfaces in the enclosure into surface elements. The view factor between each pair of elements must then be calculated. Radiation can leave an opaque surface through reflection and emission; upon reaching a surface it may either be absorbed or reflected. Using this information, the radiative exchange can be calculated by finding the difference between the emitted energy from the surface and the incoming energy from other elements. Each surface in the enclosure is assumed to be isothermal and have both uniform radiosity and uniform irradiation. Surfaces are assumed to be opaque, and gray, that is their emissivity, absorptivity, and reflectivity are independent of wavelength. A full description of the model can be found in from Bergman et al. [40].

The **Zonal Method (ZM)** is an extension of the REM developed by Hottel and Cohen [41] that models enclosures with participating media. The volume within the enclosure is broken into isothermal elemental volumes and surfaces are divided into surface elements. Exchange factors can be determined between zones and an energy balance for each zone can be written in terms of the heat fluxes from all other zones.

The **Rosseland** model simplifies the Radiative Transport Equation (RTE) for optically thick mediums and assumes that energy emitted from other locations in the domain are absorbed and have no influence on local transport. This means the model is not valid near walls. Goldstein and Howell [42] proposed a temperature boundary slip for near wall treatment to overcome this limitation. The Rosseland model introduces a new diffusion term into the energy transport equation with a strong temperature dependent diffusion coefficient. When the Rosseland approximation is introduced, the conduction and radiation heat flux combine to calculate the temperature field in the energy equation [43].

The **Differential Approximation or P1 model** is also a simplification of the radiation transport equation. This model assumes radiation intensity is isotropic or direction independent at a given location in space. It adds an additional transport equation to the simulation, making it more computationally expensive. A full derivation of the model can be found by Modest [44]. The P1 model only considers opaque diffuse walls. Any open boundaries are treated as fully transparent.
boundaries: they absorb all outgoing energy, and the incoming energy is treated as a black body temperature [45].

The application of the **Finite Volume Method (FVM)** to the RTE was introduced by Raithby and Chui [46]. In this model, the domain is divided into discrete volumes where transport properties are held constant within each control volume at each step. Control volumes are further subdivided into finite solid angles. Radiation is conserved within each solid angle for each control volume. Kamel et al. [47] proposed a method called FTn-FVM which decreased the false scattering seen in traditional FVM.

The **Monte Carlo Method** (MCM) assumes intensity is proportional to the angular flux of photons. It simulates the physical interactions between photons and their environment. A photon is tracked from a source through the system until its weight falls below a minimum value, at which point it is assumed it has been fully absorbed into the medium. Every time the photon interacts with a surface, it is either reflected, absorbed, or scattered, at which point the information about the photon is updated. This creates a ‘history’ of interaction between the photon and the system. To get an accurate picture of radiation in the system it is recommended to create multiple histories, with the default set to 10,000 [45]. Monte Carlo’s main advantage is that it simulates the process of radiation transport as opposed to directly modelling it. Emission, reflection, scattering, and absorption of photons is random on the microscopic scale and the MCM approximates this behaviour by treating radiation as a surface phenomenon, resulting in very accurate results. This accuracy comes with a large computational cost. To decrease this cost, a coarser mesh can be created for the radiation field, and a finer mesh for the flow field with the assumption that the radiation field has fewer sharp changes than other transport variables [45]. Despite its advantages, there are still issues with the model. Howell [48] documents a simulation with gray gas between parallel plates containing weakly absorbing media and found that very few rays were absorbed in the media compared to the walls. To resolve this, many rays had to be cast. Additionally, if the solution is optically thick, many rays are exchanged between neighbouring volumes which significantly increases computational time. This process could be modelled more efficiently by a diffusion approximation [49]. A more thorough description of the MCM can be found by Siegel & Howel [49] and Modest [50].
The **Discrete Ordinance (DO)** Model, first introduced by Chandrasekhar [51], solves the radiative transfer equation (RTE) for a finite number of discrete solid angles. The angular discretization is specified by the user. The DO model transforms the RTE into a transport equation for radiation intensity, similar to solving for fluid flow and energy equations [43]. The RTE is solved at discrete points in space for discrete number of directions with a weighting factor in each direction [52]. Integration of intensity over all the solid angles multiplied by the weighting factors results in a known intensity. Raithby [53] has criticized the method for failure to accurately conserve radiation in some simulations.

The **Discrete Transfer** model was developed by Lockwood et al. [54]. It discretizes the equation of transfer along rays and assumes that the scattering is isotropic. The path along a ray is discretized using sections formed by breaking the ray at element boundaries. The physical quantities in each element are assumed to be uniform. The rays are traced through the domain the same way photons are tracked in the Monte Carlo model. Elements must be small enough so that the radiation field is uniform inside of them. All physical quantities of interest are found at fixed points due to fixed sampling and ray discretization. This differs from the Monte Carlo model which calculates surface or volume averages [43]. In non-gray models, each band is treated as a separate calculation. Ray tracking is only done once, and the results for each band are combined to give the total radiative and heat transfer [43]. This can result in a large computational time for a non-gray model. In gray models, where the radiation field is homogenous and high spatial resolution is required, the discrete transfer model can be very efficient and provides accurate results with sufficient angular discretization. A problem with the discrete transfer model is a lack of error information which can sometimes resolved through sub-angular sampling [43].

Hischier et al. [55] applied the Rosseland, P1, and MCM models to solve for radiation in a high temperature pressurized porous solar air receiver. They found that when the radiative source term was plotted against the radial direction of the receiver, the P1 results matched the MCM results reasonably well; however, the Rosseland model led to an over prediction of radiation near the centre of the receiver because of the relatively small optical thickness. The P1 model was used because of its confirmed accuracy and lower computational cost.
Modest [44], [50], [56], [57] has a thorough comparison of most radiation models. He concluded that the MCM, FV, and DO methods are the only ones that can be improved through mesh refinement or casting additional rays. He also concludes that the P1 approximation is easier to apply than the FV and DO models and can yield accurate results for optically thick media.

Optical thickness is a dimensionless quantity that represents how well a path of gas can attenuate radiation of a certain wavelength. It is represented by the following equation:

\[ \tau = \int_0^S K_\lambda(S\cdot) dS\cdot \]  

where \( \tau \) is the optical thickness/opacity of a layer with thickness \( S \) and is a function of all the values of the absorption coefficient \( K_\lambda \) between 0 and \( S \). A high value means that there is a large absorption of radiation [45]. ANSYS [45] advises that for problems that are optically thin or purely transparent cases, the Monte Carlo and Discrete Transfer models are the most appropriate.

From this review, it is clear that each computational model has relative advantages and disadvantages. Many of these computational models have been used to investigate volumetric receivers and porous media, with the application of absorbing incident sunlight. Existing studies in this field are reviewed in the next sections.

### 1.2.2 Volumetric Solar Receivers

There are two types of volumetric receivers. The first is a closed loop pressurized window receiver system for a Brayton Cycle, as seen in Figure 1-9. It combines solar energy with natural gas by using concentrated sunlight to preheat compressor outlet air before entering the combustor of the gas turbine making pressurized volumetric receivers an alternative to burning fossil fuels [58]. This system results in a reduced heliostat field size and is currently cheaper to implement [59][35]. The biggest design challenge is a semi-transparent window that divides the high-pressure operation from ambient air while minimizing reflection, reradiation, and convection losses [60]. The highest efficiencies seen in this type of receiver are around 70%, and power outputs as high as 500kW [16].
The second type of volumetric receiver is the open loop atmospheric receiver system for a Rankine cycle where atmospheric air is heated through a metal or ceramic receiver, then used to produce steam to feed a Rankine turbine-generator system, as seen in Figure 1-10 [35]. Some studies into improving open loop receivers are summarized below.

Open loop receivers with metallic absorbers were pioneered by Fricker in the 1980s [61]. He developed thin mesh wires over which atmospheric air was drawn. When tested with a parabolic dish, outlet temperatures up to 840°C were seen and thermal efficiencies between 70-90% were predicted. Further attempts to improve metallic absorbers were investigated by making variations to receiver size and geometry. Larger power outputs were seen [62]; however, these designs were limited in their ability to be effectively cooled, resulting in hots spots which led to flow instabilities and irreversible damages to the receivers [63].
Ceramic absorbers were investigated after the design limitations exhibited by metallic absorbers. Ceramic was chosen because of its greater durability, mechanical tolerance, and resistance to higher solar thermal fluxes and higher thermal gradients which led to the reduction in receiver aperture size and reduced infrared losses [16].

Chavez et al. [64] studied a porous ceramic absorber with peak outlet temperatures of 730°C and an efficiency of 54%. Some design factors cited for the lower efficiency were the coating paint on the receiver blocking some of the pores. Additionally, the temperature of the receiver was measured to be 1350°C when the outlet temperature was 730°C, resulting from large radiative heat losses. The authors estimated that efficiencies around 80-85% could be seen for outlet temperatures of 550°C given an optimized structure.

Hoffschmidt et al. [65] developed a modular hexagonal ceramic cup receiver. The cups were free to expand owing to the space between them. Separating the absorber materials made it possible for the front to be fed by return air from the waste heat recovery boiler. It also made module replacement easier. It had a 680-800°C outlet temperature, and could withstand temperatures up to 1000°C. It was made of SiSiC ceramic combs and had an opening porosity of 49.5% [66]. It was found that with an outlet temperature around 750°C, and fluxes of 270-520kW/m², efficiencies were around 70-75%. Temperature differences over the absorber aperture were up to 450°C [67]. This absorber was installed in a 1.5MW solar power plant in 2009 [68]. A diagram of the receiver can be seen Figure 1-11.

![Figure 1-11 Structure of the volumetric ceramic receiver developed by Hoffschmidt et al. [65]](image-url)
Marcos et al. [69] discussed the importance of capturing waste heat in the air exhaust after thermal energy was transferred to the working fluid and recirculated the air through the receiver. They estimated that current technologies are losing 5-15% of energy potential because they are not optimizing this process.

Menigault et al. and Variot et al. [70], [71] proposed a two-slab volumetric receiver containing a front slab made of solar transparent glass beads or a silica honeycomb, while the second slab contained silicon carbide particles. This was to allow for deeper solar penetration through the first slab into the second layer. Re-radiation emission from the second slab was absorbed by both slabs. This resulted in a maximum temperature within the interior of the volumetric receiver as opposed to at its surface. Testing using a solar furnace showed efficiencies around 90%, with an outlet temperature of 700°C. Pitz-Paal et al. [72] also investigated multi-layer receivers with square glass channels covering a ceramic foil receiver. They achieved an increase in efficiency of 10% relative to pure ceramic receivers with outlet temperatures up to 1000°C.

A technology review by Ávila-Marín [16] identified configurations and materials of volumetric receivers that are most promising for electricity production. Regarding material, it was found that metals could achieve outlet temperatures between 800°C-1000°C. Stainless steels and base-nickel alloys with high chrome content are the best for receivers because of their capacity to form oxides which are black and highly absorptive, and metal ceramics could achieve higher temperatures (1200-1500°C). Al₂O₃ (aluminum ceramic) is the ideal material because of its high melting temperature (2000°C) and low price, but it is white and has poor optical qualities. Coating techniques could improve optical properties while maintaining mechanical properties. SiC was identified as the best material because it had better absorptivity and optical properties than aluminum ceramic.

The effects of mass flow density on receiver output were also reviewed. The mass flow density is determined by the pressure difference between two sides of a receiver. A pressure drop is produced by a blower through the receiver. Instabilities occur when a pressure drop causes different mass flow densities. This can lead to unstable gas flow, which results in local overheating and failures (i.e., cracking or melting) [73]. This can be controlled by reducing the
mass flow, which is linked to a lower pressure loss. However, if the flow is unstable, a lower mass flow results in a higher-pressure loss and local overheating [16]. This type of instability has only been seen in highly porous honeycomb structures [63].

Freudenstein [74] outlined the main applications of volumetric solar receivers, stating that for medium temperatures (<800°C), open loop receivers for steam generation in the Rankine cycle were best for electricity production or industrial process heat. High temperature (>800°C) open loop receivers were best for indirect Brayton cycle power production and industrial process heat. Finally, high temperature (>800°C) closed loop receivers were best for direct Brayton cycle power production.

High efficiency volumetric receivers have only been tested in labs so far. For the technology to be successful, Ávila-Marín identified key factors that need to be addressed: the improvement of performance and reliability of the key components in the CSP receivers, material durability under high fluxes, and system performance under fluctuating irradiation conditions [16]. Current CSP plants have efficiencies as low as 30-40%, with inlet temperatures less than 600°C [35]. Volumetric receivers can reach efficiencies around 70-90%, and outlet temperatures as high as 1200°C showing great potential for improvement.

The next sections will look at the existing work that has been done into volumetric solar receiver design to improve performance, with a specific focus on porous receivers.

1.2.3 Uniform Porosity Foams

Porous materials are increasingly being investigated for their volume efficient abilities to transfer heat. Below are various studies investigating the usage of constant porosity foams in concentrating solar receivers.

Wu et al. [75] used the P1 radiation model to simulate the temperature distribution of the fluid and solid phases in volumetric solar air receivers. Air velocity, porosity, mean cell size, and the thermal conductivity of the ceramic were all analyzed for their effects on the temperature field. It was found that by increasing the inlet velocity, the outlet temperature would decrease, and the
solid temperature became more uniform. The maximum temperature location also occurred deeper inside the absorber. The study found that the front temperature of the absorber decreased as porosity increased because as porosity increased, radiation was absorbed deeper into the receiver, removing heat from the front surface. The thickness of the thermal non-equilibrium region also increased slowly with the porosity. The thermal non-equilibrium region is defined as the region where temperature is changing. It was found that the extinction coefficient, the sum of scattered and absorbed radiation, decreased with an increasing porosity and an increasing mean cell size. Porosity seemed to have a higher influence on the front temperature of the absorber, where the cell size had more influence on the thickness of the non-equilibrium region. The thermal conductivity of the ceramic foam did not have a significant effect on the performance of the absorber. The study concluded that a feasible way to optimize the air receiver is to minimize the thermal radiation losses by decreasing the absorber’s front temperature. Based on the experiments and observations made, a ceramic foam with a mean pore diameter of 1-2 mm and a high porosity is the most favourable choice for the absorber microstructure.

A similar conclusion was also found in research from Dyck et al. [76] who utilized the Monte Carlo Ray Tracing (MCRT) method on various porosities and pore diameters. MCRT involves random trajectories of many rays and tracks the rays as they intersect with surfaces. Each ray has the same amount of energy and a specific, random direction. The ray’s behaviour is determined by the surface’s properties. This study found that extinction coefficients had a strong negative correlation to porosity, and a weak positive correlation with mean pore diameter. From these studies, it was concluded that graphite foams could be modelled as a black body. Wang et al. [77] also used the MCRT method to investigate the thermal performance of porous media receivers. MCRT was used to obtain the heat flux distribution on the fluid inlet surface of the porous receiver. This distribution was used as the wall heat flux boundary condition of thermal performance analysis. A local non-equilibrium thermal equation (LNTE) model with Rosseland approximation was used to investigate temperature distributions. A physical model was created from a silicon ceramic porous media receiver and was placed in the focal point of a solar dish. It was found that heat flux distribution has a large impact on temperature distribution. With uniform heat flux, temperature distribution was consistent in each vertical section of the receiver. Non-uniform heat flux resulted in inconsistent temperature distributions within the
receiver. For both uniform and non-uniform heat flux, maximum temperatures were located at the fluid inlet because of radiation absorption at the surface. Higher maximum temperatures were seen in the non-uniform heat flux. The lateral walls and outlet of the receiver were assumed to have minimal heat loss; therefore, the major heat loss was occurring at the front face, confirming that large temperatures at the front surface lead to heat losses. This study analyzed temperature with and without consideration of radiation. There was a major difference in temperature observed showing that radiation has a significant effect on the thermal performance of the receiver.

Wang et al. [77] also conducted a parametric study considering increasing porosities. In this study, a porosity range of 0.3-0.6 was tested. It was observed that the maximum temperature and the thickness of the non-equilibrium region of the solid phase increased with the porosity, as well as the front surface temperature. A mass flow range of 0.08-0.14 kg/s was investigated. It was observed that the maximum temperature of the solid phase and equilibrium temperature decreased with an increasing flow mass. The temperature difference between the solid and fluid in the thermal non-equilibrium region phase also increased with an increase in flow mass. A solid emissivity range of 0.3-0.9 was investigated, and it was seen that temperature of the solid phase decreased with the emissivity increasing. Finally, pore diameters in the range of 0.75-1.5mm were investigated with a fixed mass flow rate and porosity. The maximum temperature of the solid phase increased with the pore diameter increasing. The non-equilibrium region decreased with an increase in pore diameter.

In another study, Wang et al. [78] utilized a Monte Carlo Model (MCM) and Finite Volume Model (FVM) coupling method to solve for radiation, conduction, and convection in the porous receiver of a multi-dish collector. MCM was used to find the concentrated heat flux distribution on the inlet surface of the porous receiver. FVM was then used within the receiver to solve solid and fluid heat transfer. Through these simulations, it was found that the temperature difference between the solid and fluid phases increased with an increase in solar irradiance, while the thermal non-equilibrium region decreased slightly. The solid phase temperature decreased when air inlet velocity was increased, and the thermal non-equilibrium region increased with increased air velocity. When pore diameter was increased, there was a sharp increase in solid phase
temperature and at a specific average diameter, the thermal non-equilibrium region decreased to nearly zero. It was found that the solid phase temperature and thermal equilibrium temperature decreased as the receiver radius increased, citing the increased mass flow rate. Finally, it was found that changing the air properties had very little influence on temperature distribution in the receiver.

Kribus et al. [79] in a review paper studied various geometric and material properties, and the effects they had on volumetric absorber performance. A one-dimensional model was created for a SiC open ceramic foam absorber with local thermal non-equilibrium and effective volumetric properties. Air entered at atmospheric and ambient temperature and incident radiation was produced with a heliostat field. The study found that the temperature of the absorber’s front face was much higher than the exit and surrounding air temperature, which was explained by the initial layers absorbing most of the incident radiation. Both convection and conduction serve as heat removal techniques but were insufficient to keep the temperature lower. Thermal balance was achieved when the absorber temperature was high enough to remove all radiation through emission or convection. A study was done analyzing how changing the convective heat removal from the absorber affected the absorbers temperature. Doubling convection led to efficiency increases, with significant decreases in the front surface temperature and an increase in the exit temperature. Increasing the conductivity of the material to a level that is typical to that of metal oxides also leads to an increase of efficiency of about 2%. Local heating of the front surface was more pronounced in materials with low conductivity as heat removal was reduced. It was found that the best efficiency occurred at a higher porosity and higher pore size. The best efficiency achieved was 86% with a pore diameter of 4 mm and porosity of 90%, however with this configuration, most of the solar radiation was absorbed in the bulk of the sample, with very little reaching the back end of the absorber. An interesting occurrence was when a volumetric profile was established, high thermal conductivity resulted in heat transport from the back of the absorber towards the front which diminished the absorber performance. As this study was conducted in only one dimension, there are likely to be inaccuracies compared to real life applications, such as the local air velocity having non-uniform lateral distribution.

Mey-Cloutier et al. [80] studied different ceramic foams including Silicon Carbide (SiC) and Zirconium Diboride (ZrB2) with porosities of 72-88% and pores per inch (PPI) of 4-18. The
results were compared against a SiC honeycomb absorber. The solar to thermal efficiency was measured by comparing the conversion of the incident solar irradiant to the heat recovered at the outlet. Contradictory to previous studies reviewed, it was observed that higher efficiency was obtained for a lower porosity, citing the enhanced strut density which allowed for higher convective heat transfer. Higher efficiency was also observed with higher PPI (with a lower pore diameter) again because of the enhanced convective exchange between the solid and air flow. The best efficiency was obtained for the absorber with the lowest porosity, and highest PPI. Comparing the materials used, the ZrB2 samples had higher solar absorptivity than the SiC samples, while maintaining the same low infrared emissivity.

Metal foam receivers have also been investigated as viable options for concentrating solar receivers. Kopanidis et al. [81] created an open cell metal foam model for two different densities: 10 and 40 PPI. Results from this study show that it is valid approach to simulate flow and heat transfer on the pore scale for estimating behaviour of larger scale behaviour. Albanakis et al. [82] analyzed heat transfer and pressure drop for nickel and Inconel volumetric foam receivers. The focal point was adjusted by changing the distance of the source from the foam sample. Results showed that the pressure drop in the nickel foam was always higher than that of Inconel. Heat transfer was evaluated in relation to an effectiveness coefficient, which showed that heat transfer of nickel was always higher than that of Inconel.

Andreozzi et al. [83] evaluated the effect of local radiative conductivity on a 2D conductive-convective-radiative model within a heated channel filled with metal foam. This study simplified tetrakaidecahedron shaped pores down to cubes, a method that has been validated by experiments done by Zhao et al. [84]. The CFD simulation was solved using an equivalent conductivity model for fluid and solid parts. Equivalent conductivity was found using the following equations:

\[ k_{\text{eff}} = \text{porosity} \cdot k_{\text{efluid}} + (1 - \text{porosity}) \cdot k_{\text{esolid}} \]  \hspace{1cm} (1.2)

where:

\[ k_{\text{efluid}} = k_r + k_{\text{efluid}} \]  \hspace{1cm} (1.3)

\[ k_{\text{esolid}} = k_r + k_{\text{esolid}} \]  \hspace{1cm} (1.4)

where kr is radiative conductivity and kc is solid or fluid conductivity.
Radiative conductivity can be conducted using the following equation

\[ k_r = \frac{q_r}{dT dn} \]  \hspace{1cm} (1.5)

where \( q_r \) is radiative heat flux, \( dT \) is change in temperature, and \( dn \) is distance into the solid.

This correlation was then used to solve the energy equation for porous materials. Simulations were done for various porosities, cell edge lengths, and surface emissivity’s. Results showed that when emissivity was increased from 0 to 0.95, there was a significant increase in heat diffusion. Radiative conductivity is calculated through combined solid and reflectance emissions. The smaller a porosity is, the more “solid surface” that it has, the larger the solid emission contribution and the larger the radiative conductivity. The smaller the porosity, the smaller the reflective contribution to radiative heat transfer, and the smaller the radiative conductivity. From this study, it was concluded that even though a smaller porosity can lead to more emitted radiation, the reflectance contribution is smaller to radiative conductivity than the solid emittance contribution; however, these results did depend on the location within the receiver domain.

Closer to the surface, emission had a bigger contribution to conductivity. Near the outlet of the receiver emission and reflectance had equal contributions to effective radiative conductivity. Overall, this study showed that there was a strong contribution of radiative conductivity to effective conductivity, proving that modelling radiation is crucial when analyzing heat capture in concentrating receivers.

Using the effective conductivity for all energy transport holds the assumption there is thermal equilibrium between the solid and gas and is one way to model heat transfer; however, the temperatures can be quite different which should be considered if the solid and gas phase temperatures are important to thermal behaviour of a system. In these cases, it is necessary to represent energy stored in each phase, as well as the exchange. Gómez et al. [85] used a thermal non-equilibrium approach to propose a complete heat transfer model for porous beds. The model considered the separate conduction of solids and gasses, heat exchange by convection between the two phases, and transport of radiation and its distribution inside a porous bed. The overall goal was to simulate heat transfer in a porous medium in Fluent and compare against experimental results. The author’s used a spherical concentrator which reflected solar rays into a quartz tube receiver containing silicon carbide spheres with air forced through. This was then
modelled in Fluent with three different methods: a modified discrete ordinance (DO) model, a standard discrete ordinance model, and an equivalent conductivity (EC) model. The DO model was chosen as it is suitable for high participative media, and it has good directional accuracy; however, it has limited capabilities in multiphase medium where temperatures and radiative properties of different phases are considered. Therefore, it was modified using C++. Details of the modifications can be found in the paper. Compared to experimental results, the equivalent conductivity model had large differences in the solid and gas temperatures at the surface, which was pronounced at the front surface. The difference was reduced in lower layers due to poor penetration and dissipation of radiation in the porous zone. The standard DO model had higher temperatures than the modified model, but heat was still lost at the surface similar to the EC model. In the EC and standard DO model, heat was captured and transported by the gas phase, with thermal equilibrium modelling immediate heat exchange through the solid and gas phases. The modified DO model, which did not assume thermal equilibrium between the phases, had a delayed transfer between the solid and gas, which allowed for more heat transfer and deeper penetration into the porous zone. Overall, the modified DO model showed better results for predicting temperature in a porous solar absorber when compared to the standard DO model or the equivalent conductivity model.

Porous material has also been employed in different types of solar receivers, such as flat plate solar receivers. Flat plate solar receivers have multiple layers of glass that are painted black to increase the absorption of radiation with thinner layers at the surface to absorb shortwave radiation, and a thin lower layer that is highly reflective and low emittance for long wave radiation [18]. In these types of receivers, removing heat quickly and efficiently is crucial. In one study [86], the addition of porous media improved heat transfer by 64.3%, increasing outlet temperature and overall efficiency.

These studies analyzing uniform porosity foams have shown that porous materials can be effective at capturing concentrated sunlight and translating this heat to the back surface of the receiver. Some studies suggested higher porosity foams are best for decreasing front temperature, and others suggesting that lower porosity foams are more efficient overall. The next section will explore a combination of high and low porosity foams, where studies with porosity gradients are reviewed.
1.2.4 Variable Porosity Foams

To address the high temperature losses seen at the front surface of absorbers, there have been some studies investigating varying the porosity of solar receivers to improve heat capture.

Avila-Marin et al. [87] tested different configurations of layered mesh absorbers. Three layered metallic wire mesh absorbers were tested with different porosities and compared to a standard ceramic foams and metal foam. The wire mesh ranged from 0.13 to 1.00 mm and mesh sizes from 0.2 to 4.0 mm. In general, the gradual porosity absorbers had better efficiencies than the uniform porosity receivers. No clear trend exists for the layered absorbers, but the 54-38% appeared to have the best thermal performance of all the absorbers for low, medium, and high incident radiation conditions; however, these results may have been influenced by the wire diameter.

Fend et al. [88] analyzed the practicality of two porous materials for volumetric receivers: a double-layer silicon carbide foam and screen-printed silicon carbide. The bottom layer had a density of 20 PPI and a top layer of 80 PPI. This was compared to a sample of 20 PPI. Another unique feature of these foams was the manufacturing. Typical manufacturing of foams results in parallel channels which are manufacturing by extrusion but have the disadvantage of linear permeability properties which is not ideal for flow stability. A process called “direct typing” was used to create 3D geometry samples out of silicon carbide, which allowed for pores that were not parallel. It was found that the layered material was approximately 10% more efficient than just the 20 PPI receiver. This could be due to the larger heat transfer surface available which was approximately 5400m²/m³ for the 80 PPI foam compared to the 1100m²/m³ for the 20 PPI foam. There was also a significant improvement of efficiency of the direct typing process (about 80% efficient) compared to the metallic parallel channel structure which only had efficiency of 40%.

Wang and Vafai [89] have also proposed a porous absorber with a variable pore structure, with a linear change in porosity or pore diameter distribution along the flow direction. They compared the results of a decreasing porosity to that of increasing porosity and constant porosity under constant radiation influx using a modified P-1 model, as can be seen in Figure 1-12. Their findings concluded that a decreasing porosity type absorber, with a high porosity gradient was
most effective at decreasing the front temperature of the absorber. When testing the effect of pore diameter on heat transfer, they found that an increasing pore diameter structure led to a decrease in temperature at the front surface, and a higher temperature at the outlet with the highest difference up to 50K at the outlet. It was also found that the pore diameter had a greater effect on the distribution of conductive flux in the solid phase than the radiative flux. From this, it was concluded that a decreasing porosity combined with an increasing pore diameter could be an optimal structure as a porous absorber, however this combination was not tested.

Figure 1-12 Porous structures with decreasing porosity (left) and increasing porosity (right) proposed by Wang and Vafai [89]

In work done by Wang [90], the MCRT Method was used to simulate the propagation of concentrated radiant energy in a variable porous structure. The acceptance rejection method (ARM) was applied to generate the free path of the photon. The heat flux and radiation absorption were measured throughout the sample. It was confirmed using this method that the decreasing type solar collector achieved the deepest radiation absorption, penetrating 2 cm into the medium while a constant and increasing type collector saw radiation penetration within 1cm. It was observed that increasing the porosity at the entrance of the decreasing type receiver results in deeper penetration of photons into the receiver without resulting in optical loss due to scattering at the entrance, a phenomenon seen when increasing the porosity of the constant porous absorber. The best performing efficiency gradient was found to have a porosity of 0.95 at the inlet and 0.5 at the outlet. This study also analysed the effect that variable pore size has on solar radiation absorption homogeneity. Contradictory to the previous study, the best results were found in the decreasing type pore layout with an inlet pore diameter of 5mm and an outlet pore
diameter of 0.5mm. This is because, similar to a decreasing porosity, a decreasing pore size effectively allows rays to enter and captures them as the pore size decreases.

Du et al. [91] utilized an optimization algorithm to design a porous medium that optimizes heat capture. They employed the MCRT radiation modelling and random bubble generation for model generation. The porosity was unvaried in each layer and gradually changed down the whole structure. Where previous studies used two or three layers of constant porous material, this study used 10 sublayers with a constant pore diameter of 1mm. When measuring absorptivity and reflectivity of the structure, it was found that the gradually varied porous media was more absorptive and less reflective, therefore retaining more radiation. In terms of radiative flux distribution, it was found that a gradually varying structure reduced flux at the front surface and reduced the flux gradient within the receiver, leading to less thermal stress placed on specific point of the receiver which can cause stress damage. It is found that a front porosity of 0.95 and a back porosity of 0.65 was optimal. In terms of gradient of change, a porosity that changed slowly in the front layers and more sharply in the back layers optimized the volumetric effect and performed better. This orientation allowed more radiation to penetrate the front of the receiver, while at the back the total solar energy decreased so the porosity needed to be smaller to increase the extinction coefficient and absorb more radiation. A gradually varying porous size was also tested from 1.4-0.6mm and a constant porosity of 0.8. Compared to a uniform structure of 1mm and porosity of 0.8, the solar radiative flux was more uniformly distributed. This study only analyzed radiative and optical properties; heat transfer fluids were not taken into consideration.

Roldán et al. [92] conducted a 2D analysis of gradually varying porous receivers that varied in depth, as well as one that varied radially and compared these results against constant porous receivers. From this study, it was found that the receiver that had decreasing porosity depth wise had the highest thermal efficiency of 82%. The temperatures reached in the gradually decreasing depth receiver were lower than the radial receivers with the max temperature of 829K in the depth decreasing, and a max temperature of 1323K in the radial receivers. However, better performance was seen in the decreasing depth porous receivers because the thermal distribution was more homogenous than the gradual radial receivers. It was found that both receivers were at
risk of overheating in the centre due to low air velocity. This was more prevalent in the receivers that changed in the radial direction.

Zheng et al. [93] coupled CFD with a genetic algorithm to optimize porous geometry to enhance convective heat transfer. The region to be optimized was divided into layers in the radial direction. The effect of changing the porosity was observed, and the optimal porosity was found to be a layered porosity varying from 0.95 to 1.0. This is based on optimal flow velocity leading to better convective heat flow. They stated two main causes for thermal hydraulic enhancement with the insertion of multiple porous layers: (1) multiple layers of porous media increase the effective thermal conductivity, (2) multiple layers of porous media increase the velocity and the velocity gradient near the wall. It was found that the optimized axial velocity distribution does not change when number of porous layers exceeds 5.

In a study by Dathathri and Balaji [94] variable porous mediums were studied with specific application for cooling small electronics. Five different configurations of porous media were considered: constant porosity, increasing porosity, decreasing porosity, and porosity that increased to the middle, then decreased again to the outlet. In the varying porosity cases, each layer had a constant porosity. A genetic algorithm was used to optimize heat transfer, and minimize the pressure drop across the porous medium. Using this algorithm, 3 different cases were created with different emphasis on priorities. When equal emphasis was placed on decreasing hot spot temperature and reducing pressure drop, the porosity that gradually increased, then decreased had the best performance. When there was an increased emphasis on minimizing hot spot temperature, a porosity gradually increasing moving away from the wall had the best performance. When increased emphasis was placed on decreasing pressure drop, a high constant porosity was the highest performing. This study is interesting to consider optimal configurations when different priorities are emphasized.
1.3 Motivation and Objectives

Solar energy is a promising solution to decrease our reliance on fossil fuels for energy production. Concentrated solar power can effectively translate sunlight from the whole spectrum into usable thermal energy. Volumetric receivers are a high performing option for efficiently absorbing incident radiation in CSP systems. Porous materials have been studied as a receiver type; however, they still face the design challenge of high temperatures at the front surface of the receiver, leading to re-radiation losses. A gradual gradient porous structure shows promising results for the effective capture of radiation while decreasing the front surface temperature of the receiver. As presented above, there are many design features that can impact the performance of these receivers such as the porosity of the porous receiver, the pore size, and the material of the receiver. Many of studies reviewed have modelled the receiver as a 2D object, or else a porous continuum, which does not accurately represent a physical model of the receiver. There is a need for a 3D porous model that accurately models experimental results.

The objective of this thesis is to develop a 3D computational model that allows for the investigation and characterization of heat transfer properties, specifically radiation heat transfer properties, of porous materials. This computational model could then be used to design an optimal porous structure that enhances heat transfer to a phase change material vessel. The focus of this thesis is on the development and calibration of the computational model, and a preliminary study on the geometric and physical properties of porous materials that lead to suitable heat transfer in their usable range.

This thesis feeds into a larger project which aims to improve the overall efficiency of CSP. Research has been done to increase the concentration ratio of parabolic dish collectors by improving ray tracing [95]. This concentrated sunlight would then be absorbed by the receiver that will be explored in this thesis. After passing through the receiver, the absorbed heat would then be conducted directly into a storage medium, such as phase change materials, instead of passing to a heat transfer fluid then to storage. This system in Figure 1-13, would result in fewer potential losses and a more efficient system overall.
A conceptualized CSP system that would concentrate sunlight to a porous medium which then directly transfer the heat to a thermal storage, reducing heat transfer losses [34].

1.4 Thesis Layout

Chapter 1 includes an introduction to CSP, some of the design challenges currently faced, the studies that have been done to address these challenges, and the motivation and objectives for this thesis.

Chapter 2 describes the methods and results from an experiment that was done with a simplified gradually decreasing porous block.

Chapter 3 lays out the numerical methods used to create a computational model.

Chapter 4 describes the computational model that was created in ANSYS Fluent and verified using temperature data from the experimental work. This chapter also includes a description of the model that was created in ANSYS CFX with more geometrically complex porous structures.

Chapter 5 lays out the initial stages of a parametric study that was conducted with these complex porous structures.

Chapter 6 summarizes the results drawn and has future recommendations to carry this research forward.
Chapter 2

2 Experimental Setup and Results

An experimental setup was created for two main purposes: to calibrate a computational model of radiation exchange between a lamp and a receiver component, and to serve as a test facility for the testing of various receivers of different material and geometry. This chapter details the experimental setup and results from this study used for the calibration and verification of the computational model, which is introduced in Chapter 4.

2.1 Experimental Setup

The test section of the experimental setup, as seen in Figure 2-1, is comprised of a halogen lamp shining on a copper plate, which is insulated from below. An object of interest was placed on top of the copper plate to receive incident radiant energy from the halogen lamp. To facilitate calibration and computational modelling, the object was fitted with several thermocouples to record the steady-state temperature. There were two different objects placed on the plate: a plain aluminum block and a machined aluminum block mimicking a porous block. A halogen lamp was used as the radiation source because of its wavelength overlap with sunlight, as seen in Figure 2-2. It also reaches very high temperatures, making it a good approximation to concentrated sunlight.
For calibration purposes, an aluminum alloy 6061 T6 block was placed under the lamp on top of the plate. This block was used as a control for a model porous block of similar dimensions. Holes that gradually decreased in size were drilled down the centre of the block. Figure 2-3 shows the machined block and internal dimensions. Figure 2-4 shows the block that was used in the experiments. Holes were added to the side of the block so that thermocouples could be inserted.
to measure internal temperature. This model block was used to approximate the properties of a gradually decreasing porous block. Because of the geometric simplicity of this block, it could be accurately modelled in CFD, which facilitated verification of the calibrated computational model.

Figure 2-3 Internal pore diameters and depths of holes in the model porous block (Note: the depth of each hole was 9.5 mm below the previous hole as indicated by the first hole depth)

Figure 2-4 Model porous block used in experiments

The surface temperature of the block under the lamp was measured using a FLIR infrared (IR) camera. To accurately measure temperatures, the surfaces measured could not be too reflective as shiny surfaces reflect light and can affect temperature readings. For this reason, all surfaces were coated in black high-temperature spray paint. Before taking measurements using the
camera, it was necessary to find the reflected surface temperature. This is used to account for any reflected radiation that could potentially affect measurements from the camera [97], [98]. To find the reflected surface temperature the camera was set to the angle at which the measurements will be made. Within the camera software, the emissivity of the camera was set to 1, and the distance was set to 0 m. The camera was then pointed at aluminum foil (a highly reflective surface) and the average temperature of the aluminum foil was taken with the camera [98]. The reflected temperature was found to be an average of 20°C and was input into the camera software. The emissivity was reset to the default of 0.92, and the distance was set as 0.9 m which was the distance between the camera and the halogen lamp.

After finding the reflected surface temperature, the lamp was turned on and the block and plate were placed under the lamp. Type-K thermocouples were used to measure the surface temperatures of the copper plate, the aluminum block top, and the sides of the aluminum block. Measurements were recorded after the temperature reading from the thermocouple reached a steady state, which was assumed when there was no observed temperature change of the aluminum block for 20 mins. This state could take up to 3 hours to achieve.

To ensure accurate temperature recordings were being taken, it was necessary to account for the emissivity of different surfaces used in the experiment and sensed by the IR camera. The emissivity of the surfaces under the halogen lamp was found using the camera software, which allowed for emissivity calibration. The temperature recorded with the surface thermocouple was input into a selected region of the camera image (i.e., the top of the block). This was then compared to the temperature the camera recorded and the camera software corrected it to match the temperature that the surface thermocouple measured. This was achieved by adjusting the measured emissivity of the surface. The calculated emissivity was displayed by the software and used in future temperature measurements.

With the IR camera properly calibrated for reflected temperature and accurate surface emissivity, final measurements were taken. To take these measurements, infrared images of the plate and block were taken at steady state, as seen in Figure 2-5. The camera was angled in such a way so that the top of the plate and block could be seen clearly by the camera. These images contained
temperature information which were exported to an excel spreadsheet with x-y co-ordinate data. The temperature across the plate top, block top, and block sides were averaged and recorded.

![Image of machined aluminum block and plate with insulation underneath taken with thermal camera. The colour bar represents the temperature in Celsius.](image)

There are some locations at which the temperatures could not be recorded using the FLIR camera including the surface of the halogen lamp and the internal hole temperatures of the drilled block. These locations were measured using type-K thermocouples. The temperature of the halogen lamp was measured by placing the thermocouple on the surface of the outer lens. The internal hole temperatures were measured by inserting the thermocouple into the pore via the holes in the side and pushing the thermocouple against the inner wall of the hole. The internal temperature was challenging to read as it was difficult to get the entire thermocouple in contact with the wall. Additionally, the spaces were too small to use surface thermocouples. Because of these challenges, the accuracy of these internal readings are subject to high levels of uncertainty. All of the locations were chosen as they provide valuable insight into how the heat is being transferred down the aluminum block, however only measurements taken using the surface thermocouple or infrared camera were used for validation purposes. Each measurement was repeated three times to ensure reproducibility. They were also repeated within the same room which had a consistent room temperature and air circulation.
Further tests were conducted to investigate if the distance or angle of the camera affected the temperatures measured. No significant differences were observed when the distance and angle were changed, and all measurements were taken under the conditions detailed in Table 2-1.

<table>
<thead>
<tr>
<th>Table 2-1 Infrared camera position information</th>
</tr>
</thead>
<tbody>
<tr>
<td>Distance of camera from aluminum block</td>
</tr>
<tr>
<td>Angle of camera</td>
</tr>
</tbody>
</table>

Additional dimensions of the experimental apparatus are provided in Table 2-2. These remained consistent through all the experiments.

<table>
<thead>
<tr>
<th>Table 2-2 Experimental feature dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Distance between copper plate and lamp surface</td>
</tr>
<tr>
<td>Dimensions of Insulation Foam underneath Copper Plate</td>
</tr>
<tr>
<td>Dimensions of Copper Plate</td>
</tr>
<tr>
<td>Dimensions of Aluminum Block</td>
</tr>
<tr>
<td>Dimensions of Porous Block</td>
</tr>
</tbody>
</table>

Each measurement device that was used had uncertainty associated with it. These uncertainties are detailed below in Table 2-3.

<table>
<thead>
<tr>
<th>Table 2-3 Measurement device uncertainties</th>
</tr>
</thead>
<tbody>
<tr>
<td>FLIR IR Camera</td>
</tr>
<tr>
<td>K-type surface thermocouples</td>
</tr>
<tr>
<td>K-type thermocouples</td>
</tr>
</tbody>
</table>

The temperature of the outer lens of the halogen lamp was measured at 537 K.
2.2 Experimental Results

Temperature results from these experiments are detailed in the following section.

**Measurement 1: Copper plate**

The first measurement taken was just the copper plate (insulated from below) with no feature on top of it. Measurements were taken with a surface thermocouple and the thermal camera. Temperature measurements are given in Table 2-4.

<table>
<thead>
<tr>
<th>Thermocouple Measurement</th>
<th>139.5°C</th>
<th>412.5 K</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal Camera Measurement</td>
<td>139°C</td>
<td>412 K</td>
</tr>
</tbody>
</table>

**Measurement 2: Copper plate with Plain Aluminum Block**

The plain aluminum block (i.e., block without holes) was placed on top of the copper plate and allowed to reach steady state, at which temperature measurements of the top of the block, the side of the block, and the copper plate were taken (see Table 2-5).

<table>
<thead>
<tr>
<th>Thermocouple Measurement – Top of Block</th>
<th>87°C</th>
<th>360 K</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal Camera Measurement – Top of Block</td>
<td>87.2°C</td>
<td>360.2 K</td>
</tr>
<tr>
<td>Thermocouple Measurement – Side of Block</td>
<td>84.3°C</td>
<td>357.3 K</td>
</tr>
<tr>
<td>Thermal Camera Measurement – Side of Block</td>
<td>84.5°C</td>
<td>357.5 K</td>
</tr>
<tr>
<td>Thermocouple Measurement - Copper Plate</td>
<td>89°C</td>
<td>362 K</td>
</tr>
<tr>
<td>Thermal Camera Measurement - Copper Plate</td>
<td>89.5°C</td>
<td>362.5 K</td>
</tr>
</tbody>
</table>

**Measurement 3: Copper Plate with Machined Aluminum Block**

The aluminum block with drilled holes was placed on top of the copper plate. Temperature measurements were taken once the system had reached a steady state temperature (see Table 2-6).

<table>
<thead>
<tr>
<th>Thermocouple Measurement – Top of Block</th>
<th>87°C</th>
<th>360 K</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal Camera Measurement – Top of Block</td>
<td>87.2°C</td>
<td>360.2 K</td>
</tr>
</tbody>
</table>
It was observed that the surface temperatures for the plain aluminum block and the drilled block were nearly identical. The side surface temperatures were found to be slightly higher in the drilled block. The temperature of the copper plate on its own was significantly higher than the temperature underneath the aluminum block. When a feature was added, the copper plate had a higher temperature than the block. This is because there was a higher surface area of plate exposed to the incoming radiation than the aluminum block.

**Measurement 4: Internal Hole Temperatures**

The internal temperatures of the model porous block (the drilled block) were taken by inserting a K-type thermocouple into the access holes and pushing the thermocouple against one of the walls to maximize the contact between the thermocouple and the wall. Results from this measurement can be seen in Table 2-7.

<table>
<thead>
<tr>
<th>Table 2-7 Internal model porous block temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bottom hole (hole 1)</td>
</tr>
<tr>
<td>Hole 2</td>
</tr>
<tr>
<td>Hole 3</td>
</tr>
<tr>
<td>Top hole (hole 4)</td>
</tr>
</tbody>
</table>

The temperatures measured were similar to those measured on the surface and exterior of the porous block. It was observed that the temperature at the bottom of the block was 2 degrees higher compared to the temperature at the top of the block. This could be because the hole at the bottom of the block was much smaller compared to the pore size at the top of the block, resulting in less heat loss due to convection.
When taking these measurements, the thermocouple was able to make contact with the wall, however the other half of the thermocouple was exposed to air, which likely had an effect on the temperature. It is therefore challenging to know how much these measurements were influenced by the surrounding air temperature, and the incident radiation from the halogen lamp. For this reason, these results were not heavily considered when validating the computational model.

**Emissivity of Material Surfaces**

The emissivity of the block top, block sides, and plate top were calculated using the methods described in section 2.1. The results can be seen in Table 2-8. The emissivity was used in computational simulations to set the emissivity of the respective surfaces.

<table>
<thead>
<tr>
<th>Surface</th>
<th>Emissivity Calculated by Camera</th>
</tr>
</thead>
<tbody>
<tr>
<td>Copper Plate Top</td>
<td>0.85</td>
</tr>
<tr>
<td>Copper Plate with Object</td>
<td>1</td>
</tr>
<tr>
<td>Aluminum Block Top</td>
<td>0.91</td>
</tr>
<tr>
<td>Aluminum Block Sides</td>
<td>1</td>
</tr>
<tr>
<td>Porous Block Top</td>
<td>0.92</td>
</tr>
<tr>
<td>Porous Block Sides</td>
<td>0.98</td>
</tr>
</tbody>
</table>

Emissivity of surfaces can be affected by many factors. This includes the roughness of the surface, the colour of the surface, the angle that the surface is at, or the surrounding temperature [98]. That is why the copper plate has a different emissivity when there is an object on top versus without, as the object effects the surrounding temperature and the light that is hitting the plate. It is noted that the emissivity calculated by the camera was 1 for the copper plate and the side of the aluminum block. Although this is not physically possible, as an emissivity of 1 would make an object a perfect black body, these values were used as they were the most accurate results measured. Additionally, it is likely that the block and plate all had emissivity’s close to 1 because of their black (painted) surfaces.

The temperature and emissivity data collected was used to validate a geometrically identical ANSYS Fluent simulation. The theory behind this model will be discussed in the next chapter.
Chapter 3

3 Numerical Formulation

A computational model was developed to investigate the impact that different materials and porous structures have on the absorption of thermal radiation. The computational model mimics the experimental setup described in Chapter 2 and is calibrated in Chapter 4 using data from the experimental setup. Because of the different features of ANSYS FLUENT and CFX, both codes were used in the computational modelling effort. Due to the robust and sophisticated approaches for radiation modelling included in FLUENT, it was used to simulate the lamp to determine the radiation field incident upon the drilled porous block for the calibration feature. CFX was then used for the study of incident radiation on porous materials due to its enhanced capability for handling complex porous structures and their associated grids. While the basic formulation applies to both CFD codes, the differences in the manner that radiation is modelled is also described below.

3.1 Governing Equations

The conservation of mass and momentum equations are presented below as they apply to a laminar flow in an inertial (non-accelerating) reference frame. All fluids are treated as Newtonian and incompressible. The equations presented in this Chapter are adapted from the ANSYS manual [102].

The conservation of mass equation is given as,

$$\frac{\partial \rho}{\partial t} + \Delta \cdot (\rho \vec{v}) = 0$$  \hspace{1cm} (3.1)

where $\rho$ is the fluid density and $\vec{v}$ is the velocity vector.

The conservation of momentum equation is given by,

$$\rho \frac{\partial \vec{v}}{\partial t} + \rho \vec{v} \cdot \nabla \vec{v} = -\Delta p + \rho \vec{g} + \mu \nabla^2 \vec{v}$$  \hspace{1cm} (3.2)
where $p$ is the fluid pressure in the control volume, $\vec{g}$ is the gravity vector, and $\mu$ is dynamic viscosity.

The conservation of energy equation in fluids is written as:

$$\frac{\partial}{\partial t} (\rho E) + \nabla \cdot (\vec{v}(\rho E + p)) = \nabla \cdot (k_f \nabla T) + S_h$$

where,

$$E = h - \frac{p}{\rho} + \frac{v^2}{2}$$

$k_f$ is the conductivity of the fluid, and viscous dissipation has been neglected. $S_h$ represents volumetric heat sources, such as radiation absorption, or others that are user-defined. $h$ represents enthalpy.

The conservation of energy for stationary solids can be written as follows,

$$\frac{\partial}{\partial t} (\rho_s h) = \nabla \cdot (k_s \nabla T) + S_h$$

where $\rho_s$ and $k_s$ are the solid density and conductivity, respectively.

For the present study, the volumetric heat source in the energy equation(s) is radiation. The radiation transfer equation (RTE) for absorbing, emitting, and scattering medium at position $\vec{r}$ and direction $\vec{s}$ can be expressed as the following equation,

$$\frac{dl(\vec{r},\vec{s})}{ds} + (a + \sigma_s) I(\vec{r},\vec{s}) = an^2 \frac{\sigma T^4}{\pi} + \frac{\sigma_s}{4\pi} \int_0^{4\pi} I(\vec{r},\vec{s}') \Phi(\vec{s} \cdot \vec{s}') d\Omega'$$

where $I$ is radiation intensity (which is dependent on position and vector), $s$ is path length, $\vec{r}$ is a position vector, $\vec{s}$ is a direction vector, $a$ is the absorption coefficient, $\sigma_s$ is the scattering coefficient, $n$ is the refractive index, $\sigma$ is the Stefan-Boltzmann constant, $T$ is local temperature, $\Phi$ is a phase function, $\vec{s}'$ is the scattering direction vector, and $\Omega'$ is the solid angle.

The absorptivity coefficient and scattering coefficient define how quickly a material attenuates radiation due to absorption and scattering respectively. Specified in units of inverse-length ($L^{-1}$), they describe the change in radiation intensity per unit length along the radiation path in a fluid.
or semi-transparent medium, as seen in Figure 3-1. These coefficients effect the absorbing and emitting portions of the RTE.

![Figure 3-1 Medium showing absorption and scattering losses][102]

The RTE is a first order integro-differential equation for $I$ in direction $\vec{s}$. A boundary condition for $I$ is needed if the RTE is to be solved in a domain. A diffusely emitting and reflecting opaque boundary can be solved as,

$$I(\vec{r}_w, \vec{s}) = \epsilon(\vec{r}_w) \sigma T^4 + \frac{\rho_w(\vec{r}_w)}{\pi} \int_{n \cdot s' < 0} I(\vec{r}_w, \vec{s'}) |\vec{n} \cdot \vec{s'}| d\Omega'$$  \hspace{1cm} (3.7)

A diffusely emitting and specularly reflecting boundary can be treated as

$$I(\vec{r}_w, \vec{s}) = \epsilon(\vec{r}_w) \sigma T^4 + \frac{\rho^d(\vec{r}_w)}{\pi} \int_{n \cdot s' < 0} I(\vec{r}_w, \vec{s'}) |\vec{n} \cdot \vec{s'}| d\Omega'$$ 
$$+ \rho^s(\vec{r}_w) I(\vec{r}_w, \vec{s})$$  \hspace{1cm} (3.8)

where $r_w$ is the wall position, $\epsilon$ is the wall emissivity, $\rho_w$ is the reflectivity of the wall (1 - $\epsilon$), $\rho^d$ is the diffuse reflectivity= (1 - $\epsilon$)*diffuse fraction, $\rho^s$ is the specular reflectivity= (1-$\epsilon$)*(1-diffuse fraction), and $\vec{s}$ is the specular direction.

As seen, the radiative transfer equation depends on 3 spatial coordinates, and 2 local direction coordinates. A full solution of the RTE is extremely time consuming. Directional approximation models, such as Discrete Ordinance and Monte Carlo, and spectral approximation models, such as Gray, Multiband, and Weighted Sum of Gray Gases, reduce computational time while still attaining accurate results.
3.2 Radiation Model

The discrete ordinance (DO) model was chosen in FLUENT for its ability to model semi-transparent media, and for its relative accuracy. The DO model solves the RTE (Equation 3.6) for a finite number of discrete solid angles. Each angle is associated with a vector direction(s) in the cartesian system. The degree of angular discretization is specified by the user, in this case it was set to a value of 3 to achieve more reliable results as recommended by ANSYS [103]. The DO model uses the RTE as a transport equation for radiation intensity and solves for the RTE for the number of directions, $\bar{s}$.

Uncoupled and coupled energy equations are available with the DO model. The uncoupled option solves sequentially, solving the equations for energy and radiation one-by-one assuming prevailing values for other variables. The uncoupled energy version uses the finite-volume scheme on unstructured meshes. The coupled method solves the discrete energy and intensity equations simultaneously, assuming spatial neighbours are known. The coupled approach is faster for simulations with optical thicknesses above 10 or high scattering coefficients [103]. The uncoupled approach was used in the present simulations as the optical thickness was low in the air domain.

Gray and non-gray modelling are available in the DO model. Gray modelling treats surfaces as gray, which means that absorptivity and emissivity are assumed independent of wavelength over the spectral regions of the irradiation and surface emission under consideration. Gray modelling was deemed appropriate for the surfaces involved in the present simulations as the materials in the study did not present significant non-gray attributes and this assumption simplified the computational model. More details of the DO model in Fluent can be found in [102].

3.2.1 CFX Computational Theory

While ANSYS FLUENT was used to calibrate the computational model due to its superior capabilities in radiation modelling, a CFX model was required to model the complex features of the porous foams considered in this study. To this end, the incident radiation field was
determined from FLUENT simulations and then used as input for the CFX simulations in the study of specific foams. Unique details about the thermal energy modelling in CFX are detailed below.

Radiation modelling in CFX solves the radiation transport equation (Equation 3.6) and calculates the volumetric source term \( S_h \) for the energy equation, radiative flux at walls, and any other quantities of interest. The only radiation model that can be used in a solid-fluid interface in CFX is the Monte Carlo Model [45]. Additionally, Monte Carlo (MC) is the only model in which a directional radiation source can be specified in CFX. Due to these requirements, and the relative robustness and accuracy of the model, the MC model was used for all radiation modelling in CFX.

The MC radiation field can be thought of as a photon gas. Applying this to the RTE, \( K_a \) is the probability per unit length that a photon is absorbed at a given frequency. The mean radiation intensity \( I \) is therefore proportional to the distance travelled by a photon in unit volume at position \( \vec{r} \), in unit time. The radiative heat flux is proportional to the incident rate of photons on surface at position \( \vec{r} \) because volumetric absorption is proportional to the rate of absorption of photons [43].

There are several values that can be found by following and tallying a selection of photons in volume elements. The mean total intensity can be found by finding the distance that a selection of photons has travelled. Additionally, the mean total radiative flux, and the mean absorbed flux can be found by calculating the distance that a selection of photons has travelled and multiplying by the emissivity of a surface. In CFX, the selection is specified by indicating the number of histories to be considered, which, by default, is 10000. These histories are then divided into groups; certain histories are selected, and their physical interactions (emission, absorption, and reflection) are tracked through the domain. Once the calculation has completed, quantities of interest, such as irradiation heat flux or absorbed radiation, are recorded for each selected history. The mean value and standard deviation of each quantity are computed in each group.
3.3 Natural Convection Model

When modelling natural convection, there are a few options. The Boussinesq model treats density as constant in all solved equations, except for the buoyancy term in the momentum equation, which approximates the impact of temperature on density using a thermal expansion coefficient. Although robust, the Boussinesq model is only valid for small temperature changes [43]. An alternative model for all temperature ranges is the incompressible ideal gas model which defines the density of air using the following equation,

\[ \rho = \frac{p_{op}}{R \cdot M_w} T \]  

(3.9)

where, \( R \) is the universal gas constant (8.314 J K\(^{-1}\) mol\(^{-1}\)), \( M_w \) is the molecular weight of the air, and \( p_{op} \) is the operating pressure specified as 101,325 Pa. In this form, the density only depends on the operating pressure, not the local relative pressure field.

With the incompressible ideal gas model, the operating density appears in the body force momentum equation as \((\rho - \rho_0) \vec{g}\). This body force term follows from redefining pressure as

\[ p'_s = p_s - \rho_0 g x \]  

(3.10)

where \( p'_s \) is the kinematic pressure, \( p_s \) is the static pressure, and \( g \) is gravity which was set as -9.81 m/s\(^2\) in the y-direction. This form allows for calculation of minor pressure differences that drive buoyant flow. The hydrostatic pressure of a fluid at rest becomes \( p'_s = 0 \) [43].

Due to its application to a wider temperature range, the incompressible ideal gas model for modelling buoyancy flow was used.

3.4 Grid convergence

Discretization is the process of replacing the continuous solution of a partial differential equation with a discrete solution at specific points fine enough to represent a continuous solution. ANSYS Fluent and CFX use the Finite Volume Method (FVM), meaning a domain is discretized into non-overlapping finite regions that fill the domain completely. In the centre of the finite volumes, solution values are evaluated and stored by the CFD code and exchange that
information between adjacent cells. Discretization error can occur where there are not enough
discrete points to approximate the continuous solution, resulting in an inaccurate solution.

To ensure that accurate results were being obtained in simulations and that grids used in
computational modelling were refined enough, grid convergence studies were done in the Fluent
and CFX domains. This examines the effects of spatial discretization error and minimizes it to an
acceptable level. This was done by computing solutions on an increasingly refined grid, until a
measured parameter changed by less than an accepted amount. To measure whether grid
convergence is achieved, a systematic approach called the Grid Convergence Index (GCI) has
been reported in the literature [104]. The result from this process is a Grid Convergence Index
number, which can be interpreted as the numerical error due to spatial discretization and can be
used to specify the uncertainty of numerical computation in a discretized domain. The details of
the developed computational simulations will be reported in the next chapter.
Chapter 4

4 Calibration of Numerical Model

A computational model of the experimental setup described in Chapter 2 was developed for simulation in ANSYS Fluent. This model was run using the Discrete Ordinance (DO) radiation model and used specifically to determine the radiation field produced by the halogen lamp used in the experimental setup. The non-uniform characteristics of this radiation field were then tested in terms of their sensitivity on the temperature of the feature used for calibration. The resulting approximation of this radiation field was then used as boundary input for preliminary calculations done in CFX on porous receiver materials.

4.1 Computational Setup and Boundary Conditions (ANSYS Fluent)

![Figure 4-1 Experimental domain setup created in Fluent with labelled features](image)
The computational domain shown in Figure 4-1 is comprised of a lamp, including the reflector, bulb and lens, a plate and model porous block, and the surrounding air projected downward from the lamp. The boundary and operating conditions are as follows:

1. The air exterior boundaries were defined as pressure inlets, which define fluid pressure at flow inlets. The condition can be used when the inlet pressure is known, but the velocity and flow direction are unknown, as is often the case in buoyancy driven flows. For this simulation, the pressure was set to atmospheric pressure, which was constant for the simulation. More details about the equations defining this condition can be found in [102].

2. The block and plate boundary conditions were all set as walls. It was required in the DO model to specify emissivity of surfaces for wall boundary conditions. As stated earlier, this determines how much radiation is emitted from a surface, and through Kirchhoff’s law of thermal radiation, how much radiation is absorbed. The emissivity for each surface, as measured in the lab, were input into the simulation, and can be found in Table 2-8. Since the block was contained within the domain (i.e., no faces adjacent to the outer boundaries of the domain), all internal faces were set using the coupled condition. All faces for the plate were set as coupled, except for the bottom face of the plate, which was set as an adiabatic wall. The diffuse fraction for all surfaces on the block and plate were set as 1, meaning that any radiation emitted from these surfaces was emitted in all directions. This is an appropriate approximation because the surfaces of the plate and block were not polished, leading to more diffuse emission. The internal faces that made up the decreasing hole sizes of the drilled block were all coupled and had emissivity’s of 1, which is an appropriate approximation because the faces were all painted black.

3. The reflector behind the lamp bulb was approximated as a highly reflective surface that was losing heat to surroundings, as was the case in the experimental setup. It was setup as a mixed convection, radiation boundary condition with a heat transfer coefficient of 10 W/m²K, and a free stream temperature of 294K (the ambient room condition). The external emissivity of the reflector was set as 1, which defined the emissivity of the external facing walls of the reflector. For the walls of the reflector that faced internally, the emissivity was set as 0.1 to approximate a non-absorbing, highly reflective surface. Additionally, the diffuse fraction was set to a value of 0 such that all reflected radiation was specular.
4. The faces of the lens were set as semi-transparent such that radiation could pass through. Additionally, the diffuse fraction was set to 0 on both sides of the lens. This was an approximation because the lens was a fully semi-transparent surface and it allowed for radiation to pass directly through the lens instead of being radiated in all directions. The emissivity of the lens was not considered, as emissivity is ignored for semi-transparent surfaces in ANSYS Fluent [103]. The faces on the outer boundary of the lens were set as opaque surfaces to reduce the amount of radiation leaving through the sides of the lens. This was a realistic approximation because in the experimental setting, the outer edges of the lens were covered by the housing of the lamp, which would have reduced the radiation leaving from these faces. These faces were given boundary conditions similar to those of the reflector, with a mixed boundary condition, a heat transfer coefficient of 10W/m²K, a free stream temperature of 294K, external emissivity of 1, internal emissivity of 1, and diffuse fraction of 1.

5. The bulb was created as a cylindrical shape to be similar to the halogen bulb in the lab. It was set as an opaque surface with an emissivity and diffuse fraction of 1, and was given a heat flux boundary condition, the value of which was determined as part of the calibration exercise described below.

All simulations were run to steady state using the pressure-based solver. For spatial discretization, a least-squares cell-based option was used for the gradient, and a second-order discretization was used for pressure. The second-order upwind method was used for momentum, energy, and discrete ordinates to obtain accurate results. The pseudo-transient under-relaxation method was used, which is a form of implicit under relaxation controlled through pseudo time step size. The explicit relaxation factors were all kept at default values. The default automatic time step size of 1 was used with a time scale factor of 1, along with a conservative length scale method which uses the cube root of the mesh volume for 3D cases [102]. Simulations were initialized through hybrid initialization. Residuals were set to $1 \times 10^{-6}$, with the maximum number of iterations set to 500. These were deemed appropriate limits to allow for steady-state conditions to be realized. Residuals stabilized around 200 iterations, and typically converged after a run time of approximately 2 hours on a dual Intel Core i7 2.9 GHz desktop computer with 16GB of RAM.
A grid convergence study was done for this Fluent simulation. The parameter of interest that was compared between grid sizes was the heat flux from the top of the plate. This was done by setting up a monitor on the plate surface that tracked the area weighted average surface integral of heat flux. From this process, it was found that a grid size of approximately 500,000 was appropriate with a GCI of 3.3%.

### 4.1.1 Materials

The plate underneath the block was copper, which had properties detailed in Table 4-1.

**Table 4-1 Properties of copper in ANSYS Fluent [103]**

<table>
<thead>
<tr>
<th>Properties of Copper</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>8978 kg/m$^3$</td>
</tr>
<tr>
<td>Specific Heat</td>
<td>381 J/kg K</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>387.6 W/m K</td>
</tr>
</tbody>
</table>

The block was assigned properties of aluminum detailed in Table 4-2.

**Table 4-2 Properties of aluminum in ANSYS Fluent [103]**

<table>
<thead>
<tr>
<th>Properties of Aluminum</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>2718 kg/m$^3$</td>
</tr>
<tr>
<td>Specific Heat</td>
<td>871 J/kg K</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>202.4 W/m K</td>
</tr>
</tbody>
</table>

Properties of glass are given in Table 4-3.

**Table 4-3 Properties of glass implemented in ANSYS Fluent [105]**

<table>
<thead>
<tr>
<th>Properties of Glass</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>2200 kg/m$^3$</td>
</tr>
<tr>
<td>Specific Heat</td>
<td>830 J/kg K</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>1.5 W/m K</td>
</tr>
<tr>
<td>Absorption Coefficient</td>
<td>200 m$^{-1}$</td>
</tr>
<tr>
<td>Scattering Coefficient</td>
<td>0 m$^{-1}$</td>
</tr>
<tr>
<td>Refractive Index</td>
<td>1.5</td>
</tr>
</tbody>
</table>
The material properties specified for air can be found in Table 4-4.

**Table 4-4 Properties of Air in ANSYS Fluent [103]**

<table>
<thead>
<tr>
<th>Properties of Air</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Operating Density</td>
<td>1.2041 kg/m³ [40]</td>
</tr>
<tr>
<td>Dynamic Viscosity</td>
<td>1.7894e-05 kg/m·s</td>
</tr>
<tr>
<td>Thermal expansion coefficient</td>
<td>3.43e-3 K⁻¹ [105]</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>0.0242 W/m·K</td>
</tr>
<tr>
<td>Heat capacity at constant pressure</td>
<td>1006.43 J/kg·K</td>
</tr>
<tr>
<td>Thermal diffusivity</td>
<td>1.99e-5 m²/s</td>
</tr>
<tr>
<td>Molecular weight</td>
<td>28.966 kg/kmol</td>
</tr>
<tr>
<td>Absorption Coefficient</td>
<td>0 m⁻¹</td>
</tr>
<tr>
<td>Scattering Coefficient</td>
<td>0 m⁻¹</td>
</tr>
<tr>
<td>Refractive Index</td>
<td>1</td>
</tr>
</tbody>
</table>

**4.1.2 Fluent Simulation Results**

Calibration of the ANSYS Fluent model was conducted by systematically increasing the heat flux from the lamp bulb until the predicted temperature of the model porous block matched the experimental measurements of block temperature. Figure 4-2 shows results of this calibration study which determined the appropriate approximation of the heat flux for the lamp.
As seen, the closest input flux to the target temperature of 360 K was an input flux of 140 kW/m², which yielded a block temperature of 358.6 K. This flux value was applied as a diffuse thermal heat flux to the bulb in the ANSYS Fluent simulation. The plate temperature at this input flux was 358.2 K, which is also close to the experimental value of 362 K. The lens temperature of the simulation was 663.8 K. This is higher than the value of 537 K measured in the lab, however the temperature of the outer lens does not influence the block itself because of the distance between the two entities. When translated to the power output of the bulb, an input flux of 140 kW/m² produces a gross bulb output of 670W (the area of the bulb was ~0.005m²), which is sensible given the 500W output rating of the bulb used in the experiments.

Figure 4-3 shows the contours of the block temperature from the ANSYS Fluent simulation.
Figure 4-3 Model porous block temperature gradient with input bulb flux of 140 kW/m²

Figure 4-4 shows the mid-plane temperature of the block and plate in the Fluent simulation. The temperature scale was adjusted between 294-380 K so that the block temperature is visible.

Figure 4-4 Midplane temperature of block with input flux of 140 kW/m² showing convective heat transfer occurring from the block.

Figure 4-5 shows the full midplane temperature of the whole simulation domain. As it can be seen, the bulb reaches a very high temperature (nearly 1400K), which is somewhat above range for these types of bulbs [107], however the output radiation flux from the lamp, and corresponding block temperature were the only parameters of interest from this verification study; therefore, the temperature of the bulb was not considered for verification since it could not be directly measured.
As noted in the introductory remarks of this chapter, the simulations described above were required to calibrate the computational model and to determine the radiation field at the base of the lens such that it could be used as boundary input in CFX for simulations of complex porous materials. This was done by exporting the radiation flux on the lens from the ANSYS Fluent simulation and imposing it on the upper boundary of the CFX simulation. This flux had a unique distribution owing to the internal reflections from the surrounding housing, as can be seen in Figure 4-6.

Figure 4-6 Map of radiation heat flux leaving lens bottom in Fluent simulation containing lamp. Note: values are negative because radiation flux is exiting the boundary.
There are a few ways to impose this heat flux as a boundary condition. It could be imposed as a thermal heat flux boundary condition, a directional radiation condition, or a diffuse radiation condition. When applied as a thermal boundary condition, the heat flux value is interpreted by Fluent and CFX as a combination of convection and radiation heat flux. When the heat flux value is input as a radiation boundary condition, the value is interpreted only as radiation heat flux [45],[103]. Theoretically, radiation from the top boundary is the only source of heat flux that has any effect on the block temperature as convection would cause surrounding air to rise and would not affect the temperature of the block below. Since there were uncertainties surrounding which was the most appropriate method of imposing this heat flux, a study was done to verify the most appropriate method.

4.2 Study of Applied Distributed Radiation Results

A study was done to ensure model consistency in the transition from Fluent to CFX. This involved exporting the radiation heat flux on the outer lens from the Fluent simulation containing the lamp, as seen in Figure 4-6, and imposing this flux as either a thermal heat flux boundary condition, a directional radiation condition, or a diffuse radiation condition. The effects of applying an equivalent uniform average flux across the entire upper domain were also tested. These tests were done in both Fluent and CFX so that differences in modelling software could be observed. The results from this study can be seen in Table 4-5.

<table>
<thead>
<tr>
<th>Platform</th>
<th>Boundary Condition Type</th>
<th>Flux Input Type</th>
<th>Block Temperature (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluent</td>
<td>Thermal Heat flux</td>
<td>Uniform</td>
<td>380</td>
</tr>
<tr>
<td>Fluent</td>
<td>Thermal Heat flux</td>
<td>Distributed</td>
<td>377</td>
</tr>
<tr>
<td>Fluent</td>
<td>Directional Radiation</td>
<td>Uniform</td>
<td>513</td>
</tr>
<tr>
<td>Fluent</td>
<td>Directional Radiation</td>
<td>Distributed</td>
<td>504</td>
</tr>
<tr>
<td>Fluent</td>
<td>Diffuse Radiation</td>
<td>Uniform</td>
<td>390</td>
</tr>
<tr>
<td>Fluent</td>
<td>Diffuse Radiation</td>
<td>Distributed</td>
<td>385</td>
</tr>
<tr>
<td>CFX</td>
<td>Thermal Heat flux</td>
<td>Uniform</td>
<td>355</td>
</tr>
<tr>
<td>CFX</td>
<td>Thermal Heat flux</td>
<td>Distributed</td>
<td>364.5</td>
</tr>
</tbody>
</table>
In Table 4-5, **Boundary Condition Type** describes how the heat flux was applied. **Thermal Heat Flux** defines flux that was applied as a thermal boundary condition. **Directional Radiation** defines flux that was applied as a directional radiation boundary condition in the negative y direction and a diffuse fraction of 0. **Diffuse Radiation** defines flux that was applied as a radiation boundary condition with a diffuse fraction of 1. **Flux Input Type** defines what type of flux was input into the simulation. **Distributed** defines radiation flux that was exported from the Fluent simulation containing the lamp. **Uniform** defines flux that was applied uniformly across the entire upper surface. The uniform flux applied was an average of the heat flux on the outer lens, and was ~17,700 W/m². It should be noted that it was not possible to apply distributed flux as a radiation boundary condition in CFX, which is why only uniform heat flux was tested as a radiation boundary condition. Finally, **Block Temperature** defines the upper surface temperature of the model porous block. This value was used to gauge the efficacy of the model at recreating experimental conditions, where the model block had a steady state temperature around 360K.

Through these tests it was observed an uneven flux distribution did slightly affect the temperature of the block, resulting in temperatures slightly lower when compared to an equivalent uniform flux. Additionally, it was observed that applying the flux as a thermal boundary condition resulted in lower temperatures in the model porous block. Diffuse radiation resulted in slightly higher temperatures, while directional radiation resulted in the highest block temperatures observed. These results are consistent with the definition of the thermal boundary condition, which considers both radiation and convection heat fluxes, while the radiation boundary condition considers only radiation heat flux. These results are observed in both Fluent and CFX. Interestingly, it can be observed that the Fluent simulation that contains a distributed thermal boundary condition has a higher block temperature than the original Fluent simulation that contained the lamp. This was unexpected as this simulation most closely matched the original Fluent simulation. One reason for this difference may be that the distributed flux applied was the radiation heat flux, as opposed to the total surface heat flux. The one iteration that came closest to matching temperatures observed in the lab was the CFX simulation which had a
distributed thermal heat flux boundary condition at the upper boundary. The conditions in this iteration are closest to the original Fluent simulation, which contained a bulb that emitted thermal heat flux. Despite slight differences in temperature, this study proved that the transition between Fluent and CFX provided consistent results between the two CFD codes.

Though differences were observed through this study, this thesis was focused on changing porous properties and analyzing the effects on heat capture as opposed to analyzing the effects of heat flux distribution. Additionally, heat transfer due to incoming radiation was of most interest for this thesis. With these criteria in mind, the simulation that most closely matched experimental conditions was the CFX simulation with uniform diffuse radiation as the input flux.

### 4.3 CFX Simulation

The CFX simulations focused on the absorption of radiant heat into porous blocks of different materials. The computational model was developed to mimic future experiments in the experimental setup described in Chapter 2. In such experiments, porous materials would be bonded to flattened tubes containing a heat transfer fluid, whose flow and temperature could be monitored to determine the heat captured by the porous material. The computational model for such an experiment contains the upper wall of the tube, the porous material bonded to the tube, and a sufficient air domain to capture losses from the porous material. Figure 4-7 shows an image of the computational domain used to mimic the future experiments. To take advantage of symmetry, the domain contains only half of the tube and porous material. The domain that was created has a block size 9.8 mm x 9.8 mm x 9.8 mm, and a plate thickness of 1 mm. The block shown in Figure 4-7 has a porosity of 75% and a pore diameter of 2.4 mm. The domain size changed slightly depending on the block that was imported, but proportions in the domain were all kept the same. This was done to keep the distance between the top of the air domain and the block consistent. With this new domain shape and boundary conditions, simulations were done to verify that it operated as expected.
Figure 4-7 CFX domain containing porous block, plate, and surrounding air. The size of block used for the verification simulations was 9.8 mm x 9.8 mm x 9.8 mm.

4.3.1 Pore Geometry Generation

The porous geometries used in this thesis were created using a method described in Dyck et al. [108]. This method utilizes the Discrete Elements method and YADE software to obtain a statistically accurate representative elemental volume (REV) model of a spherical void phase (SVP) material. The geometry created using this method is permeable and spatially periodic in all three principle cartesian co-ordinates. Variables such as the porosity, pore size, and standard deviation of the pore diameter can be controlled by the user. The geometries created can then be exported and manipulated in Solidworks before being transferred to ANSYS for meshing. Figure 4-8 shows an example of a negative and positive volume of the geometries created. These two volumes were then paired, with the negative volume set as a fluid and the positive volume as a solid.
4.3.2 Computational Setup and Boundary Conditions

The simulations were solved using a high-resolution advection scheme. All simulations were solved as steady state with an auto timescale control option in both the fluid and solid domain. A conservative length scale option was used with a timescale factor of 1.0. More details on these solver settings can be found in the CFX theory guide [43].

Porous Block

The outer block faces of the porous block were all set as symmetry boundary conditions. This resulted in zero heat flux crossing the boundary. The interface conditions of the block with the surrounding air and the plate underneath were set to have conservative heat transfer, with an emissivity and diffuse fraction of 1. Since emissivity was equal to absorptivity, a value of 1 was used to ensure that all incident radiation was absorbed, which allowed for a more accurate determination of how heat flux, materials, and geometry affected the absorption of the radiation, as opposed to emissivity. A diffuse fraction of 1 was used under the assumption that the porous geometries are not polished and would re-radiate in all directions.

The material properties of the porous block changed depending on the simulation, as will be outlined in the next chapter. The validation studies were all done with aluminum which had properties detailed in Table 4-6.
Table 4-6 Properties of aluminum in CFX [45]

<table>
<thead>
<tr>
<th>Properties of Aluminum</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>2702 kg/m$^3$</td>
</tr>
<tr>
<td>Molar Mass</td>
<td>26.98 kg/kmol</td>
</tr>
<tr>
<td>Specific Heat</td>
<td>903 J/kg K</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>237 W/m K</td>
</tr>
</tbody>
</table>

Air

The air top boundary was set as an opening with a surrounding temperature of 298K and a relative pressure of 0 [Pa]. The thermal radiation model was turned on in air and set to the Monte Carlo model with the number of histories set to the default of 10,000. The gray spectral model was used. A radiation source was added that was set at various fluxes depending on the test being done. For model verification, the incoming radiation was set as a diffuse source of 17,700 W/m$^2$. This value was chosen according to the results seen in Section 4.2. The two outer air walls in the x-y plane, and front wall in the y-z plane were set as symmetry. The back outer air wall in the y-z plane was set as an opening. The bottom air boundary in the x-z plane was also set as an opening. The full buoyancy model was used, setting air as an ideal gas with the default properties given in Table 4-7.

Table 4-7 Properties of Air in CFX [45]

<table>
<thead>
<tr>
<th>Properties of Air</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference Density</td>
<td>1.1839 kg/m$^3$ [40]</td>
</tr>
<tr>
<td>Molar Mass</td>
<td>28.96 kg/kmol</td>
</tr>
<tr>
<td>Dynamic Viscosity</td>
<td>1.831e-05 kg/m-s</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>0.0261 W/m-K</td>
</tr>
<tr>
<td>Specific heat capacity at constant pressure</td>
<td>1004.4 J/kg K</td>
</tr>
<tr>
<td>Refractive Index</td>
<td>1</td>
</tr>
<tr>
<td>Absorption Coefficient</td>
<td>0 /m</td>
</tr>
<tr>
<td>Scattering Coefficient</td>
<td>0 /m</td>
</tr>
</tbody>
</table>
Plate

The plate absorbs the heat captured from the porous block and transfers it to a working fluid. The plate modelled in the simulation represents the top of a square channel containing flowing water. To simplify the model, this water channel was modelled indirectly by placing a convection coefficient on the bottom of the plate. A diagram of this water channel can be seen in Figure 4-9, with the dotted line representing the portion of the channel that was directly modelled in the simulation. The width of the channel was determined by doubling the dimensions of the porous block.

![Diagram of water channel](image)

Figure 4-9 Cross sectional view of the water channel beneath plate that could be used in experimental validation studies. Dotted line represents portion of channel wall that was modelled in the CFX.

The dimensions were used to calculate the convection coefficient using the Nusselt number correlation for fully developed laminar flow in tubes with non-circular cross sections [40]. From this process, it was found that a value of 390 W/m²K was an appropriate convective coefficient, with an external convective temperature of 283 K. Details of this calculation can be found in Appendix A. The inner three faces of the plate in the x-y plane and the front face in the y-z plane had symmetry boundary conditions. The plate wall at the back in the y-z plane was set as adiabatic.

The plate underneath the porous block was assigned as copper which had properties detailed in Table 4-8.

Table 4-8 Properties of copper in CFX [45]

<table>
<thead>
<tr>
<th>Properties of Copper</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>8933 kg/m³</td>
</tr>
<tr>
<td>Molar Mass</td>
<td>63.55 kg/kmol</td>
</tr>
<tr>
<td>Specific Heat</td>
<td>385 J/kg K</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>401 W/m K</td>
</tr>
</tbody>
</table>
It was necessary to ensure that the domain that was being modelled was large enough such that the temperature of the porous foam was not being influenced by boundary conditions, particularly boundary conditions that were open to surrounding air. It was also necessary to ensure that the domain was not too large such that the surrounding air was being discretized and using up unnecessary computation time. To achieve this, a far field boundary study was conducted. Two boundaries were gradually moved closer to the porous foam: the upper air boundary with the diffuse incoming radiation flux, and the side boundary which was open to surroundings. All other aspects of the simulation remained the same. The heat flux balance across the plate was measured to see if the boundary distance affected the results. This was done by measuring the heat flux exiting the plate divided by the heat flux entering the plate. The actual heat flux values were also compared with the changing boundaries to track changes in the values of the heat flux balance. The results of the heat flux balance measurements can be seen in Figure 4-10.

As it can be seen, the boundary distance had almost no effect on the heat flux balance in the plate. The heat flux entering and exiting were almost always within 0.5% of one another, which was deemed an appropriate difference. The actual values of the heat flux balance were also unaffected by the changing boundary distances. It was observed that having the air boundaries too close to

![Figure 4-10 Percentage of the heat balance (%) on the plate in the simulation with the changing height of the upper air boundary. It was used for the far field boundary study.](image)
the porous block did not allow for enough space for the surrounding air flow to fully develop before interacting with the block, as can be seen in Figure 4-11.

Figure 4-11 Velocity vector field of air around porous block. Left image: smallest domain modelled – air flow not fully developed before entering the block. Right Image: Final domain size with fully developed air domain.

To allow for the surrounding airflow to fully develop, a slightly larger far field air domain size was used which was approximately 2 block lengths in the x-direction, and 3 block lengths in the y-direction. For the 9.8 mm block, this resulted in a total domain width of 9.8mm, a height of 30 mm in the y-direction, and a depth of 20mm in the x-direction. This size allowed for enough
space for the surrounding airfield to fully develop before entering the porous block, while also balancing the domain mesh size.

To determine the appropriate residual levels, a residual target test was done that involved setting the residuals to $10^{-4}$ and measuring the heat flux from the bottom of the plate. The residuals were then set to $10^{-5}$ and run again. It was found there was a large difference in heat flux from the bottom of the plate between $10^{-4}$ and $10^{-5}$ residual targets. The same test was run again with the residuals set to $10^{-6}$. It was found that the heat flux measured from the bottom of the plate between $10^{-5}$ and $10^{-6}$ was within 1% of one another and a residual target of $10^{-5}$ was deemed appropriate. The same test was run for finding the number of iterations needed until convergence. The simulation was run for 200 iterations. The block temperature and flux from the bottom of the plate were recorded. This process was repeated until the values were no longer changing, which occurred around 1000 iterations. This resulted in a running time of approximately 12 hours on a dual core Intel i7 3.60 GHz desktop computer with 16 GB of RAM.

A grid convergence study was done for this CFX simulation. The parameter of interest that was compared between grid sizes was the heat flux from the bottom of the plate. From this process, it was found that a grid size of approximately 2,200,000 cells was appropriate with a GCI of 1.4%.

4.3.3 Results from Verification Study on Porous Block

Results were measured on the block by placing monitors on 7 different locations on the porous block on the inner symmetric face. A diagram of these locations on the porous block can be seen in Figure 4-12, indicated by the yellow lines.
At each of the 7 locations, 20 temperature samples were taken across the width of the block. As it can be seen, there are some locations across the block width where there is void space, these values were null and were not incorporated into the block temperature. The temperature values were then averaged to give the temperature of the porous block at each height. The temperature of the block with an input diffuse flux of 17,700 W/m\(^2\) can be seen in Figure 4-13.
As it can be seen, the block temperature was warmer near the top, and it linearly decreased closer to the plate. The temperature gradient was small, approximately 3 K. This is because of the thermal conductivity of aluminum. The radiation flux from the air to the block was 792.05 W/m². The convective loss from the block was 17.7 W/m². The heat flux from the bottom of the plate was 12,528 W/m². These values show that porous blocks can effectively absorb high amounts of radiation, with minimal convective losses, and effectively translate that to output thermal flux.

Figure 4-14 shows the contour temperature profile of the solid and fluid domain.

Figure 4-14 Temperature contour of block and surrounding air with diffuse flux input of 17,700 W/m². Left: air temperature. Right: solid block and plate temperature

Figure 4-15 shows the vector plot of the air domain surrounding the block.
This temperature profile is expected, as it shows the receiver which is warmer near the top of the receiver, and a gradual decrease of temperature deeper into the receiver, which is in line with the literature reviewed. As it can be seen, air flow is present throughout the block due to buoyancy forces caused by the temperature increase of the block.

With the boundary conditions and solver settings appropriately set and verified, a parametric study was performed to determine how changes to the porous block affected heat capture. Details of these parametric studies will be detailed in the following chapter.
Chapter 5

5  Parametric Study of Incident Radiation on Permeable Porous Materials

A parametric study was conducted to initiate the framework for the optimization of a porous structure for the efficient capturing of concentrated solar thermal radiation. The first part of the study investigated the effects of changing the input radiation flux, for three different input fluxes. The second part investigated the effects of changing the porous material, for three different common receiver materials. The final part of the study investigated geometric changes to the porous block. Three different porosities, one different pore diameter, and an increasing and decreasing porosity gradient were considered.

5.1  Effects of Changing Input Heat Flux

A study was done to analyze the effects of changing the input radiation flux on the temperature of the block and the heat flux from the bottom of the plate. Directional radiation was used for these studies. The reason for changing from diffuse to directional radiation is that direct radiation is a better approximation of what would be present in a CSP system. The mirrors in CSP focus diffuse radiation from the sun into directional radiation that is much more concentrated.

Three different directional input fluxes were chosen: 17,700 W/m², 60,000 W/m², and 100,000 W/m². The first input flux was chosen as it was the output flux from the Fluent lamp simulation. 60,000 W/m² was found to be the output flux from a concentrating dish in one study [95], however higher fluxes are possible so 100,000 W/m² was tested as well. These fluxes were added as a directional radiation source in the negative y-direction in the CFX simulation. The same porous block that was used in the diffuse input flux study was also used in these studies, with a porosity of 0.75 and a pore diameter of 2.4mm. More details about this block can be found in Table 5-4. The temperatures of the block at each input flux are given in Figure 5-1.
Directional radiation produced higher temperatures in the block. The block was approximately 12 K warmer when a directional flux of 17,700 W/m$^2$ was applied as compared to diffusely applied radiation. This is due to the reason that more radiation is striking the block directly in the directional radiation case, compared to the diffuse radiation.

As the incoming heat flux increased, the temperature of the block increased as well. Increasing the flux by approximately 40,000 W/m$^2$ increased the temperature at the top of the block by approximately 100 K. This increase in temperature with an increasing input flux was expected and is also observed in other studies [78]. As the input flux increased, the difference between the top block temperature and the bottom block temperature increased as well. With an input flux of 17,700 W/m$^2$ the difference between the temperature at the top of the block and the bottom of the block was only 5 K. With an input flux of 60,000 W/m$^2$ this difference increased to 16 K, and at 100,000 W/m$^2$ the difference increased to 25 K. This increase in temperature gradient causes more heat transfer through the block, but the higher temperatures also contribute to higher convective losses. A breakdown of different heat fluxes is given in Figure 5-2 to Figure 5-4.
Figure 5-2 Radiative flux absorbed by the block from differing input fluxes. As input flux increased, absorbed radiation flux increased.

Figure 5-3 Convective losses from the porous block with differing input fluxes. As input flux increased, convective losses increased.

Figure 5-4 Heat flux at the bottom of the plate with differing input fluxes. As input flux increased, heat flux from plate bottom increased.
As it can be seen, a higher input flux resulted in an increase in the absorbed radiation flux. Increasing the input flux by a factor of 40,000 W/m\(^2\) increased the absorbed radiation flux by about 2000 W/m\(^2\). A higher incident flux also resulted in higher convective losses from the block. Due to the higher temperatures of the block, more heat was being removed from the block by the air, contributing to a larger gradient within the solid itself. As expected, the heat flux from the bottom of the plate increased with a higher input flux. Increasing the flux from 17,700 W/m\(^2\) to 60,000 W/m\(^2\) resulted in a flux increase of \(~38,000\) W/m\(^2\). Increasing the flux from 60,000 W/m\(^2\) to 100,000 W/m\(^2\) resulted in a flux increase of \(~33,000\) W/m\(^2\). This difference could be attributed to the slightly higher incident flux increase that occurred between 17,700 W/m\(^2\) and 60,000 W/m\(^2\).

Due to the higher fluxes and favourable temperature gradient observed in the 100,000 W/m\(^2\) case, and because of its relevance to an incident solar flux, this input flux value was carried forward into the subsequent parametric analyses.

### 5.2 Effects of Changing Porous Material

This parametric analysis considered the effects of changing the material properties of the block on the block temperature and heat flux of the system. Three different materials were chosen: aluminum, graphite, and silicon carbide (SiC). These materials were chosen because of their diverse thermal properties, specifically thermal conductivity. Additionally, these three materials have been used in previous solar thermal receiver studies [81][88][109]. The properties of each material are outlined below.

The properties of aluminum in CFX can be found in Table 5-1.

<table>
<thead>
<tr>
<th>Table 5-1 Material properties of aluminum in CFX [45]</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Density</strong></td>
</tr>
<tr>
<td><strong>Thermal Conductivity</strong></td>
</tr>
<tr>
<td><strong>Specific heat capacity</strong></td>
</tr>
<tr>
<td><strong>Molecular weight</strong></td>
</tr>
<tr>
<td><strong>Max Operating Temperature</strong></td>
</tr>
</tbody>
</table>
The properties of silicon carbide can be found in Table 5-2.

Table 5-2 Material properties of silicone carbide [110]

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>4840 kg/m³</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>23 [111] W/m K</td>
</tr>
<tr>
<td>Specific heat capacity</td>
<td>650 J/kg K</td>
</tr>
<tr>
<td>Molecular weight</td>
<td>40.096 g/mol</td>
</tr>
</tbody>
</table>

The properties of graphite can be found in Table 5-3.

Table 5-3 Material properties of graphite [40]

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>2210 kg/m³</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>1950 W/m K</td>
</tr>
<tr>
<td>Specific heat capacity</td>
<td>709 J/kg K</td>
</tr>
<tr>
<td>Molecular weight</td>
<td>12.01 g/mol</td>
</tr>
<tr>
<td>Max Operating Temperature</td>
<td>2273 K</td>
</tr>
</tbody>
</table>

It should be noted that the maximum operating temperatures listed for each material are the absolute maximums. It is necessary to work significantly below these temperatures to stay within the operational range of the material where the physical properties are as listed.

There are relative advantages to each material. Aluminum has a high thermal conductivity, but a relatively low melting point, meaning it cannot withstand the higher temperatures expected in CSP receivers. Silicon carbide has a very low thermal conductivity, but a very high operating temperature, making it advantageous for the high temperatures of solar thermal receivers. Graphite has the advantage of having a high thermal receiver, and a high operating temperature, potentially making it ideal as a CSP receiver.

The porous block in the simulation was set to one of these three materials and run under a direct radiation heat flux of 100,000 W/m². A porosity of 0.75 and a pore diameter of 2.4mm were considered. The temperature results are shown in Figure 5-5 and Figure 5-6.
Figure 5-5 Block temperature (75% block porosity) for different materials. Temperature gradient was smaller for higher thermal conductivity materials.

Figure 5-6 Solid temperature contour of 75% porosity block with differing materials.

Material of block from left to right: Aluminum, Graphite, SiC

It can be observed in Figure 5-5 that the material of the block influences the operating temperature. The SiC block had a higher surface temperature compared to the aluminum and graphite foam blocks; it also had a lower temperature at the bottom of the block meaning that the heat was not being conducted through the SiC block as effectively as the aluminum and graphite blocks. This was expected because of the lower thermal conductivity of SiC. Additionally, the higher temperatures observed in the SiC sample resulted in more heat lost to convection, which contributed to more heat loss through the block and a larger temperature gradient.

It was observed that the graphite foam has a slightly higher temperature at the bottom of the block compared to aluminum (~2 K). This temperature difference can also be observed at the top...
of the block where the aluminum block was ~17 K warmer than the graphite foam block. This shows that despite having a lower top surface temperature, the graphite foam block had a higher bottom surface temperature when compared to aluminum. Again, this can be attributed to the higher thermal conductivity of graphite, which allowed for heat to be conducted through the block much more effectively.

SiC has a thermal conductivity that was approximately 10 times lower than that of aluminum. Graphite has a thermal conductivity that was a factor of 10 larger than aluminum. However, there was not a major difference in the temperatures observed between aluminum and graphite, but there was a major difference in temperatures observed for aluminum and SiC. This finding suggests that thermal conductivity influences the temperature of the block up to a certain value, after which the thermal conductivity does not majorly contribute to increased heat capture performance. This is consistent with the observation in other studies [79].

The different heat fluxes within the system are given below in Figure 5-7 to Figure 5-9.

![Figure 5-7 Absorbed radiation flux in porous blocks for different block materials. Absorbed radiation flux increased with thermal conductivity.](image)
Figure 5-8 Convective losses from porous blocks for different block materials. Convective losses decreased as thermal conductivity increased.

Figure 5-9 Heat flux at the bottom of the plate for different porous block materials. Heat flux from the plate bottom increased with thermal conductivity.

The material of the porous block did affect the heat flux observed in the system. Graphite had the highest absorbed radiation, followed closely by aluminum, and then SiC. In terms of convective losses, the graphite and aluminum porous blocks had very similar values, while SiC had higher losses. Graphite had the highest heat flux from the bottom of the plate, followed closely by aluminum, and then SiC. These differences can again be attributed to the differences in thermal conductivity between the three samples. The higher thermal conductivity of aluminum and graphite allowed for more radiation to be absorbed and effectively conducted through the porous block. This led to lower convective losses because the block did not have a high top surface.
temperature. SiC had a higher top surface temperature due to its lower thermal conductivity, which resulted in higher convective losses. The higher heat flux from the bottom of the plate for the aluminum and graphite is due to the blocks moving input radiation flux into output thermal heat more efficiently because of the higher thermal conductivity.

It is again interesting to observe that the difference between the aluminum block and the SiC block were much greater than the aluminum and graphite blocks. This further reinforces that thermal conductivity has a major effect on the thermal performance of the block only up to a certain point, beyond which, the performance does not majorly improve.

From this study, it was observed that graphite was the best performing material, owing to its favourable temperature gradient and output heat flux. Further, graphite has a much higher operating temperature that would allow it to perform at the higher temperatures expected in CSP. For this reason, graphite was used as the testing material in the subsequent parametric analysis, which is detailed in the following section.

### 5.3 Effects of Changing Geometric Parameters of Porous Block

This parametric analysis focuses on the effects of changing the geometric parameters of the porous block. Note that this parametric analysis lays the framework for more thorough investigations to characterize the influence of porous geometry (i.e., the pore size and porosity) on the overall heat transfer. In the current study, three different porosities were first investigated: 0.7, 0.8, and 0.9, with all blocks having a pore diameter of 2.4mm. A pore diameter of 1.8mm was also studied with a porosity of 0.75. Finally, a porosity gradient varying from 0.92-0.71 was investigated with a pore diameter of 2.4mm. These porosities and pore sizes were selected because they are representative of the geometries tested in previous studies [75], [83], [89], [91].

As mentioned earlier, YADE was used to generate blocks. When these blocks were generated, they had different dimensions for different porosities and pore sizes. To accommodate this, the dimensions of the overall domain were also adjusted such that the distance between the top of the air domain and the top of the block were kept consistent, which was ~2 cube lengths above the
cube. The width of the domain did not affect the results as the outer faces of the air and cube had symmetry boundary conditions.

The properties of each block are listed in Table 5-4

Table 5-4 Geometric properties of different porous blocks tested

<table>
<thead>
<tr>
<th>Geometry</th>
<th>Pore Size (mm)</th>
<th>Porosity</th>
<th>Side Length (m)</th>
<th>Area to Volume Ratio (m²/m³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2.4</td>
<td>0.7</td>
<td>1.0E-02</td>
<td>1549</td>
</tr>
<tr>
<td>2</td>
<td>2.4</td>
<td>0.8</td>
<td>9.5E-03</td>
<td>1467</td>
</tr>
<tr>
<td>3</td>
<td>2.4</td>
<td>0.9</td>
<td>8.9E-03</td>
<td>1090</td>
</tr>
<tr>
<td>4</td>
<td>2.4</td>
<td>0.75</td>
<td>9.8E-03</td>
<td>1597</td>
</tr>
<tr>
<td>5</td>
<td>1.8</td>
<td>0.75</td>
<td>10.0E-03</td>
<td>2128</td>
</tr>
<tr>
<td>6</td>
<td>2.4</td>
<td>0.92-0.71</td>
<td>9.8E-03</td>
<td>2481</td>
</tr>
<tr>
<td>7</td>
<td>2.4</td>
<td>0.71-0.92</td>
<td>9.8E-03</td>
<td>2481</td>
</tr>
</tbody>
</table>

The results from these analyses are presented below, starting with the absorbed radiation flux for different block porosities and then different pore diameter (geometries 1-5 as listed in Table 5-4). The pore diameter is indicated in the top left-hand corner of each plot.
Absorbed radiation increased with porosity and decreased with the pore size.

As it can be seen in Figure 5-10, the absorbed radiation flux was higher for a higher porosity. The absorbed radiation takes a small dip at 0.75 porosity; however, this decrease is very minor (less than 1%). At porosity of 0.90, the absorbed flux was much higher. It is approximately 2500 W/m² higher than the absorbed flux at 0.80 porosity while porosity 0.80 is only ~500 W/m² higher than 0.70 porosity. This trend is to be expected because at the higher porosity, there is more opportunity for radiation to penetrate deeper into the block and be absorbed at multiple internal faces instead of at the upper surface, which is the case for lower porosity cubes. When comparing different pore sizes, the 1.8 mm pore diameter cube absorbed heat flux is 25% (~1500 W/m²) lower than the 2.4 mm pore size.

The convective losses from the porous cubes are shown in Figure 5-11 for geometries 1-5.
Convective losses increased with porosity and decreased with the pore size.

The convective losses from the block are due to buoyancy induced air flow moving through the block. This represents heat that is lost to the surrounding air. As the porosity of the block increased, so did the convective losses. The convective losses were much higher for 0.9 porosity cube due to its higher permeability. However, convective losses are much smaller than the absorbed radiation flux, which shows that the porous cube is an effective way to capture incident solar radiation. Considering the different pore diameters at porosity 0.75, the smaller pore diameter had lower convective losses, which is due to the drop in permeability with decreasing pore size.

The heat flux from the bottom of the plate for geometries 1-5 is shown in Figure 5-12. This value represents the useful heat output from the system.
Figure 5-12 Heat flux from bottom of plate for different block porosities and pore diameters. Heat flux from plate bottom decreased as porosity increased and decreased slightly with the pore size.

Note that because the area of the bottom of the plate is much smaller than the porous block, the flux value (W/m²) appears higher than that shown in Figure 5-10. The heat flux from the bottom of the plate decreases as porosity increases. This can be explained by the fact that as the porosity increases, the contact area between the block and the plate decreases. This reduces the area over which conduction of heat can occur from the porous block to the heat spreading plate below.

Looking at the different pore diameter, the heat flux was slightly lower with the smaller pore size; however, this difference was less than 100 W/m² (approx. 0.08%) smaller. This small difference could be accounted for by the lower input flux radiation that was absorbed by the block with the lower pore diameter.
The contours of block temperature at different porosities are given in Figure 5-13.

These temperature contours show that higher temperatures were seen at the top of the block for a higher porosity. In the case of the 0.90 porosity block, higher temperatures were seen (~10 K higher than 0.80). It can also be seen that the higher porosity blocks had lower bottom block temperatures, which was again more pronounced in the 0.90 porosity block (~5K lower than the 80% block). A larger temperature gradient can also be seen as the porosity increased. Where the 0.70 block has a temperature difference of ~4 K between the top and bottom of the block, the 0.80 block a 6 K difference, the 0.90 block has a 20 K difference. This gradient can be explained by the lower strut density for higher porosity. This inhibited heat from being efficiently transferred from the top surface to the bottom surface and contributed to the larger temperature gradient. Within each horizontal layer of the 0.90 porosity block, the temperature is much less uniform compared to the other porosities. This can be seen in in the hot spot that formed at the top of the 0.90 porosity block. Again, this is likely due to the lower strut density which does not allow for heat to be effectively conducted within each layer of the block. The higher convective losses that were seen in the 0.90 porous block also may have contributed to the relatively large temperature gradient, as this resulted in more heat being removed from the block, meaning it was not conducted through the rest of the block. This effect is emphasized at higher temperatures that are seen at the top of the 0.90 porous cube. Based on these temperature contour profiles, the porous cube with the lowest top surface temperature and highest bottom surface temperature had porosity 0.70. The porous cube with the highest top surface temperature and lowest bottom surface temperature had porosity 0.90.
The temperature contours of the block with porosity 0.75 and different pore diameters are shown in Figure 5-14.

As it can be seen, there are many more pores in the 1.8 mm pore diameter sample. This resulted in a much larger cell count when meshing the domain (~6,000,000 cells). This also resulted in a much longer computational running time of approximately 36 hours. As it can be seen, the temperature between the two blocks is relatively similar, with a very similar top and bottom surface temperature, and overall temperature gradient. There is only a 0.6 mm difference between these two pore diameters, which may account for the relatively similar temperature contours.

The velocity vector profiles for geometries 1-3 are shown in Figure 5-15. These profiles were taken from one of the outer symmetry faces of the domain.
As the porosity increased, the void space within the block increased, which resulted in much more air flow into the block for a higher porosity. This allowed for more air to enter to the block, and explains the higher convective losses seen in the higher porosity blocks. The velocity of the air inside of the block and exiting the domain is slightly higher as the porosity increased. This is likely because the temperature of the air increased due to the block temperature being higher as well. This contributed to greater buoyancy driven flow, and higher air velocities.

Figure 5-16 shows velocity vector plots for different pore diameter blocks.
The vector plots show that there is greater air flow within the smaller diameter pore size. This is because there are many more pores for the 1.8mm pore size. Looking at the magnitude of velocity of the air within the 1.8 mm pore diameter block, the air has a velocity close to zero, so it is unlikely this air flow is having a huge effect on the block temperature. This was confirmed by the lower convective losses that were seen in the lower pore diameter block.

Results from the study analyzing different pore diameters suggest that based on the two pore sizes tested, a lower pore size does not absorb as much radiation, but it does have lower convective losses. Comparing the heat transfer from the bottom of the plate, the pore size did not seem to have a significant effect. The pore size did not seem to have a huge effect on the temperature gradient within the block or the overall temperatures within the block. Comparing these observed results to the literature, other studies [75] [77] concluded that the maximum temperature of the solid increased with pore diameter and that the non-equilibrium region decreased as the pore diameter increased. Although these results were not observed in this study, it is likely because the pore diameters tested were not very different, and multiple pore diameters were not tested. Because only two pore sizes were tested, and there was not a significant
variation in pore size, more pore sizes need to be tested to see if the results observed in this study hold true.

In terms of porosity, results suggest that porosity had a larger effect on the temperature gradient seen within the block. This is in line with the literature reviewed [75]. A high porosity block had a higher top surface temperature, and a lower temperature at the bottom of the block. A lower porosity block had a lower top surface temperature and a higher bottom surface temperature. The increase in maximum temperatures and the increase in thickness of the thermal non-equilibrium region at higher porosities was also observed in other studies [77]. Looking at the heat fluxes, a higher porosity model can absorb more incoming radiation but is less efficient at transferring that flux through the block to a heat spreader below. A lower porosity model is unable to absorb as much radiation but can transfer that heat through the block to a heat spreader more effectively. This observation was also observed in another study reviewed, which suggested that lower porosity foams were more efficient at converting incoming heat to useful outgoing heat [80]. As stated in Chapter 1, the radiative conductance model suggests that a smaller porosity has a larger solid emission which results in a larger overall radiative conductivity [83]. This is in line with the results observed in this study, which found a better performance in terms of output flux from the lower porosity cube.

Since the high porosity block had a higher absorbed radiation flux, and the lower porosity block had a higher output flux, this suggests that an ideal porous block would have a high porosity at the top of the block that gradually decreases. This would allow more incoming radiation to enter the block, be effectively absorbed, and then transferred to a heat spreader.

To test this, a gradient porosity block was tested. A block was tested that had a decreasing porosity gradient of 92% at the top of the block to 71% at the bottom of the block, gradually decreasing as the radiation moved deeper into the block. This gradient was also reversed, with an increasing gradient of 71% at the top of the block to 92% at the bottom of the block. The pore size of both blocks was 2.4mm. The heat flux from this analysis is presented in Figure 5-17 to Figure 5-19 and discussed below.
Figure 5-17 Absorbed radiation for different porosity gradient directions. The block with decreasing porosity gradient absorbed more input radiation.

Figure 5-18 Convective losses from porous block for different porosity gradient directions. The block with decreasing porosity gradient had slightly higher convective losses.

Figure 5-19 Total heat flux from bottom of plate for different porosity gradient directions. The block with decreasing porosity gradient had higher output flux from the plate bottom.
The results show that the decreasing porosity gradient absorbed more radiation. This is because the higher porosity is exposed to the incoming radiation, and the radiation can penetrate deeper into the block. The decreasing porosity block has only moderately higher convective losses. This is likely because air can move in and up through the block as it is rising because there is more void space at the top of the block in the decreasing block. The decreasing porosity block also has slightly higher flux from the bottom of the plate. This is because there is more solid to solid contact between the block and the plate which leads to higher conduction and more flux from the plate bottom. These trends suggest that the decreasing porosity block is more effective at absorbing incoming radiation and conducting it through the block to a heat spreader, with only moderately higher convective losses than the increasing porosity.

Comparing the values of heat flux from these gradient blocks to the uniform porosity blocks, both increasing and decreasing gradient blocks had a higher absorbed radiation than the 0.70, 0.75, and 0.80 uniform porosity blocks; however, the 0.90 porosity block absorbed over 2000 W/m² more radiation flux than both gradient blocks. Both gradient blocks had lower convective losses than the uniform 0.90 porosity block, but they had higher convective losses than the 0.70, 0.75, and 0.80 porosity blocks. Both blocks had the lowest heat transfer from the bottom of the plate, with the decreasing block being ~1000 W/m² lower than the 0.9 porosity uniform block, which was the worst performing from the uniform porosity blocks. The increasing porosity block had a heat flux that was ~1500 W/m² less than the 0.90 uniform porosity block. These results are surprising because it was expected that the decreasing porous gradient block would produce better results than the uniform porosity blocks because of its ability to absorb large amounts of radiation and effectively transmit it to an attached heat spreader.
The temperature variations in the block with porosity gradient is shown in Figure 5-20.

![Temperature contour of solid for different porosity gradients.](image)

Figure 5-20 Temperature contour of solid for different porosity gradients.

Left: decreasing porosity gradient (92% at block top to 71% at block bottom),
Right: increasing porosity gradient (71% at block top to 92% at block bottom)

The decreasing porosity gradient once again has a small area of the top surface which is warmer than the surrounding layer. This is likely because of the low strut density at the top of the block, which does not allow for efficient heat conduction to the surrounding layer. The temperature gradually decreases from the top to the bottom layers in both gradients with a similar temperature at the bottom of the blocks. The increasing gradient block has a much larger area that is has a high uniform temperature at the top of the block, while the decreasing gradient has a much larger area that is a uniform temperature at the bottom of the block. This again is due to the high strut density at the top of the increasing gradient block which allows for efficient conductivity in the top layers. As the strut density decreases near the bottom of the block, this reduces the block’s ability to conduct heat. The decreasing gradient block has a large temperature gradient at the top of the block that becomes more uniform near the bottom of the block for similar reasons.
Figure 5-21 shows the vector plot for the two porosity gradient blocks.

![Vector plots of air domain for different porosity gradients.](image)

Left: decreasing gradient (92-71%), Right: increasing gradient (71-92%)

Figure 5-21 shows that there is more air flow near the top of the block in the decreasing gradient porosity because there is more void space in this area. The velocity in this area is slightly higher because of the higher temperatures near the top of the block. There is more air flow near the bottom of the block in the increasing porosity gradient block because of the larger void spaces that exist there; however, the velocity is slightly lower compared to the same void space at the top of the decreasing block. The magnitude of velocity in both cases is very similar, however the increasing block seems to have more stagnant air within the block itself. These velocity profiles explain why higher convective losses were seen in the decreasing porosity case compared to the increasing porosity case.
Based on the heat flux and temperature results, the best performing block overall appeared to be the 0.70 uniform porosity with pore diameter 2.4mm. Despite absorbing less radiation, it had some of the lowest convective losses and the higher heat flux from the bottom of the plate. Additionally, it had the lowest top surface temperature and highest temperature at the bottom of the block. Higher porosity blocks had larger temperature gradients and did not transfer heat as effectively to the plate below, including the porosity gradient geometries. This is not in line with what was found in other studies, which suggested that a decreasing porosity gradient was able to decrease the top surface temperature of the absorber and translate that heat to a higher outlet temperature more effectively than uniform porosity blocks [89]. These studies looked at more extreme gradients, for example 0.95-0.5 [90], which may have led to more favourable results. Additionally, only one pore size was tested for the porosity gradient. More testing needs to be done with different pore sizes to see how these variable effects the observed results.

What this study has concluded is that the model developed through this thesis is able to produce results that have been found in other studies and that this model could be used in future studies to analyze more variations in porosity, pore size, and porosity gradients.
Chapter 6

6 Summary

The objective of this study was to develop a computational model that can be used to optimize the geometric properties of a porous geometry for capture and transfer of incident solar radiation. For this purpose, experiments were first conducted using a simplified “porous” geometry to attain temperature data. A geometrically identical computational model was created and iterated upon until temperature results matched the experimental results. This computational model was created using ANSYS Fluent because of its superior radiation modelling techniques, specifically its ability to model semi-transparent media needed for accurate modelling of the halogen lamp. A transition was made between ANSYS Fluent and CFX because of the ability CFX had to handle more complex porous geometries. A study was conducted to investigate how results differed between Fluent and CFX. It was found that despite slight differences in temperature results, CFX accurately predicted the temperatures recorded from the experimental study.

A computational model in CFX was developed with more geometrically complex porous blocks. This model had a plate attached to the porous geometry. The plate was used to simulate heat removal at the bottom of a receiver, where heat would be removed via a working fluid. Heat flux was measured from the bottom of this plate as the useful thermal output from the porous receiver. This model was first verified using diffuse input radiation flux.

A parametric study was conducted using the computationally verified CFX model. First, three different input radiant fluxes were studied. It was found that increasing the input flux increased the block temperature, increased the temperature gradient, and increased the output flux from the bottom of the plate. From this study, it was concluded that the input flux of 100,000 W/m² gave a more realistic temperatures that would be seen in a CSP system, and all further studies were conducted with this input radiant flux. Next, three different materials were tested including aluminum, SiC, and graphite. It was found that graphite performed the best, with the lowest top surface temperature, the highest bottom surface temperature, and the highest heat flux from the bottom of the plate. All further tests were conducted using graphite as the porous block material.
Simulations were then conducted by changing the geometric properties of the porous blocks. Three different porosities were tested: 0.7, 0.8, and 0.9 all with a pore diameter of 2.4 mm. It was found that changing the porosity increased the amount of absorbed radiation; however, it also increased the convection losses, and the temperature at the top of the block. The increase in porosity also led to a decrease in the flux from the bottom of the plate. Two different pore diameters were tested: 2.4 mm and 1.8 mm with a porosity of 0.75. It was found that decreasing the pore diameter decreased the amount of absorbed radiation and decreased the convective losses from the block. The lower pore diameter also led to a very slight decrease in heat flux from the bottom of the plate. Pore diameter did not seem to have a significant effect on the temperature of the block, or the temperature gradient.

Finally, two porosity gradients were studied. A decreasing gradient was studied which varied from 0.92 at the top to 0.71 at the base. An increasing gradient was also studied which varied from 0.71 at the top to 0.92 at the base. Both gradients had a pore diameter of 2.4 mm. It was observed that the decreasing gradient (0.92-0.71) had higher absorbed radiation, slightly higher convective losses, and higher output flux from the bottom of the plate when compared to the increasing gradient block. Compared to the uniform porosity blocks with the same pore size, the gradient blocks absorbed more radiation than the lower porosity uniform blocks; however, the gradient blocks had lower heat flux from the bottom of the plate compared to the uniform porosity blocks. This trend was not expected; however, since only one pore size (2.4mm) was studied for these gradient blocks, a definitive conclusion cannot be drawn. Overall, based on the parameters considered in this research work, the block with uniform porosity of 0.70 and a pore diameter of 2.4 mm performed the best. This block had the lowest top surface temperatures and the highest bottom surface temperatures of all the blocks tested. It did have the lowest absorbed radiation flux but had the highest output flux from the bottom of the plate. Since relatively few blocks were studied with minimal variation in porosity and pore diameter, more investigations need to be conducted over a wider range of pore sizes and porosity gradients to find the optimal porous geometry for the efficient heat capture from concentrated sunlight. The computational model developed through this thesis provides the computational framework for such investigations.
6.1 Recommendations for the future work

There are many considerations to develop an optimized porous geometry.

- First, a thorough experimental validation of computational model developed in CFX. This experimental model could involve a porous sample heated from above that is bonded to a heat transfer surface (i.e., a water channel) to steadily remove heat from the porous block. Changes in the system could be measured by measuring changes to the output temperature of the water channel.

- Next, higher input fluxes could be computationally modelled that are in line with some of the expected input fluxes in a CSP plant (which can be as high as 3 MW/m² [77]).

- A wider range of pore sizes and different pore sizes with different porosities could be studied to conclude how pore size effects porous geometry performance.

- A wider range of porosities must also be tested, likely lower than 0.70 as higher than 0.90 is not realistic for manufacturing purposes.

- In terms of porosity gradients, larger ranges of porous gradients (i.e., 0.5-0.9 or 0.3-0.9) could be studied, as well as more pore sizes with porosity gradients. A gradient pore size with consistent porosity, and a gradient pore size with gradient porosity could all be studied to find if there is a configuration that leads to the optimal heat capture and transfer.

Overall, this thesis has provided the framework through which a computational model could be used in further studies to optimize the geometric properties of a porous receiver for application in concentrated solar power.
References


Appendices
Appendix A

Calculation of Convective Heat Transfer Coefficient at Bottom of Plate

First, the hydraulic diameter was found.

\[ D_h = \frac{4A_c}{P} = 7.97e - 3 \] \hspace{1cm} (A.1)

\[ \frac{b}{a} = 3.92 \approx 4 \]

From the correlation given by Bergman et al. [40], the Nusselt value was found to be 5.33 for a uniform input flux.

\[ Nu_D = \frac{hD_h}{k} \] \hspace{1cm} (A.2)

The thermal conductivity of water at 10ºC was found to be 0.57864 W/m K [40]. From this, the value of \( h \) was found to be 387.08 W/m²K which was approximated as 390 W/m²K.
## Curriculum Vitae

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