Numerical Analysis of Novel Solar Central Receiver Using Different Types of Heat Transfer Fluids

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Session 2018-20

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Dedication

I would like to wholeheartedly dedicate my thesis to my beloved parents, who have been a constant source of inspiration and strength with continuous moral, spiritual and emotional support.

To my sister, mentor and friends who shared their words of advice and encouragement to help me finish this study.

Abstract

Solar energy is considered as a cheap source of renewable energy for meeting daily energy demand. To harness the solar energy efficiently, solar power tower (SPT) has gained huge attention currently, apart from highest capital cost and technical maturity. Central receiver is a major component which converts solar flux into thermal energy, although it carries radiation and convection heat losses. In this study a new design of central receiver is introduced to meet the current challenges related to receiver such as thermal losses. The proposed design utilizes double glazing around an absorber which behaves as radiation trap and reduce radiative and convective losses. The absorber of proposed receiver utilizes milli channels for the passage of pressurized air and metallic tube structure to pass liquid phase heat transfer medium. The respected design is analyzed numerically by using different heat transfer fluids such as therminol oil, molten salt (solar salt) and liquid metal (molten sodium). The methodology adopted in numerical analysis was finite volume method (FVM) and SIMPLE algorithm used to couple pressure and velocity terms. Sensitive analysis is carried out on different flow parameters of receiver such as temperature, pressure and turbulence kinetic energy by altering mass flow rate of heat transfer fluids and solar flux condition. At higher mass flow rates of liquid sodium, receiver temperature drops significantly as compared to solar salt and therminol oil. Liquid sodium has shown the thermal efficiency of **82.3%,** which is considered as quite significant in receivers' performance and development.

Keywords : Renewable energy, power tower, central receiver design, receiver analysis, thermal analysis, multi flow heat exchanger, liquid sodium.

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1. Hassan B , Naveed A et al, “Numerical analysis of novel solar central receiver using different types of heat transfer fluids in Proceedings of the 3rd International Conference on Sustainable Energy Technologies (ICSET 2021)” August 10, 2021.

List of Abbreviations

CSPs	Concentrated solar power systems
SPT	Solar power tower
CRS	Central receiver system
PTC	Parabolic trough collector
PDC	Parabolic dish collector
LFR	Linear fresnel reflector
HFC	Heliostat field collector
MWe	Megawatt electric
FEA	Finite element analysis
FVM	Finite volume method
MW_{th}	Megawatt thermal
Tabs	Absorber temperature
Re	Reynold’s number
T_{air}	Air temperature

Chapter 1: Introduction

1.1 Overview of research

The existing receiver designs in central receiver system have shown a lot of major issues regarding its operation throughout a process. Different researchers have introduced new design ideas related to receiver design for betterment in system overall performance. Although concentrating collector technology has already proven its worth to counter the future energy challenges. In this study a new receiver design is introduced to counter the potential challenges related central receiver system .Evolution of central receiver system technology with time will be described in following sections of this chapter.

1.2 Problem statement

Currently global warming is a major issue for an earth environment. Carbon dioxide (CO_2) is considered as major contributor to global warming. Volumetric concentration of carbon dioxide inside an environment has increased drastically in past few decades due to excessive burning of fossil fuels inside a manufacturing industry, power sector and transport sector. Demand for cheap energy for daily utilization has been increasing rapidly in recent times. Renewable energy has shown a great potential to replace conventional fuels and fulfilment of the base load demands. Although renewable energy sources have some key challenges related to initial cost and technical feasibility/maturity. Apart from hydro and wind energy, solar energy is also the reliable source of renewable energy. Total energy consumed on earth within a year is less than the total energy imparted on earth by sun in one hour. The average heat transfer from sun radiations to earth is approximately equals to $\mathbf{1.2\times10^{17}\ W}$ of solar power [1]. Presently, Photo voltaic (PV) panels are installed to capture solar energy. These systems are the best energy source for meeting remote areas power requirements and localized power demands. While PV installations cannot meet off sun hours power demand. So, power grid cannot directly relay on PV systems to meet base loads. Another technology which is considered as most reliable for capturing solar energy to meet base loads in off sun hours are the concentrated solar power systems (CSPs). CSPs has shown huge potential to counter the growing energy demand. Power tower (PT) is one of the types of CSPs which concentrates solar flux and attains high temperature. Apart from its potential to tackle current requirements, power tower also faces technical and financial challenges. Lot of research is under way to meet the economic and technical challenges of solar power tower.

1.3 Solar collectors

Systems which directly converts solar flux/direct normal irradiance (DNI) into thermal energy are called solar collectors. These systems can further be divided into two types as concentrating collectors and non-concentrating collectors. Non concentrating collectors achieve less temperature as compared to concentrating collectors. Flat plate collector is a type of non-concentrating collector which utilize same absorber area as that of interception area [2]. Concentrated collectors work with large interceptor/reflector area as compared to absorber area. Ratio of aperture area to receiver area is called geometrical concentration ratio. Concentrated collectors are further divided in to four types such as Linear Fresnel collector, Parabolic trough collector, Parabolic dish collector and Heliostat field collector. These collectors operate at different concentration ratios which are given in table 1-1.

Table 1-1 Collector concentration ratios of concentrating collectors [3]

Sr no	**Type of concentrating collector**	**Concentration ratios**
1	Parabolic trough collector	70-80 suns
2	Linear Fresnel reflector	>60 suns
3	Solar power tower	>1000suns
4	Parabolic dish collector	>1300suns

Working principle for concentrated systems are different in nature, some are point focus and some are line focus. Parabolic dish collector and solar power tower are point focus while parabolic trough collector and linear Fresnel reflector are line focus. Basic schematic models for concentrating collectors are shown in figure 1-1.

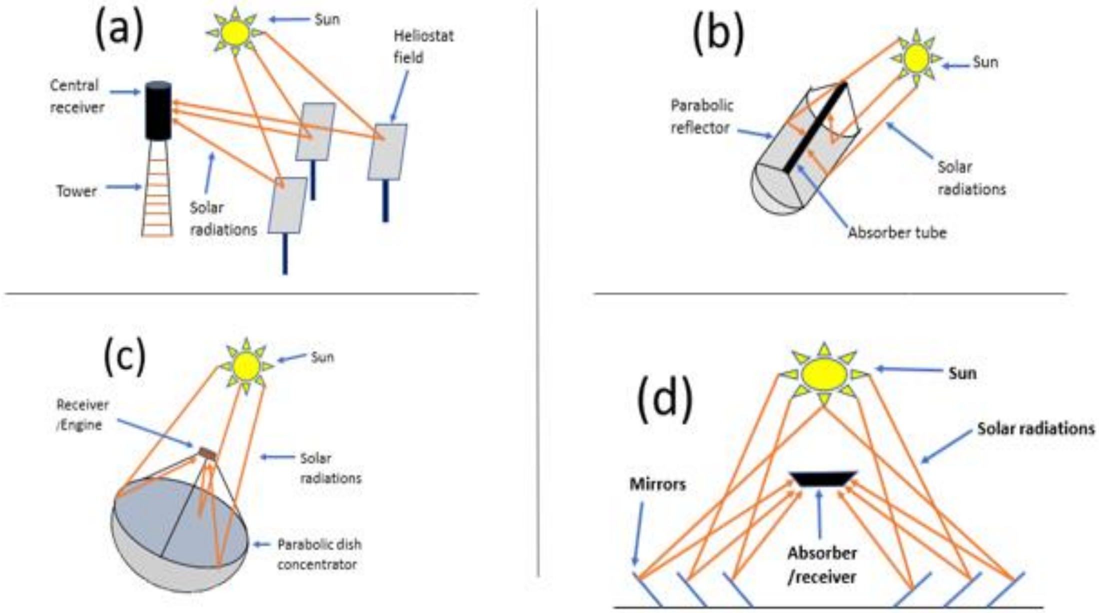

Figure 1-1 CSPs (a) HFC (b) PTC (c) PDC (d) LFR [4]

Usually, CSPs take large area for their installation so area requirement per unit hour of energy is mentioned in table 1-2.

Table 1-2 Area requirement for concentrating collectors [3]

Sr no	Type of concentrating collector	Area (m^2/MWh)
1	Parabolic trough collector	4-6
2	Linear Fresnel reflector	6-8
3	Solar power tower(heliostat field collector)	8-12
4	Parabolic dish collector	30-40

For continuous operation of CSPs , Thermal storage and backup systems are integral part to each CSP system, this enhances system performance and plant capacity factor [4].Thermal storage system requires good thermal properties of storage media, cost effectiveness and environment friendly. Thermal storage system depends on many key features such as storage material, storage time and type of storage systems. Thermal energy storage can be sensible heat storage , latent heat storage or chemical heat storage depending upon type of application [2]. Different types of materials are used as heat transfer fluid medium or thermal energy storage medium, these are currently being utiized at different comercial palnts or on test sites such as molten metals ,molten salts and compressed air. Each material has its own advantage depending on its usage , either short term or long term thermal energy storage conditions, thermal statbility or system compatability [2,5–7].

1.3.1 CSPs installation cost

Concentrated solar power systems can be used in different process, so their installation cost varies relative to system scale. To simplify the understanding of initial cost for each system, the capital cost is interrelated to electricity generation capacity which is shown in a table 1.3. Another cost which is important to CSPs performance, is an operational and maintenance cost, this effects levelized cost of energy. so operational and maintenance cost for each CSPs is also given in table 1-3.

Table 1-3 CSPs capital cost relative to capacity [3]

Sr no	**Type of CSP system**	**Capacity (MW_e)**	**Capital cost (US$/m2)**	**Operation and maintenance cost (US $/kW h)**
1	Parabolic trough collector	10-200	424	0.012 − 0.02
2	Linear Fresnel reflector	10-200	234	low
3	Solar power tower	10-150	476	0.034
4	Parabolic dish collector	0.01-04	-----	0.21

1.4 Heliostat field collector

Heliostat field collector is point focus collector. The overall system basically consists of three major areas, first area is heliostat field, second one is solar tower and third one is receiver or absorber with a cavity [8,9]. Tower holds receiver at the top of it usually height of solar tower is in range of **70 to 150 m** [5]. Heliostat field is further classified into two types, one is surrounded field and other is polar field. Both heliostat fields have their own advantages and disadvantages depending on solar positions and heliostat field design layout. Optimum heliostat field design depends on minimum shading and blocking losses [2,8]. Each mirror in heliostat field is a two-axis tracking flat mirror or slightly curved mirror (concave shape), with major function of reflecting direct normal irradiations toward receiver [2,5,8]. **Curved mirror** has ability to reflect more sun radiations as compared to flat mirror. Heliostat field mirrors area usually fall in a range of **50 m^2 to 150 m^2** [5]. Concentrating collectors concentrate a lot of solar flux towards the top of tower at receiver in order to achieve high working temperatures. Heliostat field collectors achieve the concentration ratios of **300 to 1500 suns** [2,5]. As higher the system working temperature goes higher would-be the system thermal efficiencies. Heliostat field collectors (HFC) are also a concentrating collector with a central receiver to absorb solar radiations, these systems are usually known central receiver systems (CRS). Central receiver systems have multiple applications such as power generation, industrial process heating and communal heating systems [4]. For power production, central receiver systems are known as solar power tower (SPT) [3]. In solar power tower, central receiver converts solar energy into steam generation or heating other heat transfer fluid which in turn runs steam Rankine power cycle. Central receiver systems (CRS) utilize multiple heat transfer fluids such as water/steam, liquid metals (liquid sodium), synthetic oils and molten salts (molten nitrate salts) with thermal energy storage system (TES) [3,4,8]. Central receiver systems are highly flexible in nature and capable for the integration of these systems with different type of thermal energy storage media either storage media is sensible, latent or chemical in nature [2,10].

Central receivers (CR) are further classified as external/surrounded field receiver and cavity/polar field receivers. Cavity receivers are usually designed to reduce thermal loses with optimum cavity opening to receive solar irradiations from heliostat field [2,8]. Central receivers are generally characterized as gas receivers, liquid receivers and solid particle receivers which are briefly discussed below [11].

1.4.1 Gas receivers

Gas receivers are used as central receiver where heat transfer fluid is in a gas phase. Depending upon method of heat transfer, these receivers are further classified as volumetric air receivers, small particle air receivers and tubular gas receivers [11]. Volumetric receivers consist of porous material or honeycomb structure that is irradiated with solar flux and air passes through it. Air gets volumetric heat transfer effects and attains high outlet temperatures. Volumetric air receivers are further classified as open loop air receiver and closed loop air receiver, open loop air receivers are used in Rankine cycle while closed loop pressurized air (windowed) receivers are used in Brayton cycle [8,11,12]. Small particle air receiver utilized small particle usually submicron level particles of carbon, which are suspended in air within pressurized air receiver cavity. Suspended particles are irradiated with sunlight to get heated; this heat is further transfer to air to run other cycle [11]. Tubular gas receivers contain multiple metallic tube which are arranged parallel with in cavity, each metallic tube contains pressurized air which is being heated due to solar irradiations [11,13].

1.4.2 Liquid receivers

Tubular liquid receivers were used in Solar 1 and Solar 2 demonstration plants [11]. Tubular liquid receivers utilize an array of thin wall tubes usually made up of stainless steel or alloys. These tubes carry working fluids and pass through high solar flux region. Tubular liquid receivers can be used as external or cavity type receiver, depending on required receiver temperature and working fluid outlet temperature. Mostly water/steam, molten salts and molten metal are used as heat transfer fluid or working fluid in tubular receivers [11,14,15]. Falling film receivers are used as another type of liquid receiver where liquid falls under its own weight over an inclined wall and get exposed to solar radiations directly or heated indirectly through the wall. These types of receivers reduced the pumping power to drive working fluid [11].

1.4.3 Solid particle receivers

Solid particle receivers utilize falling particles in the form of curtain, which directly irradiated with solar flux within cavity in order to reduce heat losses. Working substance in these receivers are sand like ceramic particles, which directly absorbs heat and transfer it to other heat transfer fluid to run Rankine cycle. Usually, these particles are directly stored in storage tank as thermal energy storage. These receivers have ability to attain high temperatures more than **800 °C**. Ceramic particles can sustain high concentration ratios of solar flux within cavity [11,16].

1.5 Thermal storage system

Thermal storage system is considered as second most, major part of solar power tower. It is a backbone of whole working system, without this system power tower cannot maintain stable and continuous operation at full load during off sun hours. Basic working mechanism for thermal storage system is charging and discharging of energy efficiently. So, following are the key parameter for the selection of thermal energy storage material.

- Material should have high capacity for energy storage.
- Chemical stability under working temperatures.
- Compatible with heat exchanger and heat transfer fluid.
- Stable at large number of charging and discharging cycles.
- Non flame able, cost effective and easily integrate-able with power block.

Thermal storage material can further characterize as sensible storage materials, latent storage materials and chemical storage materials. Some of important storage materials with their working temperatures range are shown in table 1-4.

Table1-4 Thermal storage materials [2]

Sr no	Sensible storage materials	Temperature range (°C)
1	Sand-rock minerals.	(200 – 300)
2	Concrete(Reinforced)	(200 – 400)
3	NaCl	(200 – 500)
4	Fire bricks (Magnesia)	(200 – 1200)
	Latent storage materials	
1	Solar salt (Hitec)	(220 – 600)
2	Silicone oil	(300 – 400)
3	Nitrite salts	(250 – 450)
4	Nitrate salts	(265 – 565)
	Chemical storage materials	
1	Iron carbonate	180
2	Methane/water	(500 – 1000)
3	Calcium carbonate	(800 – 900)

1.6 Commercial solar tower installations

The United States Department of Energy, projects in California exhibited that solar power tower generates electricity at utility scale for 24 hours a day in a year between 1980 to 1990. Currently after doing a lot of research, power tower has ability to generate electricity in a range of **30 to 400 MW** of electricity at utility scale. The Sierra Sun Tower (two tower system) installed in a year of 2009 at a location of Mojave Desert generates an electricity for five thousand houses. In California, USA at a year of 2010, another solar power tower (three tower system) was installed with generating capacity of **392 MW** electricity. The annual efficiency (solar to electrical) for solar power tower lies in between 20 to 35 %. For power tower generating more than **30 MW** of electricity, it is cautionary to augment power tower with natural gas fired combined cycle power plant or coal/oil fired Rankine cycle power plant. For experimental and demonstrational purpose plants with generation capacity of **1 MW** are installed in Greenway in Turkey and Dahan plant in China. Another operational plant is installed in year of 2016 at a location of south Africa with a generating capacity of **50 MW** of electricity. Other commercial systems which were installed include PS10,PS20 and Gemasolar plants. Under a lot of research and development in a power tower technology, it is expected in future for at least installation capacity of **995 MW** of electricity mainly in China [3]. Basic working mechanism for power tower technology containing power block, thermal storage and heliostat field are shown in figure 1-2.

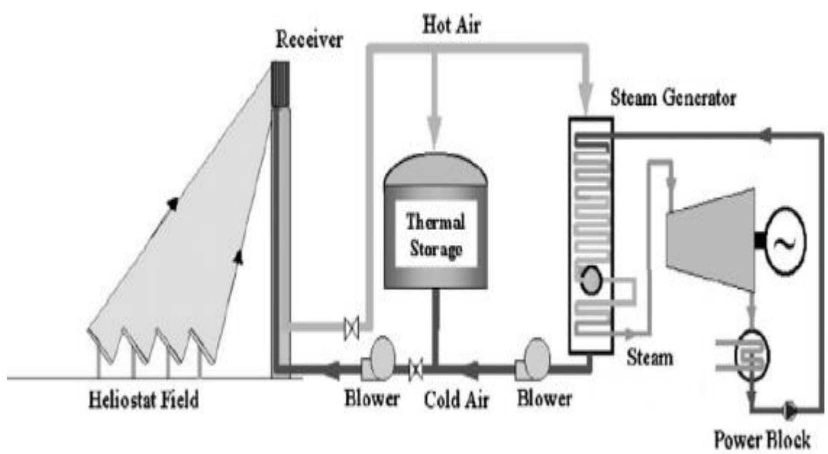

Figure 1-2 Power tower working mechanism [12]

1.6.1 PS 10 commercial plant

The first commercial solar power tower, pilot plant was installed in Spain with a generation capacity of 11 MW. It is also named as Planta solar 10. Land area it covers about 55 ha. Its annual electrical power generation capacity is measured to be 23,400 MW h/year [3].

1.6.2 Solar two

Solar two is a conversion of solar one, which was installed in Barstow, California with a capacity of 10 MW. It was the largest experimental project for power tower commercialization. Solar one started test and production in 1982 for 6 years. Solar two is the conversion of solar one pilot plant into molten nitrate salt technology. Solar two utilized same heliostat field as for used by solar one but with different receiver. The receiver of solar two was external tubular receiver with 24 panels. In solar two nitrate salt has substituted the solar one oil/rock thermocline thermal storage. This new thermal storage assisted 3 hours of full load on turbine. Temperatures for cold storage and hot storage was set 285 ℃ and 565 ℃ respectively. Thermal storage volume for hot and cold salt storage tank was about 900 m^3 each and it has capacity of 105 MW_{th}**.** Net electrical output for solar power was about 10 MW [17].

1.7 Aims and objectives

This study completely focusses on new receiver design. The new design which has a potential to eliminate external losses with better system operations under fluctuating solar conditions. Optimized geometry with complete reduction of cavity to reduce installation cost which is major challenge for deployment of CSP technology. Other objectives are as follows

- To have a smart design with multi functions.
- To get thermal stability under operations.
- To get maximum heat transfer in between parallel heat transfer fluids and absorber.

1.8 Scope and limitations

The scope of this study is to analyze the new proposed design of central receiver. The performance of this receiver is monitored on a base of analyzing different parameters such as thermal nature of materials used in receiver and thermo hydraulics effects involved inside a receiver flow passages.

The limitation of this work includes the radiation exchange with glass and its reflection and rarefaction effects.

1.9 Thesis Outlines

Chapter 2 includes literature review about different receiver which are tested at different experimental sites, also it relates Numerical analysis of different types of receivers to generate efficient working receiver. Numerous numerical models which are adopted by different researcher to analyzed receivers designs for better performance in future installations.

Chapter 3 details about the mechanical design of proposed receiver, its working mechanism and materials adopted for its smooth operation. It also includes the strategy used to evaluate the proposed receiver. Numerical analysis methodology which is adopted to reveal the behavior of proposed design under altering conditions.

Chapter 4 includes the results and discussions portion. The behavior of receiver is discussed in detail related to obtained results.

Chapter 5 covers the recommendations which are important for further betterment in design and results accuracy.

Summary

For a growing economy of a world, it is a necessary requirement to have a sustainable with reliable source of cheap energy. Renewable energy particularly solar energy has shown a huge potential to meet that demand. This chapter include brief and basic details related to Solar power systems. Above chapter has discussed the basic types of concentrated solar power systems i-e PTC, LFR, PDC and HFC (SPT) and also describe how these systems operate especially solar power tower. Basic working mechanism for power tower integrated with thermal storage has discussed related to charging and discharging cycles. Some important details which are necessary for understanding of CSPs is discussed in this chapter related to each CSP system Capital cost , capacity MW_e, land requirement, operational and maintenance costs. Types of receivers related to working fluids and thermal storage are detailed briefly for a smooth and continuous operations. The installation of experimental and commercial Solar power towers, which are operational worldwide has mentioned relative to their total generation capacity and technology used.

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Chapter 2: Literature Review

The entire study is consisted upon the receiver design of the solar power tower and strategies to analyze the different effects. Different types of receivers have been discussed in previous chapter (Chapter 1). The quality research, which has been done on solar receiver related to its performance is mentioned in this chapter. Basic working mechanism of any type of receiver is to convert solar flux into thermal energy with minimum external and internal losses. For this purpose, different techniques which have been adopted by different researchers and these will be discussed in following topics. Power tower is an expensive technology that's way it has few experimental sites for its receiver testation or experimentation. The analysis of any receiver design, the strategy which is best suited to adopt is mentioned in below section 2.1.

2.1 Central receiver analysis

Basically, central receiver absorber utilizes solid absorber material with at least liquid phase heat transfer fluid somehow, few receivers operate on solid phase heat transfer like suspended particles or falling particles. The whole system is cover by some solid phase insulation and solid structure support. The analysis of receiver is done by numerical schemes, which appears to be more cost effective before doing experimental analysis. Numerical methods completely elaborate the minor details of new generated geometry and its operational performance with respect to time. Parametric studies done by numerical analysis is more cost effective and more detailed as compared to experimental analysis, however it lacks exact accuracy while results approach towards the exact values. The approached results are sufficient to understand performance under real conditions.

2.1.1 Design and modelling strategy

Designing of solar receiver is crucial step before doing numerical analysis, this step includes the selection of materials , operating conditions such as pressures and mass flow rate, also a solar flux. The performance of power tower receiver is usually determined by two key parameters such as optical efficiency of receiver and thermal efficiency of receiver. Stress factor of safety and temperature factor of safety are also important design parameter for high temperature receivers [1].

2.1.1.1 Radiation flux calculations

After design, second step is its evaluation under pre-determined set of parameters. Methods of computational fluid dynamics (CFD) are usually utilized on 2-D or 3-D geometry with an exchange of thermal energy in between radiations, conduction in solid and convection in liquid phases. In all solar receiver cases, initially solar flux is computed, after that it is injected in energy equation as a heat source for thermal analysis. Solar flux from heliostats upon receiver is well determined by some commercial or non-commercial software's such as FORTRAN language, DELSOL, HELIOS, Soltrace, Monte Carlo ray-tracing tool (open source), Tracer, TracePro (import CAD files), FRED and ANSYS Fluent (discrete ordinate irradiance method) [1].

2.1.1.2 Radiation heat transfer calculations

The radiation heat transfer problem inside a receiver is govern by two methods such as surface to surface method and radiation transfer equation. First mentioned method is not as effective because it rarely captures volumetric effects. Radiation transfer equation defines propagation of radiation inside a medium relative to wavelength , direction and location. Directional approximations applied to radiation transfer equation further gives simplified models such as P-1, Rosseland, discrete ordinate, and Monte Carlo methods. However, each model has its own limitations [1].

2.1.1.3 Conduction and convection calculations

Conduction and convection analysis within a receiver required to solve flow and energy equation (governing equation). Radiation flux is imported as a thermal model, acting as a boundary condition which is computed initially. After applying thermal model flow domain is selected with appropriate turbulence model if it has high Reynold's number inside a domain. K-e, K-ω or K-ω SST models are available for problem solution under different flow conditions [1]. Schematic diagram for receiver performance analysis is shown in a figure 2-1.

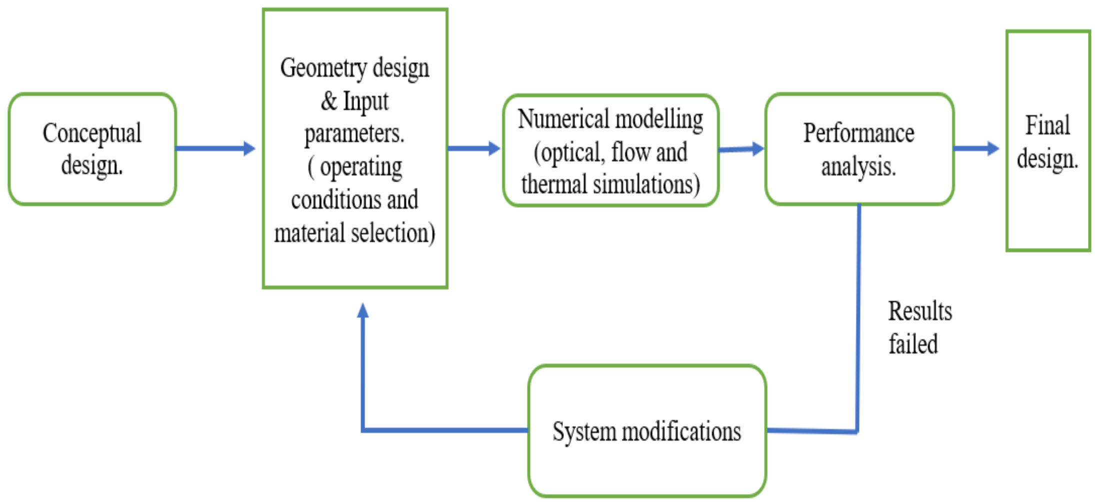

Figure 2-1 Schematic diagram of receiver analysis [1]

The installation of commercial power towers in Spain such as PS10,PS20 and gemasolar has shown some problems so, it is more demanding to have some receiver design changes to solve the problems raise in tubular receivers so, the researchers introduce the concept of volumetric heat transfer. Technical feasibility of power tower has tested after installation of demonstrated plants in 1980s. Volumetric receivers are not completely commercial but different test plants have installed such as in Julich , Germany a power tower (pre commercial) of 1.5 MW_e started production in 2009. Different volumetric receivers which have test at different test sites are mentioned in following section 2.2.

2.2 Volumetric receivers

Volumetric receivers are more reliable due to their three-dimensional heat transfer characteristics. The three-dimensional heat transfer in volumetric receiver is attained due to porous structure or regular honeycomb structure. Porous structures are usually made up of wire mesh or ceramic material which has ability to transmit more flux inside the core relative to its surface. Absorber temperature is crucial for reliable operation as ceramic SiC can sustain a temperature of 1400 °C, but its catastrophic temperature limit is 1700 °C, although some metals are also capable of generating air outlet temperature of 1000 °C. Flow stability is a major concern in volumetric receivers, porous structure cause variations in pressure drops which in turn fluctuates the mass flow rates, which further deteriorates uniformity in outlet temperature of absorber so, the predictable temperature dose not achieve. Calculation of critical solar flux is very important for working on a volumetric receiver as at higher solar flux above the critical flux, more will be a stable flow [2]. Most important application for tested volumetric receiver are as 1) Integration of volumetric receiver with Rankine Cycle as open loop system, where atmospheric air is heated through porous material of metal or ceramic and further utilized to generate steam. 2) Combination of volumetric receiver with gas turbine in combine cycle as close loop system. Antonio L et al classified volumetric receivers as shown in table 2-1.

Table 2-1 Volumetric receivers' classification [2]

Sr no	**Receiver name**	**System type**	**Material.**
1	Phoebus-TSA	Open loop	Metal
2	SOLAIR	Open loop	Ceramic
3	REFOS	Closed loop	Metal
4	DIAPR	Closed loop	Ceramic

2.2.1 Open loop metallic absorber volumetric receiver

These types of receivers utilize metal wire mesh as absorber for achieving volumetric heating effect. Receivers which include this technology are given in table 2-2.

Table 2-2 Open loop metallic Volumetric absorbers [2]

Sr no	**Receiver name.**	**Absorber (material)**	**Absorber diameter (mm)**	**Receiver Efficiency**	**Gas temp (°C)**	**Test site**	**Power output (kw)**
1	Mk-I	AISI 310	62	70–90	842	Swiss Alps	3
2	Sulzer 1	AISI 310	875	68	550	PSA	200
3	Sulzer 2	AISI 310	875	79	550	PSA	200
4	Catrec 1	X5CrAl20 5 + Ce	940	80	570	PSA	200
5	Catrec 2	X5CrAl20 5 + Ce	756	70	460	PSA	200
6	SIREC	Alloy 230	875	48	710	PSA	250

2.2.2 Open loop ceramic absorber volumetric receiver

Different receivers open loop ceramic absorber have also been test at different sites. The data about these receivers has shown in table 2-3.

Table 2-3 Open loop ceramic volumetric receiver [2]

Sr no	Receiver name.	Absorber (material)	Absorber diameter (mm)	Receiver Efficiency	Gas temp (°C)	Test site	Power out (kw)
1	SANDIA FOAM	Al_2O_3	875	54	730	PSA	200
2	CeramTec	SiSiC	950	59	782	PSA	200
3	Conphoebus-Naples	SiSiC	706	60	788	PSA	200
4	Selective receiver	SiSiC	835	62	620	PSA	200
5	HiTRec II	re-SiC	Hexagonal shape	72	800	PSA	200
6	SOLAIR 3000	re-SiC	Square shape	75	750	PSA	3000

2.2.3 Closed-loop ceramic and metallic volumetric receivers

These receivers are pressure based volumetric receivers. A special window of fused silica is used in this technology to separate inside pressurized air from outer ambient condition while it allows solar flux to transmit through it for continuous operation of receiver. Window of receiver also faces some key challenges which are necessary to take under observation such as pressure bearing strength, cooling of window and thermal expansion of glass relative to absorber material. Different types of receivers have been used utilizing different window designs with respect to desire operation [2]. These receivers are given in table 2-4.

Table 2-4 Pressurized (close loop) volumetric receivers [2]

Sr no	Receiver name.	Absorber (material)	Receiver Efficiency	Gas temp (°C)	Test site	Power out (kw)
1	PLVCR-5	SIRCON	71	1050	Sandia	3
2	PLVCR-500	SIRCON	57	960	PSA	500
3	DIAPR 30–50	Alumina–silica	71	1200	Weizmann Institute Science	50
4	REFOS	Inconel 600	67	800	PSA	350
5	SOLGATE	Inconel 600 and SiC	70	960	PSA	400

Volumetric receivers are in their development phase, Julich plant has shown a big potential of these receivers although these receivers are not commercially available. Other demonstrated commercial, small scale tubular receiver plants with their net output generation is shown in table 2-5.

Table 2-5 Tubular receiver commercial plants [2]

Sr no	Power tower name	Capacity(MW)
1	PS10	11
2	PS20	20
3	Gemasolar	17

2.3 Receiver design enhancements

L. Marocco et al [3] studied the heat transfer in between tube receiver and liquid metals. It is shown in the work that liquid metals differ from other fluids and their flow nature deteriorates from Reynold's analogy. According to this Reynolds analogy, turbulent Prandtl no dose not remains constant near to unity. This work utilizes non uniform heat flux with K-e-K_θ-e_e four equation turbulence model and its effects relative to wall thickness ratio, solid to liquid thermal conductivity ratio and length to diameter ratio relative to peclet no. Prandtl no is 0.025 which shows the nature of liquid metals such as gallium indium tin, mercury and lead-bismuth eutectic.

Zhirong Liao et al [4] proposed a new design of central receiver containing as idea of heat pipe. The overall receiver consists of heat pipe and receiver assembly. Receivers' assembly further utilizes inlet and outlet header with multi tube passes. Heat pipe has two sections, one is evaporator and other is condenser. At evaporator solar radiation concentrates and vaporizes the heat transfer fluid which is further raised up into condenser section position inside a receiver's tube, where cross flow heat transfer fluid inside a receiver takes the thermal energy. The study is also numerically solved which shows the receiver thermal efficiency of **88.5** %, after doing some optimization relative to tube passes and applied solar flux, receiver thermal efficiency increases up to 91%.

Non uniform heat flux causes hot spots which are crucial for receiver performance. Forced convection inside receiver plays important role during whole operation. Higher the forced convection inside receiver tubes, lower would be tubes outer surface temperature. Low temperature at the outer surface further decreases the losses and risks of local hot spots under non uniform heat flux. So, a lot of research has done to increase forced convection by using inside fins with different geometries. Zhang-Jing Zheng et al [5] has introduced another concept of inserting some porous junk made up of some metal inside the receiver tubes. Four different types of geometries, which are further known as enhanced receiver tubes. are studied numerically under turbulent flow conditions. The study concludes the effects of enhance receiver geometries upon tube temperature decrement, thermal conductivity ratio effects on flow nature and the better position of porous insert depending upon thermo hydraulic effects of heat transfer fluid. It is also concluded that that concentric filling of tubes with porous material enhanced heat transfer in between fluid and tube walls. M.R. Rodríguez-Sánchez et al [6] introduced another idea to enhance tube base external receiver performance. The idea consists of concentric tube or bayonet tube receiver, where external tubes combine to form a cylindrical receiver as compared to external (conventional) receiver. This new receiver is analyzed numerically by applying same boundary conditions as compared to external receiver. The overall study concludes that in bayonet receiver tube the film temperature and wall temperature are less as compared to tubular receiver which appears to be significant in less corrosion and low thermal stress. These low temperatures in bayonet tubes allows to use cheaper and easily available materials, while higher thermal efficiency bayonet design allows to reduce the receiver dimensions as well as heliostat field size which further reduce capital cost of power tower.

Flow oscillation also drives the heat transfer within tubes. The variation of velocity inside a tube or pipe influences majorly a heat transfer. Another effect which shows drastic enhancement in heat transfer inside a tube is flow reversals and pulsation in a flow. Although these effects generate a vortex's which are very difficult to model and analyze. Xuefeng Wang et al [7] has studied a pulsating flow inside a tube of constant temperature and concluded that Womersley number has a major effect on heat transfer. Optimization of Womersley number and pulsation fluid frequency has a great effect on maximum heat transfer. M.R. Rodríguez-Sánchez et al [8] introduced another receiver design utilizing the mechanism of variable velocity parameter. This study concludes that the effect of variable velocity reduces the risk of power tower receiver from overheating and other numerous advantages of this receiver is to reduce the levelized cost of energy.

Development of numerical model for volumetric receivers is important for further development. As it is considered that volumetric receivers have great potential in performance and efficiency. Develop numerical model must include porosity ratio and Reynold's no functions. Zhiyong Wu et al [9] has worked numerically on ceramic porous material behaving as volumetric structure (three dimensional). Packed tetrakaidecahedron structure represents ceramic foam for analysis. The analysis covers the details about the convective coefficient within entrance region, local thermal non equilibrium region (LTNE) and local thermal equilibrium region (LTE) regions. The effects of superficial velocity changes, cell size, porosity ratios and solid temperature are covered in whole analysis. The whole study overall concludes with a correlation for local and volumetric heat transfer coefficient relations. Which is further a function of Reynold no , cell size and porosity. This correlation for Nusselt no is very important for future developments in the field of volumetric receivers.

External receivers usually serve a molten salt for continuous operation and thermal storage. The construction geometry consists of number of tubes connecting with manifolds to get a shape of cylinder or rectangular shape. These structures mostly operate at higher temperatures more than 600 ℃, at these temperatures receiver faces a reradiation's and natural convection effects. Joshua M. Christian et al [10] introduce some geometries for external receivers and studied their external effects while considering flat plate as a base case. The introduce geometries are as 1) base case flat receiver 2) Radial Finned Receiver 3) Linear Vertical Fin Receiver 4) Horizontal Slate Fin Receiver. Idea of these geometries is to enhance ray absorption with increase geometric ratio while maintaining aperture area with different optical intercept area. These geometries are modeled and analyzed using numerical techniques. The overall study concludes that Horizontal Slate Fin Receiver shows a maximum thermal efficiency of 95.5% more than 4.5% than base case.

Another concept to enhanced open receiver efficiency is to alter the external surface to re capture emitted radiations. Oliver Garbrecht et al [11] introduce a concept of pyramid structures to enhance the thermal efficiency. The structure utilizes the double cones and a passage for heat transfer fluid in between these cones to carry out thermal energy. Initially the absorber surface is analyzed numerically by radiation trace program and temperature contours are created. These temperature trends then coupled with fluent to solve the thermo hydraulic nature of absorber heat transfer fluid, while radiations emission is measured. Overall study concludes that pyramid structure is behaving as best radiation trap although it has high temperature at apex where convection losses get increase, but receiver has shown the thermal efficiency of **91.2%**, which is considered to maximum up **to yet in literature**.

Pressurized air receiver is getting attentions in CSPs technology. Solar assisted combined cycled plant has reached a thermal efficiency of 60%. Pressurized receivers in Brayton cycle heat the pressurized air up **to 1000 ℃** while these receivers have a thermal efficiency up to 30% at the operating temperatures in between (600-700)℃. Different designs in receiver technology have utilized the fused silica window with ceramic channels or foam inside, with operating at temperature of 1200 ℃ while window design also bring forth some design challenges. Tubular receivers for pressurized air made up of metal and ceramic has been deployed but less thermal efficient as compared to cost. Shenzhou Chu et al [12] has implemented another novel idea for solar receiver design to utilize the double spiral tube in conical fashion. Material for double spiral tubes is selected to be a stainless steel 310s. The whole experimental study concludes that this double spiral receiver has ability to generate a maximum temperature of 908 ℃ under a flux of 4.19 KW, mass flow rate although effects the pressure drop and outer temperature or thermal efficiency under constant DNI values, but the thermal inertia of receiver kept parameters under stable conditions.

Solar receiver works under fluctuating conditions such as varying solar flux, wind speeds and transient material behaviors. To understand the nature of receiver under transient condition it is necessary to develop a numerical model either it is FEM model or CFD model. These models play crucial role in development of receiver design and research studies, particularly the dynamic model plays vital role in predicting the real performance of solar receiver. Andreas Fritsch et al [13] studied the tubular receiver where single tube is considered initially for development of CFD and FEM models. The FEM model for single tube (one dimensional) model results completely agree with CFD model results while considering constant convection coefficient . Complete receiver model is analyzed to observe the radiation exchange in between the surfaces of solar receiver, also numerical model is validated using solar 2 experimental data. The overall study concludes with a development of CFD and FEM models and calculation of Nusselt no within tubes and its effects on outer tube temperatures due to non-homogenous flux and variations in Nusselt no. The study also signifies that over estimation in Nusselt no reduces the tube wall temperatures trends.

Power tower operations are critical for plant working parts/components, usually molten salt (60/40wt% $NaNO_3$-KNO_3) are deployed for solar power tower operational continuity. Although solar power tower commences several shut down and startups per day. These startups and shut down cause harsh conditions for working parts such as working stresses and thermal stresses. These transient working conditions are crucial from design and operation point of view. The development of robust and efficient transient conditions such as shutdowns and startups enhanced plant components life in term of thermal fatigue and mechanical fatigue. Basic plants operations include 1) warm up the system and working fluid 2) receiver filling 3) absorber operations 4) close ups (shut down). Dimitri Doupis et al [14] studied the effects of transient operational condition by developing FEA models and analyzed the effects such as mechanical stresses and thermal stresses. In this study ISAAC Dynamics (software) is used to study transient (dynamic) model of molten solar central receiver. This model includes everything such as inlet header , piping , manifolds and outlet headers. The overall simulations operation includes process circulations, draining and initial filling. The overall study includes different start conditions like 1) warm start 2) hot start (case 1) 3) hot start (case 2). The detailed study of operational behavior of molten solar central receiver with development of CFD dynamic model, concludes that it is feasible to cool molten salt central receiver before draining to reduce the chances of thermal shock to hot components. The rate of temperature changes is considered in molten salt central receiver as another parameter to reduce the risk of fatigue failure. So, fatigue life and creep life are major factors for overall power tower life and system performance under fluctuations, although optimization is necessary in between metal tube filling temperature and rate of temperature change in components.

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Summary

The chapter 2 includes the details of literature review related to power tower technology such as the types of receivers which have installed worldwide and other types of receivers which are not commercial yet, but these are under design and experimentation phase at different power tower experimental sites such as volumetric receivers. Furthermore, this chapter includes the details of receiver design analysis by utilizing different numerical tools and the enhancements in receivers design by different researchers. The overall research in power tower technology has mentioned in detail to increase the thermal efficiency of receiver and decreasing the levelized cost of energy which is the major focus in present study. The details about the operations inside the power tower related to shut down and start up are given in this chapter with a detailed dynamic numerical model.

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Chapter 3: Design and Analysis

For the implementation of conceptual design, the physical geometry with proper dimensions is necessary for required operation. The materials used for proposed design will be discussed in the following chapter. The receiver operation should be in a define working limits for lower the risk to receiver failure. The 3-D model of proposed solar tower receiver is discussed in detail with respective geometry and detail diagram for each part in section 3.1.

3.1 Solar receiver model formulation

The basic design of proposed receiver is a combination of different parts which are assembled. To ensure complete trap for solar irradiations the double glazing is provided externally. The glazing is divided in to two glass covers, one is considered as outer glass cover and other is an inner glass. The basic dimension for outer glass is shown in figure 3-1.

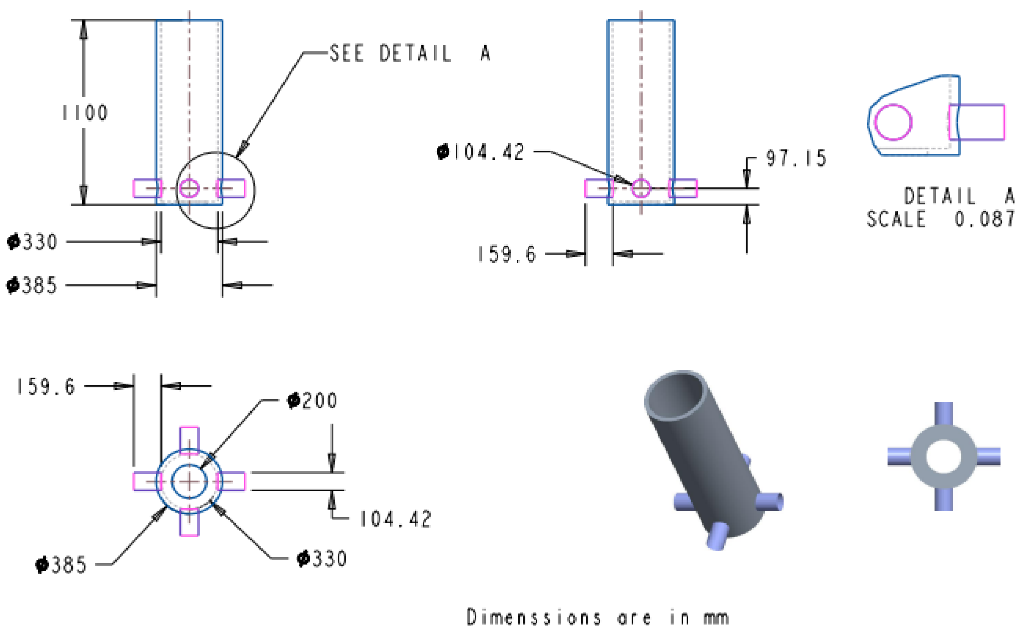

Figure 3-1 Outer glazing drawing sheet

Figure 3-1 shows the details for outer glass with respective dimensions and four inlets to intake air. Similarly figure 3-2 shows the details of inner glass with respective dimensions in millimeters.

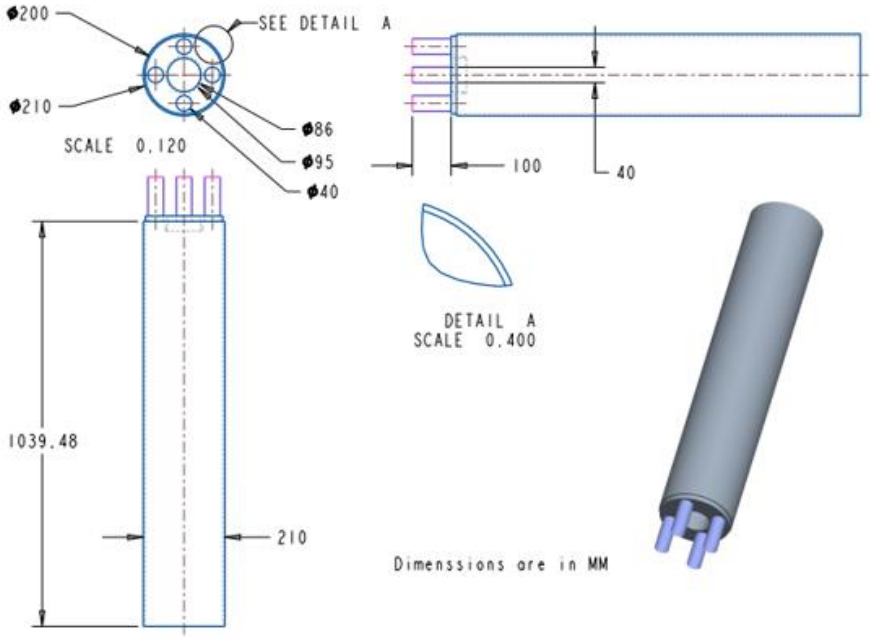

Figure 3-2 Inner glazing drawing sheet

The cover plate which is essential for providing rigidity to both glazing's and to return the incoming air into ceramic absorber. Its details are shown in figure 3-3.

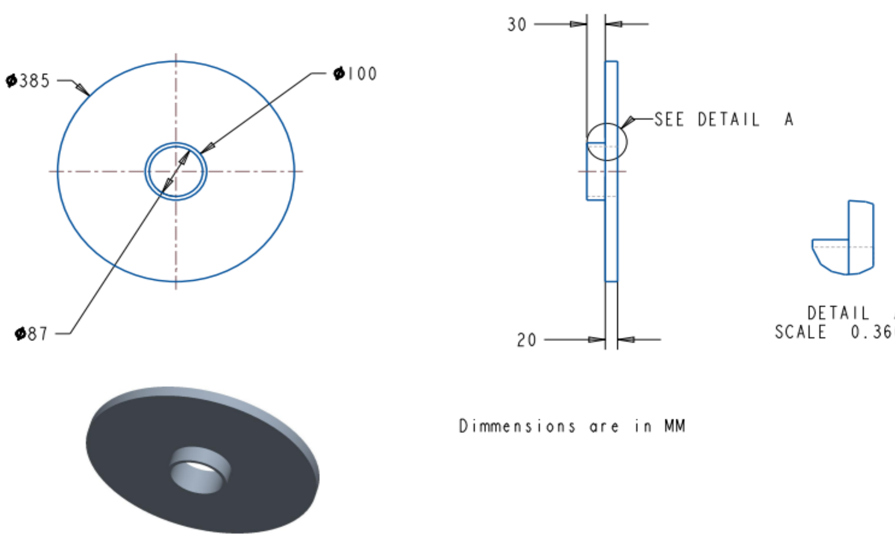

Figure 3-3 Cover plate drawing sheet

Important part which directly absorbs the incoming solar radiation flux is absorber and its dimensions are given in figure 3-4.

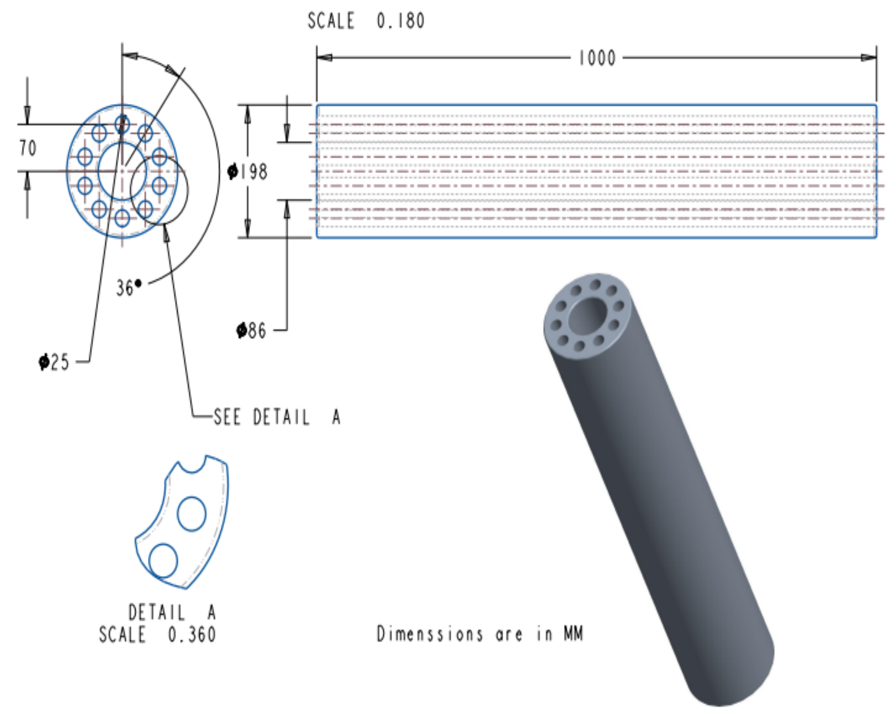

Figure 3-4 Receiver absorber drawing sheet

Metallic tube/pipe which is essential for passing heat transfer fluid throughout the receiver. Its dimensions are shown in figure 3-5.

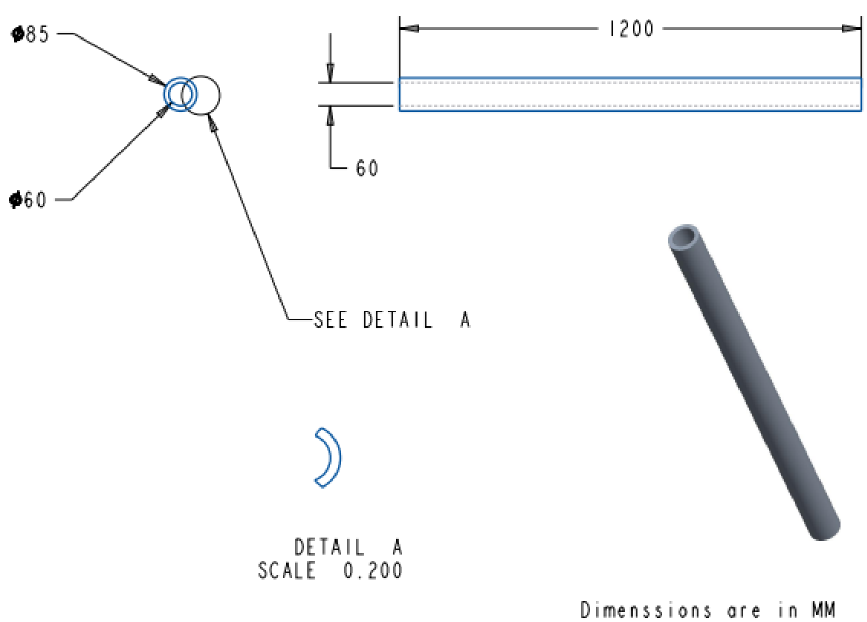

Figure 3-5 Metallic pipe drawing sheet

Assembly view for proposed receiver is shown in a figure 3-6.

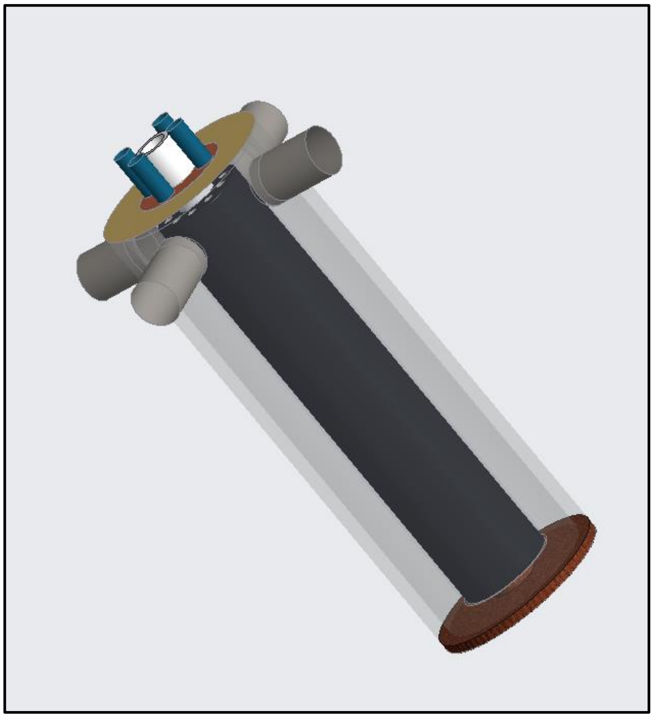

Figure 3-6 Assembly of solar receiver

Exploded view of respective receiver is shown in a figure 3-7.

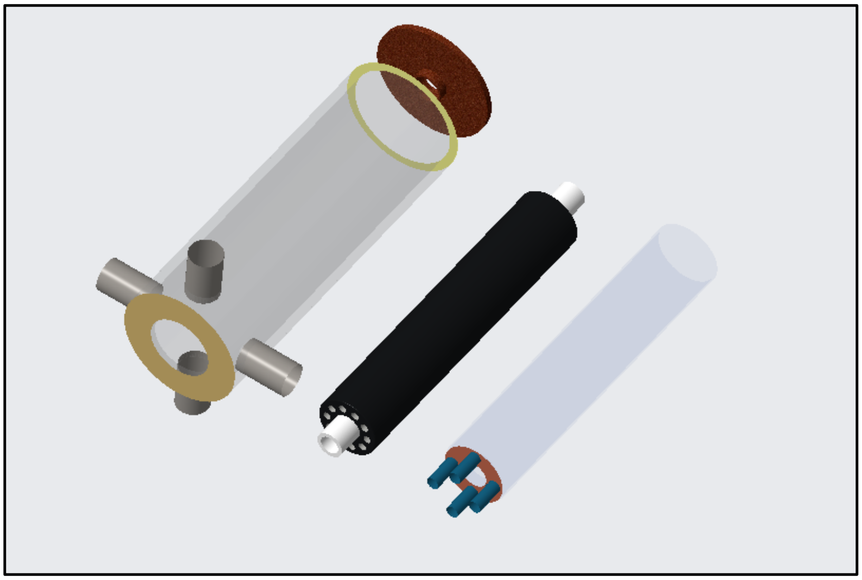

Figure 3-7 Exploded view of solar receiver

3.2 Materials for proposed receiver

Different materials are used in respective design according to number of parts being used. The outer and inner glass covering material is quartz glass. The optical properties for quartz glass are given in table 3-1.

Table 3-1 Quartz glass optical properties [1]

Sr no	Material	Transmissivity for (350-700) nm	Absorptivity for (350-700) nm
1	Quartz glass (fused silica)	0.94	0.04

Material for solar absorber, glazing's and central pipe is Ceramic (SiC), Quartz glass and Tungsten. The thermal and material properties for above mentioned substances are given in following table 3-2.

Table 3-2 Material properties [2–4]

Sr no	Materials	Density kg/m^3	Specific heat cp (j/kg.k)	Thermal conductivity K (W/m.K)
1	Ceramic	3100	750	120
2	tungsten	19300	133	167.36
3	Quartz glass (fused silica)	2200	740	1.38

Heat transfer fluids which are being used in proposed receivers are therminol oil , molten salt (Solar salt) and molten metal (liquid sodium). Another working fluid which is used inside receiver is pressurized air. Each heat transfer fluid is utilized in combination with pressurized air alone for each case. Material properties for heat transfer fluids and air is given in a table 3-3.

Table 3-3 Heat transfer fluid properties [5,6]

Sr no	Materials	Density (kg/m^3)	Specific heat Cp- (j/kg.K)	Thermal conductivity K (w/m.K)	Viscosity (kg/m.s)
1	Therminol oil	749	2485	0.0814	0.00017
2	Liquid sodium	927	1300	80	0.0007

3	Solar salt	2014	1450	0.4671	0.01382
4	Air	1.225	1000	0.03	0.0000181

3.3 Receiver heat transfer mechanism

Dynamics inside a proposed receiver depends upon flow of two working fluids simultaneously. The proposed design provides multi pass to pressurized air in combination with single pass to heat transfer fluid. The multi pass gives an advantage to pre heat incoming pressurized air parallel to cooling of double glazing. The basic work mechanism inside a proposed receiver is given in a following figure 3-8.

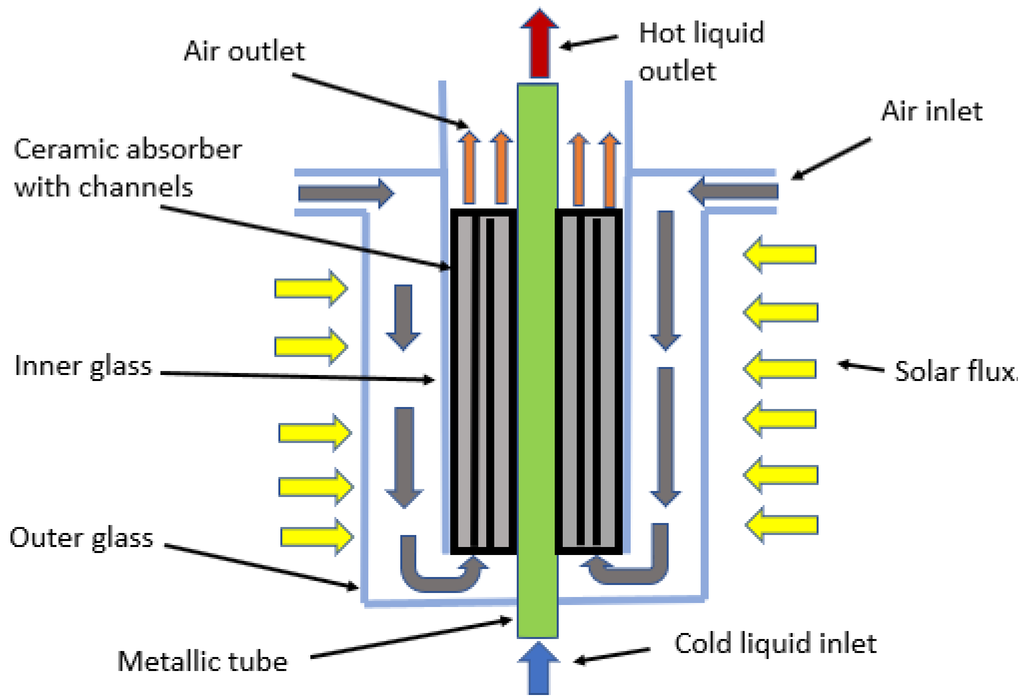

Figure 3-8 Solar receiver flow dynamics

Solar radiations after reflecting from the heliostat field strike the outer glass, where it is transmitted to the second glass, which then further transmits flux to the absorber. The flux is absorbed by the absorber, which causes it to heat up. Convection and radiation losses are protected by both glass covers. Pressurized air enters the receiver through the outside periphery, runs between the two glasses for the length of the receiver, and then enters the ceramic absorber located inside the inner glass. The air channels in the absorber are cylindrical in shape, and air passes through them. Air is heated as it passes through the absorber, and this energy is extracted from the receiver.

3.4 Analysis methodology

The objectives of proposed design, either it satisfied or not are done using numerical scheme. Thermal behavior and flow nature of fluids are essential physical phenomenon to analyze under real set conditions. The nature of physical system can be easily pre-judge by utilizing or solving different set of governing equations and boundary conditions, which although required high computation power. The numerical analyses give details of thermal and hydraulic behavior of receiver under pre-determine working conditions, which are essential to design process. The numerical analysis is also considered as finite element analysis (FEA). The finite element analysis further utilizes major three methods such as 1) finite volume method, 2) finite difference method and 3) finite element method.

The analysis of proposed (new) design for solar power tower receiver in this study is done using **finite volume method**. The basic step for finite volumetric method deploys geometry development, domain discretization (meshing) and solving set of equations with some best suited solving scheme. The assumptions which are taken in account for analysis in current study are as . 1) The flow inside the absorber is in a steady condition. 2) The properties of the absorber are homogeneous and isotropic. 3) Heat transfer at the receiver's outer surface is ignored in favor of the thermal behavior of the absorber material and heat transfer fluids. 4) Because the connection between the absorber and the metallic tube is so tight, thermal resistance between them is ignored. 5) For each temperature range, the thermal characteristics of heat transfer fluids are assumed to be constant (averaged).

3.4.1 Physical geometry

The analysis for whole receiver needs a lot of computation power under couple simulation scheme, so the overall geometry simplified to a geometry which is shown in a figure 3-9. This geometry only provides details of internal heat transfer mechanism and thermo hydraulic behavior of receiver. As the present geometry neglects external heat transfer mechanism and it is assumed that re emissions are completely absorbed by external glazing and suffers total internal reflection phenomenon. External effects either forced convections or natural convections are completely neglected due to double glazing separations between absorber and environment.

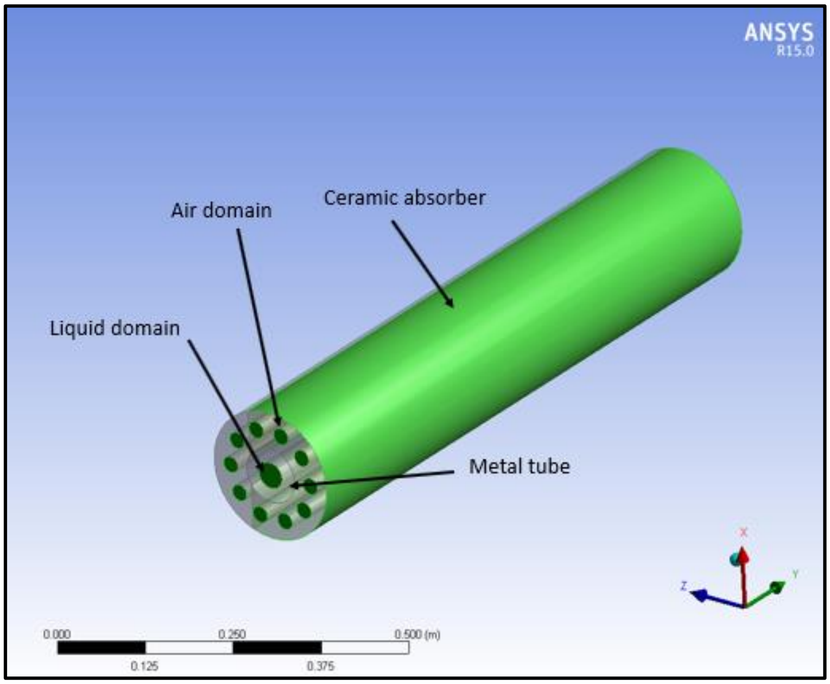

Figure 3-9 Analysis geometry

3.4.2 Governing equations

The domain of geometry is completely discretized by Finite volume method. Pressure and velocity terms are solved by SIMPLE algorithm's and second order upwind scheme is deployed. Set of equations in this study includes Energy equation, Continuity equation, momentum equations and K-ω turbulence equation. These equations are given as follows.

1. Continuity equation.

$$\frac{D\rho}{Dt} + \rho(\nabla . V) = 0 \qquad \text{---------- (1)}$$

- ρ Density
- V velocity

By considering density of fluid is constant, the continuity equation reduces to equation (2) for Cartesian coordinates.

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \qquad \text{--------- (2)}$$

for cylindrical coordinates.

$$\frac{\partial V_r}{\partial r}+\frac{V_r}{r}+\frac{\partial V_\theta}{\partial \theta}+\frac{\partial V_z}{\partial z}=0 \qquad \text{--------- (3)}$$

2. Momentum equations.

Momentum equation in r-component.

$$\rho\left(\frac{\partial V_r}{\partial t}+V_r\frac{\partial V_r}{\partial r}+\frac{V_\theta}{r}\frac{\partial V_r}{\partial \theta}-\frac{{V_\theta}^2}{r}+V_z\frac{\partial V_r}{\partial z}\right)=-\frac{\partial p}{\partial r}+\mu\left(\frac{\partial^2 V_r}{\partial r^2}+\frac{1}{r}\ \frac{\partial V_r}{\partial r}-\frac{V_r}{r^2}+\frac{1}{r^2}\frac{\partial^2 V_r}{\partial \theta^2}-\frac{2}{r^2}\frac{\partial V_\theta}{\partial \theta}+\frac{\partial^2 V_r}{\partial z^2}\right)+\rho g_r \qquad \text{-------- (4)}$$

Momentum equation in θ- component.

$$\rho\left(\frac{\partial V_\theta}{\partial t}+V_r\frac{\partial V_\theta}{\partial r}+\frac{V_\theta}{r}\frac{\partial V_\theta}{\partial \theta}+\frac{V_r V_\theta}{r}+V_z\frac{\partial V_\theta}{\partial z}\right)=-\frac{1}{r}\frac{\partial p}{\partial \theta}+\mu\left(\frac{\partial^2 V_\theta}{\partial r^2}+\frac{1}{r}\ \frac{\partial V_\theta}{\partial r}\ -\frac{V_\theta}{r^2}+\frac{1}{r^2}\frac{\partial^2 V_\theta}{\partial \theta^2}+\ \frac{2}{r^2}\frac{\partial V_r}{\partial \theta}+\frac{\partial^2 V_\theta}{\partial z^2}\right)+\ \rho g_\theta \qquad \text{--------- (5)}$$

Momentum equation in z-component

$$\rho\left(\frac{\partial V_z}{\partial t}+V_r\frac{\partial V_z}{\partial r}+\frac{V_\theta}{r}\frac{\partial V_z}{\partial \theta}+V_z\frac{\partial V_z}{\partial z}\right)=-\frac{\partial p}{\partial z}+\mu\left(\frac{\partial^2 V_z}{\partial r^2}+\frac{1}{r}\ \frac{\partial V_z}{\partial r}+\frac{1}{r^2}\frac{\partial^2 V_z}{\partial \theta^2}+\frac{\partial^2 V_z}{\partial z^2}\right)+\rho g_z \qquad \text{---------- (6)}$$

3. Energy equation.

$$\rho c_p\frac{\partial T}{\partial t}+\rho c_p\left(V_r\frac{\partial T}{\partial t}+\frac{V_\theta}{r}\frac{\partial T}{\partial \theta}+V_z\frac{\partial T}{\partial z}\right)=k\left[\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right)\right)+\frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2}+\frac{\partial^2 T}{\partial z^2}\right]+\mu\varphi \qquad \text{---------- (7)}$$

- $\mu\phi$ is a viscous dissipation function.

4. Reynolds number correlation.

$$Re=\frac{\rho u L}{\mu} \qquad \text{----------- (8)}$$

- ρ is density.
- u is flow velocity.
- μ dynamic viscosity.
- L characteristic length.

3.4.3 Grid independent test

Finite element analysis accuracy depends upon many factors such as time step and element size. By keeping other variables constant, accuracy of results changed a lot by varying grid sizes. The results of FVM analysis should be independent of element size. In this study structural mesh with hexagonal elements are generated overs physical domain. Convergence criterion is set to be 10^{-6} for simulations in all cases. Grid independent test is performed by changing element lengths. So, it is concluded that after doing grid independent test, the 2mm length of grid size gives accurate results with optimum computation power. Results of grid independent test are represented in a form of graph, which is shown in a figure 3-10. The overall 6123468 elements and 6142000 nodes are used for each case analysis.

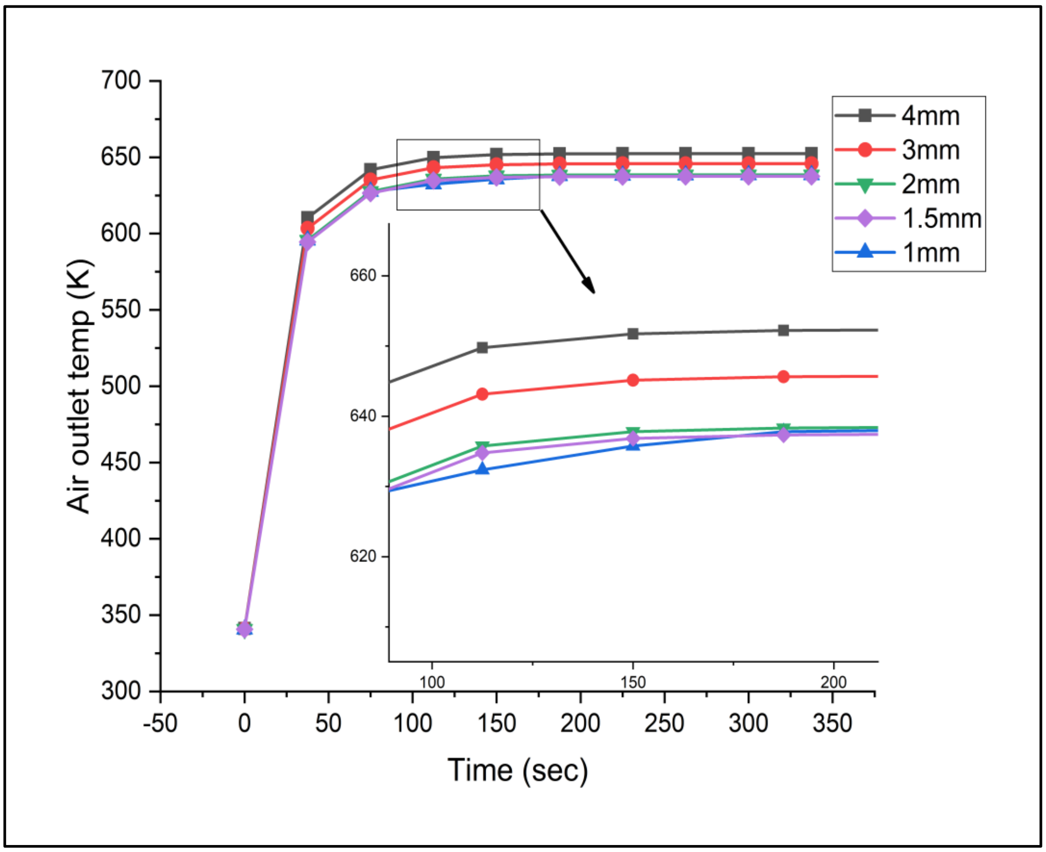

Figure 3-10 Results of grid independent test

Summary

This chapter include details of receiver design and analysis strategy, which is utilized for numerical scheme. The design parameters and dimensions with detail mechanical drawings are given in first part with their respective materials used. The overall mechanism for heat transfer and hydraulics for heat transfer fluids are discussed briefly. The physical geometry with numerical analysis details is given in this chapter such as FEA method, governing equations, solution scheme and turbulence model, as these are essential for the analysis of proposed receiver. Grid independent test which is a key parameter for numerical analysis is performed and its details are given in last section of chapter.

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Chapter 4: Results and Discussion

The results obtained after doing numerical analysis of proposed receiver has too much significance for proper receiver working and deployment of respective design. Interpretation of obtain results mark a major impact on development of smart design under harsh working conditions. The behavior of receiver is analyzed by its thermal efficiency, and thermo hydraulic nature of heat transfer fluids with absorber domain interaction. The results obtained for the proposed solar power tower receiver are included in following sections.

4.1 Thermal behavior of absorber

Thermal effects of absorber mark as a key parameter for receiver continuous and stable working. Usually, receiver suffers non-homogenous temperature distributions which further deteriorates overall plant performance. Absorber is considered as the major component in central receiver design because it directly converts the solar flux into thermal energy and works at a temperature more than 500 ℃. In this study behavior of absorber is analyzed under static and dynamic thermal loadings.

4.1.1 Static thermal analysis

The material of absorber is ceramic, and its catastrophic failure temperature is considering to be 1600 K or 1500 ℃ [1]. The temperatures of absorber should be less than pre-determined temperatures limit. Static temperature test is important because in this test absorber attains the maximum temperature under same solar flux condition/ working conditions, which is extremely important for receiver operations under critical state of blockage of heat transfer fluid or solidification of molten salts inside inlet header or outlet header. The temperature contours for proposed receivers' absorber are shown in a following figure 4-1. The result of static thermal loading shows that absorber is completely behaving normal and at its maximum temperature is well below the catastrophic failure temperature. The top side attains higher temperature as compared to bottom side because flux directly interact with top side, but the material has attained quite good temperature homogenization. The maximum temperature attains is 1350 K, which quite below the failure temperature limit 1600 K. The minimum temperature attains at back end is 1332

K. The temperature difference between these two sides is almost 18 K, which is important and quite beneficial for smoother and safer working/operation. The less temperature gradient is significant for avoiding local hot spots in absorber.

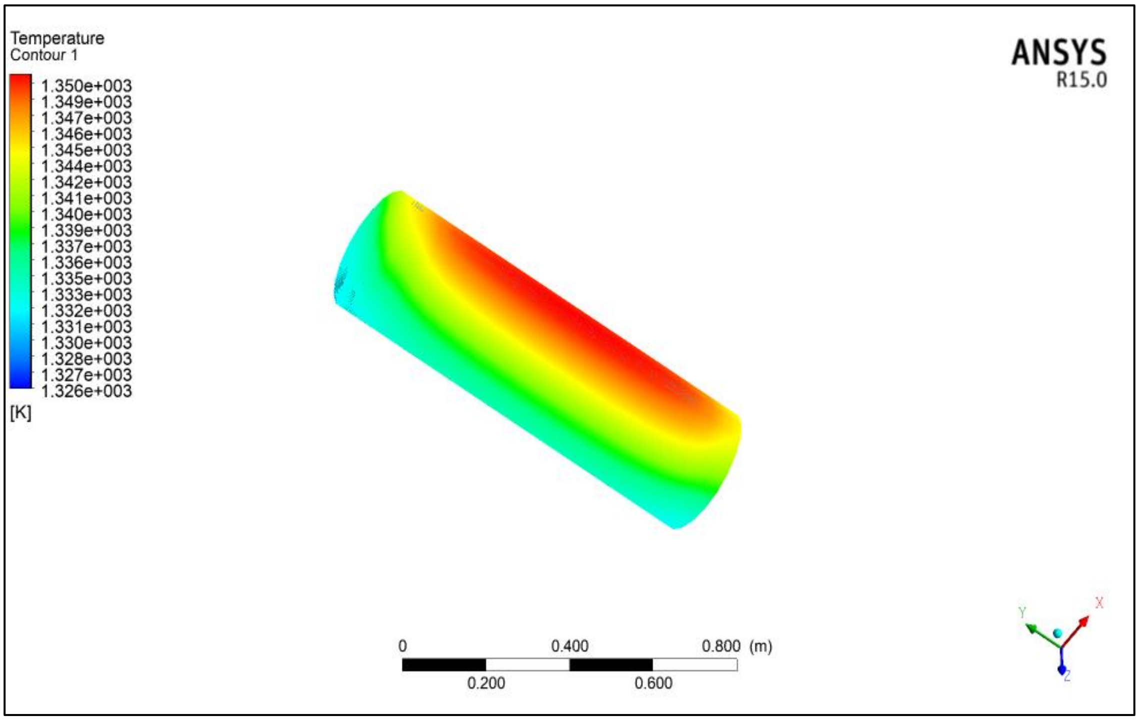

Figure 4-1 Static thermal test temperature contours

4.1.2 Dynamic thermal analysis

Dynamic test resembles the original working condition. This test results shows the behavior of receiver under normal working and gives details of heat transfer fluid behaviors relative absorber temperature distribution at some respective time of inspection. Dynamic nature of absorber is another important parameter for absorber working. So dynamic test shows the temperature trends either absorber temperature rises or decreases during the dynamic loading/conditions. The temperature of absorber should be less than the failure temperature for normal and safer operation. The behavior of absorber under dynamic conditions are shown in a figure 4-2. The temperature trends are circumferentially uniform or homogenous while temperature trends change axially but under some uniform trends. The maximum temperature attain in absorber under dynamic conditions is 1174 K at the outlet of heat transfer fluids, while the minimum temperature attains in analysis is 893 K at inlet of absorber.

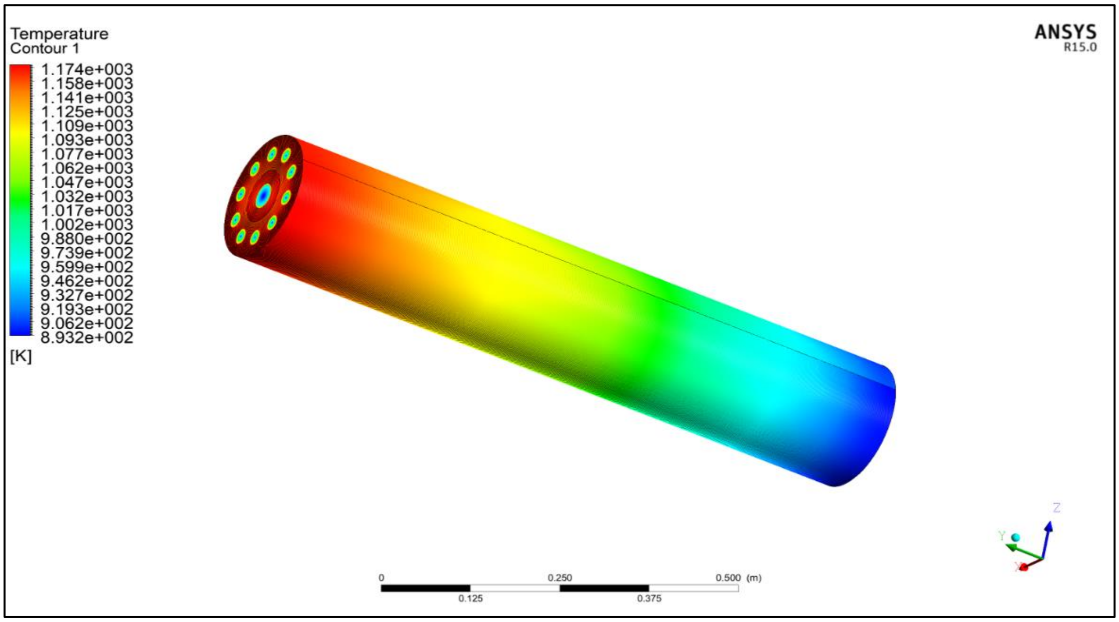

Figure 4-2 Dynamic loading test temperature contours

The comparison between the static and dynamic temperature contour are shown in figure 4-3.

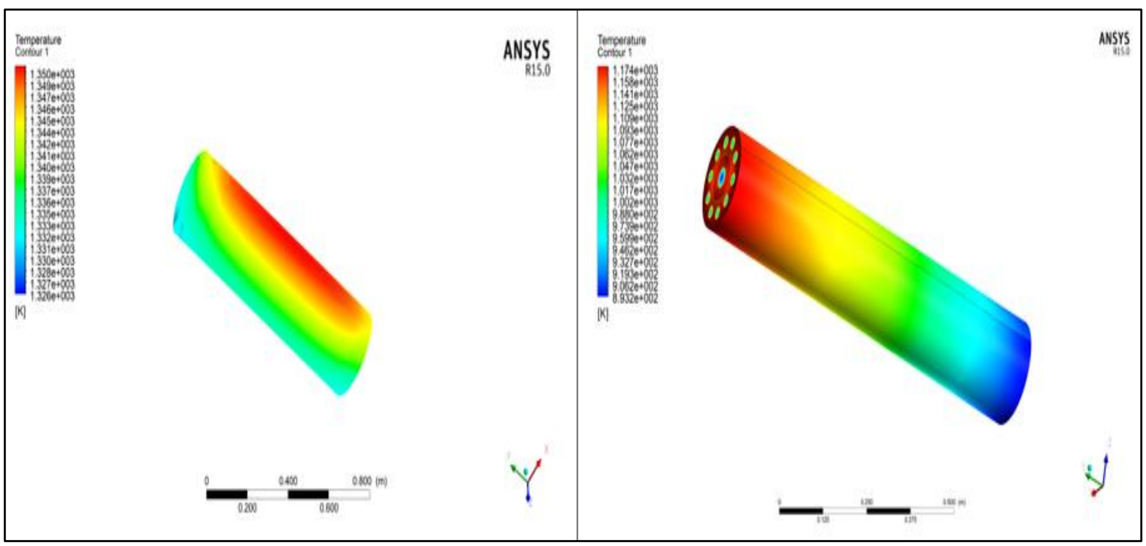

Figure 4-3 Static and dynamic temperatures comparisons

4.1.3 Heat transfer fluid effects on absorber

Interference of heat transfer fluid with absorber wall is considered important for receiver performance and heat transfer mechanism. The overall heat transfer inside absorber although depends on the behavior of thermal performance of working fluid. The temperature trends longitudinally and circumferentially consider critical for absorber performance, relative to type of heat transfer fluid used. In present study three different heat transfer fluids are being used in combination with pressurized air such as therminol oil, molten salt (solar salt) and molten metal (liquid sodium).

Since the heat flux interacts directly with the ceramic absorber, its temperature alters across time and space. Temperature fluctuations in the absorber are also influenced by the thermal parameters and mass flow rate of the heat transfer fluid. Temperatures are lower at the beginning of the absorber than at the end. At the entrance point of the absorber, the pressured air and heat transfer fluids are at their lowest temperatures, therefore it absorbs more heat than the end side. Figure 4-4 illustrates the temperature trends of the ceramic absorber along its length after 200 seconds, when the findings have reached a stable state.

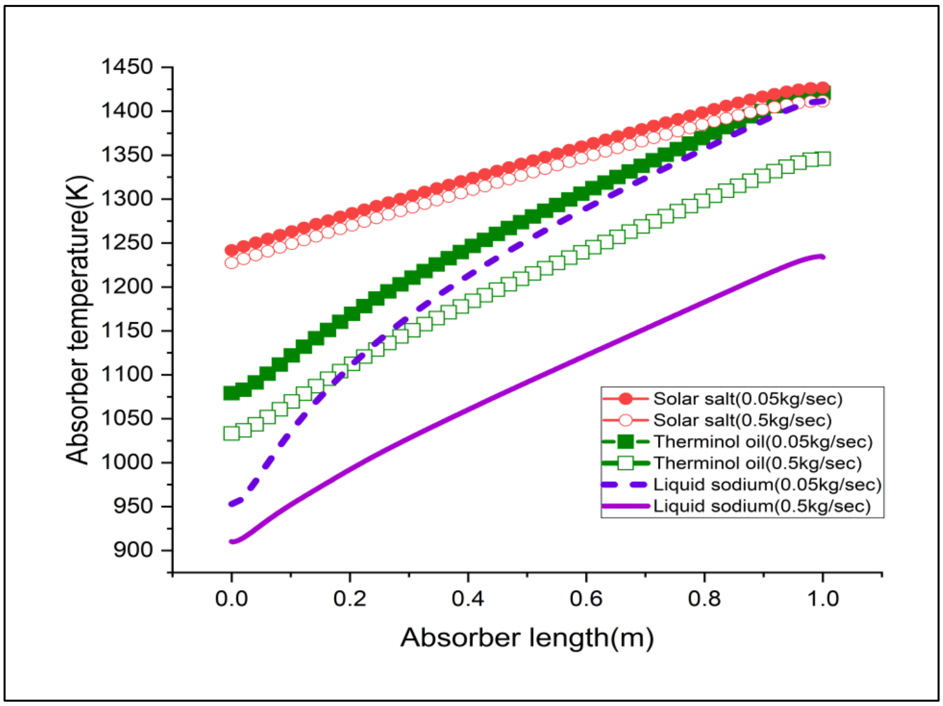

Figure 4-4 Ceramic longitudinal temperature trends

It is clearly shown in above figure 4-4 that each heat transfer fluid has different effects on absorber temperatures with respect to altering mass flow rates. In solar salt case, absorber entry temperature is high as compared to other cases, while its changing (higher) mass flow rate do not drop the entry temperature significantly. For therminol oil case, the changing temperature trends behavior is quite intermediate in between the other two cases. Entry temperature of absorber for therminol oil is well below relative to outlet and higher mass flow of solar salt. Higher mass flow rate of therminol oil case clearly shows that the temperature drops at inlet more but less significant as compared to outlet of absorber where temperature of absorber remarkably drops. In last case of liquid sodium, it is proving very significant and more adoptable heat transfer fluid for power tower receiver sufficiency. The inlet temperature for lower mass flow rate of liquid sodium is quite below as compared to same mass flow rate of other two cases and outlet temperature for absorber at higher mass flow rate of liquid sodium is dropping absorber temperature drastically.

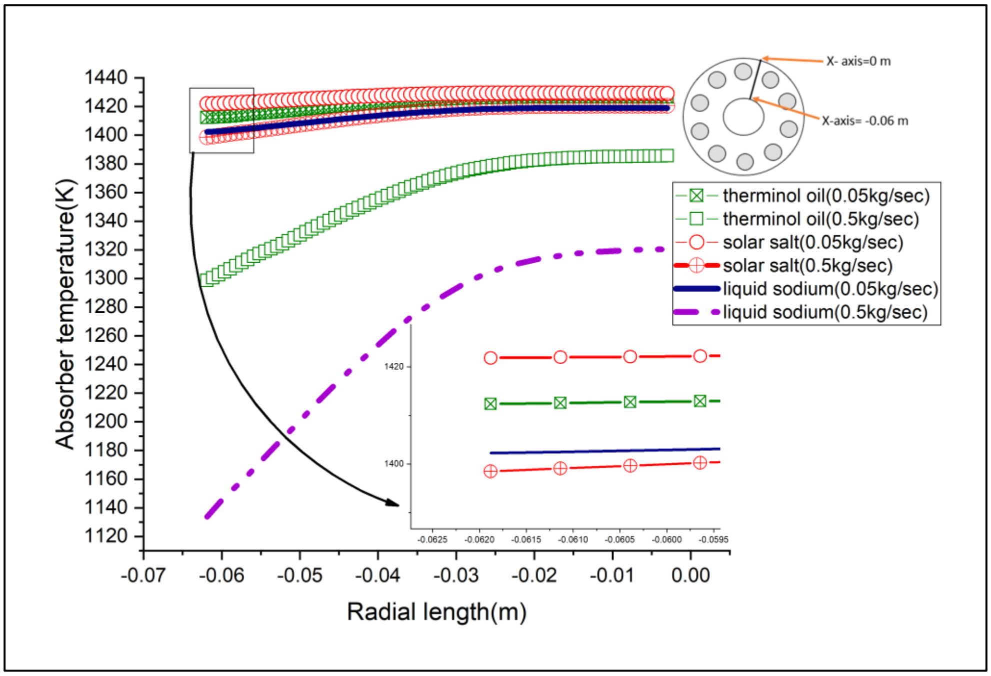

Figure 4-5 Ceramic radial temperature trends

The temperature of absorber is higher at the end section of absorber for each case. So, it is necessary to analyze the temperature trends along the cross-section of end side of absorber. The temperature rise of the absorber drives the stronger convective and radiative losses. The highest absorber temperature is measured in the case of solar salt, whereas the lowest absorber temperature is measured in the case of liquid sodium. Furthermore, as compared to therminol oil and solar salt, when the mass flow rate of liquid sodium increases, the absorber temperature drops tremendously. Temperatures are rising on the absorber's outer surface and increasing the risk of material breakdown. Figure 4-5 presents the temperatures along the radius at the absorber's end side, to counteract the impacts of heat generation inside the air channels.

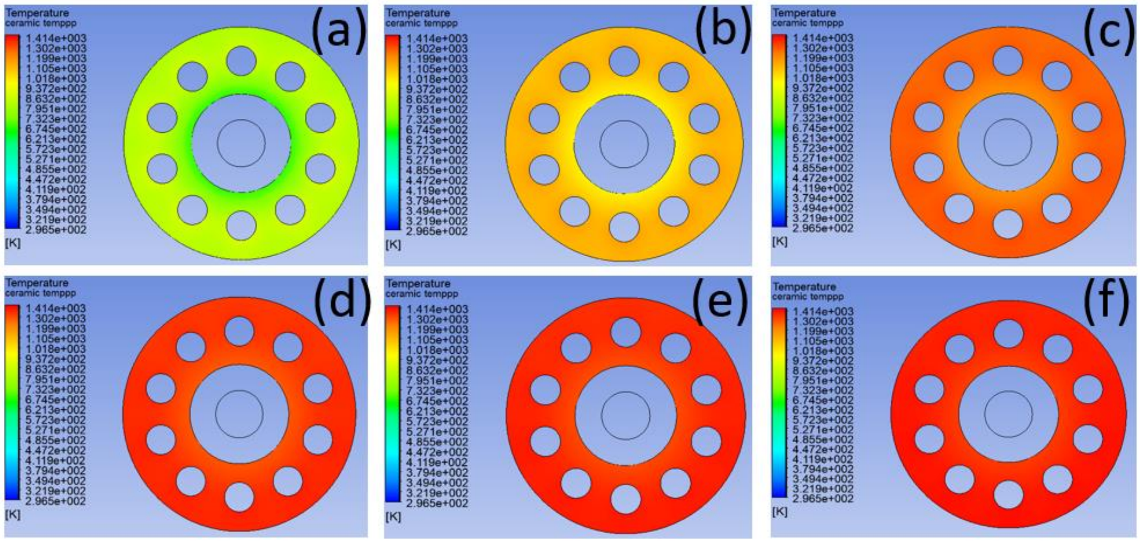

Figure 4-6 Ceramic transient temperature contours

From figure 4-5, it is shown that as heat moves towards the core of the receiver, the temperature at the inner radius of the absorber is lower than at the outside surface. The temperature rises near the air passages before dropping slightly at the outer surface. This is due to eddy viscosity, which enables heat to be generated within air channels. The temperature distribution along the radius is governed by the mass flow rate. The temperature distribution for each example is reduced as the bulk flow rate increases. Enhanced mass flow rate of solar salt is not as effective as increased mass flow rate of liquid sodium, which resulted in a major drop in temperature. Temperatures of the absorber alter throughout time, becoming consistent after 200 seconds.

The analysis of proposed receiver is transient in nature, in order to get time dependency nature of receiver. The outlet cross-sectional temperature contours for receiver absorber are presented in figure 4-6. The results completely depict the steadiness of all critical parameters of receiver after 200 seconds of flow time.

4.2 Thermo-hydraulic effects of pressurized air

Pressurized air behaves as a primary heat transfer fluid inside a receiver, so its behavior inside receiver is too significant to predict the overall performance. Basic function of central receiver is to provide air at enough temperature to perform continuous desire operation under pre-determined conditions. Heat transfer between air and absorber wall depends upon Reynolds no and nature of flow type, these effects will be discussed in following sub sections.

4.2.1 Thermal behavior of air

Behavior of pressurized air inside milli channels depends upon development of thermal boundary layer and velocity boundary layer. In order to have an idea of air outlet temperature after passing through the receiver, the temperature trends are presented in a figure 4-7.

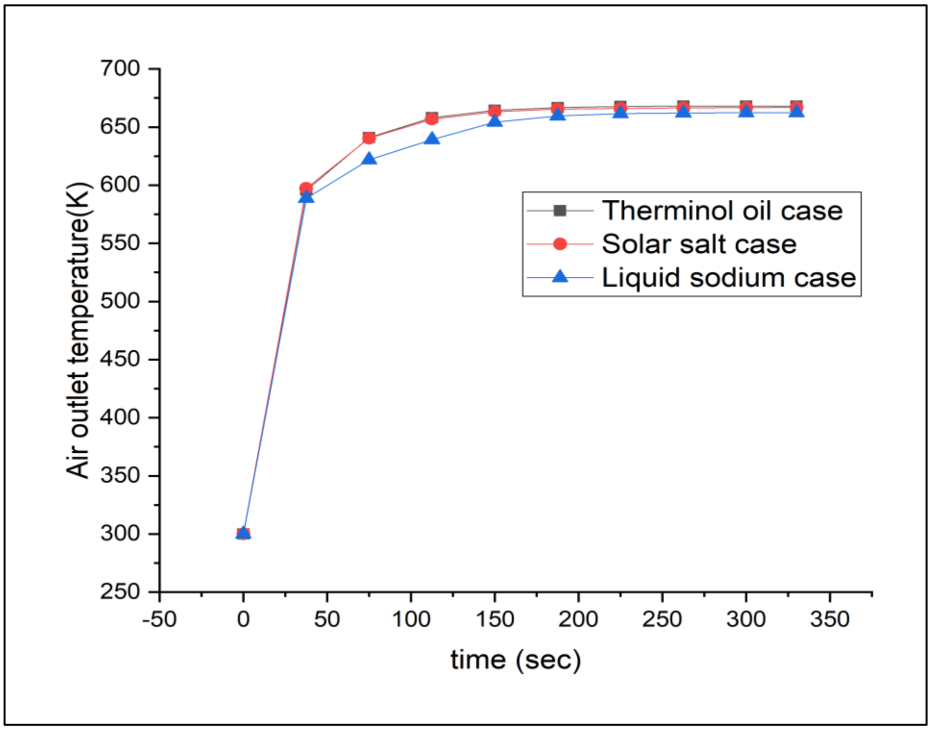

Figure 4-7 Outlet air temperature trends

It is shown in a graph of figure 4-7 that for each case, the thermal behavior or outlet temperature of pressurized air is almost same and gets uniform after an interval of 200 seconds of flow time. The achieved outer temperature for pressurized air is about 650 K.

4.2.2 Thermal boundary layer development inside the air domain

As air domain consists of milli channels, so the thermal boundary layer development can be predicted by following the behavior of streamlines inside any of milli channels. The temperature trends for streamlines inside an air channel are graphed in a figure 4-8. The five streamlines at different distances from channel wall are being analyzed.

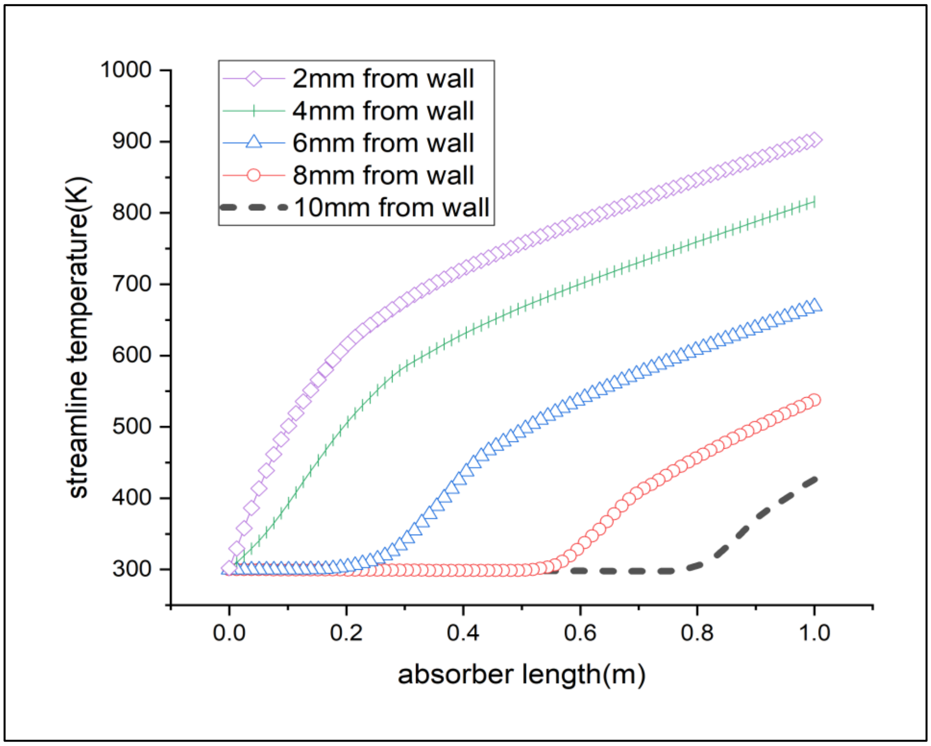

Figure 4-8 Thermal boundary development for air domain

The above shown results are taken at a flow time of 200 seconds. It is shown clearly that the streamline which is near to wall has a temperature change effects near the start of channel and as streamlines get away from wall, these have temperature change effects after some starting length respectively. The streamline which is at the center of channel has a temperature change effects at a length of 800 mm from the start of air channel. This trend of center line shows that thermal boundary layer has developed completely at a length of 800 mm from start of channels and now flow is completely thermally developed flow inside an air domain.

4.2.3 Velocity layer development inside the air domain

Nature of fluid flow has a significant effect on the heat transfer and pressure drop inside the milli channels. The analysis of a receiver is dependable on the nature of fluid flow with respect to mass flow rate which is considered as constant as mentioned in above chapter. The procedure of predicting the development of velocity boundary layer, the scheme technique is used as before in case thermal boundary layer development prediction. The velocity change profile of each streamline completely shows the flow nature with respect to channel length as given in figure 4-9.

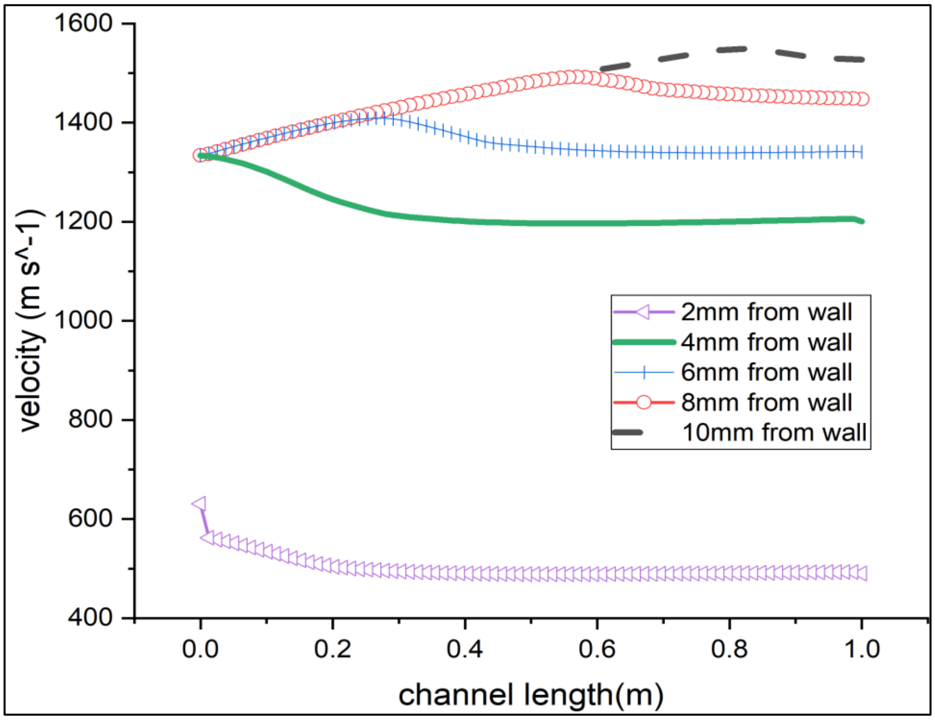

Figure 4-9 Streamlines velocity profile

It is clearly shown in above result that streamline which is very near to wall has a no slip condition and it is in a complete stagnation state. As wall distance increase, the velocity of streamlines increases relative to their respective distances and now it is observed that the streamline which is at center has more increase in velocity compared to near wall streamline but after 800 mm from the starting point there is small drop of velocity in center streamline, which is an indication of complete development of boundary layer and some back pressure with eddy effect inside boundary layer.

Velocity trends of streamlines are also supported with dynamic pressure trends which are mentioned in a figure 4-10.

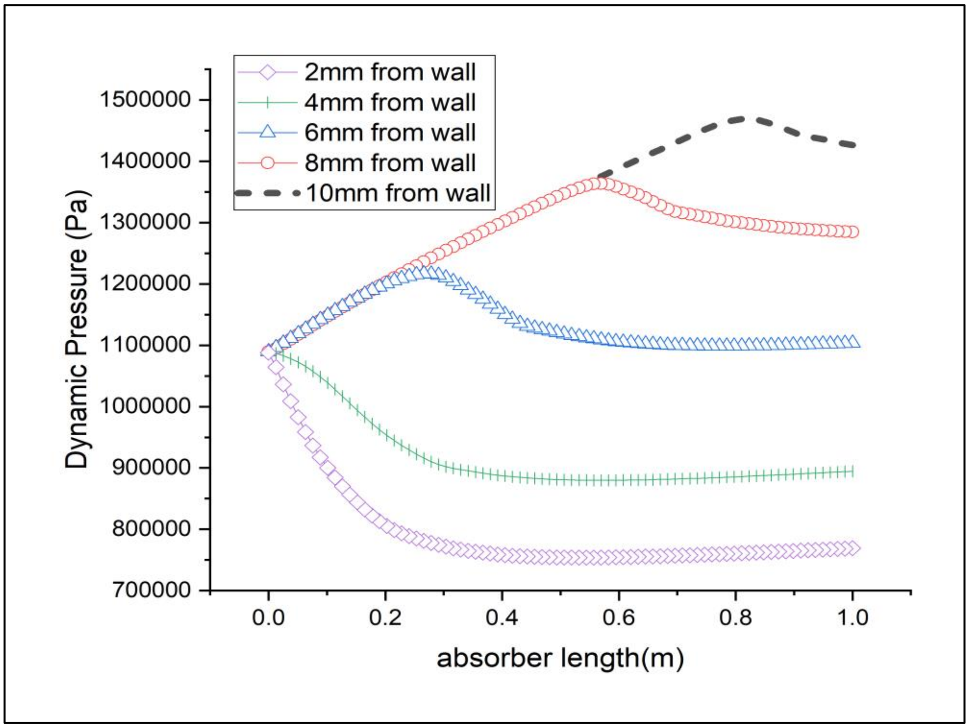

Figure 4-10 Dynamic pressure trends

For receiver performance and compressor power, the pressure drop inside air channels is an important parameter. The dynamic pressure of the streamline near the wall is found to be significantly lower. The dynamic pressure of the streamline near the channel's center has grown. The velocity profile of air inside the channel is also linked to dynamic pressure. Figure 4-10 shows that air has increased dynamic pressure in the channel's middle while maintaining stationary towards the wall. The relative total pressure within air channels varies both radially and longitudinally. Because of wall friction and fluid viscosity, relative total pressure of streamlines near the wall has a sharper pressure drop than the center of the channel, as seen in figure 4-11. With flow along the absorber length, the viscose effect increases towards the walls and reaches the channel center at a length of 800 mm. This indicates that the velocity boundary layer has emerged and that the flow has reached its full potential after 800 mm. Because of viscous forces, streamlines near walls experience detrimental pressure gradients. Due to eddy propagation, the viscosity effects become quite strong.

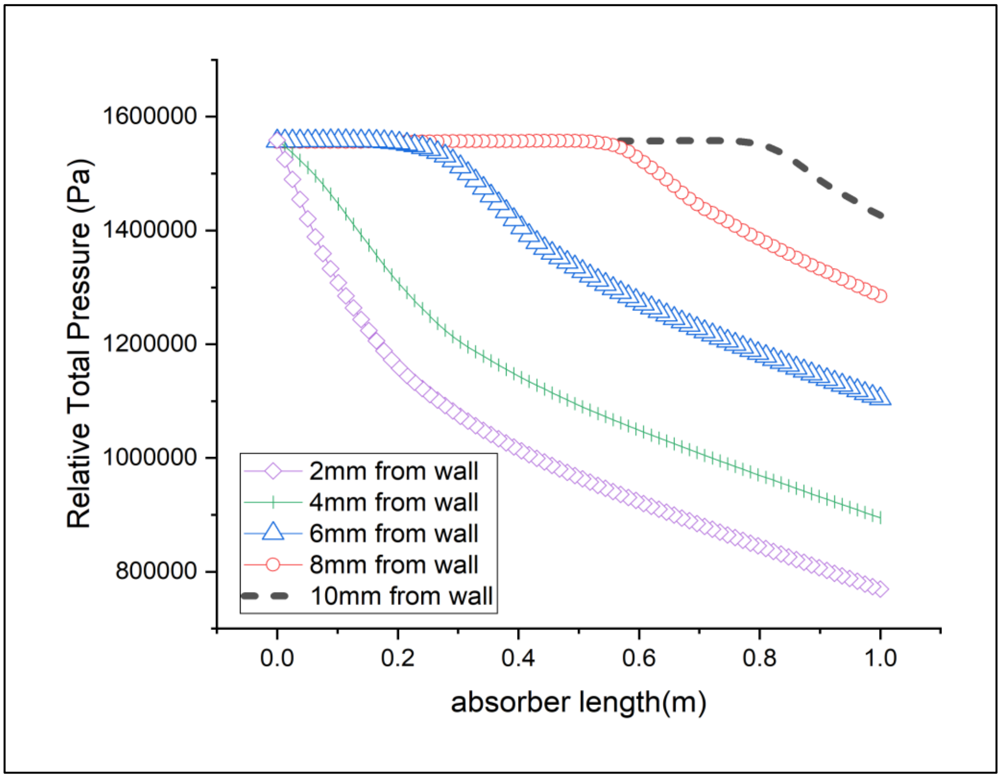

Figure 4-11 Relative pressure trends

The nature of fluid flow and the streamline distance from the wall have a significant impact on eddy propagation within the boundary layer. Figure 4-12 depicts eddy generation as a function of channel length. Streamlines close to the channel wall do not create a lot of eddy viscosity, whether the flow is in transition or turbulence, because they are entirely linked to the wall. Streamlines 4 mm away from the wall are in the transition zone between 0.2 m and 0.4 m, where eddy effects are modest, but after 0.4 m, they enter the turbulence zone, where eddy effects are substantial. At a length of 0.3 m, streamlines 6 mm away from the channel wall enter the transition area directly; beyond this length, the boundary layer height is less than 6 mm, and after 0.4 m, the eddy viscosity effects dramatically increase. At 0.5 m from the entry, streams 8 mm from the channel wall enter the turbulence region of the boundary layer. For this streamline, the eddy viscosity effects are too significant. At 0.8 m from the inlet, a streamline 10 mm from the wall (near the center) enters the turbulence region, where the eddy viscosity curve is too sharp. Because the boundary layer height is less than 10 mm beyond 0.8 m, eddy effects for the streamline near the channel center are zero.

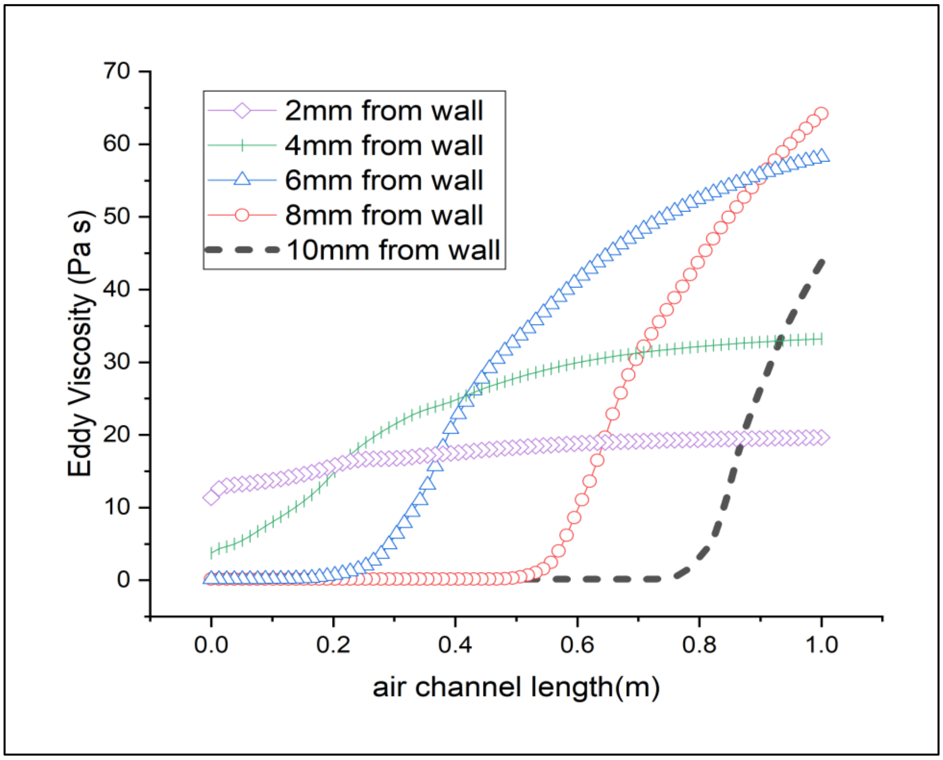

Figure 4-12 Eddy propagation in air domain

Eddies in fluid can also be used to generate energy. Turbulence kinetic energy is transformed to heat as the distance from the air channel wall increases. Figure 4-13 depicts the heat generation as a function of channel length. Streamline close to the wall produces increased turbulence and becomes a source of heat generation from the start of the flow. When streamlines go any further away from the wall, they produce no heat, but when turbulence effects kick in, each streamline produces heat, which is dependent on how early the streamline interacts with the turbulence region of the flow. In comparison to other streamlines, one towards the channel's center generate less heat.

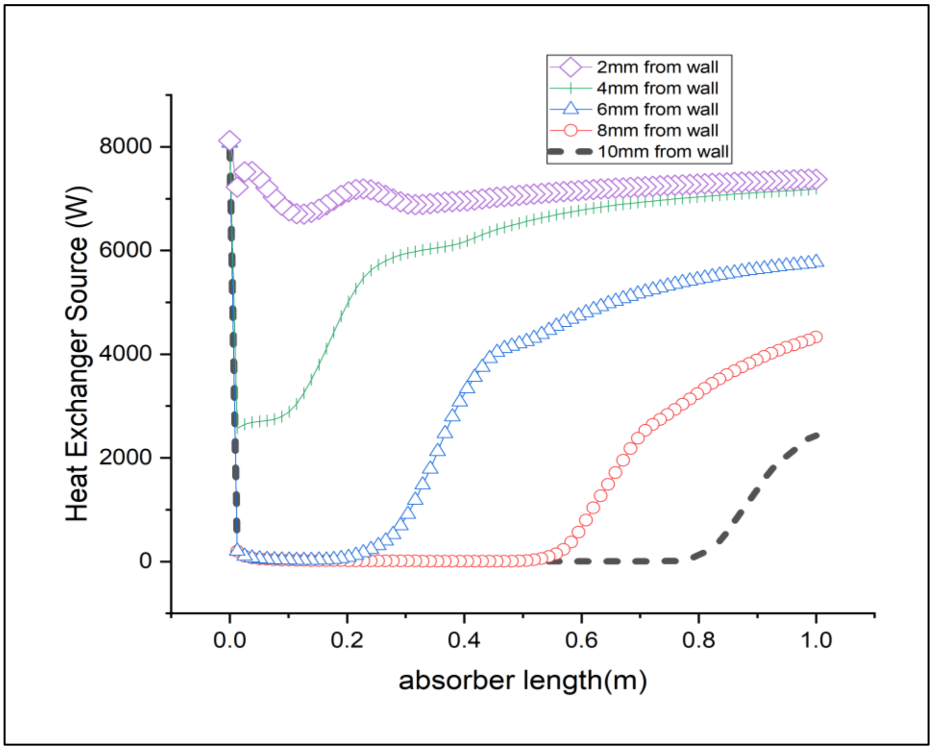

Figure 4-13 Energy generation in air domain

Temperature contours for air domain at outlet after 200 secs of flow time is shown in figure 4-14.

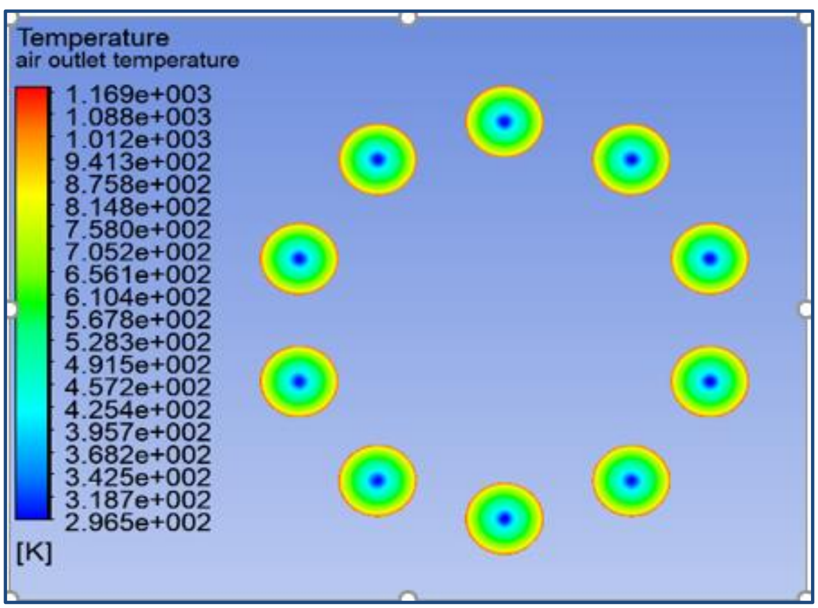

Figure 4-14 Temperature contours of air at outlet

The cross-sectional outlet temperature for pressurized air for different cases is shown in a figure 4-15.

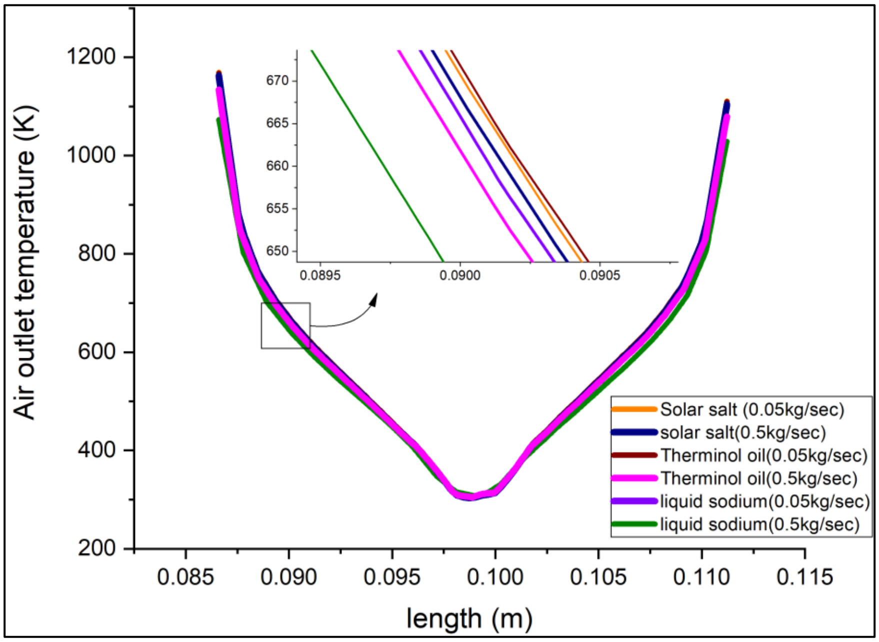

Figure 4-15 Air outlet temperatures

4.3 Thermal behavior of heat transfer fluid

The proposed receiver utilizes heat transfer fluid parallel to pressurized air, to extract more heat from receiver. The analysis is carried on three different heat transfer fluids to get an idea of best suitable heat transfer fluid for receiver performance. The utilized heat transfer fluid is therminol oil, molten salt (solar salt) and molten metal (liquid sodium). The overall performance of receiver is highly dependent on thermal behavior of heat transfer fluid. The temperature trends for different utilized heat transfer fluids relative to their mass flow rates are presented in a graph of following figure 4-16.

Figure 4-16 depicts the outlet temperatures of various heat transfer fluids in relation to their mass flow rates. In contrast to other cases, the outlet temperature of liquid sodium is high. Figure 4-4 illustrates that the absorber temperature is lower when the mass flow rate of liquid sodium is 0.5 kg/sec. Thermal losses would be reduced if the absorber temperature remained lower. The overall efficiency of the proposed absorber is influenced by several factors, including the temperature of the absorber, the temperature of the outside surface, and the temperature of the heat transfer fluid outlet. External double glazing of the receiver lowers convection and radiation losses. As a result, the efficiency of the proposed receiver is solely determined by the temperature of the pressured air output and the temperature of the heat transfer fluid outlet temperature.

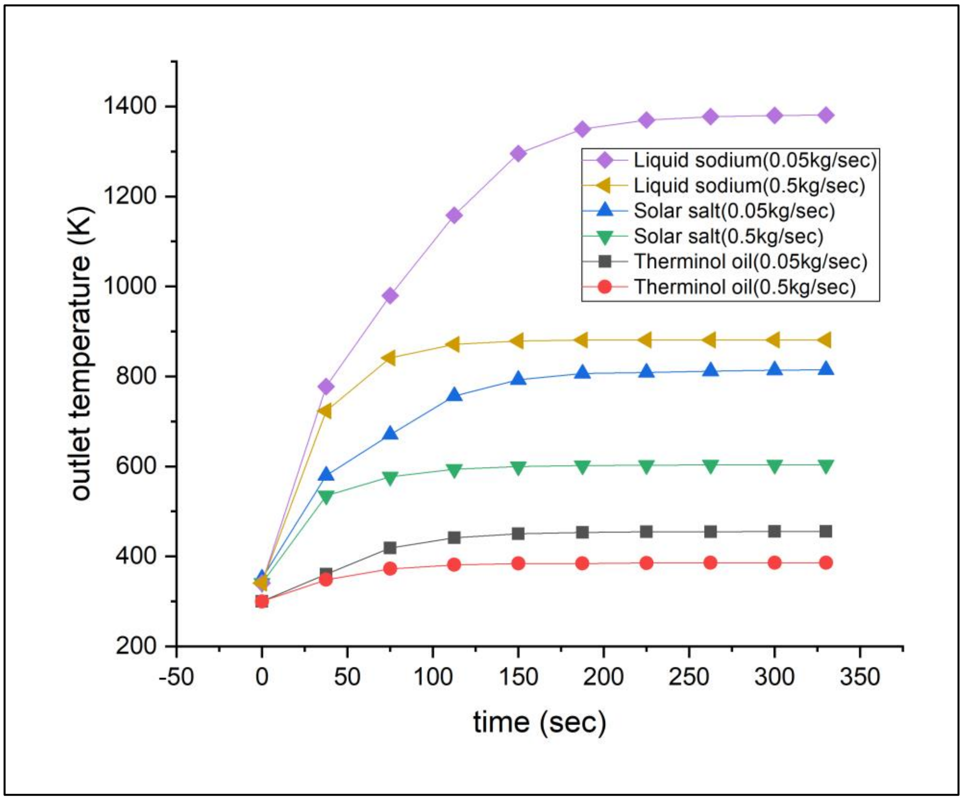

Figure 4-16 Heat transfer fluid temperatures

It is clearly shown in graph that liquid sodium is attaining outlet temperature more than 800 K at different mass flow rates, which is quite beneficial in thermal efficiency. While therminol oil attains less outlet temperature as compared to solar salt, which has intermediate temperature trends relative to other cases. Therminol oil enters at room temperature relative to other cases, which is more advantageous to other scenarios, as it reduces the cost of trace heating which is necessary for other cases like molten salt and molten metal.

The outlet temperature trends for different heat transfer fluids with respect to varying mass flow rates are depicted in a figure 4-17.

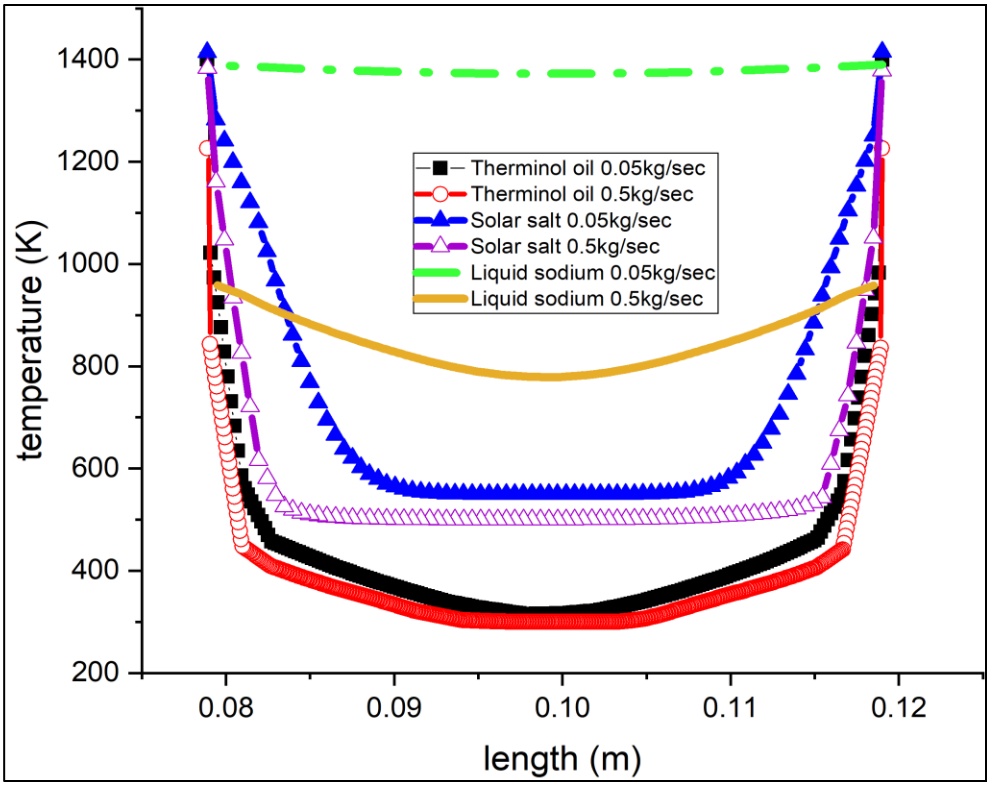

Figure 4-17 Heat transfer fluid outlet temperatures

The temperature contours for heat transfer fluids used in analysis are shown in a figure 4-18 at a flow time of 200 secs.

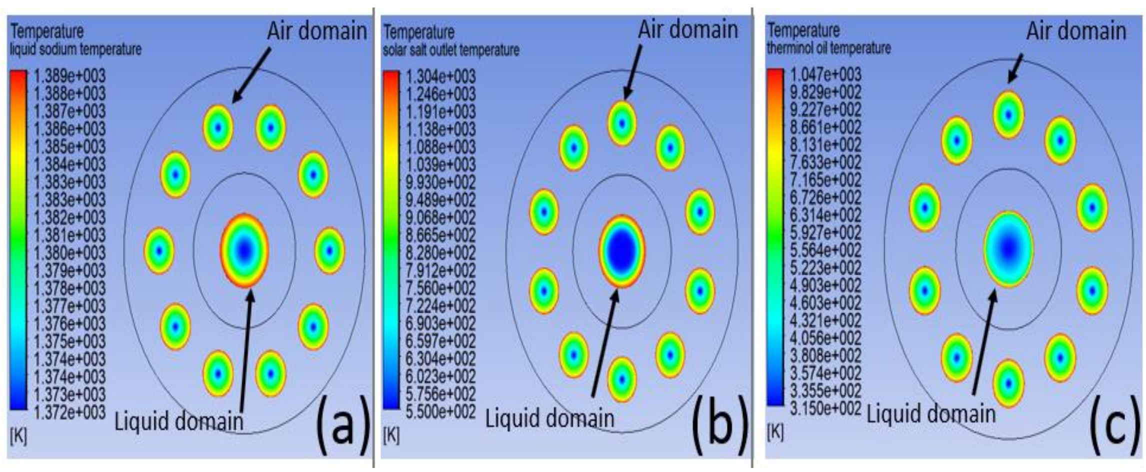

Figure 4-18 Temperature contours (a) Liquid sodium (b) Solar salt (c) Therminol oil

The performance of receiver in terms of thermal efficiency is necessary for prediction of receiver performance. In order to have an idea of heat losses the second law efficiency plays key role relative to heat transfer fluid outlet temperatures. The performance of proposed receiver is given in following table 4-1, in terms of thermal efficiency including heat absorbed by pressurized air and heat transfer fluid used, while viscous heat dissipation is removed from total heat gain.

Table 4-1 Thermal efficiency of Receiver

Sr no	**Cases**	**Thermal efficiency. η**
1.	Liquid sodium case.	84.1
2.	Solar salt case.	75
3.	Therminol oil case.	74.6

4.4 Receiver glass analysis

Quartz glass is used in proposed receivers. In current study, the optical analysis is not performed but there are some useful data which is necessary to mentioned in this chapter from literature. This data should be linked to chapter 5 for future work and recommendation section 5.2. Akihiko Sakamoto et al [2] study optical nature of quartz glass and provide the time dependent thermal properties graph at different temperature for varying wavelengths. Refractive index curve as a function of density is given, which is important for optical and total internal reflection analysis. Also T. Nishitani et al [3] provides the use full properties for optical analysis of quartz glass.

Summary

This chapter includes the details of obtain results which are considered compulsory for predicting the performance and behavior of solar tower receiver. Total results obtained after analysis are further characterized on the base of their relationship with the thermal and thermo-hydraulic nature relative to pressurized air, absorber material and heat transfer fluid used respectively. The detail results of proposed receiver also give an information regarding comparison in between and reliability. The time variant analysis for different parameters like temperatures of absorber and outlet temperatures of working fluids are mentioned in detail relative to development of thermal and velocity boundary layers.

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Chapter 5: Conclusions and Recommendations

5.1 Conclusions

The suggested receiver design can feed high temperature working fluids for a variety of operations and processes. The second law indicates that the higher the working fluid temperature (source temperature), the higher the efficiency. Different heat transfer fluids have been investigated in this work based on a new central receiver design. In comparison to the other two heat transfer fluids, liquid sodium has the superior heat transfer efficiency. Solar salt has a lower thermal efficiency than liquid sodium, but it facilitates plant operation easier because the same working fluid can also be used for thermal storage. Therminol oil is nearly as effective as solar salt, but it has the advantage of not hardening within the operating temperature range, unlike the other two heat transfer fluids. In contrary to other available receiver designs, this one is modular. It can be scaled up to meet the needs of the application. The suggested design can be used as a preheater for boilers in power plants, combined cycle plants prior to the gas turbine, cement plants preparatory to the kiln, and process plants. The proposed system has the significant advantage of simultaneously providing two high-temperature heat transfer fluids for executing distinct major tasks.

5.2 Recommendations

The present study has provided a complete technical details of proposed receiver design. The feasibility of design suggests for its implementation and manufacturing, although the analysis design is simplified because of provided facilities. To achieve more accuracy in provided details for future work some recommendations are as follows

1. Radiation analysis over the proposed receiver geometry is to be done by a given applied solar flux, which is 10 KW/m^2. The obtain temperature trends then couple with the fluent. This will require high computation cost.
2. Separate optical analysis for double glazing and in combination with absorber for better understanding the total internal refection phenomenon and radiation absorbance with solid interface interaction.
3. Perform heat transfer analysis in between the glazing and input that outlet air temperature to inlet condition of milli channels for above mentioned coupled simulations.
4. Perform economic analysis of proposed receiver as it is shown that cavity for solar receiver has been completely removed by the new design idea. So, the major problem has been removed, which is usually faced in CSPs technology due to its capital cost and levelized cost of energy.

There are some other suggestions which are mandatory to enhanced performance, these are as follows.

1. Increase the numbers of milli channels with some decrease in diameter, at present it is 25mm.
2. Increase the pressure of air to have more heat transfer, in this study its 2 bar.

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Appendix A-Publication

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Numerical analysis of novel solar central receiver using different types of heat transfer fluids.

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ABSTRACT

Central receiver system (CRS) converts solar flux into thermal energy, apart from this energy conversion process it also carries radiative and convective heat losses. Volumetric receivers and tubular receivers are usually used in energy conversion systems. These receivers have multiple drawbacks according to plant/system overall performance and cost. Final output of concentrated solar power system highly depends on receiver efficiency. In order to improve thermal performance, a new design of receiver is purposed to counter the energy losses with maximum heat transfer. This conceptual design utilizes milli channels and multi flow of working fluids, to enhanced thermal efficiency. Respective design is analysed numerically with different heat transfer fluids. A precise analysis is done on different flow parameters such as temperature, pressure and turbulence kinetic energy. The analysis of receiver considers transient effects as well as spatial effects. After that, results show that liquid sodium increases thermal efficiency of proposed solar receiver as compared to solar salt and therminol oil.

KEYWORDS: Solar power tower (SPT), Central receiver design, Liquid sodium heat transfer fluid (HTF), Thermal analysis.

1 INTRODUCTION

Solar power tower(SPT) is usually divided into three coupled systems such as heliostat field collector(HFC), solar tower and power block(Tian and Zhao 2013; Behar, Khellaf, and Mohammedi 2013). Central receiver (CR) and power tower (PT) are considered to be same, which utilizes heliostat field collector (HFC) for solar radiations collection(Behar, Khellaf, and Mohammedi 2013; Zhang et al. 2013).Central receiver can further be classified as solid particle receiver, gas receiver and liquid receiver, depending upon type of working fluid being used(Ho and Iverson 2014).

Volumetric receivers are characterized as gas receiver. These receivers utilize porous structure to enhanced heat transfer between solid absorber and flowing gas (particularly air)(Behar, Khellaf, and Mohammedi 2013; Ho and Iverson 2014). Volumetric receivers usually have higher thermal efficiencies as compared to tubular receivers for same boundary flux, due to lower front surface temperature(Ávila-Marín 2011). Volumetric receiver thermal efficiency can further be enhanced by altering geometry parameters , decreasing thermal conductivity of ceramic material and by increasing convection coefficient(Kribus et al. 2014). . Zhiyong Wu et al studied air passage inside the porous foam of ceramic and concluded with a local (wall and volumetric) convective heat transfer coefficient correlation relative to its dependence on cell size, Reynolds number and porosity(Wu et al. 2011). Andreas Fritsch et al compare CFD and FEM models of single tube of receiver utilizing molten salt and liquid metal, also develop numerical model for solar two receiver and compares results with experimental data for