

**NUMERICAL INVESTIGATION OF HYDRODYNAMIC
AND THERMAL PERFORMANCE OF SILICON
CARBIDE AS A SOLAR VOLUMETRIC RECEIVER**

BY

MUHAMMAD FAIZAN

A Thesis Presented to the
DEANSHIP OF GRADUATE STUDIES

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the
Requirements for the Degree of

MASTER OF SCIENCE

In

MECHANICAL ENGINEERING

DECEMBER 2021

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS
DHAHRAN- 31261, SAUDI ARABIA
DEANSHIP OF GRADUATE STUDIES

This thesis, written by **MUHAMMAD FAIZAN** under the direction of his thesis advisor and approved by his thesis committee, has been presented and accepted by the Dean of Graduate Studies, in partial fulfilment of the requirements for the degree of **MASTER OF SCIENCE IN MECHANICAL ENGINEERING**.



Dr. Khaled S. Al-Athel
Department Chairman



Dr. Suliman S. Al-Homidan
Dean of Graduate Studies

08 / 12 / 2021

Date



Dr. Abdulrahman S. Almerbati
(Advisor)



Dr. Bekir S. Yilbas
(Member)



Dr. Abdullah M. Al-Sharafi
(Member)

©Muhammad Faizan

2021

Dedicated to my parents and teachers

ACKNOWLEDGEMENT

All praises and thanks be to Allah Almighty, who is the Creator of everything. Whatever He has wished has happened; and what He has not, has never happened.

I express my sincere gratitude to King Fahd University of Petroleum and Minerals, which facilitated me to conduct this research.

I am thankful to my diligent advisor Dr. Abdulrahman S. Almerbati who offered me this very interesting research topic and was always there to guide me in my work. He facilitated me in every way possible and gave me every tool I needed in my research work. Thanks to my thesis committee members Dr. Bekir S. Yilbas and Dr. Abdullah M. Al-Sharafi for agreeing willingly to be a part of my committee, reviewing my research work, and suggesting useful improvements.

Special thanks to my colleagues for helping me out in pre-processing steps of simulation. I feel obliged to thank many other fellow students who helped me time and again with their valuable suggestions.

Finally, I thank my parents and siblings for their constant motivation throughout my Master's degree and especially throughout my research work.

TABLE OF CONTENTS

ACKNOWLEDGEMENT	v
TABLE OF CONTENTS.....	vii
LIST OF TABLES	x
LIST OF FIGURES	xi
NOMENCLATURE	xv
ABSTRACT.....	xviii
ملخص الرسالة.....	xx
CHAPTER 1 INTRODUCTION	1
1. 1. Background	1
1. 2. Importance.....	3
1. 3. Current Interest.....	5
1. 4. Scope of Study	6
CHAPTER 2 LITERATURE SURVEY.....	8
2. 1. Gas receivers	9
2.1.1. Volumetric gas receivers.....	9
2.1.2. Small particle gas receivers	12
2.1.3. Tabular gas receivers	12
2.1.4. Tubular liquid receivers	14
2.1.5. Falling-film receivers.....	17
2. 2. Developing challenges	21

2. 3. Manufacturing challenges	26
2. 4. Suitable materials	26
2. 5. Solar absorptance	28
2. 6. Solar radiation propagation	28
2. 7. Pressure losses.....	30
CHAPTER 3 POROUS VOLUMETRIC RECEIVER ASSESSMENT	33
3. 1. CT Scan Imaging.....	33
3. 2. Model Refining	33
3. 3. Computational domain	37
3.3.1. Computational domain for the Constructal approach	40
CHAPTER 4 NUMERICAL STRATEGY.....	42
4. 1. Mathematical modeling.....	42
4.1.1. Continuity equation.....	43
4.1.2. Momentum equation	43
4.1.3. Volume-averaged governing equations	45
4.1.4. Energy equation for the liquid domain	48
4.1.5. Energy equation for volume-averaged energy equation	48
4.1.6. Energy equation for solid domain.....	49
4.1.7. Radiation heat source	49
4.1.8. Porous zone modeling for dual-cell approach	50
4.1.9. DO radiation modeling	51
4.1.10. Performing parameters	54
4.1.11. Overall heat transfer coefficient.....	54
4.1.12. Global Nusselt number.....	54

4.1.13. Other expressions and temperature-dependent properties	55
4. 2. Solution methodology	55
4.2.1. Mesh independent tests	56
CHAPTER 5 RESULTS AND DISCUSSIONS.....	61
5. 1. Validation	61
5. 2. Laminar analysis of porous SiC foam.....	64
5.2.1. Thermal Analysis	72
5.2.2. Thermal Effectiveness	77
5.2.3. Flow Analysis	81
5. 3. Turbulent analysis of porous SiC foam.....	83
5.3.1. Thermal Analysis	85
5.3.2. Thermal Effectiveness	92
5.3.3. Flow analysis	96
5. 4. Constructal design of porous SiC foam	100
5.4.1. Thermal Analysis	104
5.4.2. Flow Analysis	128
CHAPTER 6 CONCLUSION AND RECOMMENDATIONS	130
6. 1. Conclusions	130
6. 2. Recommendations and future considerations.....	134
REFERENCE.....	137
CURRICULUM VITAE	160

LIST OF TABLES

Table 3.1: Detail of porous SiC foam samples used for analysis.	38
Table 5.1: Numerical variables. DNI refers to direct normal irradiance.	64
Table 5.2: Numerical variables including DNI, Reynolds Number and Porosity (Ψ).	84
Table 5.3: Description of geometry parameter for proposed configurations.	103

LIST OF FIGURES

Fig. 3.1: Geometry cleaning processes include a) uncleaned sample having faulty facets, b) after cleaning in Geomagic Wrap package, c) exact patches define by coloring, d) deviation analysis and its contour ranges dark blue (-5 mm) to red (5 mm), e) sample after eliminating deviation issues and f) the final CAD model of porous SiC foam.....	36
Fig. 3.2: a) Graphical illustration of a square cross-sectional channel having porous SiC foam, b) green arrow shows inlet of air flows normal to the wall and c) concentrated radiation applied on the top highlighted face.....	38
Fig. 3.3: Schematic of physical problem. Blue (cold) arrows show inlet direction of fluid while red arrows show the outlet direction of fluid.	39
Fig. 3.4: Dimensional illustration of volumetric receiver.....	41
Fig. 4.1: Graphical illustration of different meshes used for mesh independence test.	58
Fig. 4.2: Detail of meshes used for grid independence test. No. of cells for very coarse, coarse, medium and fine meshes are approximately 7.5, 9, 11, and 14 million.	59
Fig. 5.1: Graphical illustration of computational domain used for validation.....	63
Fig. 5.2: Variation of temperature distribution along channel.....	64
Fig. 5.3: Temperature distribution in form of contours along the channel	64
Fig. 5.4: Vectors represent velocity magnitude according for $Re = 50$, $\Psi = 0.77$ and $DNI = 200 \text{ kW/m}^2$	67
Fig. 5.5: Location (plane) where contours for pressure and temperature have been plotted.	68
Fig. 5.6: Velocity contours for a section of the flow channel. Gray areas represent solid phase (porous SiC foam) and colored region is fluid. Contours is plotted for $\Psi = 0.62$ and $DNI = 200 \text{ kW/m}^2$	69

Fig. 5.7: Temperature contours of small portion of channel at top side. Areas under solid black lines are solid portion and outside area is fluid. Contours is plotted at $\Psi = 0.62$ and $\text{DNI} = 200 \text{ kW/m}^2$	71
Fig. 5.8: Total heat flux at wall of porous SiC foam against Reynolds number for a) $\Psi = 0.62$, b) $\Psi = 0.695$, c) $\Psi = 0.77$ and d) $\Psi = 0.845$	73
Fig. 5.9: Overall heat transfer coefficient in channel having porous SiC foam against Reynolds number for a) $\Psi = 0.62$, b) $\Psi = 0.695$, c) $\Psi = 0.77$ and d) $\Psi = 0.845$	75
Fig. 5.10: Global normalized Nusselt number (Nu/Nu_0) of porous SiC foam against Reynolds number for a) $\Psi = 0.62$, b) $\Psi = 0.695$, c) $\Psi = 0.77$ and d) $\Psi = 0.845$	77
Fig. 5.11: Total thermal effectiveness (η_1) of channel having porous SiC foam against Reynolds number for a) $\Psi = 0.62$, b) $\Psi = 0.695$, c) $\Psi = 0.77$ and d) $\Psi = 0.845$	79
Fig. 5.12: Normalized maximum gain temperature (η_2) in channel having porous SiC foam against Reynolds number for a) $\Psi = 0.62$, b) $\Psi = 0.695$, c) $\Psi = 0.77$ and d) $\Psi = 0.845$	81
Fig. 5.13: Pressure drop in channel having porous SiC foam against Reynolds number for a) $\Psi = 0.62$, b) $\Psi = 0.695$, c) $\Psi = 0.77$ and d) $\Psi = 0.845$	83
Fig. 5.14: Temperature contours of a small portion of the channel taken from the upper portion. Areas under solid black lines are solid portions and the outside area is fluid. Contours is plotted at $\Psi = 0.77$ and $\text{DNI} = 250 \text{ kW/m}^2$. a), b) and c) show temperature distribution for air while d), e) and f) for water.	87
Fig. 5.15: Total heat flux at wall of SiC receiver against porosity and Reynolds number for a) air and b) water.....	89
Fig. 5.16: Global normalized Nusselt number (Nu/Nu_0) of SiC receiver against porosity and Reynolds number for a) air and b) water.	91

Fig. 5.17: Total thermal efficiency (η_1) of channel having SiC receiver against porosity and Reynolds number for a) air and b) water.	94
Fig. 5.18: Normalized maximum gain temperature (η_2) in a channel having SiC receiver against porosity and Reynolds number for a) air and b) water.	95
Fig. 5.19: Velocity magnitude of small portion of channel taken at $\Psi = 0.77$ and DNI = 250 kW/m ² . Areas under black lines (grey portion) shows solid portion (porous SiC foam). a), b) and c) show velocity distribution for air while d), e) and f) for water.	97
Fig. 5.20: Pressure drop in channel having porous SiC foam against SiC receiver against porosity and Reynolds number for a) air and b) water.	99
Fig. 5.21: Schematic of validation's numerical domain	101
Fig. 5.22: Temperature variation along the flow direction for validation purposes.	101
Fig. 5.23: Numerical representation for boundary conditions of the volumetric receiver.	103
Fig. 5.24: Averaged temperature variation along the flow direction as a function of axial coordinate for configurations 1 – 10.	108
Fig. 5.25: Averaged temperature variation along the flow direction as a function of axial coordinate for configurations 11 – 20.	109
Fig. 5.26: Averaged temperature variation along the flow direction as a function of axial coordinate for configurations 21 – 30.	110
Fig. 5.27: Averaged temperature variation along the flow direction as a function of axial coordinate for configurations 31 – 40.	111
Fig. 5.28: Averaged temperature variation along the flow direction as a function of axial coordinate for configurations 41 – 50.	112
Fig. 5.29: (a), (b), (c), (d), (e) and (f) show temperature [K] contours for configuration 3, 8, 13, 18, 23 and 4, respectively. Sub-figures. names 1 and 2 show solid and fluid.	114

Fig. 5.30: (a), (b), (c), (d), (e) and (f) show temperature [K] contours for configuration 48, 43, 38, 33, 28 and 49, respectively. Sub-figure names 1 and 2 show solid and fluid.	115
Fig. 5.31: Thermal equilibrium location corresponds to length of receiver and effective length for achieving thermal equilibrium for configuration 1 – 25.....	118
Fig. 5.32: Thermal equilibrium location corresponds to length of receiver and effective length for achieving thermal equilibrium for configuration 26 – 50.....	119
Fig. 5.33: Effective area (A_{eff}) for achieving thermal equilibrium for all proposed configurations.....	121
Fig. 5.34: Exit temperature (T_{exit}) taken as mass-averaged at air outlet of receiver for all proposed configurations.	124
Fig. 5.35: Maximum temperature (T_{max}) achieved within numerical domain for all proposed configurations.	125
Fig. 5.36: Exit temperature (T_{exit}) corresponding to its effective area for all proposed configurations.....	127
Fig. 5.37: Pressure drop across over the complete length or pressure difference inlet to outlet for all proposed configurations.	129

NOMENCLATURE

Symbols

A	Surface area/Area [m^2]
A_0 and A_s	Turbulent model constant
A_h	Area of inlet [m^2]
a	Spectral absorption coefficient
c_p	Specific heat [J/kg K]
C_μ	High Reynolds number Limiter in $k - \epsilon$ /RTM
$C_1, C_2, C_{1\epsilon}, C_{3\epsilon}$	Controlling constants for turbulent dissipation rate
d	Diameter [m]
d_c	Cell Diameter [m]
D_h	Hydraulic diameter [m]
DNI	Direct Normal Irradiance
P_k	Production of turbulent kinetic energy based on mean velocity
G_b	Production of turbulent kinetic energy based on buoyancy force
I	Radiation intensity [W/sr]
k	Turbulent kinetic energy [m^2/s^2]
k_s	Thermal conductivity of solid [W/m K]
k_f	Thermal conductivity of fluid [W/m K]
L	Length of receiver [m]
$\dot{m}$	Mass flow rate [kg/s]
Nu	Nusselt Number
n	Refractive Index
$\vec{n}$	Vector direction
N_θ	Number of discretization based on Zenith angle
N_ϕ	Number of discretization based on Azimuth angle
q_r	Radiation source [kW/m^2]

q''	Heat flux [kW/m^2]
Re	Reynolds Number
$\vec{r}$	Vector position
S	Strain rate [$1/\text{s}$]
$\vec{s}$	Position [m]
T	Temperature [K]
T'	Fluctuating temperature (K)
U_i, U_j	Velocity magnitude according to i and j arrays [m/s]
U^*	Local velocity [m/s]
U'	Fluctuating velocity [m/s]
$\bar{U}$	Volume averaged velocity [m/s]
$\bar{U}'$	Volume averaged turbulent velocity term [m/s]
V	Volume of receiver [m^3]
X	Cartesian coordinates
Y_M	Fluctuating dilatation

Greek letters

α	Absorptivity
α	Thermal diffusivity [m^2/s]
ϵ	Emissivity
ϵ	Turbulent dissipation rate [$1/\text{s}$]
$\delta_{ij}, \bar{\delta}$	Unit tensor
ρ	Mass density [kg/m^3]
σ	Stephen Boltz constant [$\text{J/m}^2 \text{ s K}$]
σ_s	Scattering Coefficient
σ_t	Turbulent Prandtl number
σ_k and σ_ϵ	Turbulent Prandtl number for k and ϵ , respectively
μ	Molecular viscosity [Pa s]
μ_t	Eddy viscosity [Pa s]
θ	Cone angle ($^\circ$)

Ψ	Porosity
Φ	Phase function
η_1	Total thermal efficiency
η_2	Normalized maximum gain temperature
Subscripts	
f	Fluid
k	Turbulent kinetic energy
in	Inlet/in
interface	Interface wall
m	Mean
max	Maximum
o	Nusselt number of channel without porous SiC foam
out and exit	Outlet/out/exit
s	Solid
sf	Mean surface temperature
t	Turbulent
Abbreviations	
CT	Computed tomography
DO	Discrete Ordinate
DNI	Direct Normal Irradiance
CAD	Computer-Aided Design
CFD	Computational Fluid Dynamics
CSP	Concentrated Solar Power
GHG	Greenhouse Gas
LTNE	Local Thermal Non-Equilibrium

ABSTRACT

Full Name : Muhammad Faizan

Thesis Title : Numerical investigation of hydrodynamic and thermal performance of silicon carbide as a solar volumetric receiver

Major Field : Mechanical Engineering

Date of Degree : December 2021

A porous volumetric receiver in the solar thermal system is a promising utility for energy harvesting applications. The hydrodynamic behavior and thermal performance of porous structures in the receiver remain critical for efficient operation. Hence, in the present study, the assessment of SiC foam in a volumetric receiver is carried out for a concentrated solar energy harvesting system. In the numerical study, a 3D model of actual SiC foam is incorporated and the discrete ordinates (DO) radiation model, which is fully coupled with conduction and convection heat transfer, is adopted. A 3-dimensional computed tomography scan (CT-scan) of porous SiC foam is imported for CFD modeling. The foam is located in a square channel having an optical transparent top wall, which allows solar radiation to enter the channel. The porous SiC adsorber responses are examined for two working fluids including air and water. The overall thermal performance of the solar harvesting system is evaluated by considering the overall heat transfer coefficient, normalized Nusselt number and thermal effectiveness based on the outlet and the maximum temperatures. All performance parameters are computed using temperature-

dependent properties for different Reynolds numbers, solar concentrations, and porosities. Assessment of hydrodynamic and thermal behavior of porous SiC foam is conducted for both flow conditions laminar as well as turbulent. Further extension of assessment involves channel's configurations which are numerically analyzed using dual-cell approach with LTNE model. Novel cone frustum (converging) shape of volumetric receiver is introduced to achieve the best-performing design of the volumetric receiver. Various configurations of cone frustum are analyzed, and comparison has been made between these configurations by varying design's parameters such as cone angle of the frustum, the ratio of inlet diameter to outlet diameter, and length of receiver. It is observed that employing porous SiC foam considerably improves the thermal performance of the solar harvesting system as compared to the channel configuration without porous SiC foam. A significant improvement is observed in the Nusselt number while demonstrating the convection is the apparent heating mode in the porous system, provided that the pressure loss of the channel with SiC foam is not substantially high. Moreover, a significant thermal enhancement is observed while comparison with the previous volumetric receiver of the same volume and inlet conditions. It is worth mentioning that considerable high exit improvement is achieved for proposed configurations as compared to the previous studied design while having the almost same pressure.

ملخص الرسالة

الاسم الكامل : محمد فيضان

عنوان الرسالة : دراسة الأداء الهيدروديناميكي والحراري لمستقبل حراري - شمسي مسامي مصنوع من مركب كربيد السيليكون (SiC) باستخدام التحليل العددي (المحاكاة)

التخصص : الهندسة الميكانيكية

تاريخ الدرجة العلمية : ديسمبر ٢٠٢١ م

يعتبر المستقبل الحجمي المسامي في النظام الشمسي-الحراري أداة واعدة لتحصيل وكسب الطاقة. يظل السلوك الهيدروديناميكي والأداء الحراري للهياكل المسامية في المستقبل عاملين مؤثرين للتشغيل الفعال للنظام. لذلك في هذه الدراسة، سيتم إجراء تقييم لرغوة (هيكل مسامي) مصنوعة من كربيد السيليكون (SiC) في جهاز استقبال حجمي لنظام تجميع الطاقة الشمسية المركزة. في هذه الدراسة العددية، تم استخدام نموذج ثلاثي الأبعاد لرغوة SiC حقيقية وتم اعتماد نموذج إشعاع الإحداثيات المنفصلة والمعروف بـ (DO) والذي يتضمن أنواع النقل الحراري التالية: التوصيل والحمل. تم عمل مسح التصوير المقطعي ثلاثي الأبعاد (CT-scan) لرغوة SiC المسامية لاستخدامها في نمذجة المحاكاة CFD. بعد ذلك تم وضع الرغوة في قناة مربعة جدارها العلوي شفاف، والذي يسمح للإشعاع الشمسي بدخول القناة. يتم فحص استجابة النظام من خلال استعمال اثنين من الموائع (الهواء والماء). يتم تقييم الأداء الحراري الكلي لنظام التركيز الشمسي من خلال النظر في معامل انتقال الحرارة الكلي ورقم Nu والفعالية الحرارية بناءً على درجة حرارة المائع عند مخرج القناة وكذلك الدرجة القصوى له في داخل القناة. يتم حساب جميع معاملات الأداء باستخدام الخصائص المعتمدة على درجة الحرارة للهواء والماء وكربيد السيليكون. تتم دراسة المعاملات عند أرقام مختلفة من Re وتركيزات عديدة للطاقة الشمسية وباعتبار قيم مختلفة لمسامية كربيد السيليكون. يتم إجراء تقييم السلوك الهيدروديناميكي والحراري لرغوة SiC المسامية باعتبار حالة تدفق المائع laminar و turbulent . يتضمن هذا البحث دراسة تأثير شكل القناة على أداء النظام الشمسي الحجمي والذي تم تحليلها عددياً باستخدام نهج الخلية المزدوجة

مع نموذج (LTNE). تم اختبار الشكل المخروطي (المتقارب) للمستقبل الحجمي لتحقيق أفضل تصميم أداء لجهاز الاستقبال الحجمي. يتم تحليل تكوينات مختلفة من المخروط المخروطي، وتم إجراء مقارنة بين هذه التكوينات من خلال تغيير معاملات التصميم مثل زاوية انحراف المخروط ونسبة قطر مدخل المائع إلى قطر المخرج، وطول القناة. لوحظ أن استخدام رغوة SiC المسامية يحسن بشكل كبير الأداء الحراري لنظام التحصيل الشمسي مقارنة بتكوين القناة عندما يكون فارغاً. كذلك لوحظ تحسن كبير في قيم Nu والذي يدل على أن نقل الحرارة بالحمل هو العامل المؤثر الأكبر على عملية التسخين بداخل النظام المسامي، والذي صاحبه ارتفاع طفيف في الضغط المفقود عبر القناة. علاوة على ذلك، لوحظ تحسين حراري كبير أثناء مقارنة التصميم الجديدة مع تصاميم تقليدية سابقة للمستقبل الحجمي والتي تضمنت مساحة مقطعية متساوية لمدخل ومخرج القناة. كذلك من الجدير بالذكر أن التصميم المقترحة الجديدة حققت تحسن كبير في الأداء للمكثفات الشمسية مقارنة بالتصاميم المدروسة سابقاً بنفس قيمة فقد الضغط الناتج عبر القناة.

CHAPTER 1

INTRODUCTION

1. 1. Background

Energy and environment are the major obsessions of the contemporary world and delicate the balance for human life. Energy production and management are the backbone of modern economy and contributes major part in development and well-being for the world. Above 50% of energy production depends upon fossil fuels which are significantly polluting the environment. Secondly, resources of fossil fuels are reducing day by day due to ever-increasing demand of energy. In recent decade, convectional energy starts relying more on different sources of clean and green energy to alleviate the primary cause of environmental pollution [1–3]. In order to build an unfailing network of clean energy to satisfy the future demands as well as the criteria of eco-friendly, the renewable energy sources have become more robust to serve this purpose. Among all renewable energy sources (wind, geothermal, solar etc.), solar harvesting system proven itself more reliable available energy source to meet the energy demands of current and incoming era [4,5]. Especially for tropical regions and middle east, solar harvesting system is more feasible and applicable as an alternative energy source. Moreover, it is also capable to become a primary source of energy by enhancing and tuning the thermal performance of system without imposing carbon-based pollution on environment [6,7].

Solar harvesting systems have been divided into two primary categories: solar photovoltaic cells and solar thermal receivers. Solar receiver systems which work on principal of the Concentrated Solar Power (CSP), was derived using idea of ancient Greece, where Archimedes used bronze shields as a defensive tactic, to concentrate the sunrays onto attacking army to fire invading ships. This myth was successfully proven by L. Sakkas in 1973, when 60 ships caught on fire while concentrated sunrays were propagated through bronzed coated mirror onto these ships from 200 ft [8]. The first document on CSP was appeared during 1860s, in which CSP technique is used to heat water through parabolic troughs. This plant was established by A. Mouchout to produce steam and became first solar-based steam engine [9]. In 1912, this solar-based steam engines were introduced to commercial market and first official working model was installed in Meadi, Egypt for pumping of 6000 gallons of water per minute for farming. Then, the CSP applications were extended to numerous areas with promising outcomes. In 1968, G. Francia used CSP for solar power plant by using field of solar collectors; in 1982, US department of energy introduced operating CSP plant which was able to produce 10 MW electricity; in 1986, world largest CSP facility was established in California USA. As of June 2010, 34 CSP plants having capacity of 880.45 MW energy were available worldwide [10,11]. As of 2019, generation of CSP increased up to approximately 34% with surplus 24% annual growth. Based on these facts, by 2030, it is expected that a considerable portion of energy will be producing by CSP. However, major percentage of energy will still be depended on fossil fuels. In order to achieve major portion of energy generation depending on CSP plants (clean and sustainable), the technology still requires tons of enhancement [12,13].

1. 2. Importance

The motives are to lower the level of GHG emissions and to depend on more secure resources. Modern societies require clean and reliable energy systems which can respond to the energy demand and population growth by mitigating the environmental risks e.g., global warming and climate change [14,15]. Thus, massive attention was recently devoted to renewable (mostly solar and wind) research and development. One of the crucial renewable technology is concentrated solar power (CSP) [16] in which the energy conversion efficiency reaches 25% [17]. Since high temperature can be achieved through CSP technique, thus, it emerged as an acceptable candidate for large scale energy production (solar tower) and other heat applications. The efficiency of solar tower technology is associated with the performance of its central element known as the solar receiver. Solar receivers have been using in solar thermal concentrating techniques are of mainly two types: linear receivers and volumetric receivers. In linear receivers, surface of receiver absorbs concentrated heat then transfer to working fluid which follows linear path, mostly consists of tubular structure. Thermal conductivity of surface is main factor to receiver performance, if linear receivers possess low thermal conductivity which result in high temperature at outer surfaces. This might lead to high loss of emission from outer surfaces. The volumetric receiver overcomes this drawback faced while using linear receiver [18]. However, volumetric receivers are designed in such a way that solar radiation is captured as much as possible. Their special configuration of design is also subjected to penetrate solar radiation into the receivers which act like a thermal cavity for trapping heat inside them and transfer heat directly to working fluids. Here, unlike linear receivers, temperature of volumetric receivers remains higher than working fluids. Furthermore, the

porous structure of volumetric receivers ensures that absorbing heat inside cavities and transferring to working fluid must be simultaneously [19].

First successful pre-commercial trial of volumetric receiver utility in tower plant having capacity of 1.5 MWe was installed in Julich, Germany and its operation started in June 2006 [20]. Earlier, testing field of concentrating solar site was originated around 1980s, in which 250 acres site with series of mirror for concentrating solar was assessed mainly to generate power of 3 MW and other purposes such as hot fluid from this testing field used for chemical process and desalination [21]. Later, application of concentration solar plant was expanded for melting salt. However, recently, various other modifications have been incorporated in CSP to achieve optimize performance including utility of ceramic foams, oxide and non-oxide ceramic receivers.

Numerous studies have been conducted on developing the CSP technology to make it economically competitive. These were mainly accomplished based on increasing the solar receiver temperature which directly improves the conversion efficiency [22–24]. However, challenges are still associated with the solar volumetric receivers including optics for high concentration, stability of the material, and porous material structure design [25]. Three types of structures, which are wire mesh, honeycomb, and metallic foam, have been extensively utilized in designing efficient solar volumetric receivers [24,26,27]. The limitation of the maximum outlet air temperature achieved with volumetric receivers made of metal reinforced the use of ceramic-based volumetric receivers [28–30].

1. 3. Current Interest

Porous volumetric receivers for solar energy harvesting have been studied experimentally and numerically to optimize the design and depended parameters [31]. For example, V. R. Patil et al. [16] experimentally investigated the performance of different reticulated porous ceramic receivers for thermal processing above 1000°C. Air outlet temperature and thermal efficiency were considered for thermal performance by varying parameters such as receiver materials, average pore size, air mass flow rate, and solar concentration ratio. Thermal efficiencies were achieved in the range of 0.22 - 0.69 having air outlet temperatures varying from 1160 to 450°C. Furthermore, results also indicated that larger pores improved the heat transfer while variable porosity of ceramic receiver decreased temperature gradients which potentially contributed to the design optimization. In another study, Z. Wu et al. [23] numerically analyzed the influence of pore size on temperature fields and fluid pressure drop inside the solar receiver. Findings illustrated that the mean pore size of the ceramic receiver is strongly related to the thermal performance. G. Barreto et al. [17,18] optimized the thermal performance of solar volumetric receivers by using cylindrical-shaped porous open-cell ceramic foam for concentrated solar power application. Findings revealed that higher values of the fluid velocity at the inlet enhanced the thermal efficiency, but it decreased the mean fluid temperature at the exit. However, with increasing porosity and pores size, the pressure drops decreased. Additionally, the solar concentration and the Reynold number are strongly related to the solar volumetric receiver performance. In general, increasing solar concentration and Reynold number directly enhanced the heat gain by the solar receiver; hence, the main challenge is to achieve a high ratio between heat gain and heat supplied. The further investigation focused on the placement of the solar

receiver by adjusting height and angle [19]. S. Du et al. [20] studied the reflectance losses of the porous volumetric solar receiver by considering several parameters including the wavelength of solar irradiation, angle of incident, porosity, and pore size. The reflectance losses were increased when wavelength and incident angle was found to be above 420 nm and 30°, respectively. They found that higher porosity and larger pore size reduce the reflectance losses.

1. 4. Scope of Study

The above literature addresses the importance of the design and structure of a solar volumetric receiver for hydrodynamic and thermal performances which can be improved by optimizing the parameters such as porosity, pore size, solar concentration, Reynolds number and design of receivers. Several researches show the geometry investigation of reflector dishes which are utilized for concentrating radiation for directing towards receiver. Similarly, pore size and porosity are also investigated but flow channel size is not developed according to pore size and porosity which is also research gap. Likewise, lack of attention has been observed towards the design of receivers especially related to length of receiver. The current study aims the investigation of influence of input and design parameters on global Nusselt number, overall heat transfers co-efficient, radiation heat flux, maximum temperature, effective size of receiver and optima according to these parameters. Input parameters are studied widely, however, design parameters are not analyzed for volumetric receiver. Comparison between different configurations of receiver's channel is also included in this study in detail. To the best of author's knowledge, the study brings novel augmented volumetric receiver configurations with detail debate of thermal performance parameters.

The present numerical study uses actual porous SiC foams as a volumetric receiver and their structures are scanned using the micro-scan facility and the structural features of the foams are imported into computational fluid dynamic (CFD) code. CFD code is used to solve the governing equations including continuity, momentum equation, $k - \epsilon$ turbulence models (realizable and standard), energy equation, Discrete Ordinate radiation equation, Local Thermal Non-Equilibrium Model and their volume-averaged equations. CFD analysis is carried out using ANSYS Fluent. Purpose to solve these governing equations is to find the temperature distribution in fluid and solid phases of the pores structure and pressure distribution in the working fluid along the channel of the solar volumetric receiver. Discrete Ordinate (DO) radiation model is employed to couple the convection, conduction, and radiation, and also includes the influence of scattering, diffusion, emissivity and absorption of radiation.

This work comprises 6 chapters. Chapter 1 (Introduction) serves as preface to this research work. Chapter 2 (Literature Survey) contains the literature survey relevant to the topic undertaken. Detail modeling strategy for CFD code is described in Chapter 3 (Porous Assessment). Chapter 4 contains the supportive theory and the formulation of governing equations (Numerical Analysis). Results of the numerical assessment are discussed in Chapter 5 (Results and discussion) in tabular and graphical forms. Chapter 6 (Conclusion and Recommendations) concludes the work and includes recommendations for future work.

CHAPTER 2

LITERATURE SURVEY

Electricity generation from low-carbon energy resources (renewable energy) is globally replacing the generation from fossil fuels. Solar energy is extensively available in the form of renewable energy, however, harvesting solar energy techniques are still costly and they involve less efficient processes. With the need to minimize the levelized cost of solar power systems as compared with other electricity generating units, the solar energy harvesting research remains significant. In general, solar harvesting is accomplished by different means such as photovoltaic cells and solar receivers [37–39]. Solar receiver could be categorized into three classes; (1) Particle receivers (entrained particles, falling curtain), (2) Surface receivers (external cavity, stills, tabular, convective plates), and (3) Volumetric receivers (ceramic foams, metallic and wire meshes, honeycomb structure, ducts, packed beds) [40,41]. The efficiency of volumetric receivers, which is mainly CSP, is associated with the performance of its central element known as the porous foam act as volumetric receiver. This receiver consists of a porous material heated by concentrated solar radiation [42].

In general, volumetric receivers are designed towards achieving the minimal heat losses of solar radiation when operating at high temperatures. The improvement in volumetric receiver performance through reshaping the absorbing elements is in progress [29,43]. To achieve sustainable development of the operation of receivers, the optimized features of the device basing on materials and structures become essential. The permeable porous

structure of receivers that act as volumetric receivers becomes preferable to harvest the incoming concentrated solar radiation. Since most of the heat is transferred from receivers utilizing convection, it implies that properties of heat transfer fluid matter [44]. Mainly, the heat transfer fluids are air or water because they are highly available and environment friendly, but other fluids such as carbon dioxide could also be taken as the heat transfer fluid due to its high heat capacity. The highest temperature can be reached inside the receivers because of the volumetric effect and, thus, the thermal-hydrodynamic behavior of the receiver at high temperature becomes one of the main concerns. In addition, it has been noted that the temperature at the exposed surface of volumetric receivers is lower than the inside temperature of receivers that leads to lower radiation losses [45].

2. 1. Gas receivers

2.1.1. Volumetric gas receivers

From 1980s, solar volumetric gas receivers have been under investigation which usually contain porous structures such as ceramics foams, honeycombs, wire meshes, etc. and are subjected to irradiate by concentrated solar radiations. Heat transfer fluids are allowed to flow and heated up to critical temperature of receivers' material, for examples 800 - 1000°C for metals and 1200 - 1500°C [46]. Consequently, these heat transfer fluids can be utilized for various applications including as a working fluid for a Rankine steam cycle and directly to gas turbine [47], and power source for storage medium [39]. Besides these applications of heat transfer fluids, solar volumetric gas receivers can directly be used into open-loop atmospheric receiver system for a Rankine cycle and closed-loop pressurized (windowed) receiver system for a Brayton Cycle. In 1991, Chavez and Chaza [48] reported the design

of porous ceramic receiver and its testing for thermal performance which is installed in Plataforma Solar de Almeria Spain. Air was used as heat transfer fluid, and highest concentrated solar radiation flux and exit temperature of air were achieved to be 800 kW/m^2 , and 730°C . However, highest thermal efficiency of 65% was found with exit temperature of 550°C and 54% decrement in the thermal efficiency was noted when exit temperature was 730°C . This drastic drop in thermal efficiency was faced because of unoptimized design factors especially large thickness of Pyromark paint and blocking some of the pores. Furthermore, maximum temperature of air in receiver was obtained up to 1350°C which was very high as compared to exit temperature and contributed to reduction of thermal efficiency due to large radiative heat losses. Conclusion has been made that optimization could improve thermal efficiency up to 85% for exit temperature 550°C . For controlling of radiative heat losses, two researcher groups [49,50] configured the volumetric receiver in form of two slab selective receiver: 1) silica honeycomb (solar transparent glass beads) which was placed at front and supposed to control irradiation and 2) SiC particles to utilize the radiation heat to elevate the exit temperature. Front slab (transparent silica honeycomb) was allowed to penetrate the solar radiation (mainly infrared emissions) towards second slab (SiC particles) to absorb these emissions. In such a way, controlled absorption of solar radiation was achieved which helped to decrease the maximum temperature due to two different materials. As a results, volumetric effect was more profound that leads to maximum temperature within receiver instead of fluid domain. Thus, final testing of the study revealed that windowed receiver achieved thermal efficiency of almost 90% with exit temperature of 700°C . Another group [51] experimentally determined that thermal efficiency has been enhanced by up to 10% while

using square glass channels covered with ceramic thin film receiver. Their findings showed incredible improved exit temperature up to 1000°C as compared to pure ceramic receivers. New re-feeding concepts was proposed by Marcos et al. [52] which incorporates some part of working fluid re-feed again into volumetric receiver to recover the waste heat. Secondly, this volumetric receiver is combined with working fluid of some power cycle and re-feeding was also coupled with exhaust air of power cycle to capture the waste heat. This configuration resulted in significant improvement of thermal efficiency up to 70% and recovered waste heat up to 15%. Further extension of this study involved optimization of air re-feeding and design parameters of volumetric receiver via implementation of numerical investigation. This optimization, in turns, augmented the thermal efficiency by more 20% (total near 90%) [53,54].

Turbulence streams of working fluid and non-uniform heating due to irregular path of porous structure of volumetric receivers often cause the excess heating at any place within fluid and solid domain which also leads to thermal instability or failure [26,42] . For solid body, thermal instability significantly reduces the durability of system. For fluids, thermal instability causes drastic change in temperature dependent thermophysical properties (especially density and viscosity) of fluid, however, these effects can be mitigated by optimizing porosity of volumetric receivers. Karni et al. [55] proposed the design of solar volumetric receiver having ceramic pins (named as Porcupine) that had capability to achieve exit temperature of fluids up to 1000°C which was almost same as maximum temperature of that receivers.

2.1.2. Small particle gas receivers

Submicron carbon particles are dispersed in pressured working fluids (mainly air) enclosed in cavities or tubes; this design of receiver known as small particle gas receivers. Then, these receivers are heated using concentrated solar radiation and recovered heat with help of pressurized working fluids which is transferred to Brayton cycles as an energy input [37,46]. First model of the small particle gas receiver was first established during 1970s [56,57]. Main advantages of this receiver that uniform absorption of solar radiation throughout the fluid body and high surface area of solid particles contributes to achieve significantly enhanced exit temperature. Moreover, this idea prevents thermal instabilities and ensures continuous recovery of heat. But higher solar concentration can damage the particles or alter their thermophysical properties as well as active chemically (oxidization) [58]. Findings of investigation on these receivers explored that thermal efficiency can be achieved up to 90% by optimizing some input parameters (solar concentration, mass-flow rate, etc.), pressure of gas, dispersion of particles, optical properties and size of particles [58,59]. It is experimentally studied while air as a working fluid in 25 kWh small particle receiver attained the temperature of 700°C. But this was achieved with optimized parameters (stable dispersion and suitable concentration of micro-particles) and conceiving appropriate window for the pressured receiver [60].

2.1.3. Tabular gas receivers

The application of solar thermal receivers operating at high temperatures for the purpose of air-Brayton cycles has been tried and tested over the past couple of years. The initial models were designed in parabolic structure and relied on the use of pipes to increase heat

transfer. Inside the liquid-metal heat pipe, heat transfer coefficient is around $30,000 \text{ W/m}^2 \text{ K}$ as compared to only $300 \text{ W/m}^2 \text{ K}$ with gases [61,62]. This helps to reduce the size of the receiver and reduce pressure drop because of the high solar flux. However, the cost is an issue with these types of receivers. Researchers have tested design parameters of 815°C outlet temperature of air and 565°C inlet temperature of air, 0.24 kg/s mass flow rate of air, 2% pressure drop, with 85% thermal efficiency [63]. Some other experiments have been done on different designs by DLR for a solar-hybrid microturbine system operating used on power of $100 \text{ kW} - 1 \text{ MW}$ [64–66]. Design concepts introduced in these studies include the use of absorber tubes with a “Profiled-Multi-Layer” design consisting of Inconel material with copper sandwiched in between in a hydroforming process to enhance the circumferential heat distribution and heat transfer to the gas, segmented silica windows to reduce convective and radiative heat losses, and hybridization. Simulations showed an increase in the potential thermal efficiency of the receiver from 68% to 81% when a window was used compared to simulations with no window. Tests performed at the Plataforma Solar de Almeria using the system, a smaller improvement from 40% (no window) to 43% (window) with receiver inlet and outlet temperatures of 600°C and 800°C , respectively, a turbine power of 70 kW , and a solar input of 273 kW [67]. The difference in the measured and the calculated values here occur due to loss of heat from the cavity walls. These receivers operating at high temperature are subjected to thermomechanical loads that are fluctuating in nature which reduce the fatigue life when added with the heat losses due to radiation and convection. Since it has been shown that solid- CO_2 is able to give higher efficiencies close to 50%, its use as heat transfer fluid is recommended. Tubular receivers have smaller tube diameter that can sustain higher pressures necessary for

supercritical phase when they use solid-CO₂ as the heat transfer fluid. Pressure may reach up to 15 - 25 MPa at the inlet of the turbine. The biggest issue with solid-CO₂ is its storage that needs to be kept separate if solid-CO₂ is used as receiver [68].

2.1.4. Tubular liquid receivers

Various configurations have been proposed for tubular liquid central receiver systems since the 1970s, but first model was successfully implemented in the 1980s and second implementation became ready during 1990s named with Solar One [69] and Solar Two [70], respectively. In conventional tubular receivers, arrays of thin-walled tubes having good thermal conductivity (stainless steel or alloyed) that are arranged in such a way to shuttle the heat transfer fluid (e.g., water/steam or molten salt) in multiple passes by exposing to incident concentrated solar radiation. Then, heated fluid is transported for storage into the power block. Both cavity-type (internal flow) and external receivers can be found with principle of tubular liquid receivers. Molten salts are also used as heat transfer fluids while using liquid-based tubular receivers, which have common approaches with current power tower receiver. These receivers have been comprehensively investigated in facilities of Sandia National Laboratories [71–73], Themis [74] and Plataforma Solar de Almeria [75]. Temperatures of the working fluid of the tubular receivers have been observed less than 600°C till now. At higher temperature near to 650 – 750°C, re-radiation impact should be carefully determined in order to select an open or an enclosed receiver design. Liquid sodium [76] and fluoride-salt [77] as working fluids have also been suggested as an alternative to molten nitrate salt to accomplish higher temperatures and thermal efficiencies. Higher thermal conductivities associated with liquid metals allows for higher incident radiation flux values (in excess of 1.5 MW/m²), as the

thermal conductivity decreases the front-to-back receiver tube's temperature that also lessens the associated thermal stresses [76]. Higher incident radiation flux also enhanced the thermal efficiency of the receivers as a smaller receiver can be constructed for the same thermal power collected. Selection of suitable tube size and wall thickness helps to improve the heat transfer while considering pumping losses. The heat transfer coefficient scales as $1/\text{diameter}$, making small diameters attractive for convection. However, pumping losses and material costs increase and required wall thicknesses decrease at a given pressure with smaller diameters, resulting in an optimum diameter [78]. Tubular receiver designs are usually comprised of several panels, which are in turn comprised of an array of tubes. Tubes in the same panel have fluid flows in the same direction and have approximately the same flux distribution. The use of numerous tubes effectively acts as a mechanism to enhance heat transfer, much like fins are used to increase surface area. Estimated efficiencies for an external, tubular receiver employing a variety of working fluids (including high temperature working fluid such as LiCl/KCl and NaCl) show the values up to 89% appear achievable [79], with design point operation reaching above 90% [80]. Final evaluation results of Solar Two's receiver reveal the similar values which become a standard for comparison. Other fluids must also be considered (such as the fluorides [81]) for obtaining reasonable working fluid melting points and higher thermal conductivities that can enhance thermal performance. The working fluid type is a crucial factor in the receiver because of operating temperature that results in higher thermal efficiency. Incorporating a selective absorption technique that improves or maintains the absorptivity while reducing emissivity can provide a significant boost to efficiency and reduce losses to achieve efficiency targets [82,83]. Receiver design is highly dependent on the selected

working fluid used to convey the absorbed thermal energy. Liquid-based receivers have high heat-transfer rates and high specific heat relative to gaseous Heat transfer fluids. Numerous Heat transfer fluids have been tested to date for use in receiver systems including water/steam [84], nitrate salt [69], and sodium [75]. Water/steam systems at elevated temperatures have been deployed at pilot facilities such as Solar One and PS10 and PS20. For systems that operate with conventional steam cycles, the turbine inlet conditions are commonly 9 – 13 MPa [76]. Analysis on both external-type and cavity-type receivers has been conducted [85] with cavity receivers generally expected to have a lower radiation heat loss and higher convective heat loss than that for external receivers. Towers for surround field systems are shorter than the tower required for a north-field design. However, a larger land area is required for surround fields than for north-facing fields. The impact of receiver area, thermal losses (including emissive, convective, reflective and conductive), number of receiver tubes in a panel, tube diameter and surface temperature for a molten salt cavity receiver have been calculated with benchmarked models [86]. Experimental demonstrations of tubular receiver panel performance have also been conducted with slightly lower thermal efficiencies obtained for external-type receivers [87]. As described above, selective absorber coatings for high-temperature central receivers that increase the solar absorptance while minimizing thermal emittance can increase the thermal efficiency. Desired features for these solar selective absorber coatings include stability at high temperatures in air, high durability (must withstand thousands of thermal cycles), low cost, and ease of application. Optical properties of commercial high-temperature paint (Pyromark 2500) have been characterized [88], and other coating

materials and deposition methods have also been evaluated for use in high-temperature central receiver applications [89,90].

2.1.5. Falling-film receivers

Gravity-driven working fluids (mainly liquids) motion is main phenomena involve in the falling film receivers. Fluid can be directly heated through solar radiation, indirectly via heated wall or direct exposure to solar radiation coupled with convection heat recovery of heated wall. Gravity-driven characteristic of these receivers reduces the requirement of pumping power. Furthermore, direct irradiation of working fluids in falling-film receivers exploits the absorption of the thermal energy by direct exposure and lessens the thermal resistance [91]. In general, this approach is well-known as a direct absorption liquid receiver where the working fluids are heated without any internal flow path or heat recovery from any solid media. However, the falling-film receivers are also coherentized for external black body (wall) heat recovery receivers. In addition, sometimes these receivers can be combined with concept of small particle receiver when gravity-driven working fluids contain suspended submicron particles [92].

Like other types of solar receivers, the falling-film receivers also possesses several critical factors which can be optimized for obtaining high thermal efficiency including thickness of fluid falling-film, absorption quality of black wall, optical and thermophysical properties of working fluids (by dispersion of submicron particles). Optical properties of working fluids highly affect the emission losses which can also control by falling-film thickness [93]. For enhancing opacities to control emission losses molten salts showed promising results while having oxide dopants [94–96]. One investigation on cobalt oxide as a

submicron particles in working fluids of falling-film receiver showed the enhancement of thermal efficiency by approximately 5%, but cost-effectiveness of this system might be questioned for enhancement [97].

Recently, a study investigated dispersed nanoparticles in working fluid to absorb radiation and developed the correlation for determining the influence of heat loss, particle concentration in working fluids, solar radiation flux, and channel height of receiver on thermal performance [98]. Film stability has been studied in numerous direct absorption receiver designs, but demonstrations have been with water under isothermal conditions [99–102]. Analytical studies have been reported of heat and mass transfer in wavy liquid films [103] and falling turbulent films [104]. A correlation to predict the heat flux required to break a falling liquid film (thermocapillary breakdown) has also been reported [105]. For the direct-exposure external DAR, a common method to improve fluid control is to increase the panel tilt as it has a strong effect on controlling fluid loss but results in taller receiver surfaces. Manifold designs and fluid stability have been studied to improve fluid control but without implementation in a demonstration plant [106]. Demonstrations of the external DAR concept using a molten carbonate salt have been reported by Bohn with thermal efficiencies on the order of 80 – 90% and heat transfer coefficients of about 3000 W/m² K [107]. However, larger heat transfer coefficients on the order of 6500 W/m² K are expected for Reynolds numbers of 3000. These early tests were conducted up to a temperature of approximately 700°C and a flux of 43 W/cm². One major concern with respect to external DARs is the influence of wind on the falling film (making it unstable and resulting in hotspots or fluid loss) and fluid fouling from environmental exposure. For

an internal DAR (which reduces wind impact on the falling film), the solar flux is incident on a curtain of the flowing liquid film along the internal wall of the cavity, [108].

The heat transfer medium enters the receiver at the top of the enclosure and travels radially down an inclined absorber sidewall. The receiver's inner absorber walls are heated by incident radiation entering through the face down aperture at the bottom of the enclosure. The liquid HTF film established on the absorber walls is semi-transparent to the radiation, allowing the walls to heat up (solar salt is transparent in the solar spectrum [93,109] with absorption bands in the mid-infrared region [110]). As the liquid working fluid travels down the absorber walls, convection heats the falling fluid. While the liquid is somewhat protected by its location in a cavity, it is exposed to the environmental conditions and wind. DARs have the potential to achieve higher receiver outlet temperatures than conventional salt-in-tube receivers [111] and is one of the primary reasons for its consideration. Initial studies of a face-down cavity receiver indicate that for a design power having 250 MW with an inlet temperature of 385°C and an outlet temperature of 620°C, the reflection losses are 1.4% and the thermal radiation losses are 3.9% [108]. Despite, the open aperture, convection losses are estimated to be about 0.8% (following [112]) without wind due to the face-down concept, where the hot air is expected to stagnate inside the cavity. In practice, the convection losses may approach the radiation losses in magnitude [76]. The highest reflection losses occur when the absorber walls approach a vertical orientation. However, increasing inclination also increases absorber wall height and overall receiver surface area for additional losses. These competing effects drive the absorber wall inclination to an optimum angle. Design point efficiencies have been projected to be as high as 94.5% for an internal, direct exposure, falling film receiver. Additional

configurations of the internal DAR approach include rotating or inclined absorber walls to increase efficiency [108]. The inclined walls can significantly reduce reflection losses but only slightly increase thermal radiation losses. Rotation mitigates excessive temperature variation in the fluid and increases film stability. The variation of the angular velocity allows different operating strategies. The standard strategy intends to apply angular velocities below the velocity for which the liquid film would start to flow upward on the inclined plane. For non-rotating absorber wall, overheating of the working fluid can be avoided by mass flow control whereby higher flow rates are supplied in areas of highest flux. Concerns for a DAR (internal or external) mainly stem from the exposure of the salt to the environment as the working fluid can be prone to contamination, film instability and wind effects. Despite these concerns, one projection is that the annual power production using a DAR can be up to 14% greater than that of a comparable tube receiver [113]. To avoid the weaknesses of direct-absorption techniques and the exposure of the receiver heat transfer fluid to the environment, indirect exposure internal film receiver designs have been proposed wherein the liquid film is on an internal surface of an inclined cavity wall. In this approach, the wall is illuminated (heated) by concentrated flux and the film in contact with the backside of this wall is the mechanism of heat transfer from the irradiated surface. Numerous potential advantages of the internal film receiver as compared to salt-in-tube and DAR receivers have been outlined [114]. Since the receiver fluid is not used as a volumetric absorber, no dopants are required. Projections of a 6% system cost reduction versus salt-in-tube approaches have been made due to its simplicity in manufacturing and assembly [115]. This cost reduction is largely due to the receiver weight reduction by replacing a tube bundle with a thin plate and a smaller pump requirement (from reduced

pressure losses). The material reduction is also expected to reduce startup times. Initial examination of this design concept yielded efficiency values near 60% at incident powers from 240 kW to 577 kW; efficiencies greater than 80% are anticipated through straightforward design improvements [116].

2. 2. Developing challenges

Overall enhancement of efficiency is not only illustrated with thermal efficiency, but it also required suitable design and structure to obtain the pressure drop in acceptable limits. The structure of solar receivers is highly attributed to their hydrodynamic and thermal performance. Many researchers have studied the hydrodynamic behaviors of solar volumetric receivers in order to distinguish those phenomena which are responsible for pressure losses and affect the heat transfer by convection [117]. The porous media of SiC receivers produce some undesirable flow instabilities that decreases the thermal efficiency and may also cause of structural damages. Consequently, many studies have been done by both approaches experimental and numerical which aim to identify the conditions and causes that initiate and promote this phenomenon [46].

S. Mey-Cloutier et al. [21] performed an experimental investigation to achieve the best thermal-to-electric power ratio by using the various structure of solar volumetric receiver. The thermal performance of the honeycomb structure was compared with the open-cell foam structure of the solar volumetric receiver. Best receiver efficiency was achieved while using lower porosity with a small pore size of the open-cell foam structure. X. Chen et al. [22] numerically determined the influence of pore parameters and geometrical dimension designs on the thermal performance of the porous solar volumetric receiver. The results of

this study concluded that a single porous layer structure exhibited a high-temperature gradient and low porosity decreased the temperature gradient, and trends of temperature gradient remained the same for mean pore size.

The reason behind constructing solar receivers from porous structures is because of minimizing energy losses due to solar reflection and thermal emission. These losses are circumvented by introducing a porous structure to trap the radiation into the pores and enhance the convection phenomena. The reflection losses were studied experimentally and numerically incorporating the porous SiC volumetric receivers [120,121]. Monte Carlo Ray Tracing method was applied to determine the total reflection. It was shown that the variation of porosity and pore size spectral behavior had a considerable effect on the magnitude of the reflection. Moreover, augmentation in thermal performance of volumetric receivers for harvesting solar energy while leading to improved design and its configuration is essentially required. Several studies have been carried out to analyze the performance of solar energy harvesting and conversion devices with the use of volumetric receivers. Gu et al. [122] investigated the thermal performance of the refrigeration system operating on solar absorption. They indicated that the performance of the solar absorption refrigeration system increased while using the large wet ratio of the surface. Overall enhancement of efficiency was not only illustrated with thermal efficiency, but it also required for a suitable design to obtain the pressure drop in acceptable limits.

On the other hand, the thermal efficiencies of solar receivers tend to reduce at elevated temperatures because of the emission of thermal radiation from material to surroundings. Hence, the synergic behavior of thermal and mechanical efficiencies of the solar receiver is an essential when optimizing device design. One study performed experiment to

optimize the porous solar receiver by considering the solar absorption, porosity, pore diameter, and material of the porous receiver (SiC and ZrB₂). The optimized values of pore diameter and porosity revealed that convective heat exchange between a solar receiver and fluid flow should remain higher [123]. The solar receiver material can be optimized according to durability, solar absorption, and radiation emissivity. Agrafiotis et al. [30] employed the monolithic honeycomb porous SiC (MH-SiC) volumetric receivers for solar energy harvesting. The comparison was made before and after the long duration of operation of the device for assessing the durability of the thermomechanical properties. The long-term exposure of MH-SiC to solar concentration produced hardness and exhibited more compressive strength. Hence, the proper selection of pore size and porosity led to improve the sustainability of the mechanical properties. The influence of geometric parameters on heat transfer fluids remains critical particularly for multi cavity tubular arrangement of solar receivers [30]. In this case, approximately 3 times higher thermal efficiency was achieved while using water (75.81%) as a heat transfer fluid as compared to air (21.1%); however, water resulted in a slightly higher pressure drop than air. Andreozzi et al. [124] incorporated the two-equation radiation model for numerical investigation of porous SiC foam. The findings revealed that cell size exhibited a significant influence on receiver efficiency.

Furthermore, the porous structure of volumetric receivers ensures that absorbing heat inside cavities and transferring to working fluid must be simultaneous. For example, A. Carotenuto et al. [125] numerically optimized the design of a multi-cavity volumetric receiver with ceramic material properties by concerning the outlet air temperature. The first

successful pre-commercial trial of volumetric receiver utility in tower plant having capacity of 1.5 MWe was installed in Julich, Germany and its operation started in June 2006 [20].

However, recently, various other modifications have been incorporated in concentration solar plants to achieve optimal performance including the utility of ceramic foams, oxide and non-oxide ceramic receivers. T. Fend et al. [24] experimentally determined that porous material, design of receiver, and operating conditions had significant effect on the thermal performance of volumetric receivers. In addition, the thermal conductivity of porous material and solar radiation distribution cannot be neglected for performance optimization.

Z. D. Cheng et al. [126] numerically investigated the performance of a pressurized volumetric receiver and solved using Monte Carlo Ray-trace (MCRT). Further complexity of this investigation includes radiation-convection-conduction coupled heat transfer mechanisms. Similarly, X. Lui et al. [127] involved various absorbing-emitting-scattering functions in their study about pressurized volumetric receiver combined with natural convection. They modeled the slot-vented room with the radiant floor heating unit for their investigation. L. Chandra et al. [128] investigated the various geometry configurations to get the best results of recirculating performance of heat transfer fluid and avoid the flow instabilities. Many flow paths have been investigated in this study including circular channel, square channel, triangular channel and hexagonal channel.

The design of volumetric receiver and its channel configurations are the most critical factor for obtaining effective heat transfer. Volumetric receivers are available in several shapes, the most common shape is cylindrical or tabular shape to avoid the pumping power [129]. Various researchers worked on the design of reflector dish that is installed for concentrating radiation which directs toward the receiver. Their motive was to optimize reflector dishes

that could lead to high concentrating radiation which results in high receiver temperature. J. A. Harriest and T G. Lenz [130] designed parabolic dish with different rim angles to direct radiation onto the receiver cavity. The shapes of receiver cavity were of five different configurations including cylindrical, elliptical, conical, heteroconical, and spherical. Their observations concluded that rim angle had more effect on performance of CSP while shapes of receiver cavity showed almost same results. Likewise, C. Zou et al. [131] proposed and optimized solar cavity receiver having helical pipe shape. They compared the performance of helical-shaped solar receiver with the cylindrical-shaped receiver and the helical-shaped solar receiver was found more efficient.

It is well-known that the channel's design/configuration plays a crucial role in heat transfer management and pressure drop. Geometry configuration for CSP should also be explored widely to avoid flow instability and heat load [132]. The geometric configurations in result of different pore sizes and porosity, optical properties (absorptivity, diffusion phase, scattering functions, etc.), thermophysical properties (thermal conductivity and emissivity), and volumetric receiver materials (steel, ceramics etc.), are studied extensively. However, in these configurations, the flow path is not developed in such a way that the maximum amount/part of solar radiation must be recovered through convection. S. Lim et al. [133] numerically optimized the design of tubular porous receivers for heat transfer application. In which, optimization parameters include design factors such as length and diameter of tube, porosity, and also thermal conductivity of porous medium.

Many aspects of the thermal properties of solar volumetric receivers are mentioned in the literature such as incident radiation propagation within the receivers, the temperature distribution throughout solid as well as in fluid. These distributions have been calculated

by coupled simulation of convective-conductive-radiative heat transfer model, but the measurement of real achievable thermal efficiency is possible with the help of experimental strategy.

2. 3. Manufacturing challenges

Production of SiC honeycomb structure is mostly involves recrystallization and infiltration which results in recrystallized SiC (reSiC) [134] and infiltrated SiC (SiSiC) [135], respectively. However, for manufacturing of SiC open-cell foam (OCF) receivers, most commonly used method is the polymer foam replication process [62]. These methods utilize different amounts of additive agents including binders, organic polymers, lubricants, plasticizers, dispersing agents and solvents, which possess significant influence on their radiative properties and thermal performance. However, behavior of these agents is still not explored in detail after and before being exposed to high concentrated radiation.

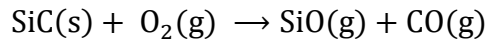
2. 4. Suitable materials

The properties of material for receivers must have high resistive towards oxidation environment, show stability even at remarkably high temperature and also capable of high absorptance of solar spectrum (frequency) band along with low emittance of infrared ray. The purpose to overcome the heat losses by infrared emission, the temperature of receivers maintains at that possible level where effectiveness become unchanged. Consequently, where receivers are not completely transfer absorbed heat by convection then high thermal conductivity is critically important as convective coefficient to conduct heat into the structure. Similarly, receivers with high convective heat transfer coefficient and low thermal conductivity can be source of temperature gradient for receiver to maintain the

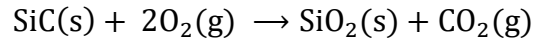
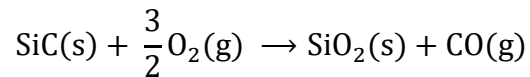
maximum temperature inside the structure. Moreover, high temperature can also be barrier for oxidation of metal oxide ceramics such as alumina (Al_2O_3) and silica (SiO_2). However, pale color of metal ceramics reduces the ability of absorptivity along with low thermal conductivity also limits the thermal diffusion. Nevertheless, SiC ceramics have remarkable ability of radiative and thermal properties, this is the reason for extensive use of SiC receivers in OCF employed as volumetric receivers.

The setting of maximum operating temperature for volumetric receivers are mainly defined by their melting point and resistance to corrosion, further oxidation of materials correspondingly to operating temperature is also considered as main factor. Several studies have been performed about oxidation activities including active oxidation, passive oxidation and active-to-passive transition on silicon carbide (SiC). The reactions belong to these oxidation processes are described below in following reaction balance.

Active oxidation:



Passive oxidation:



Active oxidation shows active oxidation of SiC that producing silicon oxide (SiO) and carbon monoxide (CO) both in gaseous phase which consequently escaped in atmosphere and cause deficiency in weight. However, in passive oxidation, silicon dioxide (SiO_2) appeared as a thin layer in solid phase on receivers that prevent the active oxidation because of its high stability at high temperature. Agrafiotis et al. [30] assessed the weight loss due

to oxidation and change in mechanical and physical properties of a set of reSiC and SiSiC receivers in the presence of solar radiation. They indicated the mechanical properties of reSiC increased while it is expose to solar heating, but it also causes of oxidation.

2. 5. Solar absorptance

Effect of high concentrated solar radiation on porous SiC receivers are rarely explored that how the absorptance property of porous SiC receiver behaves with time of operation and high temperature. Excess heat loadings can harm the materials used in fabrication of porous SiC receivers. Many researchers set the value of absorptance for porous SiC receiver constant for complete obtained range of temperature distributions in their numerical studies and the values of solar absorptance are also not related to solar radiation characteristics. Wu et al. experimentally determined the average value of solar absorptance for porous SiC OCF receiver by conduction series of cyclic test using solar spectrum.

2. 6. Solar radiation propagation

Due to technical complexities and limitations of solar radiation propagation for volumetric receiver systems, it is not widely discussed in research. Solar radiation propagation can be achieved theoretically by two approaches: the probabilistic method of Monte Carlo (MC) [1,136] and the Beer-Lambert law [41]. Monte Carlo is less accurate analytic model for calculation of orientation and eradication of solar radiation along the receiver length, it is usually applied on calculation of radiation transfer within homogeneous media. And the Beer-Lambert law is generally used as phenomenological expression for determining the solar radiation propagation in receivers. It is described by Eq. (1).

$$\frac{I(x)}{I} = \exp(-\beta x) \quad (1)$$

Where $I(x)$ represents the non-absorbed local flux, I is the incident flux and β is the eradication coefficient. the eradication length (τ) is given by the Beer-Lambert law which is expressed as $\tau = 1/\beta$. This model does not include the incoming light effects and its propagation, that accounted its drawback.

The fundamental difficulties of OCF structures produces the qualities of MC methods to elaborate the solar radiation propagation challenge. Homogeneous media that compromising two continuous phase the porous medium (solid) and heat transfer fluid, is considered to overcome the extra computational resources for these receivers. Concentrated solar propagation and radiation emitted within structures is taken as source of radiation term in energy equation of solid. The radiation term has been assessed by several models including the Roseland diffusion equation, the P-I model and the two-flux approximation [137,138]. The Roseland diffusion equation is mainly used for short distance when radiation travel in optically thick media and it also known as simplest radiation model. The P-I models is applied for calculation of diffuse and collimate contribution intensities that further describe the solar radiation propagation. The two-flux approximation is specifically used when isotropic assumption is made for both positive and negative directions of one-dimensional radiation propagation. The two-flux approximation is also applied to measure intensity of radiation by using radiation transfer equation.

2. 7. Pressure losses.

Many researchers have studied the hydrodynamic behavior of solar volumetric receivers to distinguish those phenomena which are responsible for pressure losses while affecting the convection heat transfer. However, improper location of the porous media in the volumetric receiver system could produce some undesirable flow instabilities that decrease the thermal efficiency and may also cause structural damages [139]. Veearagavan et al. [98] developed the analytic model to design the solar volumetric receivers. Furthermore, they showed that the analytic model can predict optimum thermal efficiency and geometry parameters especially the length of receivers. Alvarez et al. [140] demonstrated the agreement between numerical and experimental findings for plate shape solar receivers. They also introduced the numerical model to determine the thermal performance using heating curves and the numerical model coupled the one-dimensional convective equation for fluid with a two-dimensional heat equation for collector. Wang et al. [141] experimentally determined the thermal performance of SiC foam located in a solar volumetric receiver. The experimental study was performed through varying porosity and pore diameter of the foam. The use of SiC foam resulted in extremely high temperatures (more than 1000° C) during the experiment without any material damage while demonstrating high thermal stability. In this case, a high-temperature gradient was observed for the foam with high porosity because of low flow velocity that led to low interphase convection. However, a smaller pore diameter while keeping porosity constant was found to be favorable for high thermal efficiency. Likewise, Barreto et al. [142] numerically studied the thermal performance of SiC foams as solar radiation receiver having various porosities and pores sizes. They showed the effect of pore size on thermal performance was more important than the

porosity; however, both had the same influence on the pressure drop. In addition, high porosity and large pore size were found to be favorable for the pressure drop.

The impact of viscous and inertia forces within flow domain of the channel structure having solar volumetric receiver become significant reason for pressure drop and it also has intrinsically relation with heat transfer rate. When viscous effect has profound impact, then the pressure losses can be correctly described by Darcy's law. However, inertial effects become significant as viscous effect, addition term that leads to Forchheimer's law will be helpful to describe more precisely. Darcy's law [143] and Forchheimer's law [144] are shown in following Eqs. (2) and (3) respectively.

$$\frac{\Delta p}{L} = \frac{\mu}{K} u \quad (2)$$

$$\frac{\Delta p}{L} = \frac{\mu}{K} u + \rho f u^2 \quad (3)$$

In above Eqs. (5) and (6), K is representing the permeability of structure and F is used for its inertia coefficient. As above equations are indicated that K has inverse effect to pressure loss. Therefore, high permeability will be favorable for low friction, i.e., receiver architecture in solid-fluid interface reduces the pressure loss. A high inertia coefficient indicates direct relation to pressure drop which is promoted by internal mixing of heat transfer fluid and result in enhancement of heat transfer and increasing of pressure loss. When the pressure losses are described by Darcy's law (Eq. 2), inertial effects are scarce which results in limitations towards achieving maximum amount of heat transfer in receivers. Fend et al. showed the results that viscous effects restrict the convection of heat in MH architectures because these effects are much dominate than inertial effect. It is

concluded in this study that pressure losses showed remarkably linear behavior in MH while using Eq. (2).

Wu et al. [145] performed experiments for examination of pressure losses in SiC receivers that is function of pore diameter and porosity. Eq. (3) satisfy the results of pressure losses in SiC receivers. In addition, the study showed that reduction in porosity and pore diameter which increased both viscous and inertial effect to the pressure losses. It can also be noted from their results that effect of inertia is more pronounced than viscous. They concluded in their study that for improved heat transfer in SiC receivers, reduction in pore diameter and porosity is required but these geometry parameters lead to pressure losses.

CHAPTER 3

POROUS VOLUMETRIC RECEIVER ASSESSMENT

This section elaborates the process and techniques of developing a 3D model of SiC samples for the CFD package. This import includes various following steps which are briefly described.

3. 1. CT Scan Imaging

CT-scan (Skyscan 1172 Micro-CT, imaging facility provided images with a pixel size of 0.68 μm resolution) was performed on various structures (having different porosity and pore size) of porous SiC foam. After several trials, a set of 3D models in building the images with optimum parameters was achieved. Images are converted into 3D facet objects with “Skyscan CTsan” software which is generally used to construct the visual 3D object from images. In CTscan computation, linear algebraic equations of radon transform are subjected to solve the filtered back-projection or Fourier slice theorem to obtain tomographic images.

3. 2. Model Refining

After constructing a 3D porous SiC foam structure, it further requires processing for eliminating the errors, which appear while converting the structure into a volume body (CAD model). Initially, each model consists of approximately 16 million facets (triangular surfaces which define model structure). Ansys Spaceclaim was used to reduce the facets by 3D wrapping function without loss of any detail. Less number of facets consumes less

computational resources for converting the 3D object into CAD volume. The wrapping function also makes the model watertight bodies and reduced approximately 90% of facets (~1.6 million). After this arrangement, the model structure is imported into the Geomagic Wrap package to further cleaning against some errors such as self-intersecting facets, overlapping facets, small tunnels, highly creased edges, and floating facets. The cleaning of models highly depends on the type of errors present in the model. The second main purpose of Geomagic Wrap software is to reduce the triangular facets by the surfacing tool that converts the number of facets into a single patch or more than one patch according to model complexity. It is highly unfavorable to use the high number of facets (~1.6 million) because it consumes tremendous computational resources and time of solution processing. Surfacing tools created patches along with the facets and connect these patches to 3D surface objects. However, the model still had a huge number of surfaces resembling the pores more than that of the uniformly distributed pores model. Because of the actual nature of the pores in the SiC foam, some surfaces had a noticeably small area ($\sim 1 \times 10^{-6}$ m that is 10,000 times smaller than the larger ones) and edge which could increase the number of mesh elements significantly. Hence, it becomes almost possible to adopt the geometric feature of the 3D porous structure to solve governing equations in the CFD package. Therefore, the Gambit package is used to further reducing the number of faces and edges. Fig. 3.1 represents the graphical illustration of various steps of building and refining of the SiC foam, which involves geometry extraction from Spaceclaim after wrapping function that eliminates small faulty facets tunnels, Fig. 3.1(a). Fig. 3.1(b) shows the model after the cleaning process which includes removing and regenerating the intersected facets, spikes, creased angles, sharp angles, and small edges, then it is subjected for patching as

shown in Fig. 3.1(c). After patching, a deviation test is performed to check the deviation of the cleaned model from the original facet model. It can be observed that a very small deviation can be found as appeared in red in Fig. 3.1(d). Then, resolving the deviation issues, the resulting model is shown in Fig. 3.1(e). Finally, CAD geometry with considerably less number of faces is obtained as shown in Fig. 3.1(f). Further merging of faces is achieved in Gambit and grids for flow analysis is also created.

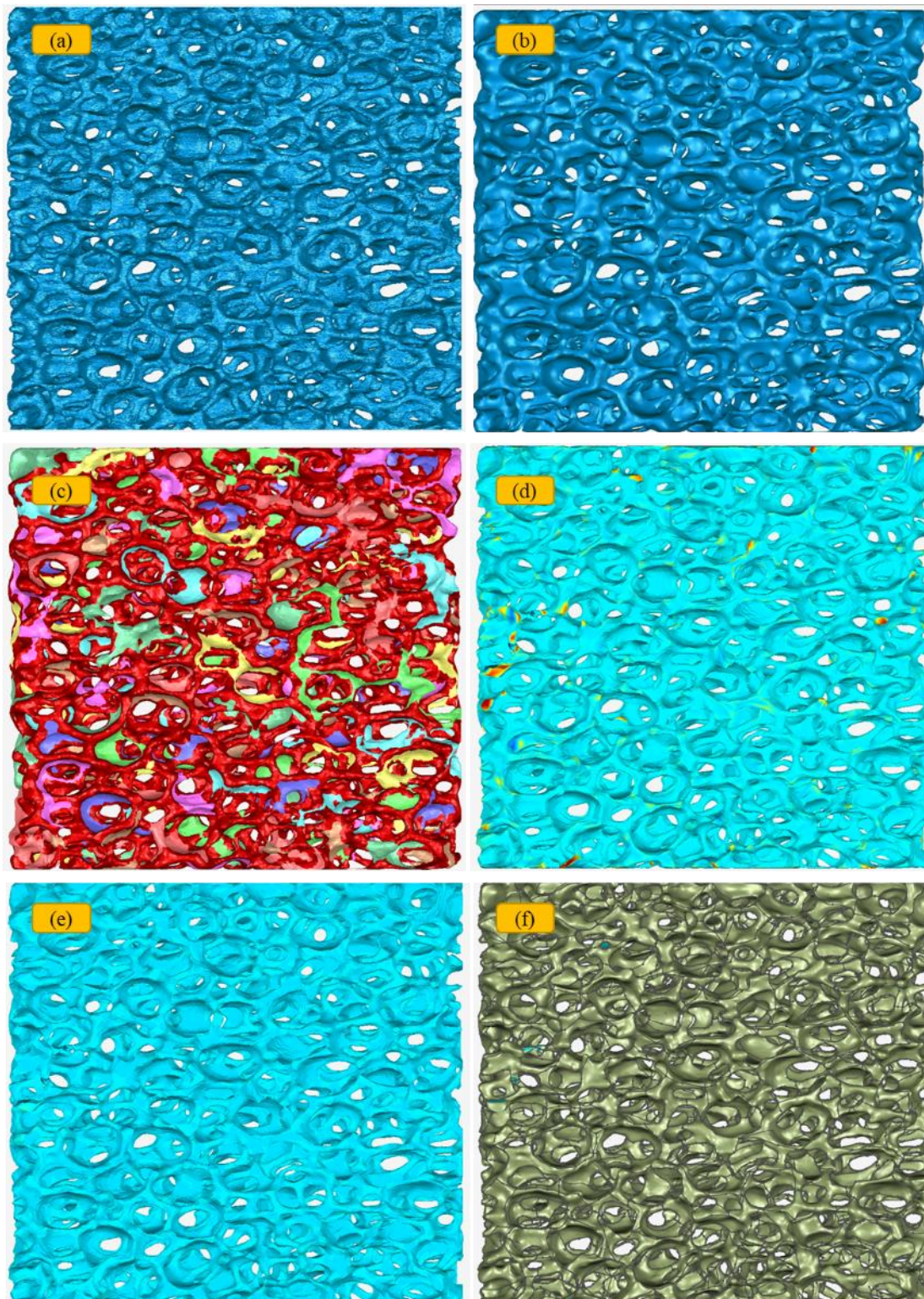


Fig. 3.1: Geometry cleaning processes include a) uncleaned sample having faulty facets, b) after cleaning in Geomagic Wrap package, c) exact patches define by coloring, d)

deviation analysis and its contour ranges dark blue (-5 mm) to red (5 mm), e) sample after eliminating deviation issues and f) the final CAD model of porous SiC foam.

3.3. Computational domain

The final structure of the 3D domain, which is extracted from Gambit is adopted for the simulations. Porous SiC foam is considered as a solid body with temperature-dependent properties of SiC and fluid domain is modeled over the solid body by using the enclosed body option of CAD software with 5mm clearance (0.02% of length) on each side as shown in Fig. 3.2. Air having temperature-dependent properties is considered as a working fluid and the flow in the channel is assumed to be steady. Porous SiC foam is employed in the enclosed fluid domain of the channel having porosity and average pore diameter of (0.62, 0.695, 0.77, 0.845) and 3mm, respectively. Detail of porous SiC foam samples are tabulated in Table 3.1. Concentrated solar radiation is applied to the computational domain from the top surface which is taken as a transparent surface and irradiates the porous SiC foam. Porous SiC foam and walls of the channel are assumed to be gray where the diffusion coefficient is set to 1. Some fraction of the radiation is reflected from SiC foam can also diffuse throughout the fluid domain. The sensible heat from solar radiation is transferred by thermal conduction within SiC foam. Concurrently, inlet airflow extracts heat from SiC foam by convection. Hence, energy balance involves all types of heat transfer including radiation coming from the concentrated solar. Although these heat transfer modes are interdependent and governed by the heat equation, most of the previous studies neglected part of them for simplicity or computational limitation. However, this study includes the conduction, convection, and radiation effects using the discrete ordinate (DO) radiation model.

Table 3.1: Detail of porous SiC foam samples used for analysis.

Sample	Porosity	Pore Diameter	PPIs
1	0.62	2.98	15
2	0.695	3.08	15
3	0.77	2.9	20
4	0.845	3.10	20

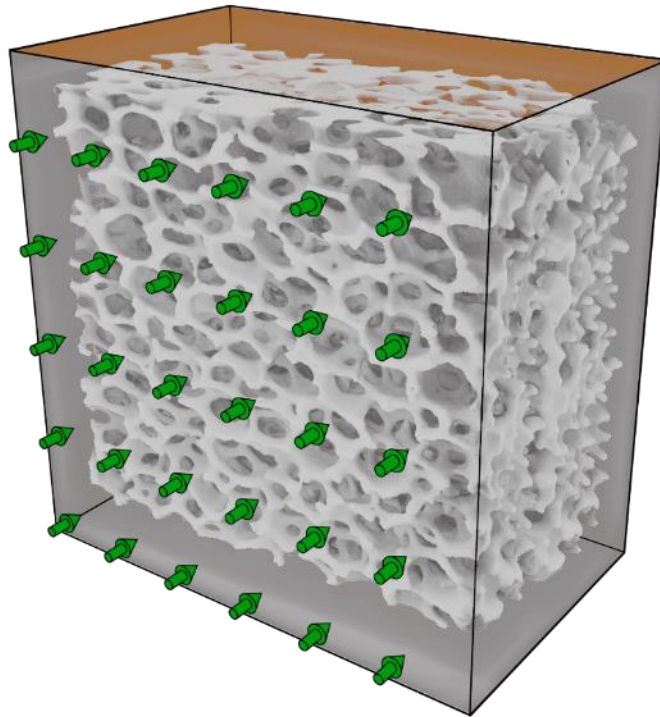


Fig. 3.2: a) Graphical illustration of a square cross-sectional channel having porous SiC foam, b) green arrow shows inlet of air flows normal to the wall and c) concentrated radiation applied on the top highlighted face.

Furthermore, in order to optimize the design of solar harvesting system for hydrodynamic and thermal performance, fluid domain thermo-flow analysis has been conducted numerically by using a cylindrical cone frustum-shaped flow domain. Flow and solar

radiation are allowed to pass through converging flow path having SiC porous as solar volumetric receiver. A schematic illustration of the flow channel and porous SiC foam is shown in Fig. 3.3. Fully developed mass-flow profile (which was computed according to inlet diameter d_{in} for every configuration) is given to inlet boundary condition. All walls are assumed to be gray and diffused. Radiation is allowed to pass through the inlet surface and subjected to scattered throughout the porous zone and heat the fluid domain. The sensible heat followed conduction heat transfer in porous SiC, convection heat recovery for fluid and radiation is considered as a heat source.

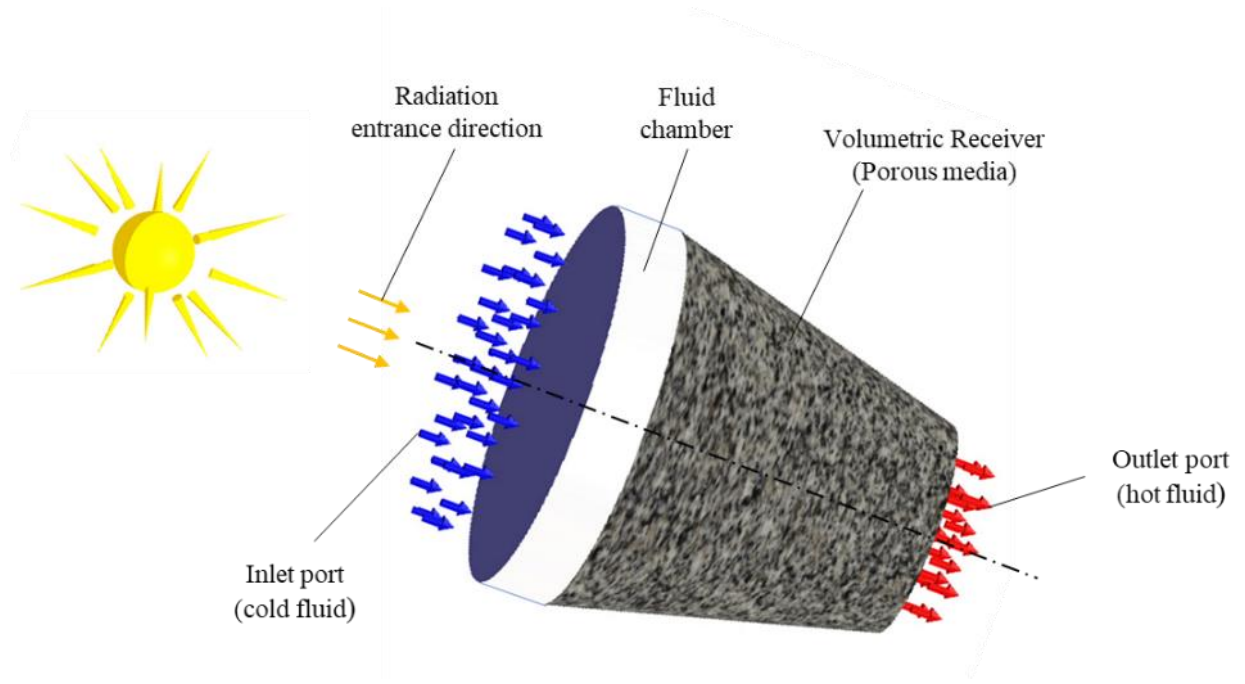


Fig. 3.3: Schematic of physical problem. Blue (cold) arrows show inlet direction of fluid while red arrows show the outlet direction of fluid.

3.3.1. Computational domain for the Constructal approach

This investigation is conducted to optimize the fluid domain using the Constructal law. For this purpose, 50 different configurations are adopted for obtaining pressure and temperature distribution (as tabulated in Table 5.3). Three main parameters were considered for geometry description of porous zone while following the volume constraint. Length (L), ratio (r) of inlet diameter (d_{in}) to outlet diameter (d_{out}) and cone angle (θ) were varied to model these configurations for porous zone. At a time, cone angle (θ) were taken with any of length (L) or ratio (r) as input for every configuration, consequently, other dimensions were calculated based on these two parameters. This study shows the enhancement of thermal performance of solar harvesting system as compared to previous work [23], therefore, volume is taken as constraint of geometry for all the configurations ($V = 9.81 \times 10^{-5} \text{ m}^3$) which is used in previous study [23] with radius (25 mm) and length (50mm) of the cylinder. Inlet diameter of the fluid chamber is designed same as the inlet diameter of porous zone with length of 10 mm. Fig. 3.4 represents the schematic for variable parameters of the computational domain.

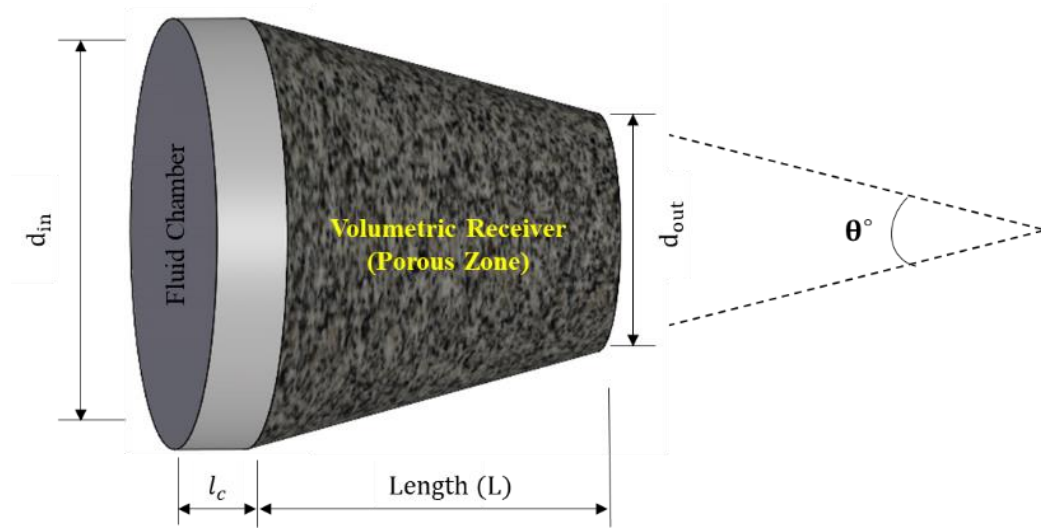


Fig. 3.4: Dimensional illustration of volumetric receiver

CHAPTER 4

NUMERICAL STRATEGY

The final CAD model of porous SiC foam is placed in a square-shaped channel to create flow boundaries as shown in Fig. 3.2. Porous SiC foam is considered as a solid body in the domain and the volume around the porous SiC foam is modeled as the fluid domain. The top wall is taken as optically transparent to allow radiation to enter into the channel and irradiate the porous SiC foam. Portion of solar radiation is reflected through porous SiC foam and scattered throughout the fluid domain. The sensible heat is thermally conducted through the solar volumetric receiver and simultaneously the heat transfer fluid (air or water) regains heat by convection.

4. 1. Mathematical modeling

The thermal performance of solar volumetric receivers is carried out numerically using three parameters for laminar flow conditions including the Reynold numbers solar concentration and porosity. In addition, turbulent flow conditions are also examined using two heat transfer fluid (air and water). For turbulent Reynolds number (varies 3000 to 4000), the $k - \epsilon$ realizable turbulent model is utilized to compute the pressure, temperature, and velocity distribution within the channel. The temperature distribution is obtained through implementing the energy equation and DO radiation model while using the temperature-dependent properties. All radiation properties such as absorption, scattering,

emission, and diffusion are also considered in the simulation of both domains (solid and fluid). Emissivity and scattering function are selected to be 0.9 and isotropic, respectively.

The absorbed solar radiation acts as a heating source in the fluid and porous foam. Hence, the hydrodynamic and thermal performance are governed by mass, momentum, and energy conservation equation with DO radiation model. The study is organized in such a way that it targets mainly pressure and temperature distribution to evaluate the normalized global Nusselt number, thermal efficiencies (based on the outlet and maximum temperatures), and total pressure drop. Since the system incorporates the turbulent flow conditions, the following set of governing equations was computed for desired parameters.

4.1.1. Continuity equation

$$\frac{\partial(\rho_f U_i)}{\partial X_i} = 0 \quad (4)$$

here ρ_f and U_i refer to density of fluid and velocity correspondingly to the cartesian coordinates (X_i), respectively.

4.1.2. Momentum equation

$$\frac{\partial(\rho_f U_i U_j)}{\partial X_i} = \frac{\partial p}{\partial X_i} + \frac{\partial}{\partial X_i} \left(\mu_f \left(\frac{\partial U_j}{\partial X_i} + \frac{\partial U_i}{\partial X_j} - \frac{2}{3} \delta_{ij} \frac{\partial U_i}{\partial X_i} \right) \right) + \frac{\partial(-\rho_f \overline{U_i U_j'})}{\partial X_i} \quad (5)$$

where μ_f and δ_{ij} denote dynamic viscosity of fluid and unit tensor in cartesian coordinates, respectively. $\frac{\partial(-\rho_f \overline{U_i U_j'})}{\partial X_i}$ shows the turbulent part of the momentum equation and known as

additional Reynold stresses, which is obtained by using $(k - \varepsilon/\text{RTM})$. For laminar flow, Reynold stresses become zero, consequently, no turbulence transport equation involve.

$$\frac{\partial(-\rho_f \overline{U_i U_j'})}{\partial X_i} = \mu_f \left(\frac{\partial U_j}{\partial X_i} + \frac{\partial U_i}{\partial X_j} \right) - \frac{2}{3} \delta_{ij} \left(\rho_f k - \mu_t \frac{\partial U_k}{\partial X_k} \right) \quad (6)$$

$$\mu_t = \rho_f C_\mu \frac{k^2}{\varepsilon} \quad (7)$$

where μ_t is eddy viscosity or turbulent viscosity. U_k corresponds to velocity obtained from k-transport equation of $k - \varepsilon/\text{RTM}$. C_μ serves as a limiter for high Reynolds numbers and is formulated in Eq. (8). In another $k - \varepsilon$ turbulent model, C_μ is constant and it also distinguish $k - \varepsilon/\text{RTM}$.

$$C_\mu = \frac{1}{A_0 + A_s \frac{k U^*}{\varepsilon}} \quad (8)$$

here A_0 is model constant and equal to 4.04. A_s is also model constant and related to strain rate [146].

Above additional turbulent terms are solved using two transport equation, turbulent kinetic energy (k) and dissipation rate (ε). The basic approach to understanding mathematics of $k - \varepsilon/\text{RTM}$ involves combining the Boussinesq relationship and the eddy viscosity (μ_t). To model realizable terms in $k - \varepsilon/\text{RTM}$, these quantities must be positive. Furthermore, it also satisfies some specific constraints on the Reynolds stresses, especially related to the physics of the turbulent flow. An additional advantage of the implementation of $k -$

ε /RTM is that it predicts more accurately when the flow path is involved in adverse pressure gradients, separation, and recirculation [146]. Modeling flow system of the porous medium, this advantageous feature of $k - \varepsilon$ /RTM supports significantly to capture turbulence along the flow path through the porous channel [147]. Mathematical modeling of $k - \varepsilon$ /RTM is described below;

$$\frac{\partial(\rho_f k U_j)}{\partial X_i} = \frac{\partial}{\partial X_i} \left(\left(\mu_f + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial X_i} \right) + P_k + G_b - \rho_f \varepsilon \quad (9)$$

$$\frac{\partial(\rho_f \varepsilon U_j)}{\partial X_i} = \frac{\partial}{\partial X_i} \left(\left(\mu_f + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial X_i} \right) + \rho_f C_1 S \varepsilon - \rho_f C_2 \frac{\varepsilon^2}{k + \sqrt{\nu \varepsilon}} + \rho_f C_{1\varepsilon} \frac{\varepsilon}{k} C_{3\varepsilon} G_b \quad (10)$$

where P_k refers to the production of turbulent kinetic energy due to the mean velocity gradient ($G_k = -\rho_f \overline{U_i U_j} \frac{\partial U_j}{\partial X_i}$). Likewise, G_b represents the production of turbulent kinetic energy because of buoyancy force. Additionally, σ_k and σ_ε refer to the turbulent Prandtl number for k and ε , respectively. While C_1 , C_2 , $C_{1\varepsilon}$ and $C_{3\varepsilon}$ are constants.

4.1.3. Volume-averaged governing equations

Due to complexity of computational domain for analysis of different channel configuration, the porous medium is modeled using a dual-cell zone approach to compute LTNE model. The following assumptions are made to govern the volume-averaged mass conservation, momentum conservation and energy equations including: 1) no-slip velocity at walls, 2) flow is assumed to be steady and turbulent, border effects were ignored because of a large domain, 3) buoyancy effects, hydrodynamic dispersion, and thermal expansion were also neglected, 4) diffusion coefficient and refractive index are taken 1 for all wall, and 5)

isotropic scattering phase function (Fig. 5.23). In addition, the numerical implementation is predicted using superficial volume-averaged continuity and Navier-stokes equations. Since Reynolds number (calculated based on inlet hydraulic diameter) lies within the range of turbulence, thus the behavior of flow is turbulent. Therefore, velocity and pressure distribution were obtained using the $k - \varepsilon$ /Standard turbulence model ($k - \varepsilon$ /STM). The temperature distribution was computed via energy equation and LTNE model while radiation heat source was dealt with DO radiation model. Reason behind the implementation of DO radiation model is that it is able to include the solution of absorption, scattering, emission, and diffusion for both domains (fluid and solid). Having the above assumptions, volume averaged transport equations are modified as follow;

$$\nabla \cdot (\Psi \rho_f \bar{U}) = 0 \quad (11)$$

$$\begin{aligned} \nabla \cdot \left(\rho_f \frac{\bar{U}\bar{U}}{\Psi} \right) = & -\nabla(\Psi \langle \bar{p} \rangle) + \nabla \cdot \mu_f \left[\nabla \bar{U} + (\nabla \bar{U})^T - \frac{2}{3} \bar{\delta} (\nabla \bar{U}) \right] + \nabla \\ & \cdot [-\Psi \rho_f \langle \bar{U}'\bar{U}' \rangle] + \Psi S \end{aligned} \quad (12)$$

Here, $\langle \rangle$ used for representation of volume-averaged quantity. Similarly, $\bar{U}$ represents volume-averaged velocity, which is also known as superficial velocity. $\bar{p}$ denotes pressure. Turbulence is incorporated by Reynolds stresses ($\overline{U'U'}$) in momentum conservation. Volume fraction (fluid-to-solid) is introduced by local porosity (Ψ). It is worth noting, this local porosity is important to describe flow through porous medium. That is reason, for fluid chamber porosity become 1 which represents no solid zone. S denotes the source term in momentum conversation which in results of pressure drop that incorporates the momentum sink parameters; permeability coefficient (K) and inertial resistance (f).

$$S = \left(\frac{\mu}{K} \bar{U} + \frac{\rho}{f} |\bar{U}| \bar{U} \right) \quad (13)$$

$$\nabla \cdot [-\Psi \rho_f \langle \bar{U}' \bar{U}' \rangle] = \mu_f [\nabla \bar{U} + (\nabla \bar{U})^T] - \frac{2}{3} \bar{\delta} (\rho_f \langle k \rangle - \mu_t \nabla \bar{U}_k) \quad (14)$$

Where, Reynold stresses are dealt with using $k - \varepsilon/\text{STM}$. $\bar{U}_k$ is in the result of volume-averaged k -transport equation of $k - \varepsilon/\text{STM}$. Similarly, C_μ works as a limiter for high Reynolds numbers which is defined in Eq. (8). The turbulent terms (k and ε) for dual-cell approach are computed using additional transport equations which are mentioned below. The mathematics of these transport equations involves the Boussinesq relationship and the eddy viscosity (μ_t).

$$\nabla \cdot (\rho_f \bar{U} \langle k \rangle) = \nabla \cdot \left(\left(\mu_f + \frac{\mu_t}{\sigma_k} \right) \nabla \langle \Psi k \rangle \right) + P_k + G_b - \rho_f \Psi \langle \varepsilon \rangle \quad (15)$$

$$\nabla \cdot (\rho_f \bar{U} \langle \varepsilon \rangle) = \nabla \cdot \left(\left(\mu_f + \frac{\mu_t}{\sigma_\varepsilon} \right) \nabla \langle \Psi \varepsilon \rangle \right) + \rho_f P_k C_1 \frac{\langle \varepsilon \rangle}{\langle k \rangle} + \rho_f C_2 \frac{\langle \varepsilon \rangle}{\langle k \rangle} (G_b - \Psi \langle \varepsilon \rangle) \quad (16)$$

The production term can be calculated using the following equation.

$$P_k = -\Psi \rho_f \langle \bar{U}' \bar{U}' \rangle \nabla \bar{U} \quad (17)$$

4.1.4. Energy equation for the liquid domain

$$\frac{\partial(U_i T_f)}{\partial X_i} = \frac{\partial}{\partial X_i} \left(\alpha_f \frac{\partial T_f}{\partial X_i} - \overline{U_j' T_f'} \right) - \nabla \cdot q_{\text{rad}} \quad (18)$$

where q_{rad} defines radiation source term. α_f and T_f refer to thermal conductivity and temperature of the fluid. The additional heat flux term $(-\overline{U_j' T_f'})$ corresponds to turbulent part and being modeled using Reynolds's analogy which relates the energy equation and turbulent momentum as:

4.1.5. Energy equation for volume-averaged energy equation

Basic two approaches were commonly utilized for handling heat transfer in porous medium while having volume-averaged equations. The first approach is local thermal equilibrium (LTE) model that treats fluid and solid as a single cell, thus the temperature distribution for both fluid and solid is found to be same. The second approach is local thermal non-equilibrium (LTNE) model that considers the dual-cell approach and solves temperature distribution according to volume fraction (porosity) and effective thermal conductivity. LTNE gives temperature distribution for both fluid and solid phases by using their heat transfer own heat transfer's equations. Since the flow system is turbulent, the energy equation can be written as below.

$$\nabla \cdot (\rho_f c_{p,f} \langle \bar{U} T_f \rangle) = \nabla \cdot (\Psi k_f \cdot \nabla \langle T_f \rangle - \overline{U' T_f'}) + h_v (\langle T_f \rangle - \langle T_s \rangle) \quad (19)$$

$$\nabla \cdot ((1 - \Psi) k_{\text{eff},s} \cdot \nabla \langle T_s \rangle) = h_v (\langle T_f \rangle - \langle T_s \rangle) + \nabla \cdot q_{\text{rad}} \quad (20)$$

where, T_f and T_s denote fluid and solid temperature, respectively. $(-\overline{U'T_f'})$ belongs to the turbulent part and is being incorporated using Reynolds's analogy which relates the energy equation and turbulent momentum as described in Eq. (14). A radiation source term ($\nabla \cdot q_{\text{rad}}$) is included in the solid energy equation as a heat source and divergence of the radiative flux. h_v corresponds to volumetric convection term and establishes the coupling between solid and fluid phases.

$$-\overline{U'T_f'} = \frac{\mu_t}{\sigma_t} (\nabla \cdot T_f') \quad (21)$$

4.1.6. Energy equation for solid domain

$$-\nabla \cdot q_{\text{rad}} + k_s \frac{\partial}{\partial X_i} \left(\frac{\partial T_s}{\partial X_i} \right) = 0 \quad (22)$$

here T_s and k_s are temperature and thermal conductivity of solid, respectively.

4.1.7. Radiation heat source

$$\nabla \cdot q_{\text{rad}} = a_f(4\sigma T_s^4 - G) \quad (23)$$

$$G = \int_0^{4\pi} I(\vec{r}, \vec{s}) \, d\Omega \quad (24)$$

here G is integral of radiation intensity (I).

4.1.8. Porous zone modeling for dual-cell approach

Porous zone modeling in ANSYS Fluent is highly dependent on effective conductivity, pressure drop relation (momentum sink), and optical properties. Due to less specific knowledge about the precise effective conductivity and also for simplification of the porous zone, it is commonly assumed isotropic media. Therefore, the effective conductivity of porous zone is usually modeled using correlations containing the thermal conductivity of fluid and solid. The Schuetz-Glicksman [148] relation for effective conductivity is considered a more accurate formula and also proven by Kamiuto and Tseng's investigation [149]. Correlations for effective conductivity of porous media are described as below.

$$k_{\text{eff},f} = \Psi k_f \quad (25)$$

$$k_{\text{eff},s} = \frac{1}{3} (1 - \Psi) k_s \quad (26)$$

$$k_{\text{eff}} = k_{\text{eff},f} + k_{\text{eff},s} \quad (27)$$

Furthermore, based on the above effectivity conductivity correlations, the relation for the volumetric heat transfer coefficient is obtained as follow.

$$h_v = (32.50 \Psi^{0.38} - 109.94 \Psi^{1.38} + 166.65 \Psi^{2.38} - 86.98 \Psi^{3.38}) k_f \frac{\text{Re}^{0.438} \Psi^{0.438}}{d_c^2} \quad (28)$$

In the above-mentioned momentum equation, the source term (S) is defined using relations resulting from pressure drop of porous medium based on experimental data [23]. This

source term involves two parameters are the permeability coefficient (K) and the inertial resistance (f). Many researchers have developed correlations of pressure drop depending on characteristics length and the porosity of porous media. Permeability coefficient (K) and inertial resistance (f) are described as below based on porosity.

$$K = \frac{d_c^2}{1039 - 1002\Psi} \quad (29)$$

$$f = \frac{0.5138 \Psi^{-5.739}}{d_c} \quad (30)$$

Where, d_c is average cell diameter of porous medium.

Optimal properties of porous media are mainly related to absorption coefficient (a) and scattering coefficient (σ_s) and their correlation is defined as below.

$$a = \frac{1.5 \propto (1 - \Psi)}{d_c} \quad (31)$$

$$a + \sigma_s = \frac{3(1 - \varphi)}{d_c} \quad (32)$$

Where, $\propto$ refers to absorptivity.

4.1.9. DO radiation modeling

Unlike other radiation models in Ansys Fluent, DO radiation model has an advantage while considering radiation parameters such as scattering terms, spatial discretization and angular discretization of the radiation beam, participation of fluid medium in radiation, and

different optical properties (semi-transparent, non-gray, and opaque walls). Furthermore, DO radiation model also provides good accuracy of solution for unstructured meshes. DO radiation model allows to solve the finite number of discrete solid angles with the help of radiation transfer equation. Each solid angle corresponds to vector direction ($\vec{s}$) according to the global Cartesian system. Control of further division of this angular discretization is available in DO radiation model [150].

Radiation transfer equation for DO model is described as

$$\nabla \cdot (I(\vec{r}, \vec{s}) \vec{s}) + (a + \sigma_s) I(\vec{r}, \vec{s}) = an^2 \frac{\sigma T^4}{\pi} + \frac{\sigma_s}{4\pi} \int_0^{4\pi} I(\vec{r}, \vec{s}') \Phi(\vec{s} \cdot \vec{s}') d\Omega' \quad (33)$$

here λ refers to wavelength, a and σ_s define spectral absorption coefficient and scattering coefficient, n is refractive index and $\Phi(\vec{s} \cdot \vec{s}')$ refers to isotropic phase function.

The total intensity $I(\vec{r}, \vec{s})$ in each direction vector $\vec{s}$ along with position $\vec{r}$ is solved through Eq. (17) and this summation corresponds to the complete wavelength band.

$$I(\vec{r}, \vec{s}) = \sum_k I_{\lambda_k}(\vec{r}, \vec{s}) \Delta\lambda_k \quad (34)$$

The scattering phase function is generally described as following: (known as anisotropic phase function)

$$\Phi(\vec{s} \cdot \vec{s}') = p(\cos \theta) \quad (35)$$

$$\frac{1}{4\pi} \int_0^{2\pi} \int_0^\pi p(\cos \theta) \sin \theta \, d\theta \, d\phi = 1 \quad (36)$$

Since interface boundary is present at the interaction of fluid and solid, Thus, Eq. (23) indicates the transfer of radiation heat for interface boundaries between fluid and solid regions.

$$k_s \nabla T_{s, \text{ interface}} \cdot \vec{n} = k_f \nabla T_{f, \text{ interface}} \cdot \vec{n} + q_{r, \text{ interface}} \quad (37)$$

where, $\vec{n}$ is a unit vector which is normal to the interface and directs the incident radiation to fluid from the solid and $q_{r, \text{ interface}}$ is occurred due to reflection and re-radiation. Hence, radiation heat flux at the interface $q_{r, \text{ interface}}$ can be expressed as below:

$$q_{r, \text{ interface}} = q_{\text{out}} - q_{\text{in}} \quad (38)$$

$$q_{\text{out}} = (1 - \epsilon)q_{\text{in}} + \epsilon \sigma T_{s, \text{ interface}}^4 \quad (39)$$

$$q_{\text{in}} = \int_0^{4\pi} I(\vec{r}, \vec{s}) \, d\Omega \quad (40)$$

Herein, ϵ and σ represent the emissivity of solid and the Stefan-Boltzmann constant, respectively. $T_{s, \text{ interface}}$ is temperature of solid at interface wall.

4.1.10.Performing parameters

4.1.11.Overall heat transfer coefficient

In general, the definition of overall heat transfer implies the effectiveness of fluid film over the solid wall to drive out the heat flux. In the current case, the area-weighted integral is applied on SiC foam surface and correspondingly mass-weighted integral is applied on interacted air film to calculate temperatures and, accordingly, the total surface heat flux on a wall is directly calculated. Hence, the overall heat transfer coefficient is expressed as:

$$h = \frac{q''}{A (T_s - T_f)} \quad (41)$$

here, h is the overall heat transfer coefficient, q'' and A denote heat flux and area of wall surface, respectively.

4.1.12.Global Nusselt number

Estimation of the ratio between convection and conduction heat transfer for the entire system is known as Global Nusselt Number. Nusselt number shows that how much convection heat transfer is effective or dominant over conduction heat transfer. Mathematically expression for Global Nusselt number is defined as;

$$Nu = \frac{q'' \times D_h}{(T_{sf} - T_m) k_f} \quad (42)$$

Here, D_h is the hydraulic diameter. T_{sf} and T_m are mean surface and fluid temperature for the entire system.

4.1.13. Other expressions and temperature-dependent properties

Following relations are implemented for pre-processing and processing of simulation. Eq. (43) is applied for the calculation of mass-flow rate based on the Reynolds number which is given at inlet condition [151]. Eqs. (44 - 47) are used for temperature-dependent thermophysical properties of the fluid while the solid properties are taken from [152].

$$Re = \frac{\dot{m} D_h}{A_h \mu_f} \quad (43)$$

$$\rho_f = 3.018 e^{-0.00574 T} + 0.8063 e^{-0.0008381 T} \quad (44)$$

$$\mu_f = \frac{0.000001458 T^{1.5}}{T + 110.4} \quad (45)$$

$$c_p = 1062 - 4.685e^{-1} \cdot T + 1.187e^{-3} \cdot T^2 - 8.332e^{-7} \cdot T^3 + 1.968e^{-10} \cdot T^4 \quad (46)$$

$$k_f = 5.786e^{-4} - 9.986e^{-5} \cdot T + 4.103e^{-8} \cdot T^2 - 9.354e^{-12} \cdot T^3 \quad (47)$$

where Re , $\dot{m}$, A_h and μ_f are Reynolds number, mass-flow rate, area of inlet, and viscosity of fluid (considered at 25°C).

4. 2. Solution methodology

The governing equations are computed considering fluid flow coupling with energy equation and DO radiation formulation. The no-slip condition is assumed for all walls in the channel. For pressure-velocity coupling, SIMPLEC algorithm with ‘Green Gauss Node Based’ spatial discretization is applied. Convergence criteria for pressure, momentum,

turbulent kinetic energy, turbulent dissipation rate, energy, and radiation are second-order, quick, third order, third order, quick, and first-order upwind, respectively. Controls for under-relaxation factors of pressure, density, body forces, momentum, turbulent kinetic energy, turbulent dissipation rate, turbulent viscosity, energy, and discrete ordinates are limited to 0.25, 0.8, 0.8, 0.6, 0.75, 0.75, 0.9 0.8 and 0.9, respectively.

The direction of incident radiation on the top plane is with a beam angle (α) of $54\pi/180$ rad. Division and pixels of angular discretization (N_θ and N_ϕ) are assumed the same for all the cases in these simulations. N_θ and N_ϕ represents zenith and azimuth angles, and their values for pixilation and division are taken 3x3 and 3x3, respectively. We chose the number of pixilation and division as recommended in previous studies [150].

4.2.1. Mesh independent tests

Several mesh sizes (very coarse, coarse, medium, and fine) are generated for grid distribution for the solution domain to test the mesh independency. The grid parameters (cell size and growth rate) are applied to ensure that the final mesh is under the acceptable value of skewness and aspect ratio is possibly kept the same throughout the domain. $k - \varepsilon$ realizable turbulent model is utilized for mesh independence test at $\text{DNI} = 250 \text{ kW/m}^2$, $\text{Re} = 3000$ and $\Psi = 0.77$. Meshes are generated in such a way that they are denser near the wall where field variables have greater influence. Tetrahedral mesh with Tgrid scheme is applied in the numerical investigation. Detail of meshes respective to their results is tabulated in Fig. 4.2 and graphical representation of meshes is shown in Fig. 4.1. It can be observed that at higher mesh density, the variation in results becomes negligible or it can be said that the solution is independent of mesh size/number. Therefore, the medium mesh

(with 12 million number of element) is used for the analysis to circumvent the high computational power. The convergence of the solution is monitored by residuals which were set to 10^{-6} for momentum, turbulence, and continuity and to 10^{-9} for the energy and radiation equation. Extra 3000 iterations were confirmed after achieving the convergence criteria.

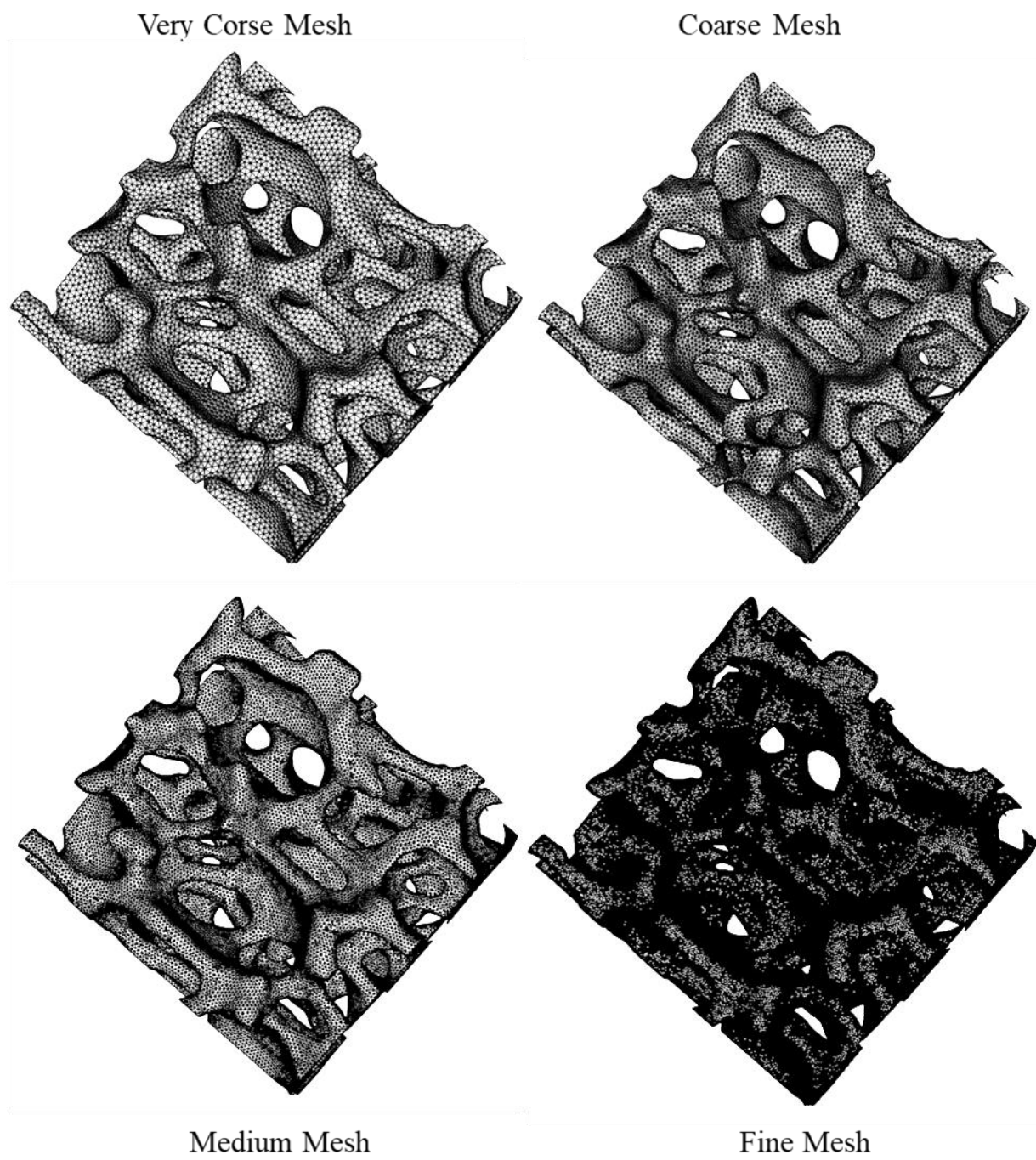


Fig. 4.1: Graphical illustration of different meshes used for mesh independence test.

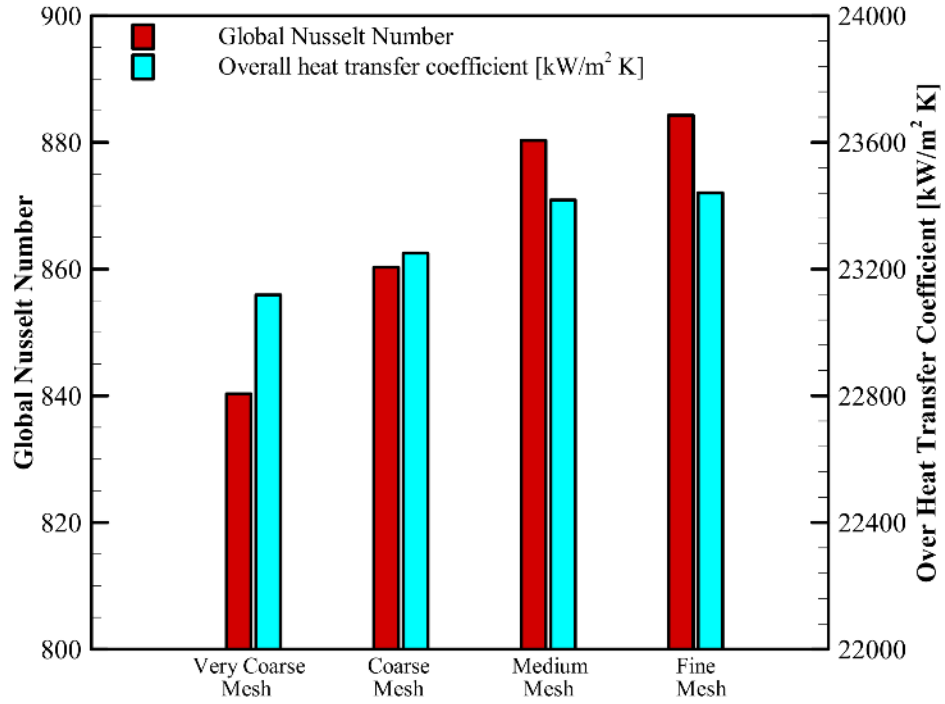


Fig. 4.2: Detail of meshes used for grid independence test. No. of cells for very coarse, coarse, medium and fine meshes are approximately 7.5, 9, 11, and 14 million.

It is worth mentioning that the predictions should be independent of the turbulence model as well. Every turbulence model has some limitations for certain type of problems. Thus, we here considered two turbulence models from $k-\Omega$ family (SST 4 equation and $k-\Omega$ standard) and two turbulence models from $k-\varepsilon$ family (standard and realizable $k-\varepsilon$) in order to check the accuracy and independence of the solution from the turbulence models. Same solution methodology and boundary conditions with temperature-dependence properties are used for this test. However, the arrangement of meshes is generated in such a way that there is satisfaction with the assignment of y^+ value. It is noteworthy that values of y^+ for wall adjacent cells are held to be well within the viscous sub-layer for $k-\Omega$ and within the log-law layer for $k-\varepsilon$. The final selection of the turbulence model is based on convergence criteria. The realizable $k-\varepsilon$ turbulence model satisfied the convergence in which residuals

were less than 10^{-6} for energy and 10^{-9} for continuity. On the other hand, the other compared models did not satisfy the values of residuals.

CHAPTER 5

RESULTS AND DISCUSSIONS

5. 1. Validation

Validation of the used model provides confidence in the accuracy of results. For this purpose, a validation study is conducted with earlier published work [153]. The physical domain and boundary conditions are modified as presented in previous work. We selected this work since the same radiation model and material properties (temperature dependent SiC and fluids properties) were considered. In addition, it also includes both experimental and numerical studies. Same inputs are used for this validation process, including No. of grid = 34,560, N_θ and $N_\phi = 3 \times 3$, DNI = 500 kW/m² and mass-flow rate = 0.00617. Dimensions of the channel domain are specified to be 0.5mm for wall thickness, 50 mm for wall length with 10 mm extension of channel for air inlet. Same convergence scheme, algorithm, and pressure-velocity coupling are implemented which are second-order spatial discretization, SIMPLEC-algorithm QUICK scheme for momentum and energy. It can be observed that temperature variation along the channel length is in good agreement with that presented in [153]. Fig. 5.2 and 5 are evidence for that agreement. Slight difference in contours of temperature at the channel end is mainly because of post-processing software and its color scheme for contours formation. The trend of the temperature curve is exactly matching, and the error is under tolerance (5%), as shown in Fig. 5.2.

Validation of the used model provides confidence in the accuracy of results. For this purpose, a validation study is conducted with earlier published work [153]. The physical domain and boundary conditions are modified as presented in previous work. We selected this work since the same radiation model and material properties (temperature dependent SiC and fluids properties) were considered. In addition, it also includes both experimental and numerical studies. Same inputs are used for this validation process, including No. of grid = 34,560, N_θ and $N_\phi = 3 \times 3$, DNI = 500 kW/m² and mass-flow rate = 0.00617. In case 5 of early work, the channel dimensions are 0.5 mm wall thickness, 50 mm channel length, and 10 mm extension of channel for airflow. Complete computational domain used for validation is shown in Fig. 5.1. DO radiation method is employed to model the radiation for validation. Same governing equation is computed to solve the velocity, pressure and temperature distribution. Same convergence scheme, algorithm, and pressure-velocity coupling are implemented which are second-order spatial discretization, SIMPLEC-algorithm QUICK scheme for momentum and energy. It can be observed that temperature variation along the channel length is in good agreement with that presented in [153]. Figures 5.2 and 5.3 are evidence for that agreement. Slight difference in contours of temperature at the channel end is mainly because of post-processing software and its color scheme for contours formation. The trend of the temperature curve is exactly matching, and the error is under tolerance (5%), as shown in Fig. 5.3.

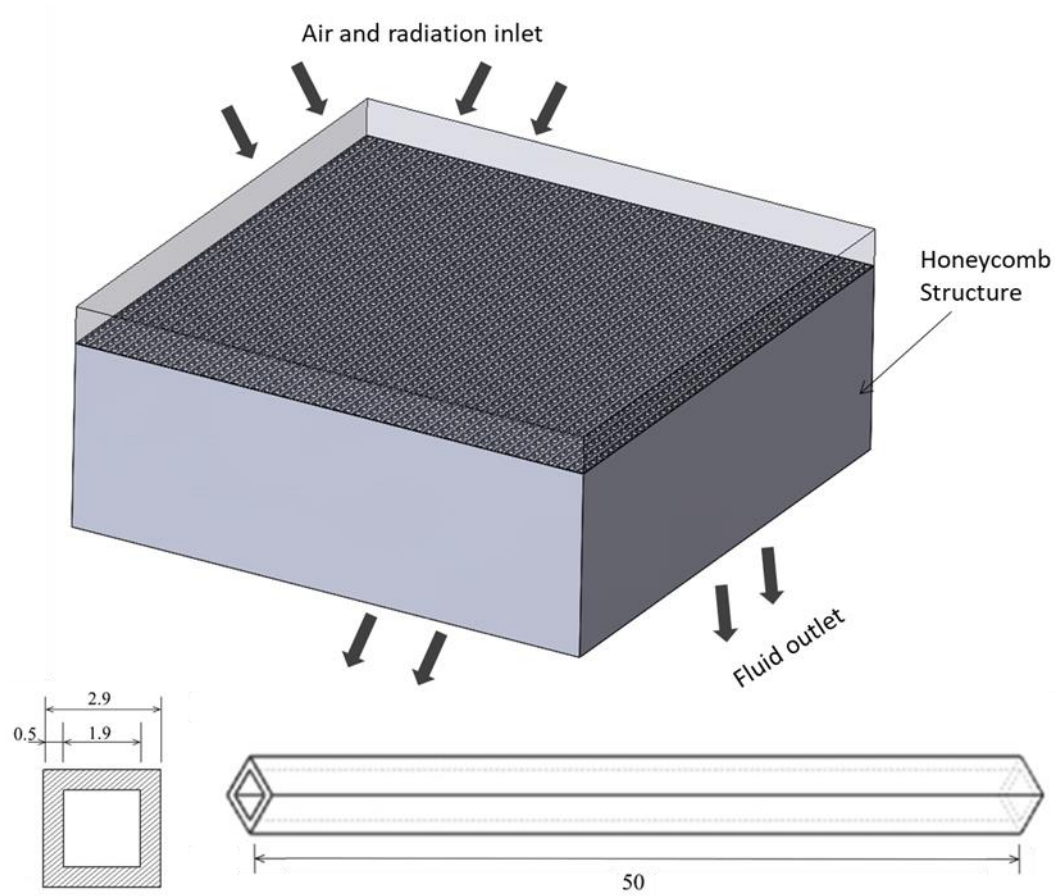


Fig. 5.1: Graphical illustration of computational domain used for validation.

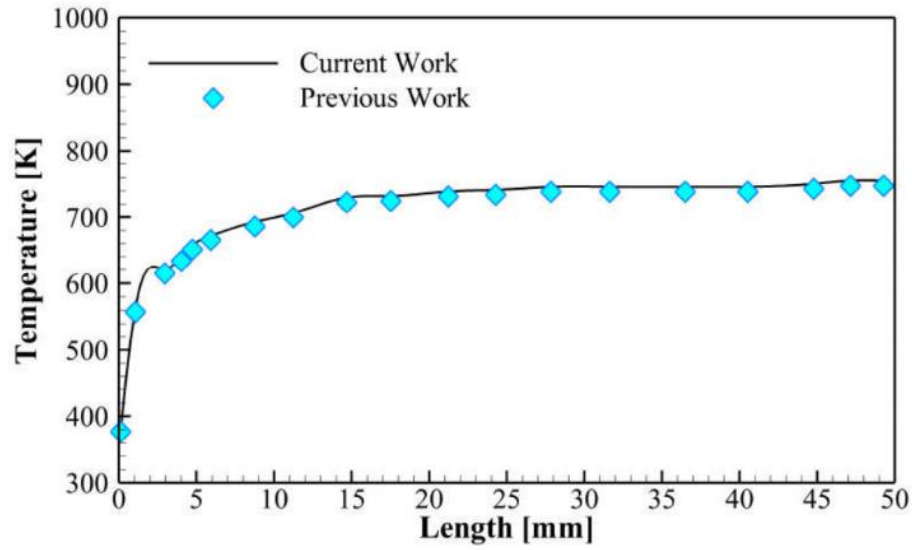


Fig. 5.2: Variation of temperature distribution along channel

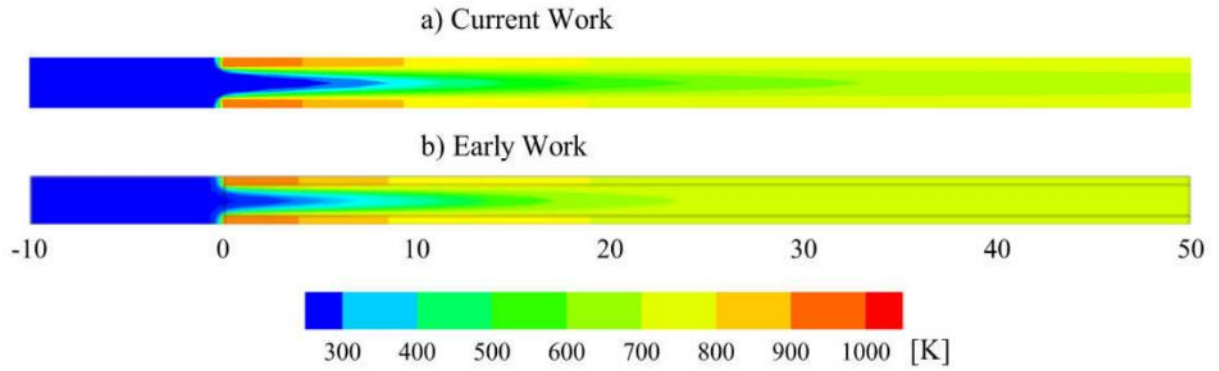


Fig. 5.3: Temperature distribution in form of contours along the channel

5. 2. Laminar analysis of porous SiC foam

The total number of cases for laminar flow condition is 64 as tabulated in Table 5.2.

Table 5.1: Numerical variables. DNI refers to direct normal irradiance.

Sr.	DNI	Reynolds	Porosity	Sr.	DNI	Reynolds	Porosity
	[kW/m ²]	Number			[kW/m ²]	Number	(Ψ)

1.	200	50	0.62	33.	200	50	0.77
2.	400	50	0.62	34.	400	50	0.77
3.	600	50	0.62	35.	600	50	0.77
4.	800	50	0.62	36.	800	50	0.77
5.	200	150	0.62	37.	200	150	0.77
6.	400	150	0.62	38.	400	150	0.77
7.	600	150	0.62	39.	600	150	0.77
8.	800	150	0.62	40.	800	150	0.77
9.	200	300	0.62	41.	200	300	0.77
10.	400	300	0.62	42.	400	300	0.77
11.	600	300	0.62	43.	600	300	0.77
12.	800	300	0.62	44.	800	300	0.77
13.	200	500	0.62	45.	200	500	0.77
14.	400	500	0.62	46.	400	500	0.77
15.	600	500	0.62	47.	600	500	0.77
16.	800	500	0.62	48.	800	500	0.77
17.	200	50	0.695	49.	200	50	0.845
18.	400	50	0.695	50.	400	50	0.845
19.	600	50	0.695	51.	600	50	0.845
20.	800	50	0.695	52.	800	50	0.845
21.	200	150	0.695	53.	200	150	0.845
22.	400	150	0.695	54.	400	150	0.845
23.	600	150	0.695	55.	600	150	0.845
24.	800	150	0.695	56.	800	150	0.845
25.	200	300	0.695	57.	200	300	0.845
26.	400	300	0.695	58.	400	300	0.845
27.	600	300	0.695	59.	600	300	0.845
28.	800	300	0.695	60.	800	300	0.845
29.	200	500	0.695	61.	200	500	0.845
30.	400	500	0.695	62.	400	500	0.845
31.	600	500	0.695	63.	600	500	0.845
32.	800	500	0.695	64.	800	500	0.845

This section of results and discussion presents the analysis of solar harvesting from the volumetric receiver for laminar conditions. After completion of the grid independence tests and validation study, all these 64 cases are computed as tabulated in Table 5.2. The volumetric receiver in the channel is heated by direct incident radiation. Simultaneously, air as a working fluid is allowed to flow through the porous SiC foam to transport the heat. For temperature and pressure distribution, CFD is carried out using the ANSYS Fluent commercial code. The thermal performance of porous SiC foam is evaluated through the overall heat transfer coefficient and thermal effectiveness. The fluid and solid temperatures are obtained by using mass-weighted and area-weighted integral in the post-processing step of ANSYS Fluent. Thermal effectiveness based on mass-weighted outlet temperature and maximum temperature in the domain are also compared for all cases. Reynolds number, solar concentrated radiation, and porosity are the variables considered for the assessment of the thermal performances of the solar volumetric receiver. Fluid flow performance is also evaluated by calculating pressure drop.

The velocity vector within the channel and the SiC foam is shown in Fig. 5.4. Porous SiC foam creates complex flow structures and velocity magnitude increases locally where the flow passages are narrowed and blocked by the neighboring structures in SiC foam. As velocity magnitude increases, the forced-convection increases leading to enhanced convection heat transfer. Moreover, the porous structure in the volumetric receiver traps the fluid and promotes temperature rise and flow mixing. The flow mixing gives rise to wake-like flow structure in pore sites. Likewise, the wake-like structures can give rise to vortices contributing to the heat transfer. However, the flow mixing and vortices lead to

high-pressure drops across SiC foam. Nevertheless, the complex flow structure results in heat transfer enhancement despite the pressure drop remains large.

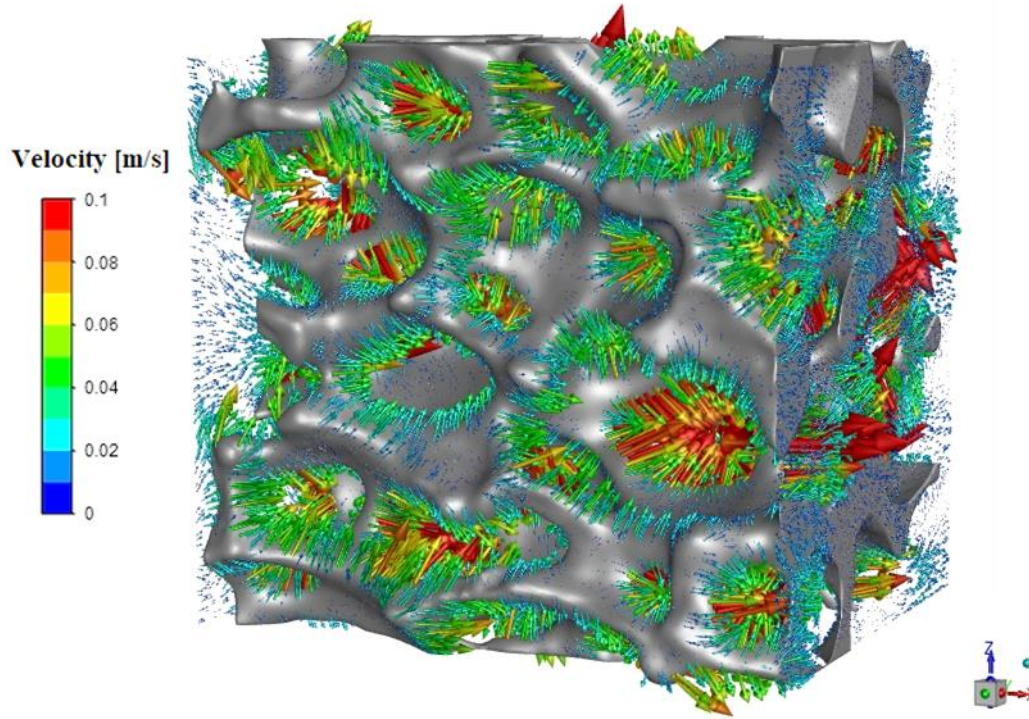


Fig. 5.4: Vectors represent velocity magnitude according for $Re = 50$, $\Psi = 0.77$ and $DNI = 200 \text{ kW/m}^2$.

Figure 5.6 shows velocity contours inside the porous SiC foam. Contours are plotted at midplane cross-section as shown in Fig. 5.5. Velocity attains large values in the region of the throttle where the cross-section of the porous structure becomes narrow. Since, porous structures are connected with geometrically unstructured paths (channels), flow behavior changes with the passage geometry of the upstream in the porous medium. In this case, as the downstream pressure increases because of partial blockage of the porous path, the

magnitude of flow velocity reduces. Hence, non-uniform behavior of flow occurs inside the foam because of the complex nature of the foam structures. In the region of the closed flow passages in the structure, flow velocity becomes stagnant and the contribution to the local pressure change becomes significant. Moreover, in some regions, flow passages are fully connected and the flow velocity remains high. This demonstrates that contribution of forced convection to heat transfer enhances in the region where the foam structure is fully connected. However, heat diffusion and natural convection dominates in the closed paths of the foam structure where the flow velocity reduces significantly.

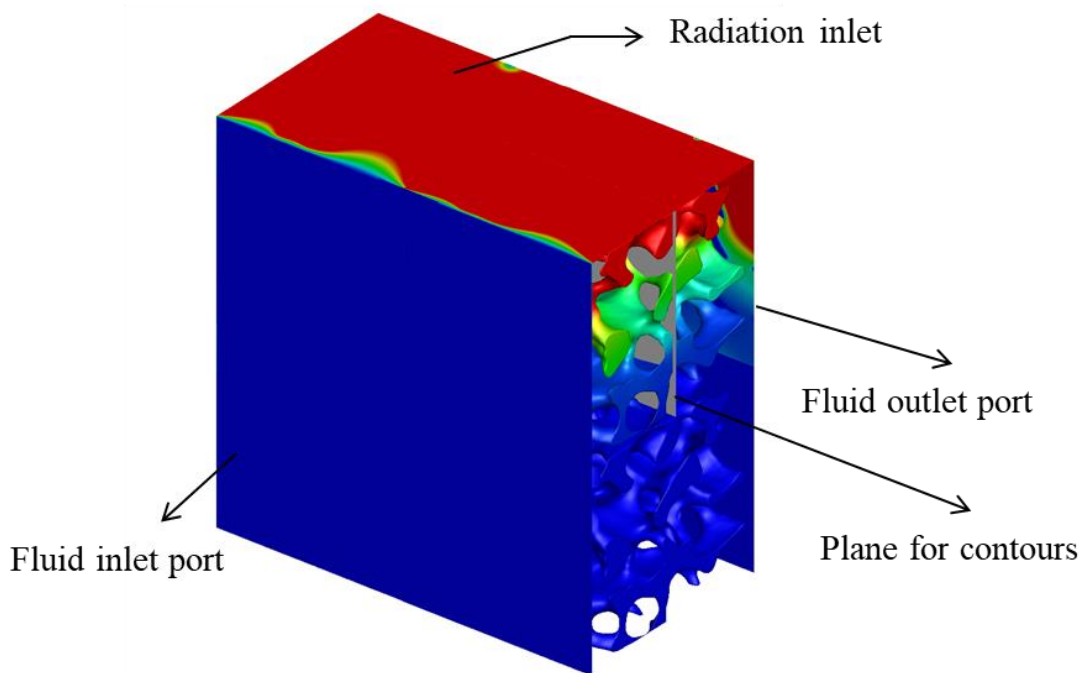


Fig. 5.5: Location (plane) where contours for pressure and temperature have been plotted.

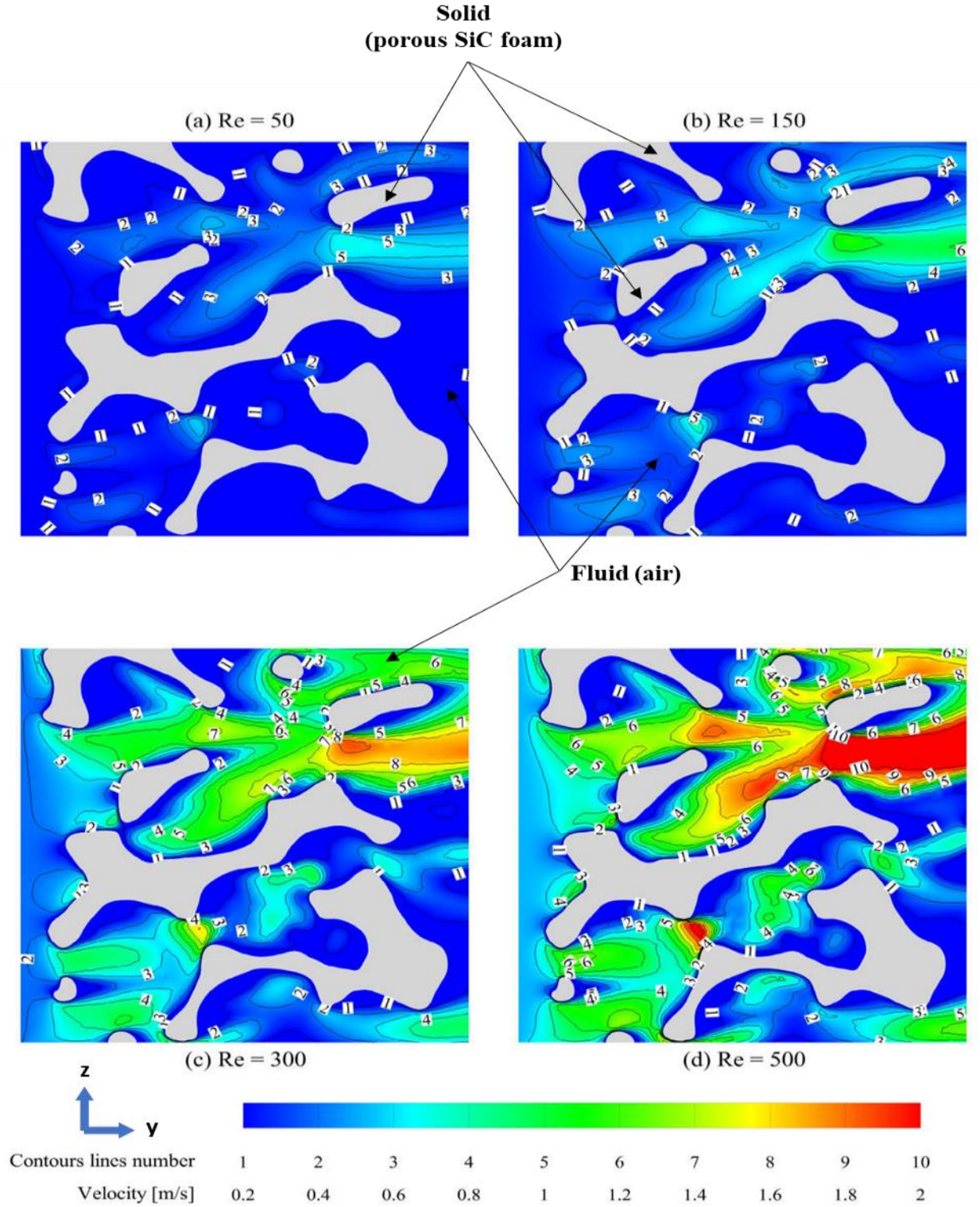


Fig. 5.6: Velocity contours for a section of the flow channel. Gray areas represent solid phase (porous SiC foam) and colored region is fluid. Contours is plotted for $\Psi = 0.62$ and $DNI = 200 \text{ kW/m}^2$

Figure 5.7 shows temperature distribution inside the porous SiC foam. The contours of temperature are taken for an upper section of the channel where $\Psi = 0.62$ and $\text{DNI} = 200 \text{ kW/m}^2$. It should be noted especially for $\text{Re} = 150$ and 300 , the temperature of solid regions is slightly higher than the working fluid. This is mainly due to coupled heat transfer between the solid and the fluid in the volumetric receiver. This can also occur in a well-performing receiver such that the incident radiation overcomes the convective contribution, i.e. energy received locally by the solid medium of porous foam is more than the energy released to the fluid through interfacial convection. Therefore, the temperature remains high near the radiation entering region. The volumetric heating effect is also observed through the temperature difference of fluid and solid. In this case, the temperature of the receiver in the top boundary, where the solar radiation is applied, becomes lower than the temperature of fluid leaving the receiver. This suggests that a highly porous structure is effective to produce a volumetric heating effect; however, pressure drop must be in an acceptable range. In addition, high thermal conductivity fluids can perform better for heat harvesting thermal energy from the volumetric receivers.

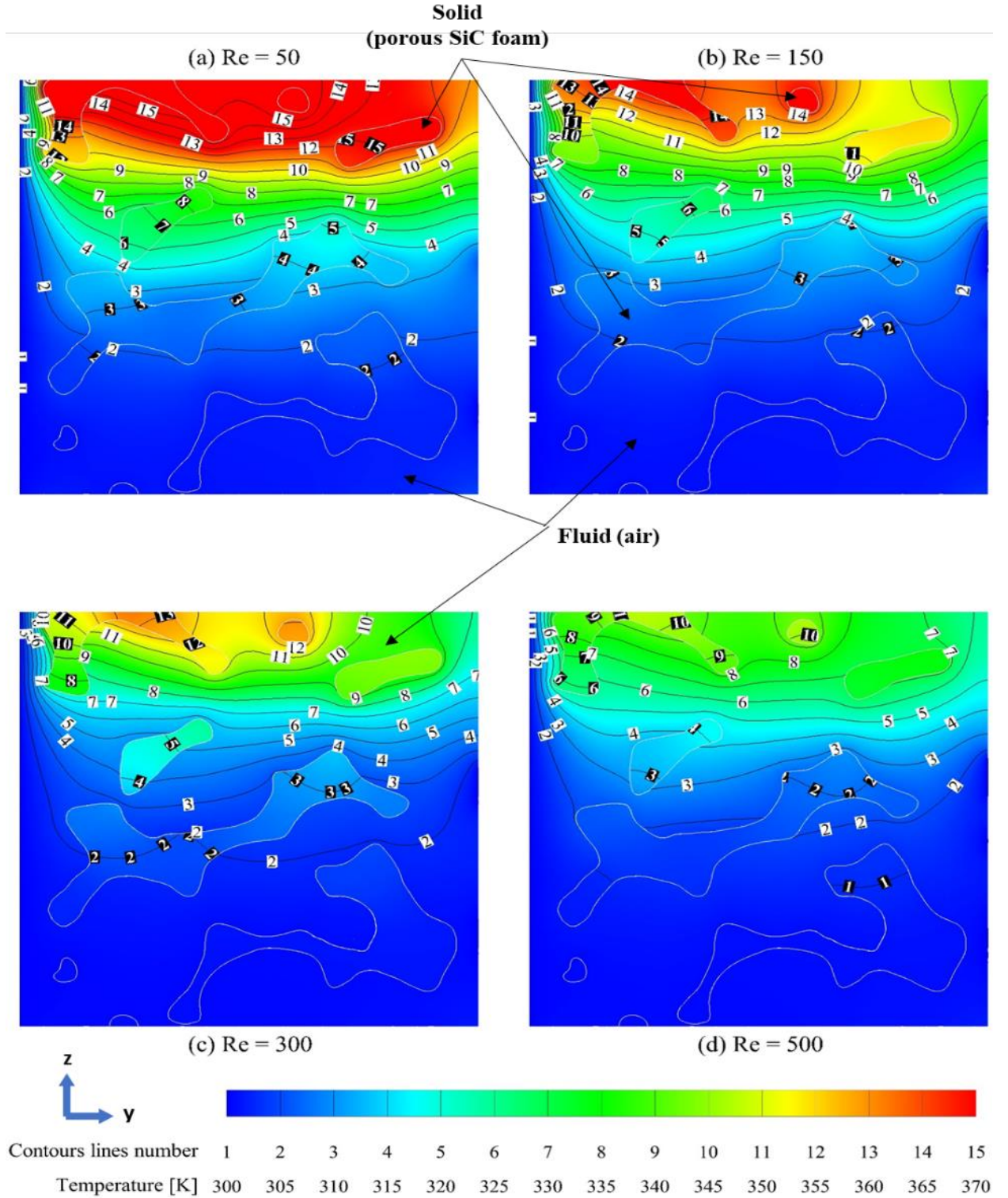


Fig. 5.7: Temperature contours of small portion of channel at top side. Areas under solid black lines are solid portion and outside area is fluid. Contours is plotted at $\Psi = 0.62$ and $DNI = 200 \text{ kW/m}^2$

5.2.1. Thermal Analysis

Total wall heat flux refers to the overall heat flux occurring in SiC foam, which includes convection, conduction, and radiation heat fluxes. Figure 5.8 shows total wall heat flux in the volumetric receiver for Reynolds number (50-500), solar concentration (200-800 kW/m²), and porosity (0.62-0.843). The influence of Reynolds number is almost negligible on total wall heat flux because the volumetric effect is significantly profound. However, solar concentration causes a gradually increase total wall heat flux and the trend remains similar for all porosities considered. Increment of total wall heat flux is directly dependent upon the heat source (solar radiation). The effect of Reynolds number is almost the same for all porosities and solar concentrations. However, for $\Psi = 0.845$, increasing Reynolds number increases the wall heat flux. In this case, the volume ratio of solid to fluid in the foam changes, which appears to result in optimal coupling of heat transfer between the solid and fluid in the foam incorporated. In addition, high fluid velocity resulted for $\Psi = 0.845$ is expected to give rise to the low duration of fluid trapping phenomena inside the foam. Hence, the convection heat transfer contribution to overall heat flux becomes notable in this case. Consequently, coupling of flow and porous structures plays a critical role in the overall heat flux in the foam material.

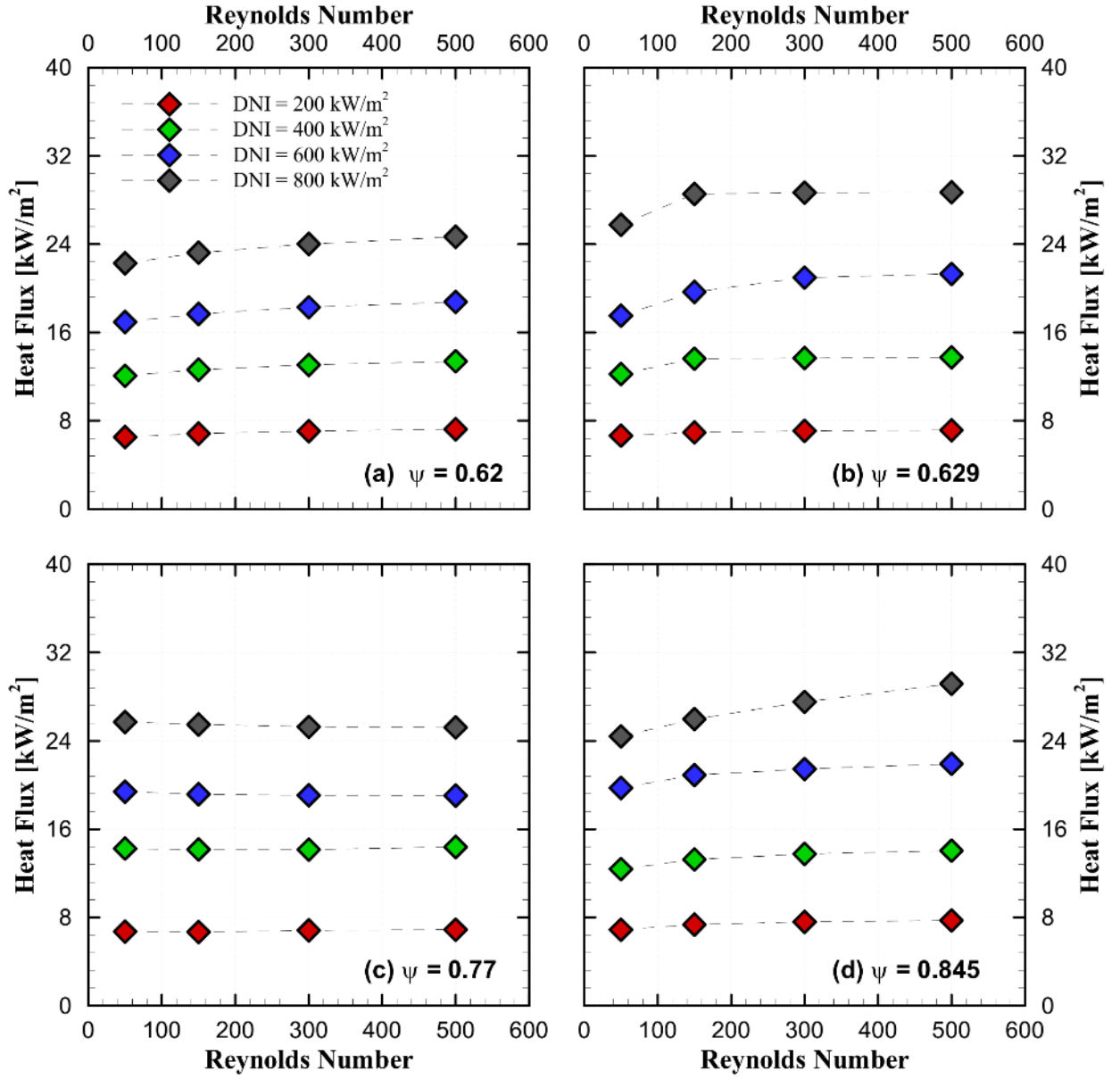


Fig. 5.8: Total heat flux at wall of porous SiC foam against Reynolds number for a) $\Psi = 0.62$, b) $\Psi = 0.695$, c) $\Psi = 0.77$ and d) $\Psi = 0.845$

Figure 5.9 shows the overall heat transfer coefficient for each case given in Table 5.2, with different Reynolds numbers, solar concentrations, and porosities. Increasing solar radiation considerably causes a decrease in the overall heat transfer coefficient. This behavior occurs due to localized heating under the radiation. Since the radiation heat model is incorporated

in the analysis, at high radiative fluxes temperature difference between the fluid and the liquid becomes small. This in turn lowers the overall heat transfer coefficient. As the flow velocity at the foam inlet increases, the overall heat transfer coefficient also increases. Thus, the heated fluid, which is carried by the flow at a high Reynolds number gives rise to the increased temperature difference between the solid and fluid in the foam. For low porosities ($\Psi = 0.62$ and 0.695), solar concentration decreases almost 33% (from 200 to 400 kW/m²) while higher porosities ($\Psi = 0.77$ and 0.845) possess approximately 11% decrease in heat transfer coefficient from 200 to 400 kW/m². However, for the intensities of 400 to 800 kW/m² and all porosities, the decrement in heat transfer coefficient comparatively becomes less because of the high heat flux created in the foam. Heat transfer coefficient is augmented to the amount of fluid flowing residing in the foam with time. In this case, the overall heat transfer coefficient remains less for the cases having large porosities because of less residency time of the fluid recirculating inside the foam. In addition, the influence of Reynolds number on heat transfer coefficient is the same for all porosity and solar concentrations.

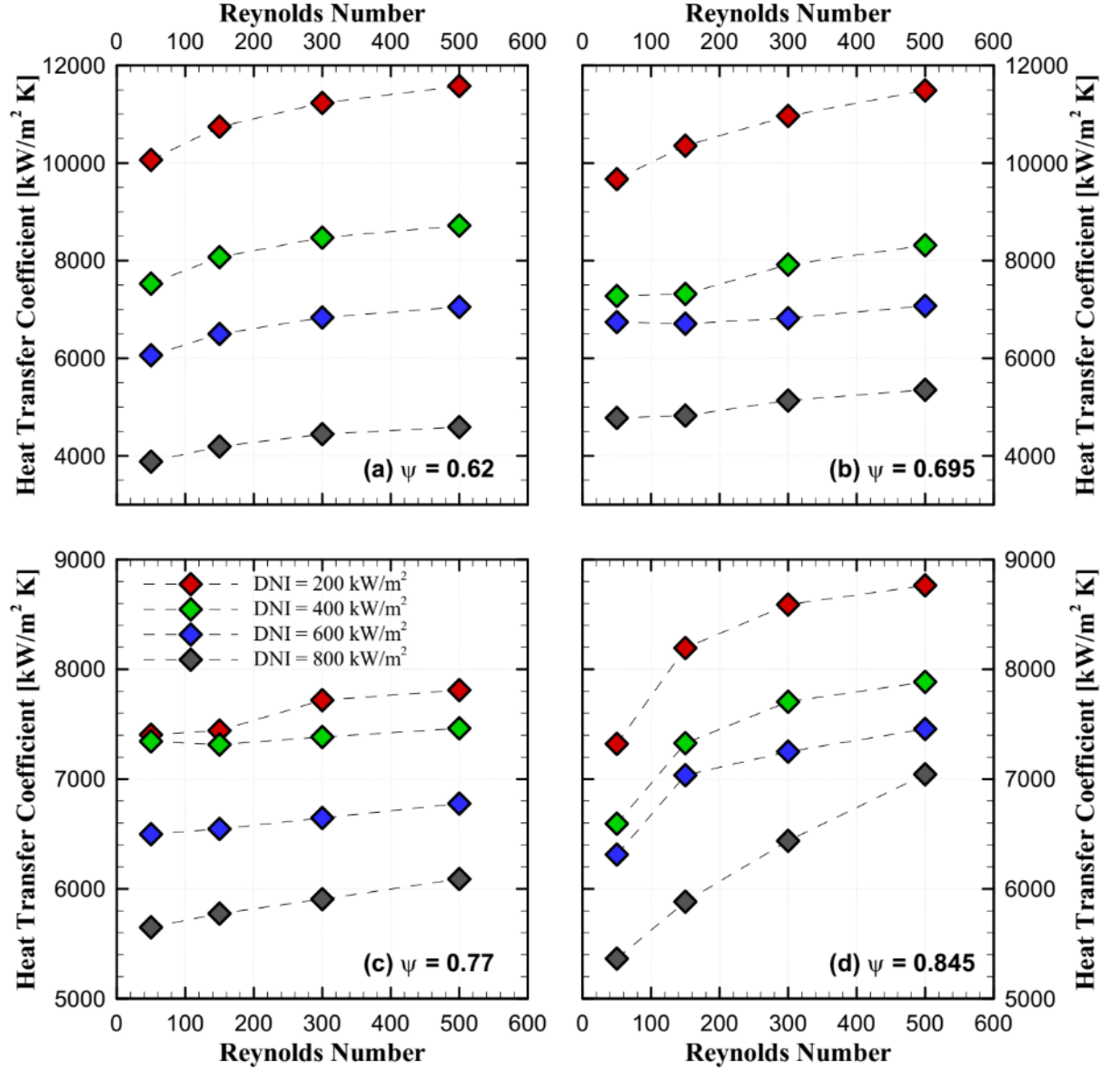


Fig. 5.9: Overall heat transfer coefficient in channel having porous SiC foam against Reynolds number for a) $\Psi = 0.62$, b) $\Psi = 0.695$, c) $\Psi = 0.77$ and d) $\Psi = 0.845$

Since Nusselt number describes the amount of convective heat over the conduction, the comparison is made to assess the heat transfer effectiveness for the cases with and without porous foam in the channel. For this purpose, variation of normalized Nusselt number is plotted against several Reynolds numbers (50 -500) and solar concentration (200 – 800 kW/m²) for different porosities as shown in Fig. 5.10. Nusselt number is normalized by

Nusselt number when the channel has no porous foam. It is worth mentioning that the Nusselt number (Nu_0), in which foam is absent, is obtained by simulating the flow field in the same channel geometry. Normalized Nusselt number attains values value greater than a hundred-fold. This indicates that the enhancement of heat transfer while using porous SiC foam, is significantly high. Moreover, increasing Reynolds number drastically decreases normalized Nusselt number because of lowering the fluid resistance period inside the foam. In addition, large inertial forces created under high Reynolds number influence flow mixing inside the foam. This creates a mixing behavior of fluid in such a way that the heat removed from porous foam is not appeared to be significant in the channel. It should be noted that the Nusselt number attains considerably small values at high Reynolds numbers than that corresponding to the low Reynolds numbers flow in the channel. For high solar concentrations, reduction in Nusselt number becomes less as Reynolds number increases. This indicates that at high heating loads, increased solar concentration gives rise to low heat transfer coefficients. Hence, heat extraction by the working fluid becomes low even at low Reynolds numbers. The trend of normalized Nusselt number appears to be similar as Reynolds number or solar concentration increase, which is true for all porosities of the foam. In addition, it is worth noting that low porosity shows the large value of Nusselt number because of improved mixing of fluid within the foam voids. The highest value of normalized Nusselt number is obtained at the lowest value of Reynolds number (50) for the solar concentration of 200 kW/m^2 and porosity of 0.62.

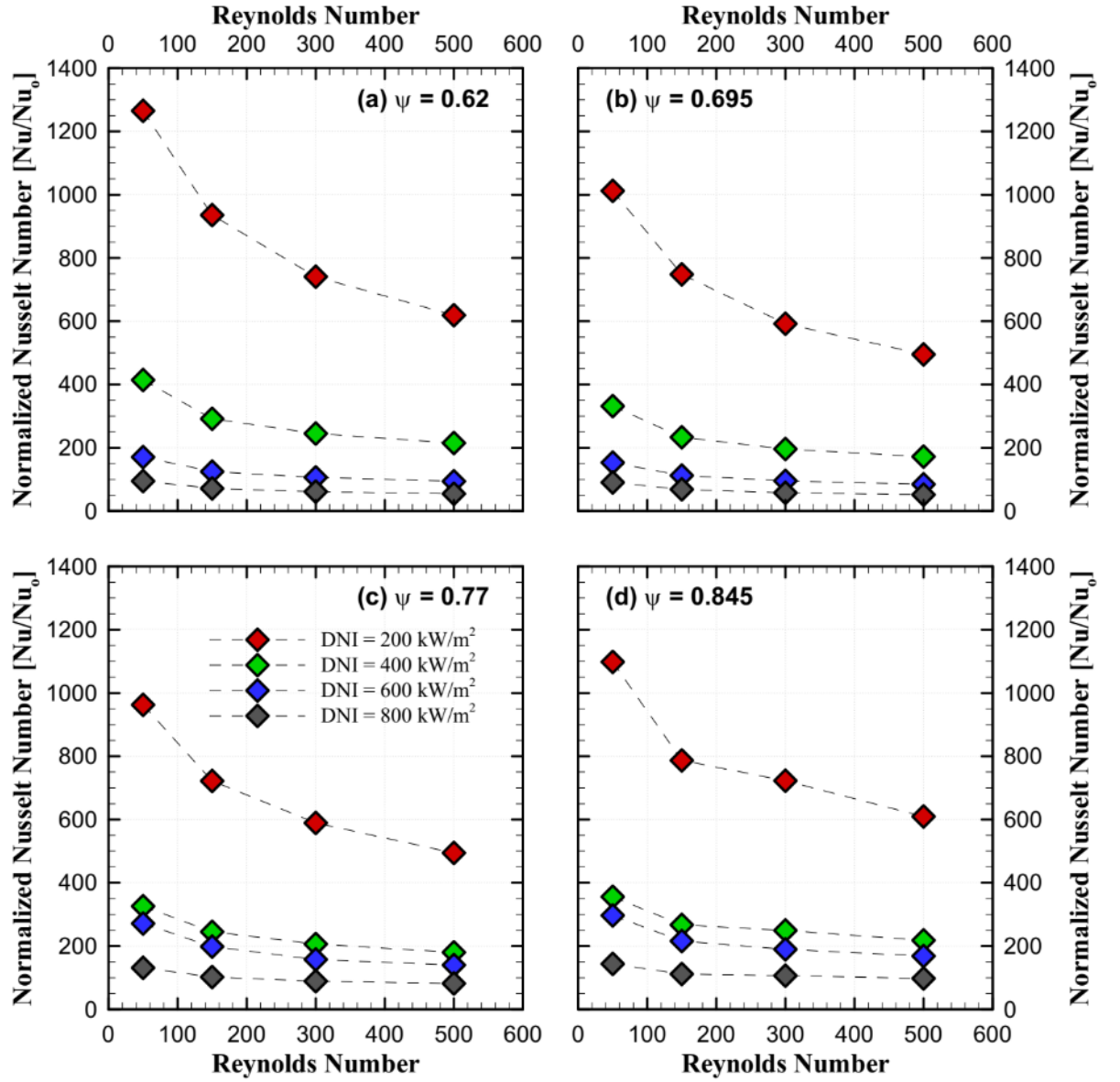


Fig. 5.10: Global normalized Nusselt number (Nu/Nu_0) of porous SiC foam against Reynolds number for a) $\Psi = 0.62$, b) $\Psi = 0.695$, c) $\Psi = 0.77$ and d) $\Psi = 0.845$

5.2.2. Thermal Effectiveness

To assess the thermal performance of the porous foam in terms of thermal effectiveness (η_1), the following dimensionless temperature is introduced:

$$\eta_1 = \frac{T_{\text{out}} - T_{\text{in}}}{T_{\text{in}}} \quad (48)$$

T_{out} represents the mass-average outlet temperature of the air, and T_{in} is the air inlet temperature. The variation of normalized temperature with Reynolds number is shown in Fig. 5.11 for various solar concentrations and porosities. For the range of Reynolds numbers $Re \geq 150$, η_1 demonstrates a slightly decreasing trend; however, at low Reynolds numbers, the normalized temperature attains high values. Hence, the system, in terms of temperature ratio, exhibits high performance at low Reynolds numbers due to the long residency time of the working fluid inside the ceramic foam. In addition, increasing foam porosity gives rise to low normalized temperature on the thermal system, which is more pronounced for higher solar concentrations. Consequently, low values of Reynolds numbers and low porosities remain favourable to obtaining the high mass average fluid temperature. However, the opposite trend is observed for solar concentration, i.e. increasing solar concentration enhances mass averaged fluid temperature. The maximum mass-average fluid temperature at the foam outlet occurs for $Re = 50$, $DNI = 800 \text{ kW/m}^2$ and $\Psi = 0.62$ and the minimum mass-average fluid temperature at the foam outlet takes place for $Re = 500$, $DNI = 200 \text{ kW/m}^2$ and $\Psi = 0.845$. The fluid outlet temperature is associated with heat transfer including convection, conduction, and radiation inside the foam; hence, foam with $\Psi = 0.62$ results in the highest outlet temperature, which is also supported by Nusselt number variation (Fig. 5.9). Similarly, for $Re = 50$ and $DNI = 800 \text{ kW/m}^2$, the porosity of foam $\Psi = 0.62$ results in 42% more mass average fluid outlet temperature than corresponding to $\Psi = 0.845$.

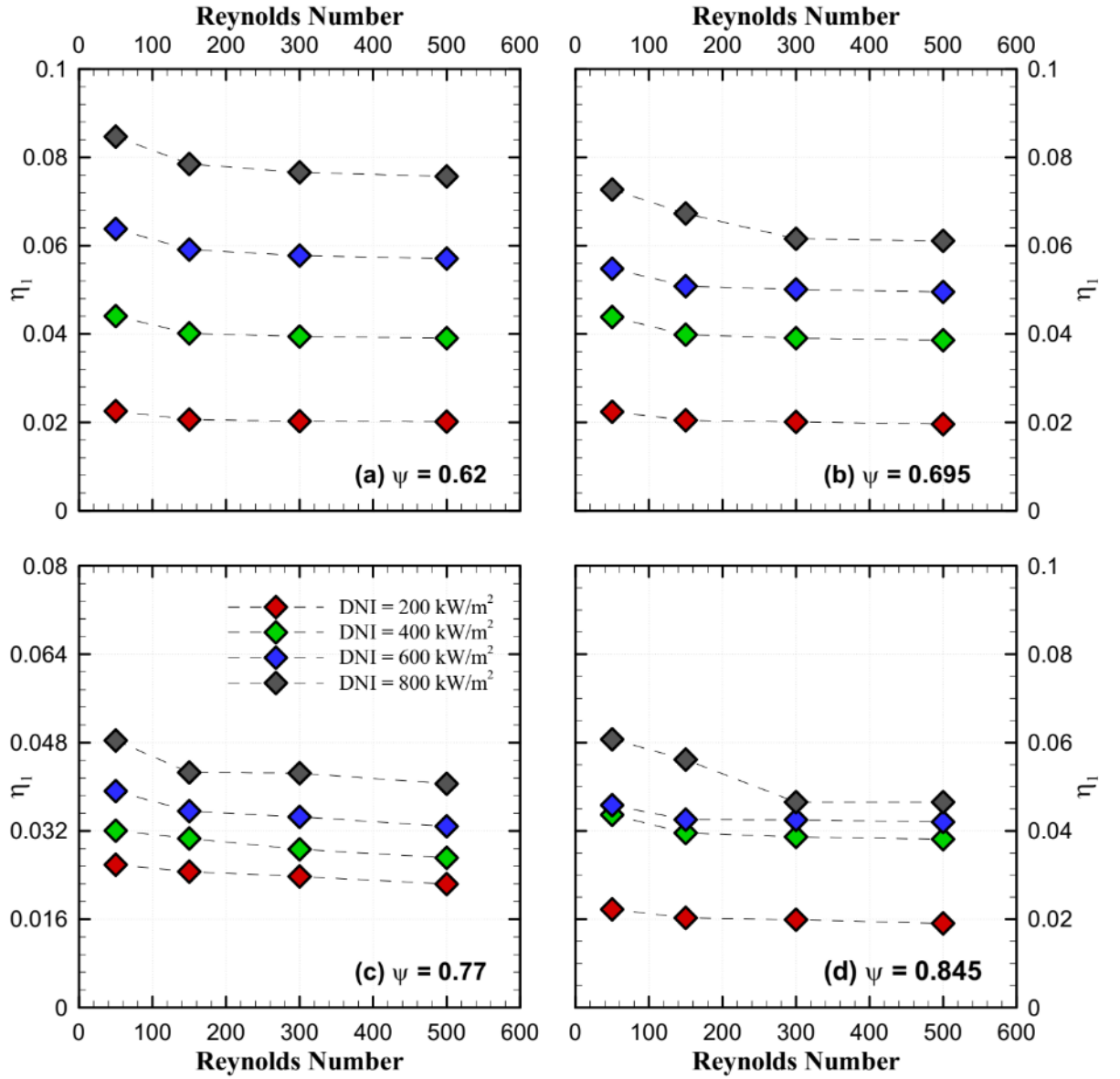


Fig. 5.11: Total thermal effectiveness (η_1) of channel having porous SiC foam against Reynolds number for a) $\Psi = 0.62$, b) $\Psi = 0.695$, c) $\Psi = 0.77$ and d) $\Psi = 0.845$

The maximum temperature increase in the pore foam is assessed through temperature ratio

(η_2) introduced:

$$\eta_2 = \frac{T_{\max} - T_{\text{in}}}{T_{\text{in}}} \quad (49)$$

Here, $T_{\max}$ denotes the maximum fluid temperature achieved in the flow channel. Figure 5.12 shows the maximum temperature ratio with the Reynolds number for various concentrations and porosities. The influence of foam porosity and Reynolds number on the maximum temperature ratio demonstrates a similar trend; in which case, the change of the maximum temperature remains considerably small. However, increasing solar concentration enhances the maximum temperature in the working fluid. Hence, localized heating under the high solar concentration can result in a large value of the maximum temperature. The nonuniform heating of the working fluid becomes unavoidable if the concentration increases considerably even though the porosity of the foam and Reynolds number of the flow are increased.

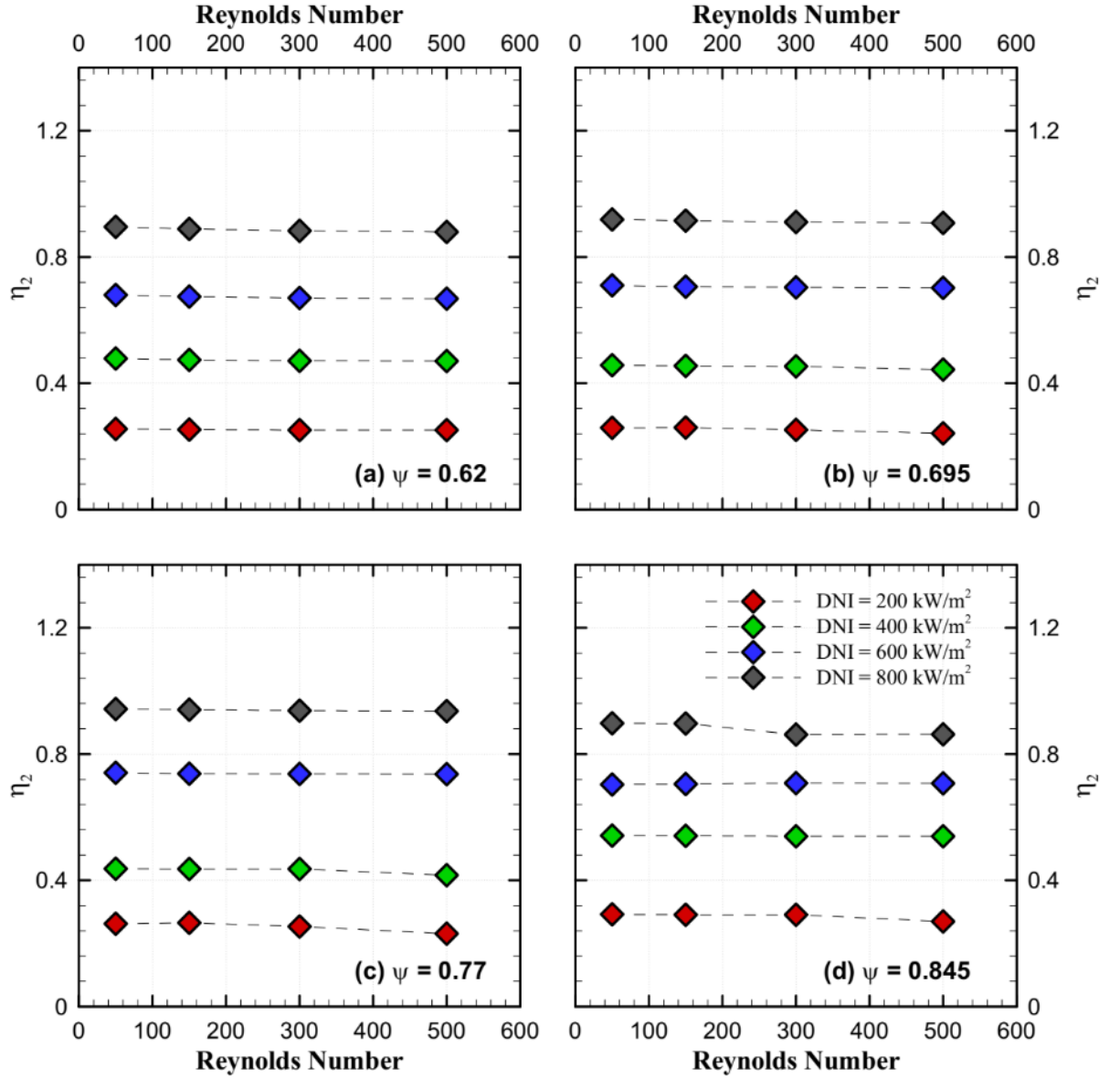


Fig. 5.12: Normalized maximum gain temperature (η_2) in channel having porous SiC foam against Reynolds number for a) $\Psi = 0.62$, b) $\Psi = 0.695$, c) $\Psi = 0.77$ and d) $\Psi = 0.845$

5.2.3. Flow Analysis

The fluid power loss due to pressure drop can be expressed in terms of fluid pumping power required. Here the pressure drop occurs because of pores in the foam under drag and fluid friction within the pores and the channel. In general, flow mixing in the pores structures and irregular paths connecting the pores main contributes to the pressure drop inside the

foam. The major challenge towards employing a porous structure is always its pumping power. Figure 5.13 depicts the pressure drop in the channel with Reynolds number for various values of solar concentrations and porosities of foam. The pressure drop increases sharply as the Reynolds number increases. This is attributed to the complex flow field generated inside the pores, which creates significant flow mixing in the pores' structures. Solar concentration increase has no considerable effect on the pressure drop, provided that this effect may become visible at a high Reynolds number. This behavior is attributed to the fluid temperature increase at high solar concentrations. Air viscosity increases with increasing temperature, which causes attainment of the high rate of shear in the fluid. Hence, pressure drop increases accordingly; however, this effect is not significant and the influence of solar concentration on the fluid pressure drop becomes small. In the case of the influence of porosity on the pressure drop, increasing porosity lowers the pressure drop. Hence, small voids ($\Psi = 0.62$) give rise to large pressure drop in the foam. It is worth mentioning that the value of the thermal performance of the system remains almost similar for both low and high porosities. However, the influence of porosity on the pressure drop becomes significant, i.e. the pressure drop becomes almost double as the porosity reduces.

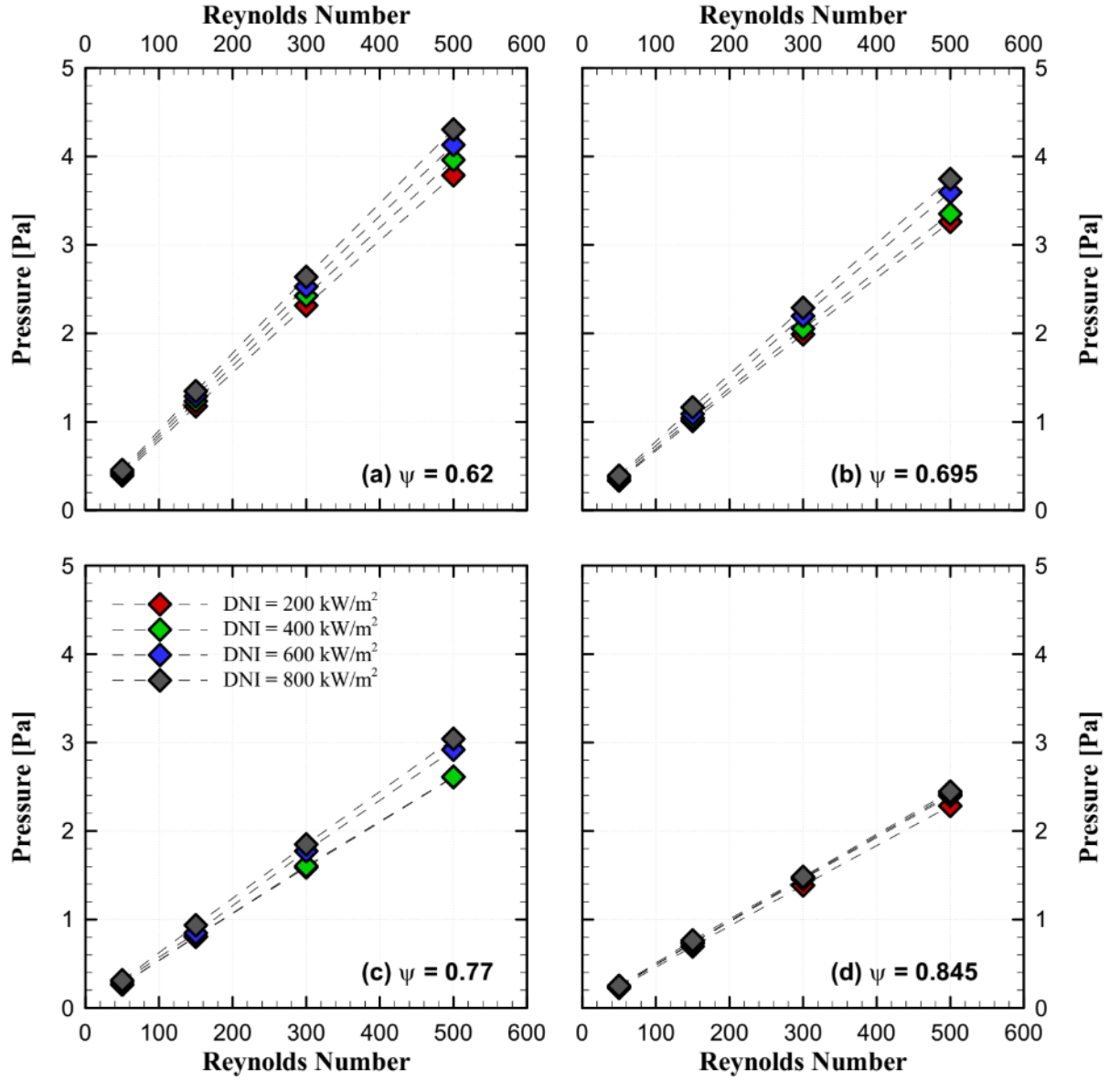


Fig. 5.13: Pressure drop in channel having porous SiC foam against Reynolds number for a) $\Psi = 0.62$, b) $\Psi = 0.695$, c) $\Psi = 0.77$ and d) $\Psi = 0.845$

5. 3. Turbulent analysis of porous SiC foam

The total number of cases for this turbulent flow conditions is 54 as tabulated in Table 5.2.

Table 5.2: Numerical variables including DNI, Reynolds Number and Porosity (Ψ).

Sr.	DNI [kW/m ²]	Reynolds Number	Porosity (Ψ)	Fluid	Sr.	DNI [kW/m ²]	Reynolds Number	Porosity (Ψ)	Fluid
1	250	3000	0.69	Air	28	250	3000	0.69	Water
2	500	3000	0.69	Air	29	500	3000	0.69	Water
3	750	3000	0.69	Air	30	750	3000	0.69	Water
4	250	3500	0.69	Air	31	250	3500	0.69	Water
5	500	3500	0.69	Air	32	500	3500	0.69	Water
6	750	3500	0.69	Air	33	750	3500	0.69	Water
7	250	4000	0.69	Air	34	250	4000	0.69	Water
8	500	4000	0.69	Air	35	500	4000	0.69	Water
9	750	4000	0.69	Air	36	750	4000	0.69	Water
10	250	3000	0.77	Air	37	250	3000	0.77	Water
11	500	3000	0.77	Air	38	500	3000	0.77	Water
12	750	3000	0.77	Air	39	750	3000	0.77	Water
13	250	3500	0.77	Air	40	250	3500	0.77	Water
14	500	3500	0.77	Air	41	500	3500	0.77	Water
15	750	3500	0.77	Air	42	750	3500	0.77	Water
16	250	4000	0.77	Air	43	250	4000	0.77	Water
17	500	4000	0.77	Air	44	500	4000	0.77	Water
18	750	4000	0.77	Air	45	750	4000	0.77	Water
19	250	3000	0.85	Air	46	250	3000	0.85	Water
20	500	3000	0.85	Air	47	500	3000	0.85	Water
21	750	3000	0.85	Air	48	750	3000	0.85	Water
22	250	3500	0.85	Air	49	250	3500	0.85	Water
23	500	3500	0.85	Air	50	500	3500	0.85	Water
24	750	3500	0.85	Air	51	750	3500	0.85	Water
25	250	4000	0.85	Air	52	250	4000	0.85	Water
26	500	4000	0.85	Air	53	500	4000	0.85	Water
27	750	4000	0.85	Air	54	750	4000	0.85	Water

This portion presents the analysis of porous SiC volumetric receivers for turbulent flow conditions. Same computational domain is used as illustrated in Fig. 3.4. Moreover, after

confirming grid and turbulent model independence and completion of the validation study, all 54 cases are numerically solved according to the defined calculation variable in Table 5.2. For turbulent flow and radiation modeling, $k - \epsilon$ realizable turbulent and DO radiation models are considered in the numerical simulations. Likewise, mass-weighted and area-weighted integral approaches are applied to calculate the field variables for fluids and solid in post-processing of ANSYS Fluent. Hydrodynamic behavior of fluids (air and water) and pressure drop in the channel are evaluated. The assessment of thermal performance is examined through global Nusselt number as well as thermal efficiencies. The maximum temperature increase in the fluid domain is also determined for two working fluids which are air and water, Reynolds numbers with the range of 3000- 4000, solar concentration ($\text{DNI} = 250 - 750 \text{ kW/m}^2$), and porosities from 0.69 to 0.85.

5.3.1. Thermal Analysis

Figure 5.14 shows temperature contours inside the solar receiver at different Reynolds numbers (3000, 3500, and 4000) for porosity $\Psi = 0.77$ and DNI of 250 kW/m^2 . The solid domain of the channel has a higher temperature than the fluid domain. The reason is due to the volumetric effect created through solar radiation absorption, i.e. the synergic effect created by the heat transfer between the solid and fluid while exposing to solar concentrated radiation. The absorption of solar radiation by SiC causes a temperature increase in the porous structure. The heat transfer between the porous structure and the working fluid via interfacial convection, conduction, and radiation lowers the SiC temperature. However, the temperature drop due to heat transfer remains slightly lower while causing attainment of high temperature in the SiC porous structure. Moreover, it can be observed that the volumetric effect is profound when the heat transfer is subjected through the water. In this

case, the temperature in the SiC porous structure attains lower values for water than the air for given conditions. Moreover, plots of temperature distributions (Fig. 5.14) show less difference among various Reynolds numbers which indicates that the receiver system can recover the effective amount of heat for all Reynolds numbers. Water, which has a higher viscosity and higher heat capacity than air, results in a high-temperature rise. Hence, the temperature difference between the fluid and SiC porous structure becomes small, which prevents the channel material from high thermal stress and stress-induced cracking.

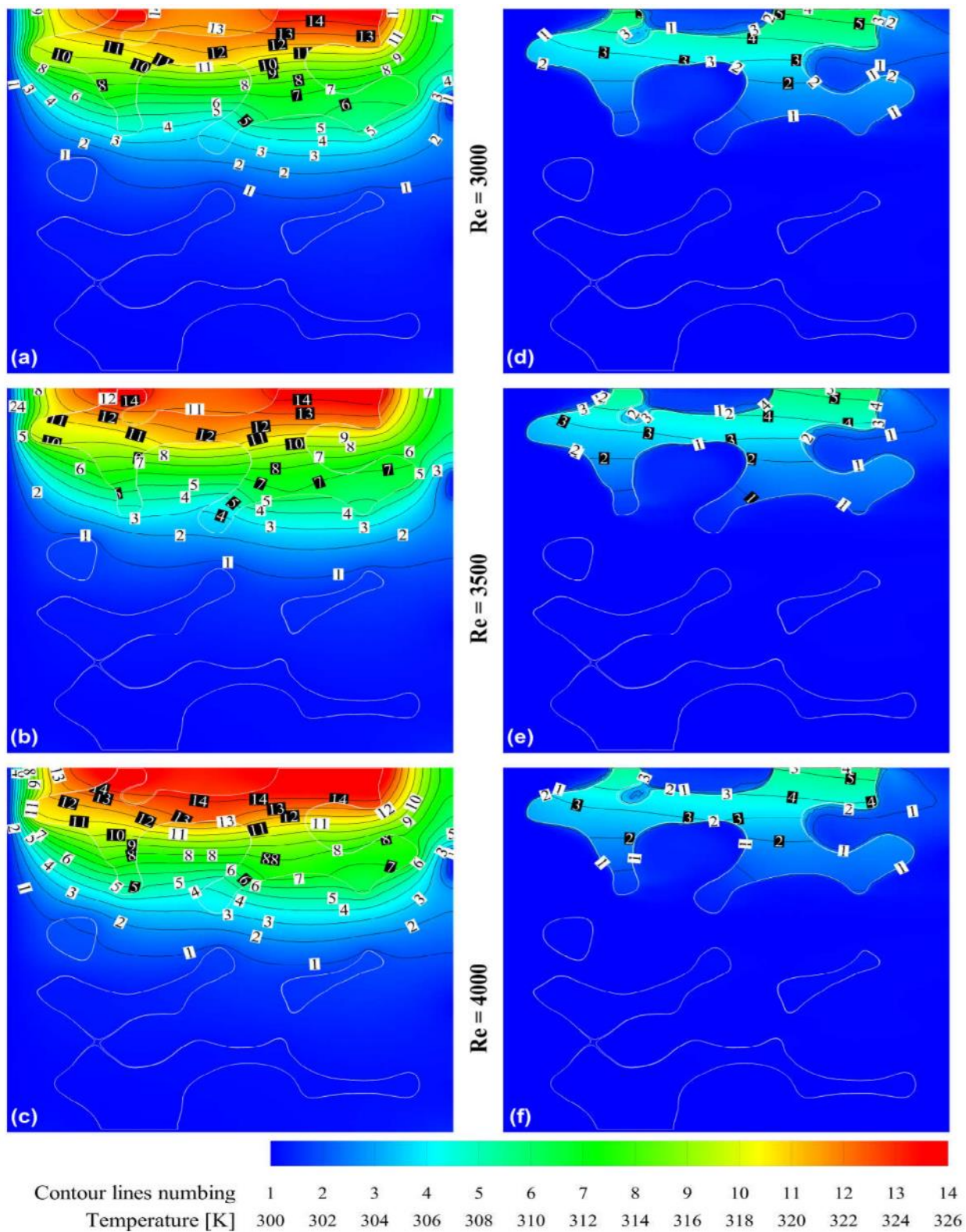


Fig. 5.14: Temperature contours of a small portion of the channel taken from the upper portion. Areas under solid black lines are solid portions and the outside area is fluid.

Contours is plotted at $\Psi = 0.77$ and $\text{DNI} = 250 \text{ kW/m}^2$. a), b) and c) show temperature distribution for air while d), e) and f) for water.

Figure 5.15 depicts the total heat flux at the surface of the SiC porous structure for which conduction, convection, and radiation are contributed. The variation of total heat flux with Reynolds number is presented in Fig. 5.15 for porosities of 0.69, 0.77 and 0.85 and solar concentrations of 250, 500, and 750 kW/m^2 . The considered Reynolds number range was 3000 to 4000 for air (Fig. 5.15a) and water (Fig. 5.15b). It can be observed that the heat flux remains higher for air compared with water for given conditions. This is because of the low density and viscosity of air, which results in high velocity for a given Reynolds number. This in turn enhances the convection heat transfer from the porous structure surfaces; hence, the rate of heat flux from the SiC porous surface increases. Moreover, the effect of Reynolds number on the heat flux for the air is almost negligible, i.e. the force convection at high velocities does not alter the considerable heat fluxes at high Reynolds numbers, i.e. increasing the flow rate, due to increase of the Reynolds number, does not alter notably the temperature distribution within the air as shown in Fig. 5.15. In the case of water, the total heat flux linearly decreases with increasing Reynolds number. Hence, the rate of heat transfer from the SiC porous structure to the fluid increases for high-density fluid. The findings reveal that the influence of Reynolds number is almost the same for all solar concentrations and porosities. For both fluids, increasing solar concentration gives rise to considerably increased total heat flux and the porosity follows the same trends for increasing solar concentration. For water, the increasing solar concentration at higher porosity reduces the enhancement of total heat flux. Hence, normalized Nusselt number reduce as compared to those occurring for lower porosities. This behavior is associated with the coupling of flow properties and porosity of the structure, which allows less

duration for the fluid to be trapped in the pore sites and become fully turbulent in these regions.

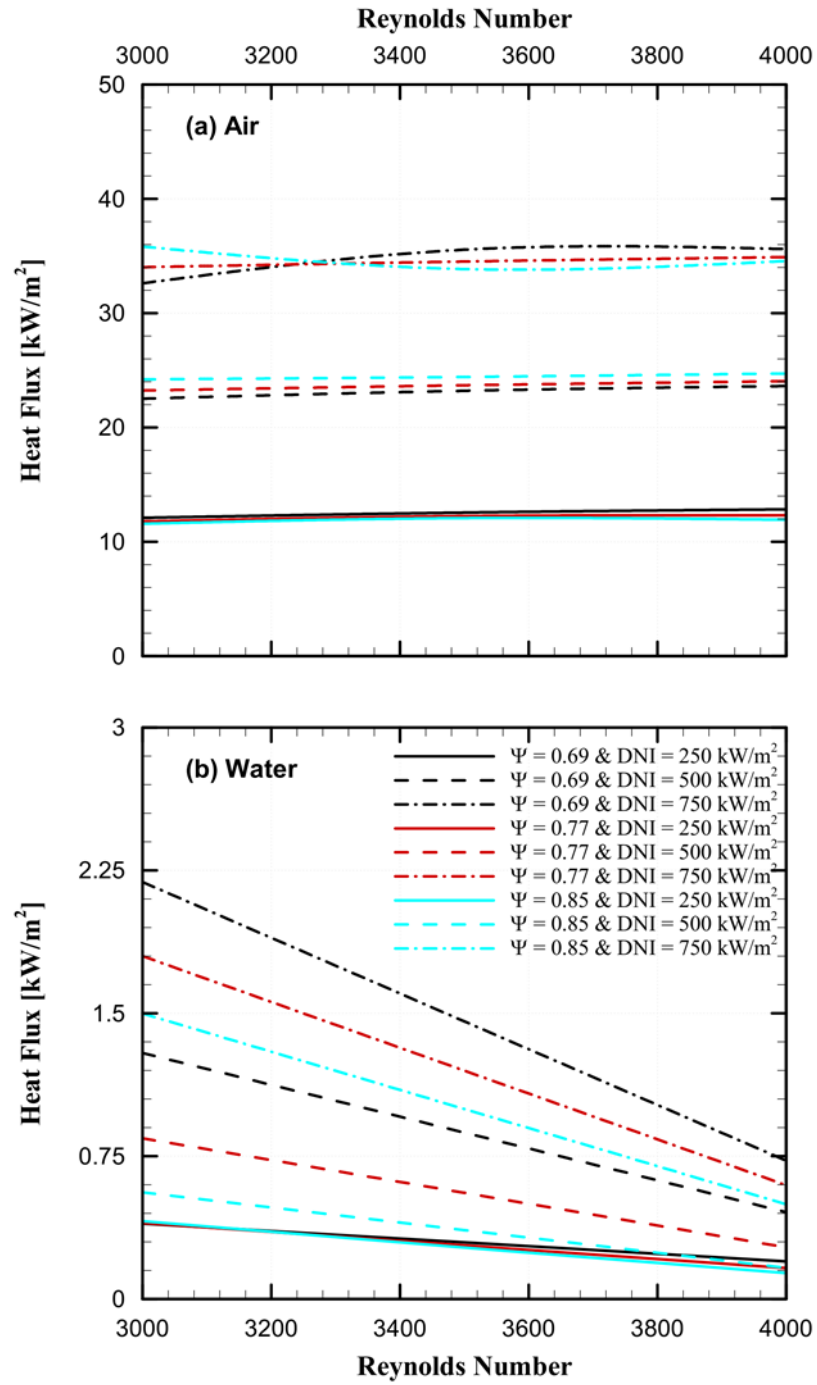


Fig. 5.15: Total heat flux at wall of SiC receiver against porosity and Reynolds number for a) air and b) water

The heat transfer rates from the solar harvesting process are assessed using the normalized Nusselt number which represents the amount of convective to conduction heat rates. Figure 5.16 illustrates the variation of global normalized Nusselt number with Reynolds number for various solar concentration values and porosities. Global, here, refers to the overall Nusselt number that is normalized through dividing by its counterpart that contains non-receiver channel configuration. When the water is considered as a heat transfer fluid, increasing Reynolds number considerably decreases normalized Nusselt number. This seems due to the lower flow resistance inside the solar receiver at high Reynolds numbers. However, air exhibits inconsistent trends for the normalized Nusselt number, which is associated with the unpredictable nature of turbulence, particularly when the pore structure promotes turbulence. At high Reynolds numbers, the mixing behavior of fluid becomes less which leads to lower heat recovery from the porous structure. Similarly, higher solar concentrations give rise to a decrease in the normalized Nusselt number for both fluids (more profound in the case of water) which is true for all porosity. This occurs because of the inertia forces, which become dominant due to the turbulent mixing of the fluid inside the pores. Higher solar concentration creates high heating loads on the receiver which results in the large difference between fluid and solid temperatures. Trends of normalized Nusselt number with increasing porosities appear to be decreasing for air, however, water shows the opposite behavior for normalized Nusselt number. For the case of air, it should be noted that lower porosity gives rise to normalized Nusselt number because of less volume ratio of the flow path to solid, which increases the residence time of fluid as well as trapping of fluid inside the pores. The highest value of normalized Nusselt number is

achieved at low Reynolds number, low solar concentration, and low porosity in the case of air; however, water results in the highest normalized Nusselt number for high porosities.

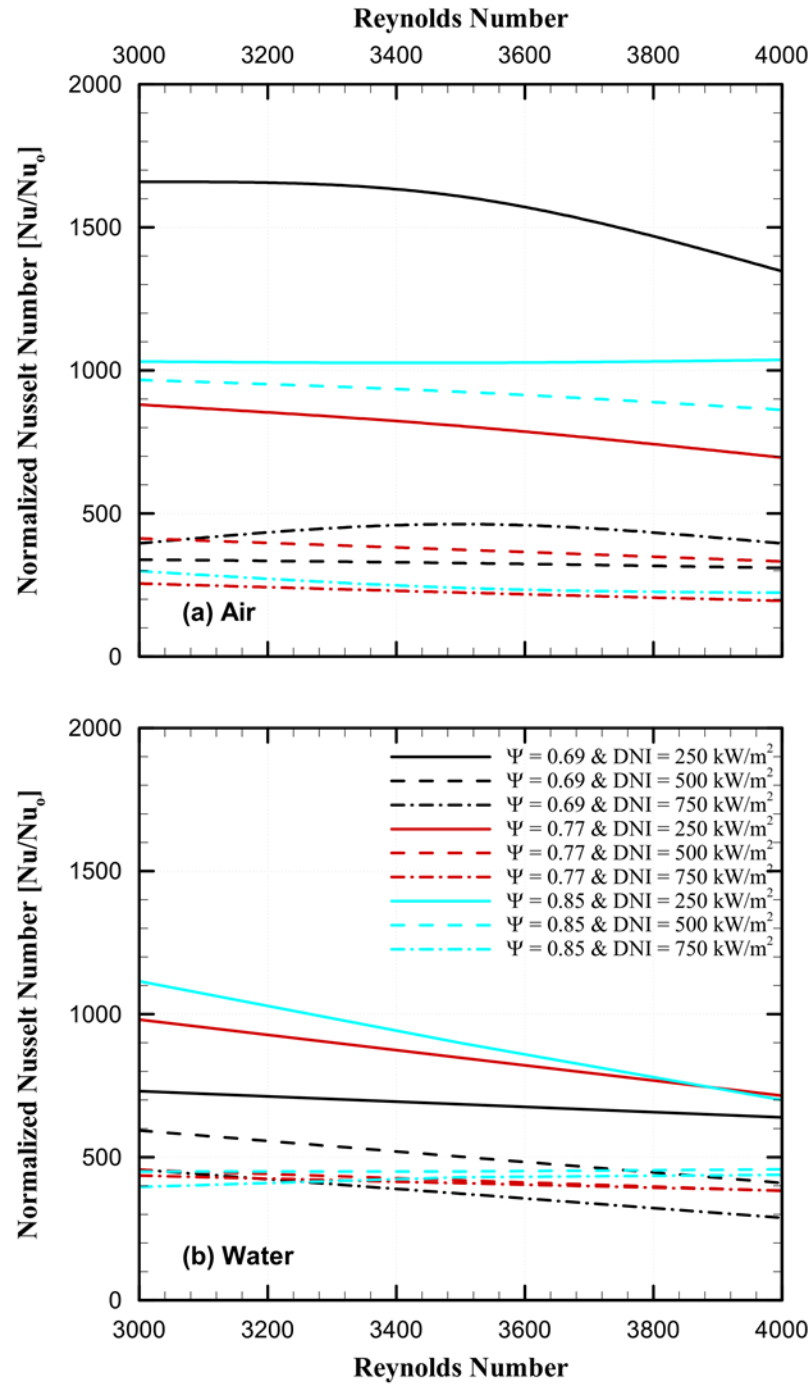


Fig. 5.16: Global normalized Nusselt number (Nu/Nu_0) of SiC receiver against porosity and Reynolds number for a) air and b) water.

5.3.2. Thermal Effectiveness

For the numerical investigations, the thermal efficiencies (η_1 and η_2) are defined based on the normalized mass-weighted exit temperature, Eq. (48), and normalized maximum temperature, Eq. (49), in the channel. Normalized exit temperature indicates the rise of temperature throughout the flow domain which is related the effectiveness of the thermal system, while normalized maximum temperature is associated with the maximum rise of temperature at any location within the flow domain. Figure 5.17 depicts the variation of normalized exit temperature following the variables given in Table 5.2. Likewise, Fig. 5.18 shows the variation of normalized maximum temperature. In case of air, the influence of Reynolds number on the normalized exit temperature is almost negligible for all porosity and solar concentration, but the increasing trend is observed for normalized maximum temperature for porosity ($\Psi = 0.69$ and 0.77) and this shows the decreasing trend for porosity ($\Psi = 0.85$). The effect of solar concentration remains the same for all porosities. This can be attributed to the optimal point of normalized maximum temperature against these porosities for the air. However, increasing solar concentration gives rise to high values of normalized exit and the normalized maximum temperatures for air. The enhancement in normalized exit temperature is observed as solar concentration increases. The variation of normalized exit temperature and normalized maximum temperature follows the same fashion for all porosities while the trend changes with changing Reynolds number and solar concentration, which is particularly true for air. In case of water, increasing Reynolds number causes a decrease in normalized exit and normalized maximum temperatures. This effect is more prominent for normalized maximum temperature. The fluid exit temperature is associated with the heat transfer through

conduction, convection, and radiation inside the volumetric receiver, thus, higher porosity ($\Psi = 0.85$) shows slightly different behavior than other porosity values, which can be supported by the heat flux variation (Fig. 5.15-b) and Nusselt number variation (Fig. 5.16-b). Variation of both thermal efficiencies with increasing solar concentration remains the same for all porosities. High radiation intensities enhance the thermal efficiencies of the solar harvesting system particularly for water as a heat transfer fluid.

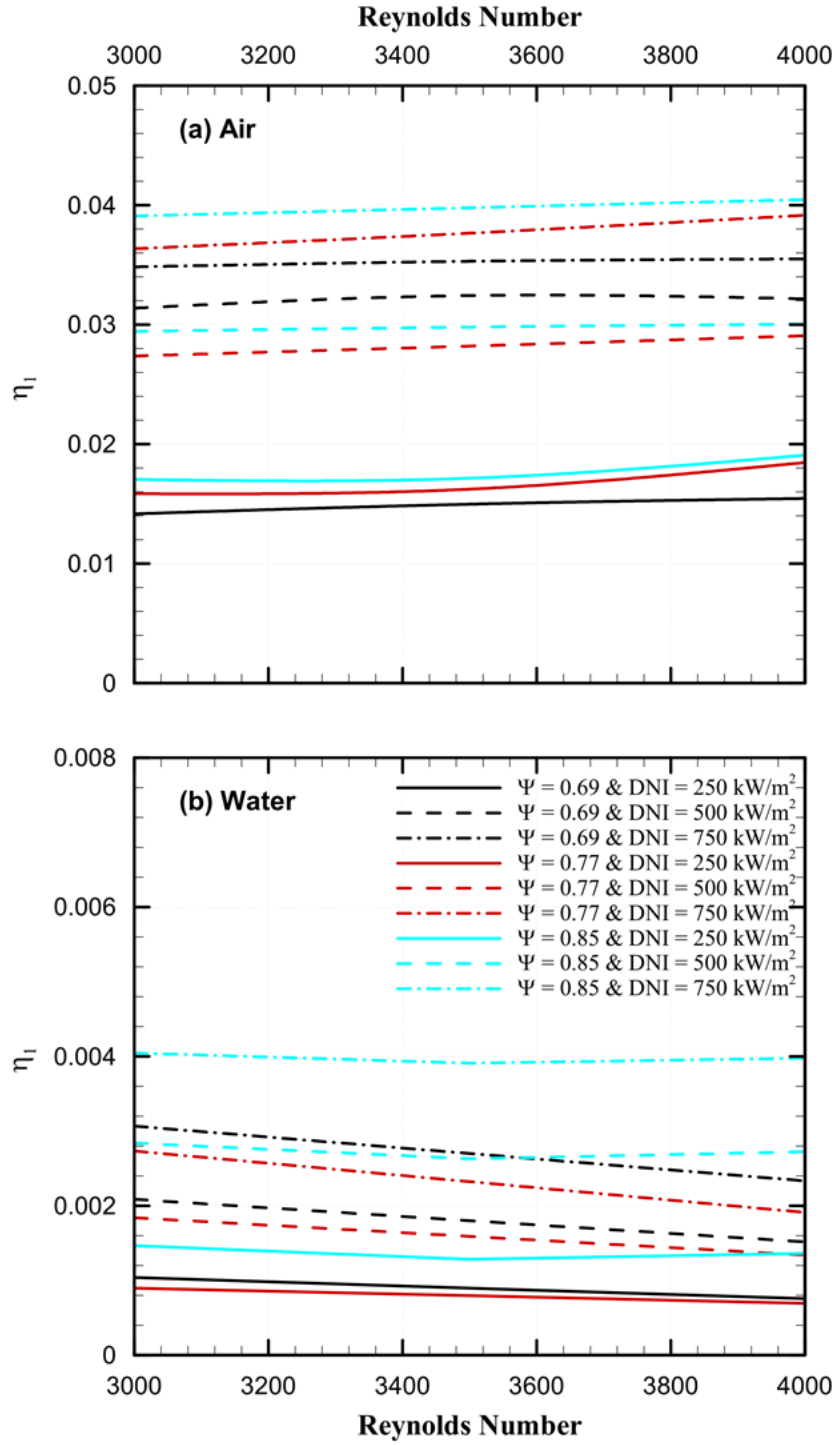


Fig. 5.17: Total thermal efficiency (η_1) of channel having SiC receiver against porosity and Reynolds number for a) air and b) water.

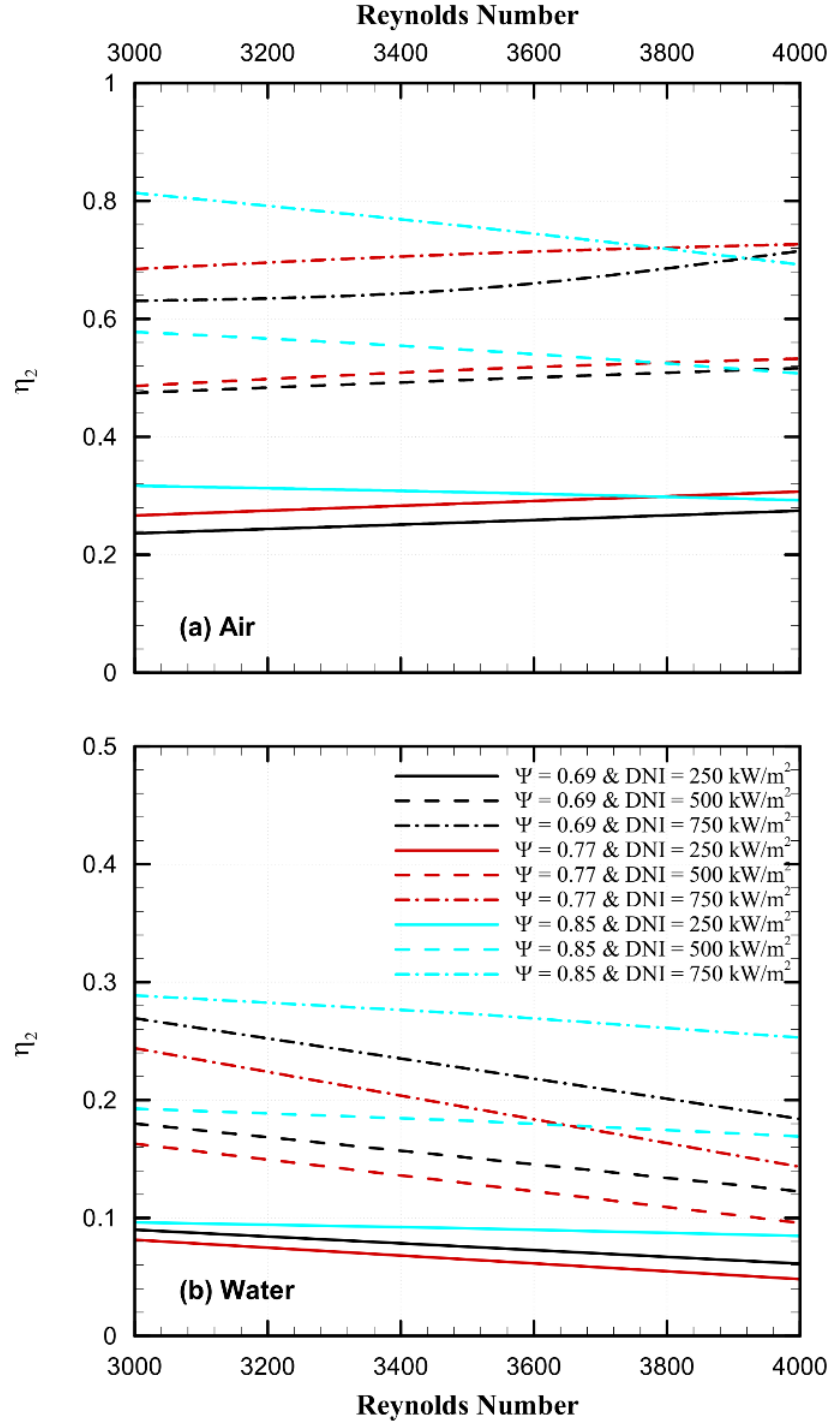


Fig. 5.18: Normalized maximum gain temperature (η_2) in a channel having SiC receiver against porosity and Reynolds number for a) air and b) water.

5.3.3. Flow analysis

Figure 5.20 shows velocity distribution within the pores of SiC receiver located in the channel for various Reynolds numbers at $\Psi = 0.77$ and $\text{DNI} = 250 \text{ kW/m}^2$. The air results in higher velocity magnitude as compared to water because of the low viscosity, provided that both fluids possess the same distribution of high and low-velocity regions. Irregular patterns in velocity distribution demonstrate that the porous medium promotes turbulence, which results in the trapping of fluids while causing the vorticity generations inside the pores. Partial blockage of the porous sites increases the downstream pressure, which results in low-velocity regions. The rapid change in the flow path within the porous structure causes the development of the flow stagnation regions. This affects the local pressure distribution within the porous sites. The high-velocity regions contribute to the formation of vorticities under the shear effect, which contributes to the local pressure variation in this region. In addition, the flow passages in some regions are fully interlinked, which in turn gives rise to the attainment of the high-velocity magnitude in these regions. This adds to the forced convection influence on the improvement of the heat transfer rates in the regions where the porous structure is fully interlinked. Nevertheless, the contribution of heat diffusion and natural convection to the heat transfer rates becomes significant in the closed paths of the porous structure where the flow velocity decreases considerably.

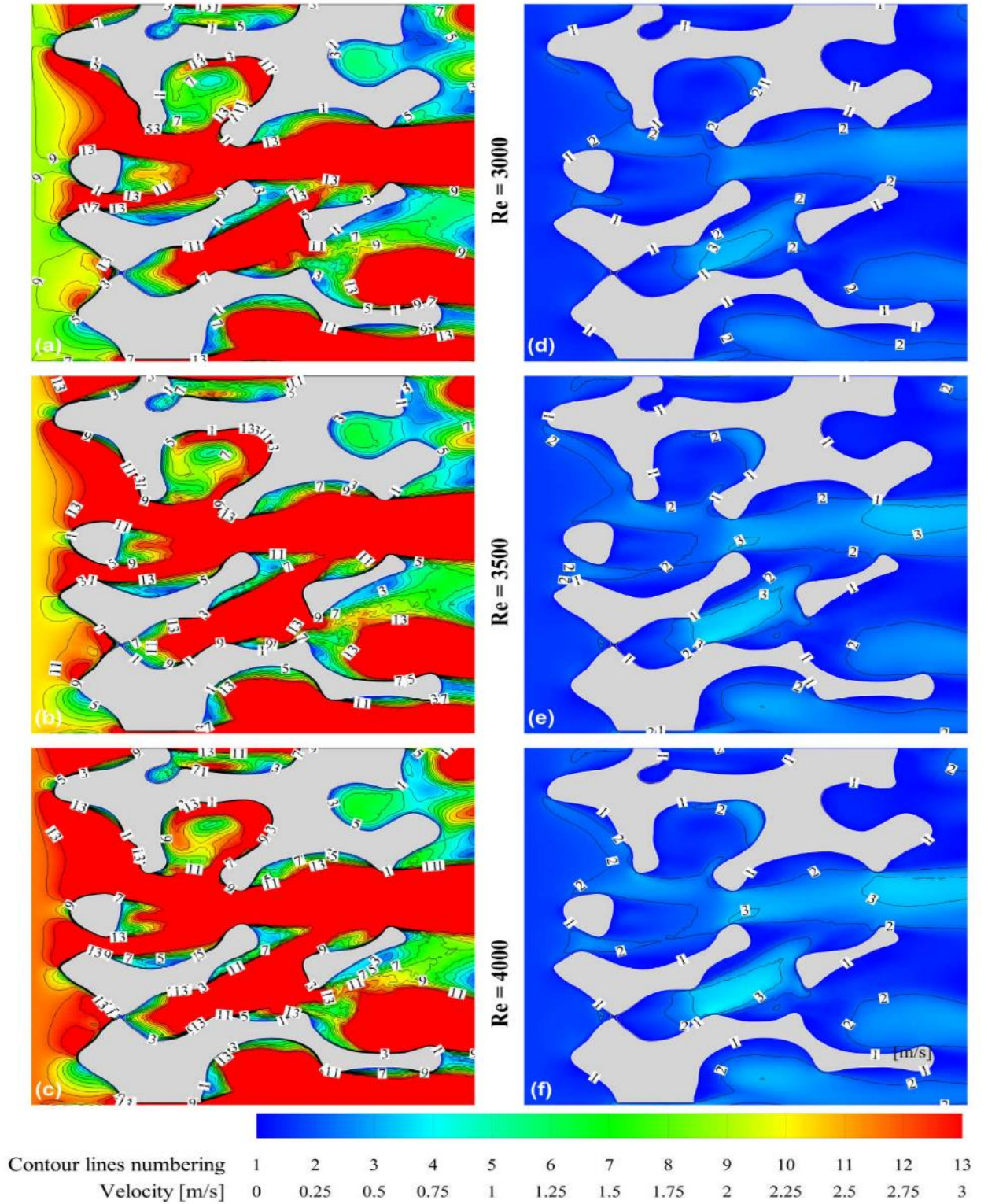


Fig. 5.19: Velocity magnitude of small portion of channel taken at $\Psi = 0.77$ and $\text{DNI} = 250 \text{ kW/m}^2$. Areas under black lines (grey portion) shows solid portion (porous SiC foam). a), b) and c) show velocity distribution for air while d), e) and f) for water.

On the other hand, the fluid pumping power is directly related to the pressure loss while transporting fluids within the channel. Here, the pressure loss at the outlet is mainly caused by the porous structure of the receiver due to the pressure drag and the fluid friction within the pore sites. The high-pressure loss in the porous structure occurs due to the irregular paths connecting the pore sites even though the irregular paths promote the flow mixing that is favorable for heat recovery through convection. Introducing the porous structures in a flow system faces the challenges of high pumping power requirements. This is the only major loss in the system; however, the thermal system shows incredibly high thermal performance than the pumping power deficit, which enables the adaptation of the porous structures at large-scale systems. The pressure losses for air and water are shown in Fig. 5.20-a and Fig. 5.20-b, respectively. It can be seen that the pressure loss for water is higher than air due to the high viscosity of water, which causes the high rate of shear strain in the fluid. The influence of solar concentration on the water is negligible while air shows some increment in pressure with increasing temperature. High solar concentration gives rise to a high-temperature rise in the channel which increases the shear rate of air significantly and the pressure loss increases accordingly. In case of water, the fluid possesses an inherently high shear rate; therefore, it has less effect on pressure loss. Low porosity refers to a less fluid volume ratio which results in more fluid trapping and increases the drag. The trends of pressure loss increase linearly with porosity, however, for air at the lowest porosity ($\Psi = 0.69$), it exhibits different behavior which supports the turbulence immaturity due to porous structure for low viscous fluids [154]. Hence, careful selection of porosity remains crucial for designing a solar harvesting system that incorporates porous volumetric receivers.

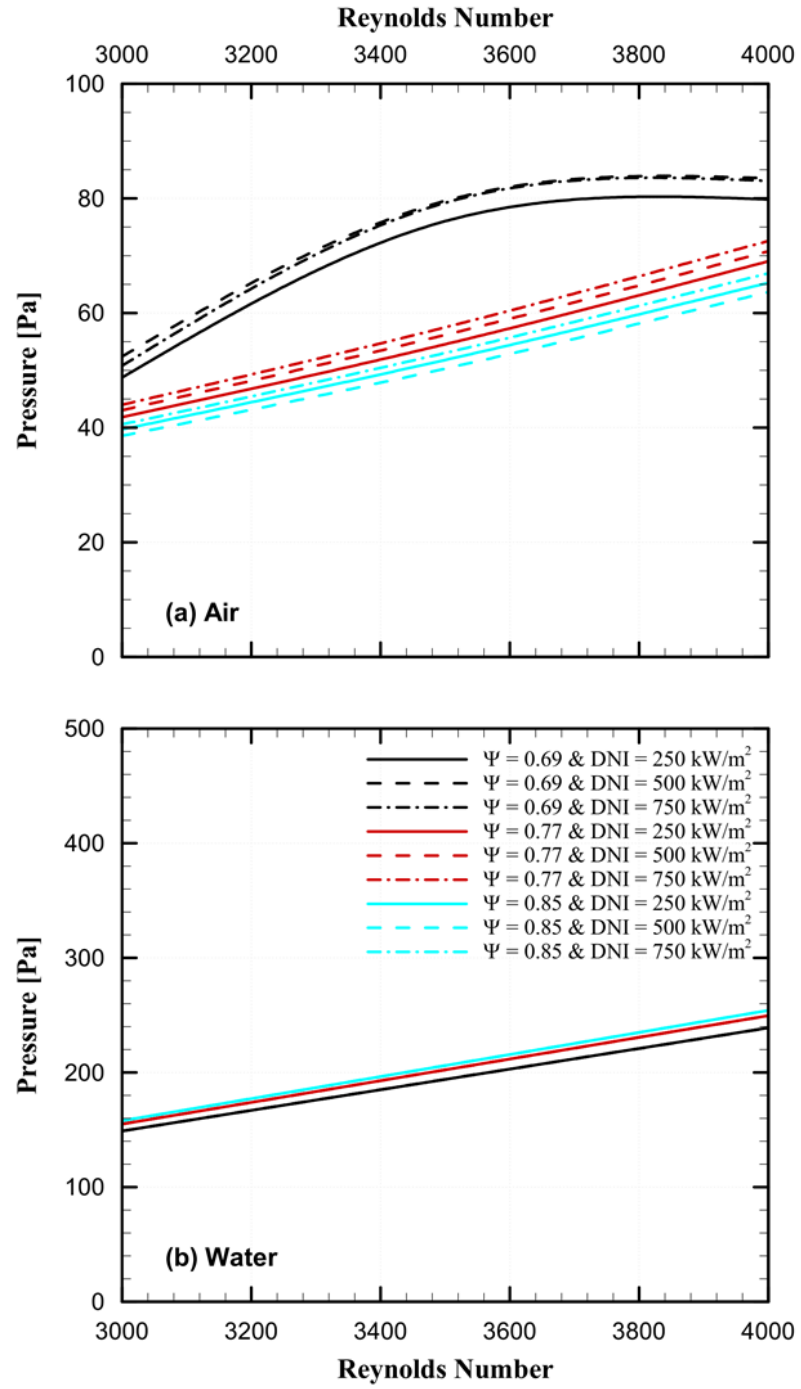


Fig. 5.20: Pressure drop in channel having porous SiC foam against SiC receiver against porosity and Reynolds number for a) air and b) water

5. 4. Constructal design of porous SiC foam

After the assessment of porous SiC foam with square cross-sectional channel, numerical analysis is further extended for evaluating the influence of channel design. For this purpose, the volume-averaged governing equations are solved by the numerical finite volume method via ANSYS Fluent 2020 (R2). Volume-averaged governing equations used dual-cell approach with LTNE for the flow configurations. The validation study is conducted here in line with early published work [23]. The physical domain is modeled as prescribed in the previous work [23]. Reason behind the selection of this early work is the use of same numerical approach ‘local thermal non-equilibrium model’ for porous zone, and boundary conditions which are also of the same interest. Furthermore, this study includes both experimental and numerical work for investigating heat transfer through SiC foam volumetric solar receiver. This validation study adopted same geometrical parameters which was a cylindrical SiC porous media having 50 mm length and 50 mm diameter (as shown in Fig. 5.21) and 80 mm extension of channel for airflow. The calculation details of solar radiation were considered including adsorption and scattering phase function using DO radiation model in ANSYS Fluent. Inlet air velocity and radiation flux were 1.3 m/s and 600 kW/m², respectively. Temperature-dependent fluid and solid thermophysical properties were selected for this numerical investigation. Porous zone followed same momentum sink parameters (K and f), effective thermal conductivity (k_{eff}), porosity (0.8) and emissivity (ϵ). Interfacial heat transfer coefficient and optical properties (adsorption coefficient and scattering coefficient) are calculated based on Eqs 28 - 32. As mentioned in the early work, 84.5 kPa is applied at outlet pressure (-16.825 gauge pressure [23]). In Fig. 5.22, it can be observed that temperature variation throughout the channel is in good

agreement with that presented in [23]. The trend of the temperature curve is exactly matching, and the error is under tolerance of 5%.

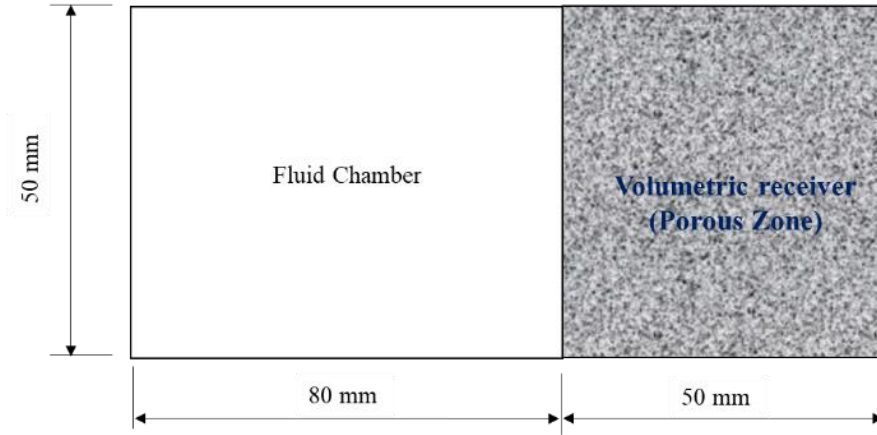


Fig. 5.21: Schematic of validation's numerical domain

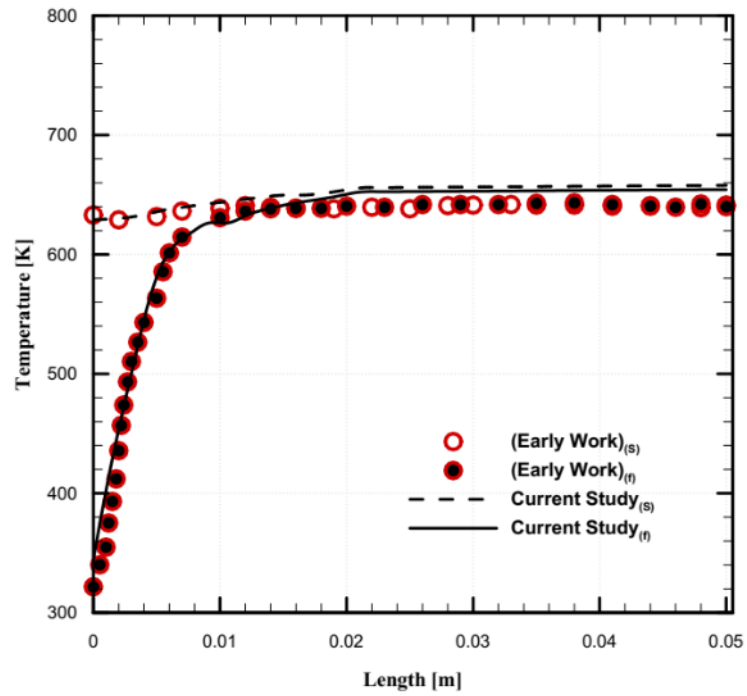


Fig. 5.22: Temperature variation along the flow direction for validation purposes.

Residuals of governing equations and mass-flow rate monitor at inlet and outlet, confirmed the convergence of the solution. The solution was converged at residuals for continuity and momentum below 10^{-6} and for energy and DO equation was 10^{-8} , while mass-flow rate monitor should show the difference of mass-flow rate at inlet and outlet below 10^{-12} . After satisfying the required convergence criteria, the solution is calculated for further 3000 iterations for confirming a stable solution. For achieving converged results, the distance-based coupled scheme was used for pressure-velocity coupling. A gradient of spatial discretization was 'Least Squares Cell-Based' while pressure, momentum, turbulent kinetic energy, turbulent dissipation rate and energy were of second-order upwind scheme. The solution controls involved pseudo transient explicit relaxation factors are 1.0 for pressure, momentum, density, body forces, Discrete Ordinates, and turbulent viscosity while 0.75 for energy, turbulent kinetic energy, and turbulent dissipation rate. Since the solution contains DO radiation model, 3x3 for divisions and 3x3 for pixels of angular discretization (N_θ and N_ϕ) were considered for each octant of angular space. N_θ and N_ϕ denote zenith and azimuth angles, and their values were taken for every case/configuration. These values are highly recommended in literature for saving computational power as well as in ANSYS Fluent's user guide [150].

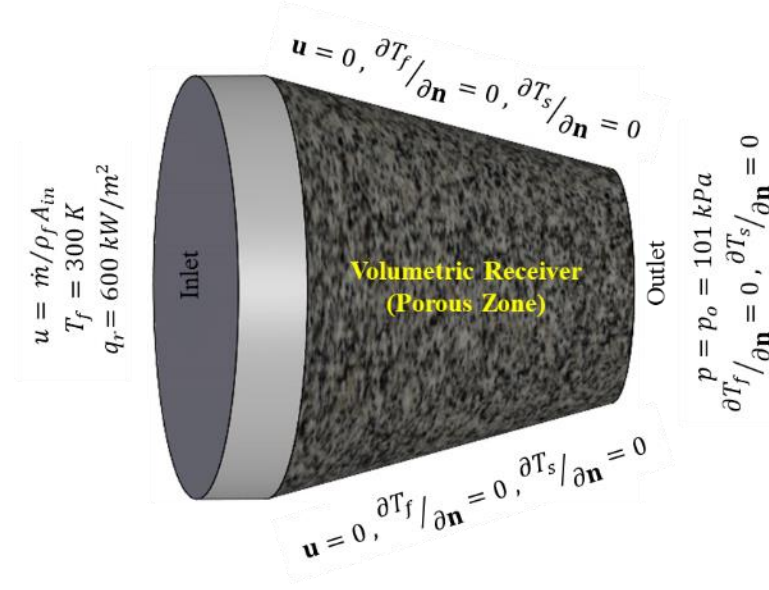


Fig. 5.23: Numerical representation for boundary conditions of the volumetric receiver.

Table 5.3: Description of geometry parameter for proposed configurations.

Configuration #	Angle (θ°)	Ratio (r) $r = \frac{d_{in}}{d_{out}}$	Configuration #	Angle (θ°)	Length (L) [m]
1	10	1.2	26	10	0.034
2	15	1.2	27	15	0.034
3	20	1.2	28	20	0.034
4	25	1.2	29	25	0.034
5	30	1.2	30	30	0.034
6	10	1.3	31	10	0.038
7	15	1.3	32	15	0.038
8	20	1.3	33	20	0.038
9	25	1.3	34	25	0.038

10	30	1.3	35	30	0.038
11	10	1.4	36	10	0.042
12	15	1.4	37	15	0.042
13	20	1.4	38	20	0.042
14	25	1.4	39	25	0.042
15	30	1.4	40	30	0.042
16	10	1.5	41	10	0.046
17	15	1.5	42	15	0.046
18	20	1.5	43	20	0.046
19	25	1.5	44	25	0.046
20	30	1.5	45	30	0.046
21	10	1.6	46	10	0.050
22	15	1.6	47	15	0.050
23	20	1.6	48	20	0.050
24	25	1.6	49	25	0.050
25	30	1.6	50	30	0.050

5.4.1. Thermal Analysis

To find the optimal design of the solar volumetric receiver, various porous configurations are investigated using the Constructal approach. These configurations are tabulated in Table 5.3. For all configurations, the applied concentrated solar flux is 600 kW/m^2 , inlet flow temperature is 300 K, the mass flow rate is 0.003221 kg/s and a porosity of 0.8 is considered for the volumetric receiver. The thermal performance of the solar volumetric receiver is analyzed using temperature distribution, effective length, effective area, outlet

temperature and maximum temperature of domain. Evolution of design is incorporated by variation of three parameters including ratio of inlet/outlet diameters, cone angle and length of receiver. While using volume of receiver as a constraint, then either ratio and cone angle can be varied, or length and cone angle can be varied. Ratio of inlet and outlet diameters (r) and receiver length (L) were varied as presented in Table 5.3. Propagation of solar radiation is same as flow inlet as shown in Fig. 3.3. Furthermore, the key interest of this investigation is to enhance the heat transfer within the volumetric solar receiver through increase temperature uniformity, exit and maximum temperatures.

Since this numerical method used the LTNE model for solving temperature distribution, the computational domain of porous zone resulted in temperature distribution of both phases (fluid and solid). In Figures 5.24 - 5.28, the temperature distribution along the flow path is depicted. These temperature plots are taken as average of corresponding nodes in their cross-sectional area along the flow path i.e., volume-averaged temperature as a function of axial coordinate. The fluid phase temperature at the inlet ($x=0$) of the porous media is not equal to 300 K. Figures 5.29 and 5.30 are evident in which the temperature values closer to the channel walls are higher than the centerline temperature and this is due to no-slip boundary condition. The variation of temperature distribution pattern remains similar for the solid phase of all configurations which is slightly increasing towards the outlet of channel. For fluid phase, the trend for temperature variation also appears the same for all configurations which is exponentially increasing at the porous media entrance and then the change of temperature become insignificant until the exit. Higher temperature of solid phase closer to the entrance is associated with the applied radiation to the SiC receiver boundary at $x = 0$. Similarly, fluid is considered transparent and only subjected to recover

heat from the SiC surfaces, mainly by convection. This is why the temperature of fluid starts with around 300K (as inlet temperature) then increases due to convection heat transfer. For solid phase, low increment in temperature along the fluid path is related to less heat recovery through convection while conduction seems to be constant throughout the solid domain.

For ratio-based configurations ($r = 1.2, 1.3, 1.4, 1.5, 1.6$) in Figs. 5.24, 5.25 and 5.26, length of receivers tend to be shorter when cone angle (θ) was subjected to increase. It is worth mentioning that temperature is remarkably increased as cone angle increases. If temperature curves of lower ratios at different cone angles are considered to compare with higher ratios, results indicate no major enhancement than the comparison between the cone angles. However, higher ratio achieved thermal equilibrium location earlier than lower ratio. The converging shape of receiver also supports to increase the temperature along the flow path. It can be observed that when using cylindrical-shaped receiver (Fig. 5.22), the fluid/solid equilibrium temperature was not rising along the flow path. But in case of ratio-based configurations, approximately 20 – 30K increment is noted because the local energy absorbed by the porous zone become larger due to volumetric effect. Another finding from these temperature curves that is the thermal enhancement from $\theta = 10$ to $\theta = 15$ is 17%, however, this enhancement is reduced at higher cone angles i.e., $\theta = 25$ to 30. This seems to occur because the cross-sectional area is drastically decreased towards channel outlet which turns in higher velocity i.e., higher velocity leads to less residency and mixing of fluid in porous zone. Highest temperature is achieved while using the highest ratio ($r = 1.6$) and highest cone angle ($\theta = 30$).

In case of length-based configurations (as shown in Figs. 5.26, 5.27 and 5.28) temperature variation as function of axial coordinate for shorter length ($L = 0.034$ m) possessed profound increasing trend while the longer length has a constant temperature line for solid phase throughout the length. In addition, larger cone angles with shorter length configurations show more enhancement for temperature as compared to smaller cone angles with longer length configurations. However, larger cone angles with longer channel length revealed better rise in temperature for solid phase than same cone angles of shorter lengths. Hence, increasing cone angles with larger lengths (converging-shaped receiver) considerably improved thermal performance. Although highest temperature is achieved while using larger cone angles and longer lengths of receiver in length-based configurations, but these configurations mostly lack the thermal equilibrium between fluid and solid phase. Therefore, shorter length configurations for length-based configurations are demonstrated more favorable.

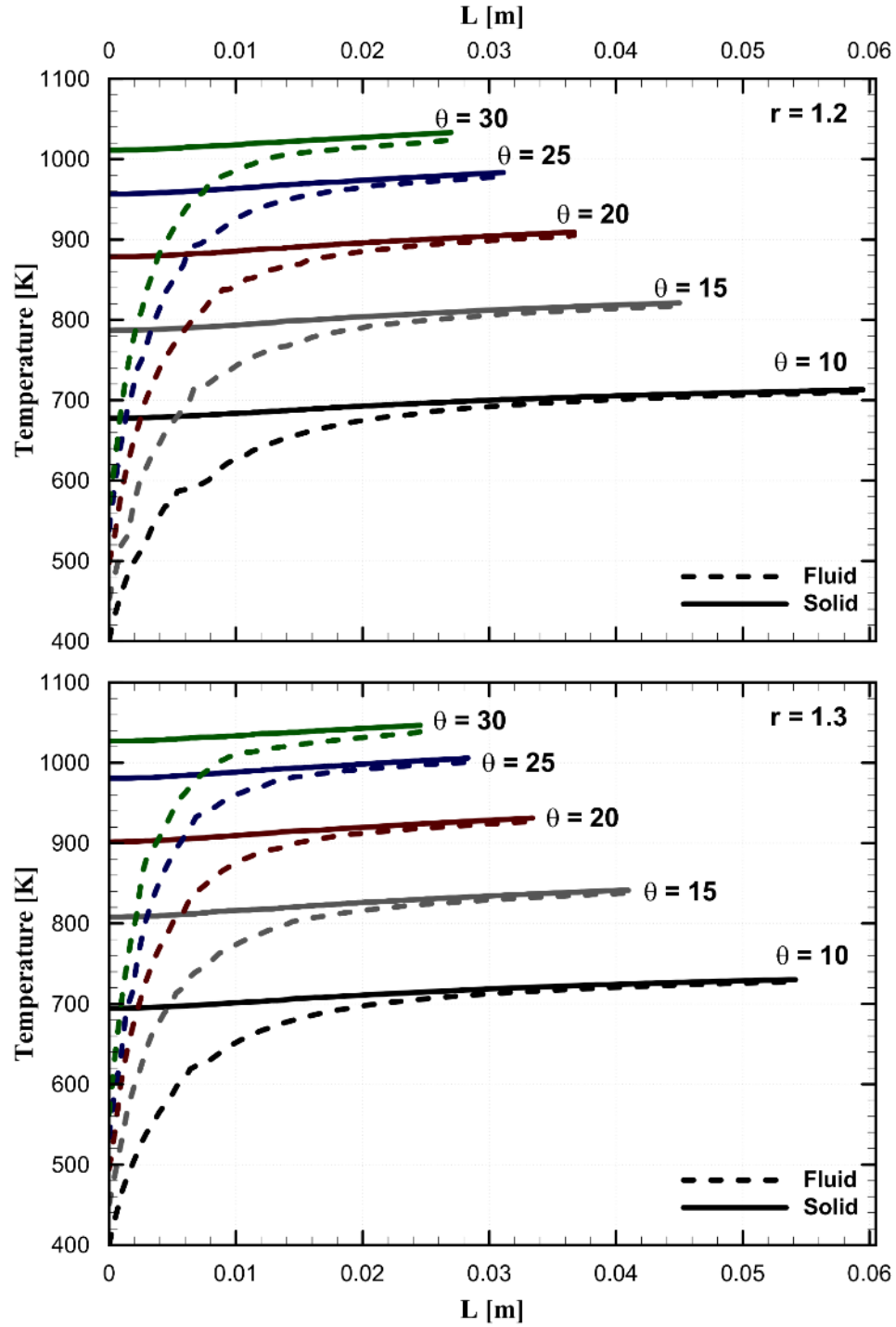


Fig. 5.24: Averaged temperature variation along the flow direction as a function of axial coordinate for configurations 1 – 10.

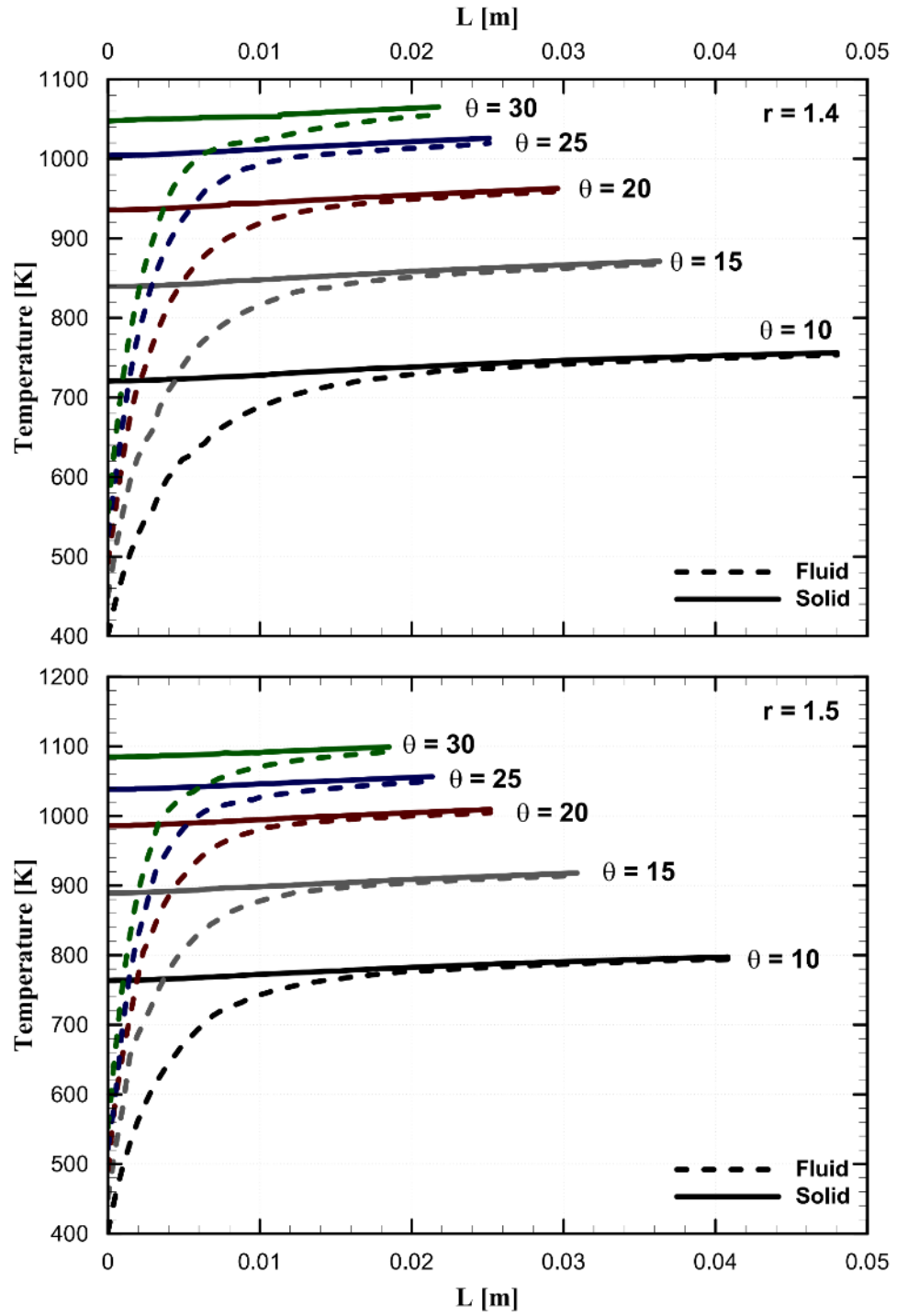


Fig. 5.25: Averaged temperature variation along the flow direction as a function of axial coordinate for configurations 11 – 20.

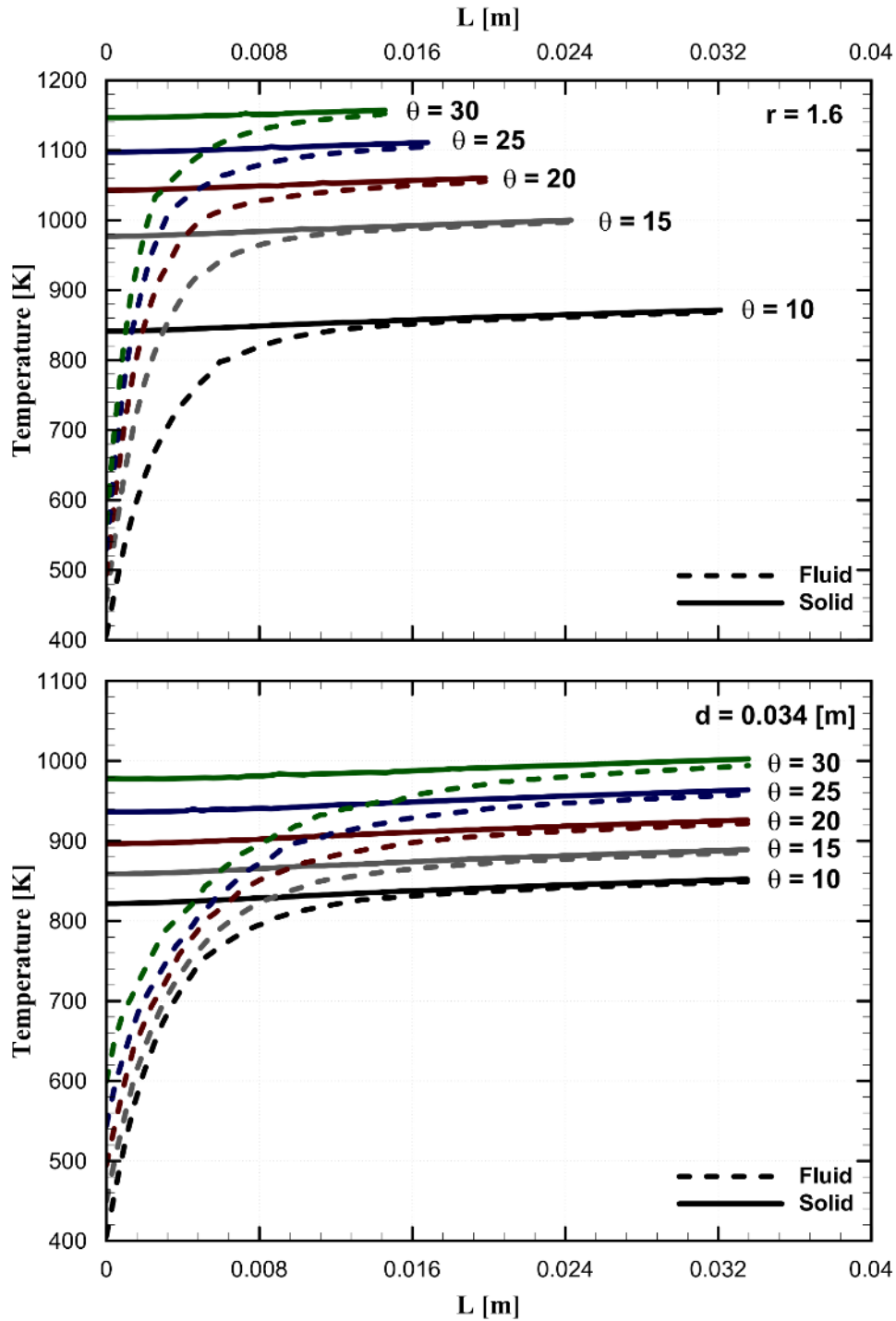


Fig. 5.26: Averaged temperature variation along the flow direction as a function of axial coordinate for configurations 21 – 30.

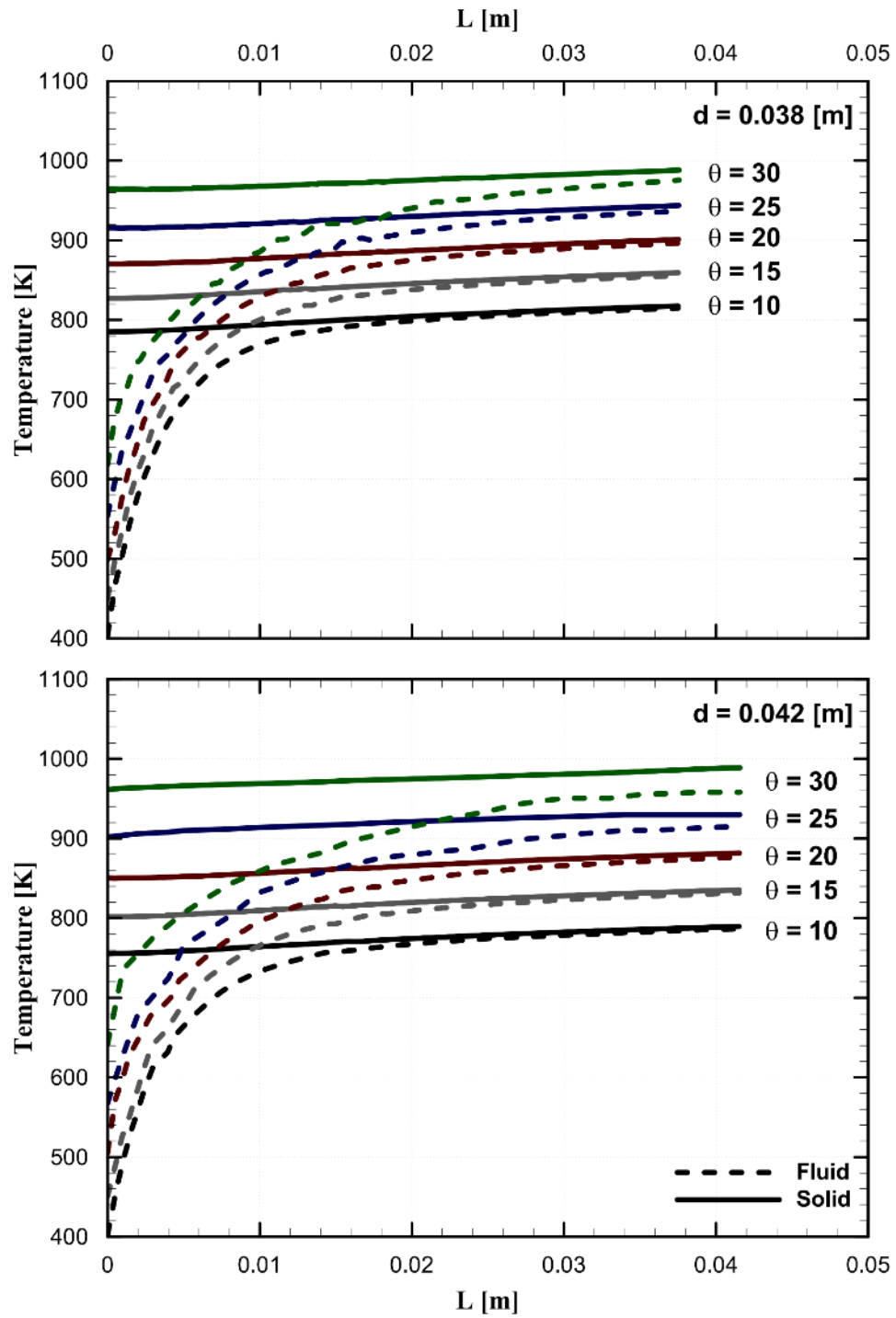


Fig. 5.27: Averaged temperature variation along the flow direction as a function of axial coordinate for configurations 31 – 40.

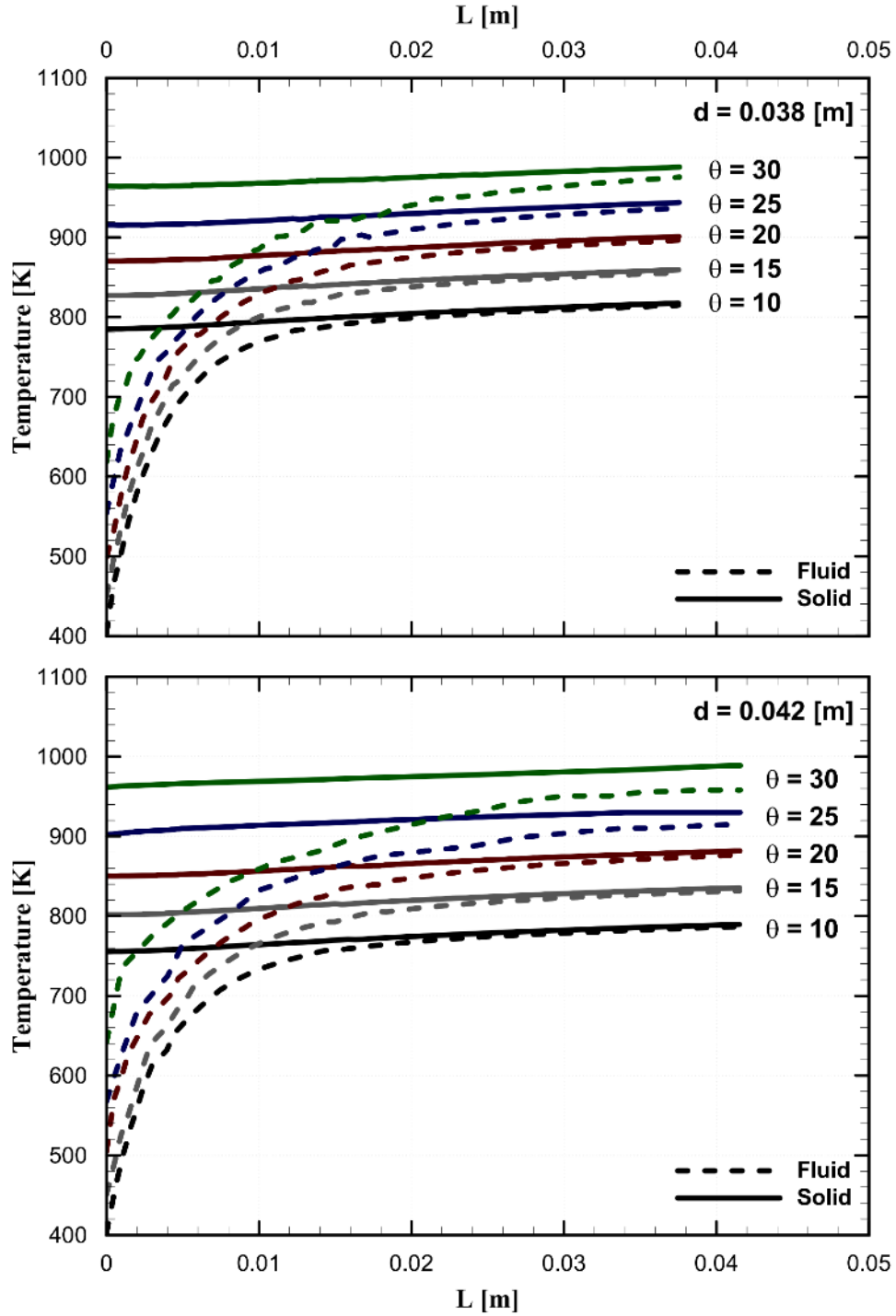


Fig. 5.28: Averaged temperature variation along the flow direction as a function of axial coordinate for configurations 41 – 50.

Figures 5.29 and 5.30 depict the temperature contours for configurations 3, 4, 8, 13, 18, 23, 28, 33, 38, 43, 48 and 49, which represent the mid-plane (parallel to flow direction).

Patterns of temperature profiles for fluid and solid phases are almost similar for all configurations. It is worth remarking that the solid phase exhibited higher temperature than fluid which is contributed to volumetric effect where absorption of radiation heat flux occurred. It can be also seen that near inlet, no-slip boundary condition at walls has more effect on the temperature profile (temperature is very less close to wall) but the converging shape also increased the temperature which provides good mixing of fluid. Ratio-based configurations indicate a significant enhancement of thermal performance as compared to length-based configurations. Furthermore, ratio-based configurations allow more heat recovery as compared to length-based configurations. Consequently, the pattern of temperature profile for thermal equilibrium is shifted towards outlet as cone angle increases. Also, it can be observed that the solid temperature is increased along the flow path, attributed to the volumetric effect because the radiative contribution overcomes the convective contribution. In the case of length-based configurations (Fig. 5.30) that lower temperature regions are wider which is not favorable for thermal uniformity. Solid phases of ratio-based configurations show continuous increment in temperature contours while less enhancement is noted in the case of length-based configurations.

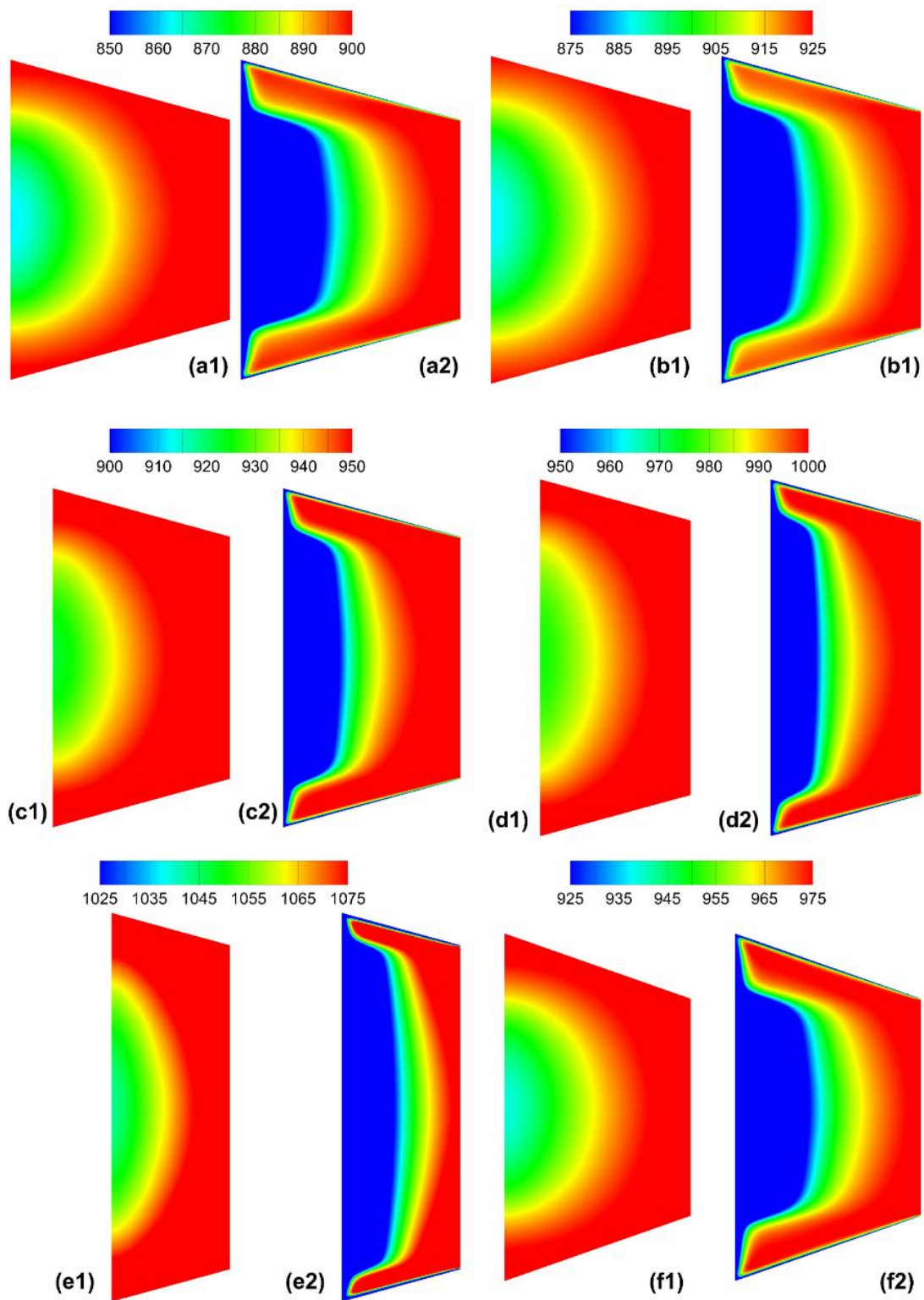


Fig. 5.29: (a), (b), (c), (d), (e) and (f) show temperature [K] contours for configuration 3, 8, 13, 18, 23 and 4, respectively. Sub-figures. names 1 and 2 show solid and fluid.

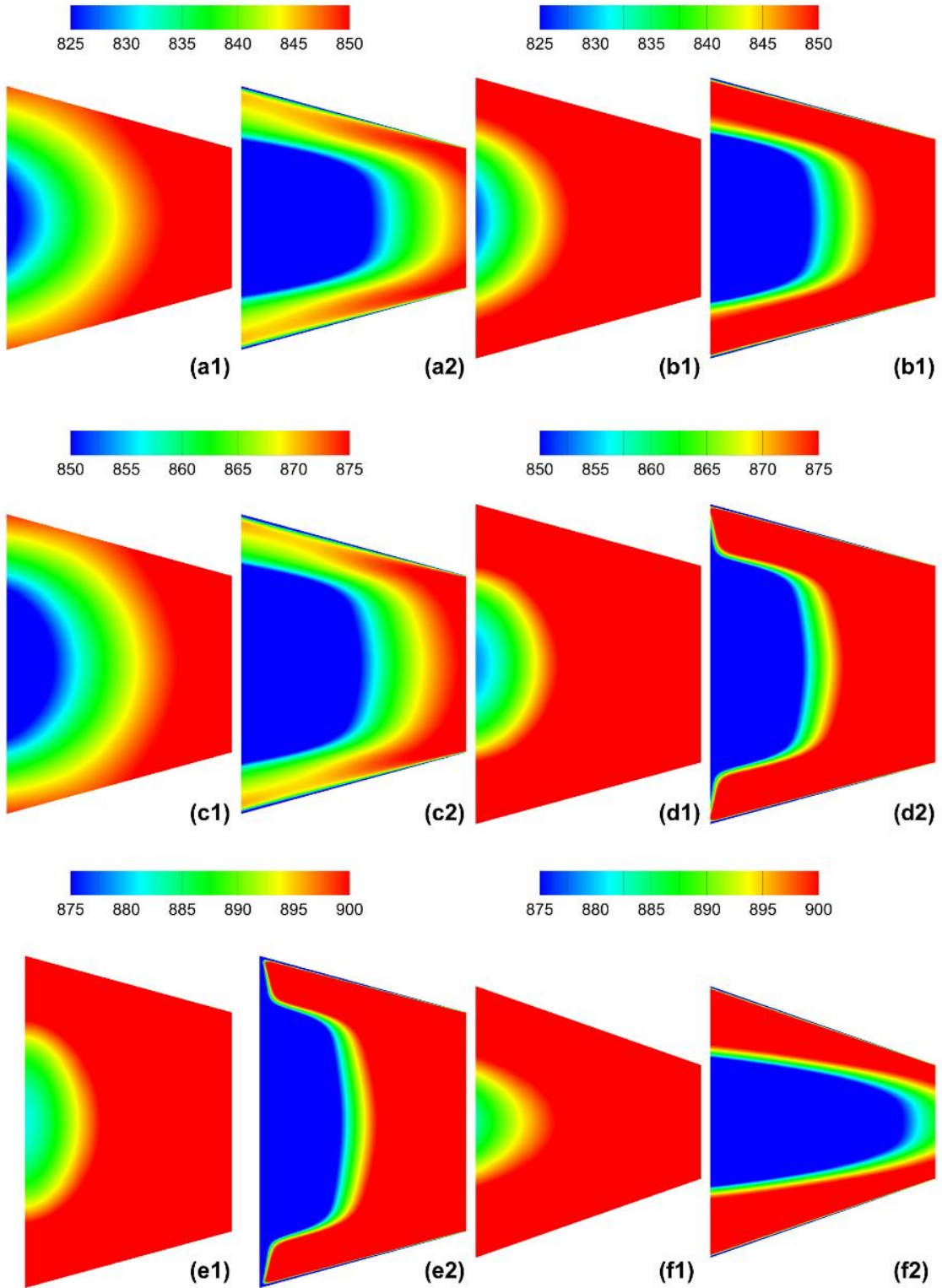


Fig. 5.30: (a), (b), (c), (d), (e) and (f) show temperature [K] contours for configuration 48, 43, 38, 33, 28 and 49, respectively. Sub-figure names 1 and 2 show solid and fluid.

In Figures 5.31 and 5.32, thermal equilibrium locations between solid and fluid phases are presented for ratio-based and length-based configurations, respectively. Achieving thermal equilibrium with thermal enhancement for proposed configurations is the ultimate goal of this investigation where optimization of receiver involves parametric analysis. It is worth to be mentioned that larger cone angles are associated with shorter length, at which thermal equilibrium and higher thermal performance attained. Proposed configurations are designed in such a way that the whole length of receiver is to be utilized to achieve higher solid/fluid equilibrium temperature. Fig. 5.31b shows that 60 - 80% of the receiver's length were utilized to attain those higher equilibrium temperature. This is mainly due to contribution of the irradiation through the porous media and the interfacial convective heat transfer from the solid to the fluid. When the radiation heat transfer predominates over the local convection which leads to the shorter thermal equilibrium length, due to the increased volumetric effect. Effective length is calculated using total length as a counterpart. The irregular trend of effective length could be of two reasons:

- while varying the ratio and cone angle, length of receivers is drastically changed which results in different geometry configurations,
- the presence of turbulent flow can cause irregular convection patterns, however, the trend of effective length between cone angles of different ratios remains same.

For length-based configurations larger cone angles and longer length shows impressive results but some (smaller cone angles and shorter length) configurations did not attain thermal equilibrium (Fig. 5.32). This depends on smaller cone angles results in smaller diameter's ratio, according to findings of ratio-based configurations that the smaller cone angles and the smaller ratio the longer thermal equilibrium length (Fig. 5.31). Like ratio-

based configurations, the length of thermal equilibrium becomes shorter as the cone angles increases. Simply, the shorter thermal equilibrium length turns into the smaller effective length. Furthermore, no configuration with $L = 0.034$ m achieved thermal equilibrium. Highest temperature is not obtained by these configurations, and they also lack the thermal equilibrium location, therefore, no further investigation is needed for finding the thermal equilibrium location.

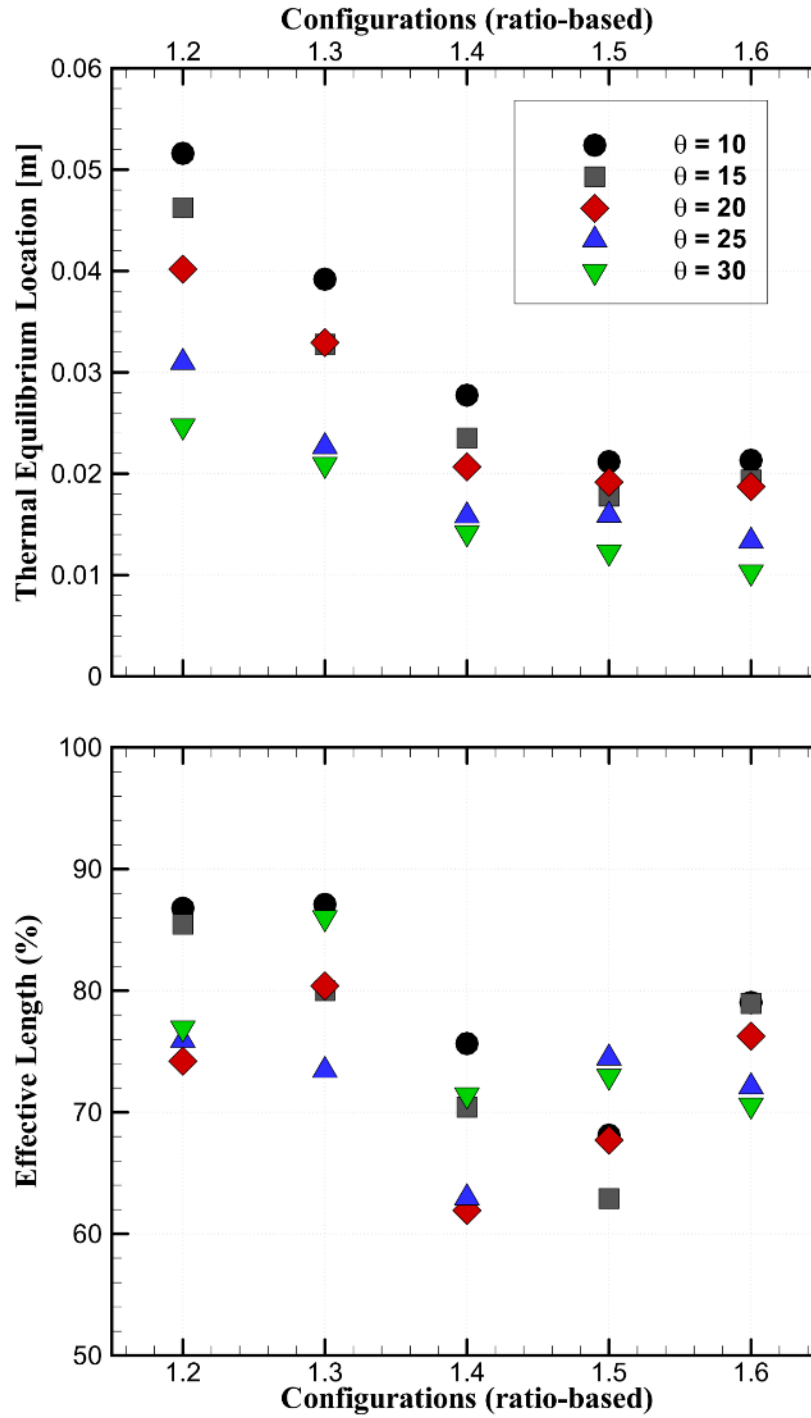


Fig. 5.31: Thermal equilibrium location corresponds to length of receiver and effective length for achieving thermal equilibrium for configuration 1 – 25.

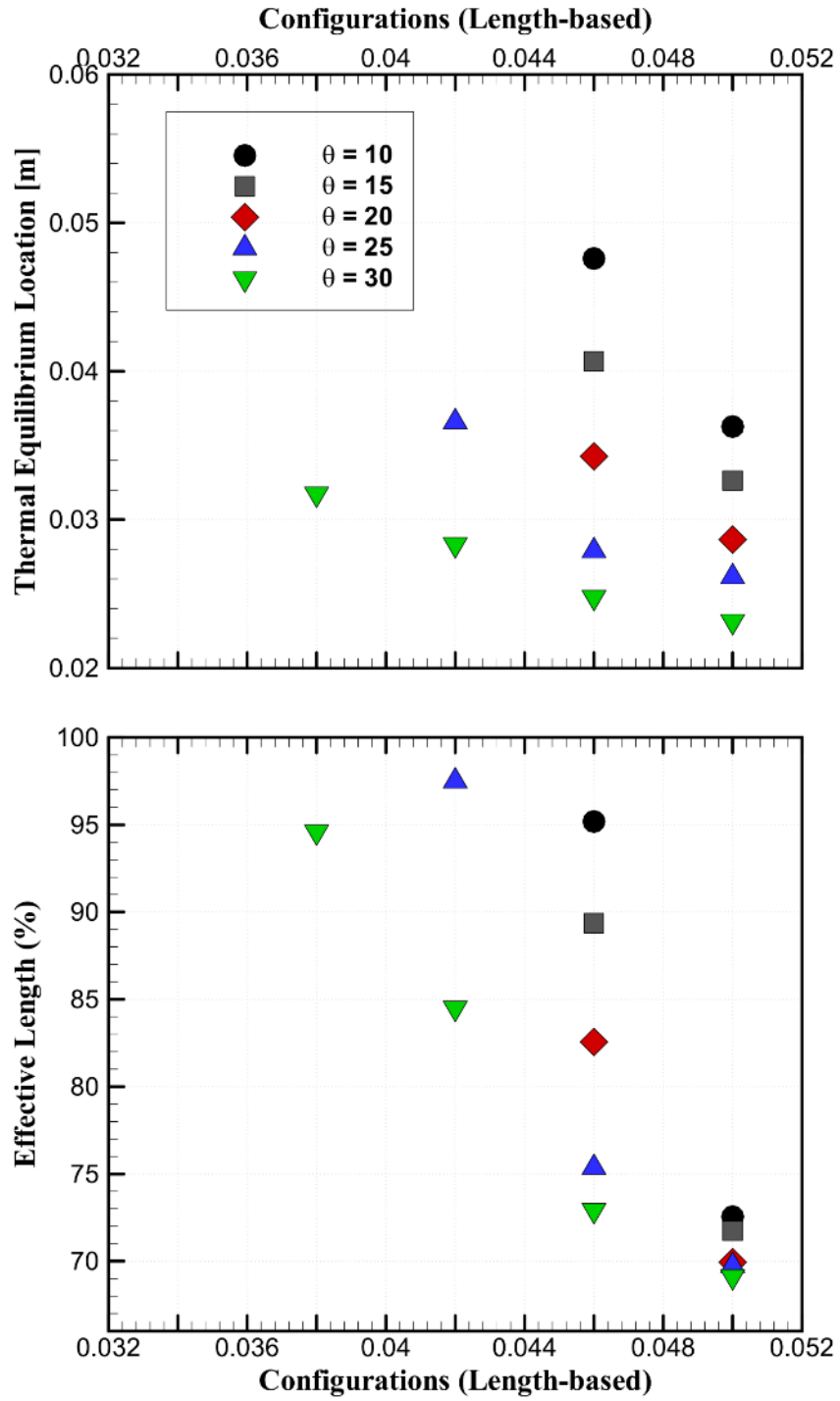


Fig. 5.32: Thermal equilibrium location corresponds to length of receiver and effective length for achieving thermal equilibrium for configuration 26 – 50.

Since the shape of the volumetric receiver is cone frustum, therefore, temperature is not same throughout the cross-sectional area. Thus, effective volume or effective axial area (used for attaining thermal equilibrium) is an important parameter to evaluate while dealing with Constructal approach. Axial area is considered using mid-plane which is parallel to flow path that can also represent the effective volume of configurations. Near the wall, the temperature of the fluid and the solid phases reaches thermal equilibrium earlier than the centerline of receiver (Figs. 5.31 and 5.32) which indicates the effective shape of the receiver for outlet. In Fig. 5.33, effective area (A_{eff}) for the proposed configurations are presented in which the effective area is considerably less than effective length due to non-uniform distribution of temperature at the cross-sectional area. Also, shorter length can not give 100% effectiveness for heat recovery, optimizing the inlet and outlet shape of receiver is equally important. Ratio-based configurations follow the increasing trend for increasing cone angle and ratio while length-based configurations show the opposite trend. Patterns of effective area are not similar to the effective length for ratio-based configurations because convection heat recovery is greater close to wall for smaller cone angle and ratio.

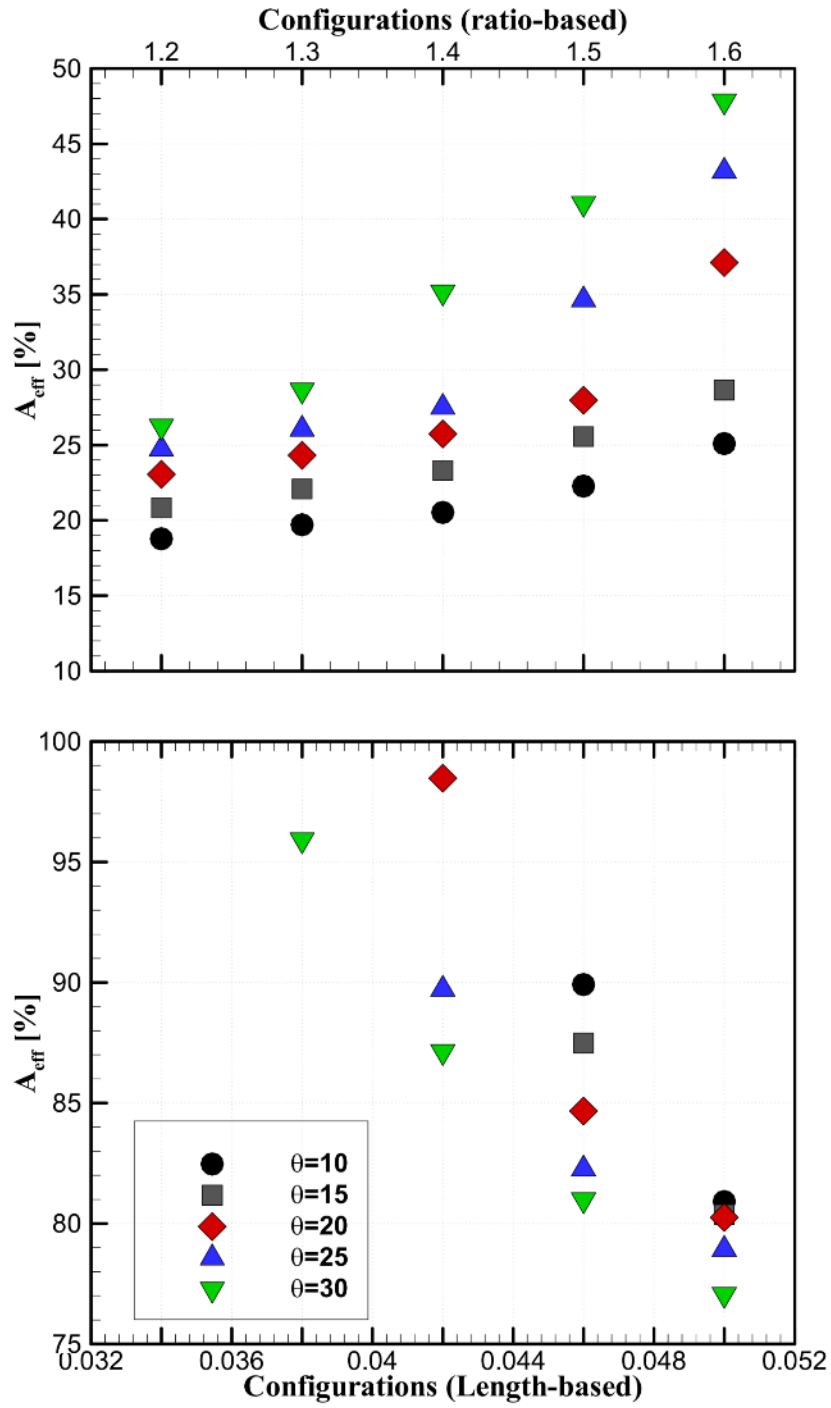


Fig. 5.33: Effective area (A_{eff}) for achieving thermal equilibrium for all proposed configurations.

Higher exit temperature is fundamental to obtain enhanced thermal performance because thermal efficiency directly depends on it. Greater the exit temperature results in higher

thermal efficiency. Figure 5.34 presents the mass-averaged exit temperature (T_{eff}) for all proposed configurations. Highest exit temperature is attained by ratio-based configurations. Larger ratios and cone angles exhibited higher exit temperatures. Moving towards smaller ratios and cone angles, a dramatic fall of exit temperature is observed. In the case of length-based configurations, shorter lengths and larger cone angles show higher exit temperature, and decrement in exit temperature is noted while moving towards longer length and smaller cone angle. An increment in exit temperature is more profound while shortening the length for smaller cone angles. Similarly, smaller cone angles significantly increase the exit temperature while increasing the ratio.

In Fig. 5.35, maximum temperature (T_{max}) is displayed for all proposed configurations. If value of T_{max} within domain exceeds the T_{exit} , then this exceeded temperature is undesirable or can affect the system's thermal performance. Ideal thermal system management avoids this undesirable temperature. It can be expressed as below.

$$\eta = \frac{\dot{m}c_{p,f}(T_{\text{exit}} - T_{\text{in}})}{q_{\text{rad}}} \quad (50)$$

$$\Omega = \frac{T_{\text{max}} - T_{\text{exit}}}{T_{\text{in}}} \quad (51)$$

$$\frac{\dot{m}c_{p,f}(T_{\text{max}} - T_{\text{exit}})}{q_{\text{rad}}} = \Omega \cdot \eta_{(T_{\text{in}}=0)} - \eta \quad (52)$$

Where, η represent thermal efficiency and Ω is a parameter that relates T_{max} to T_{exit} . It can be remarked that if T_{max} becomes greater than T_{exit} , then obtained efficiency deviates

from ideal efficiency. In proposed configurations, $T_{\max}$ remains approximately equal to T_{exit} which indicates no loss of thermal efficiency.

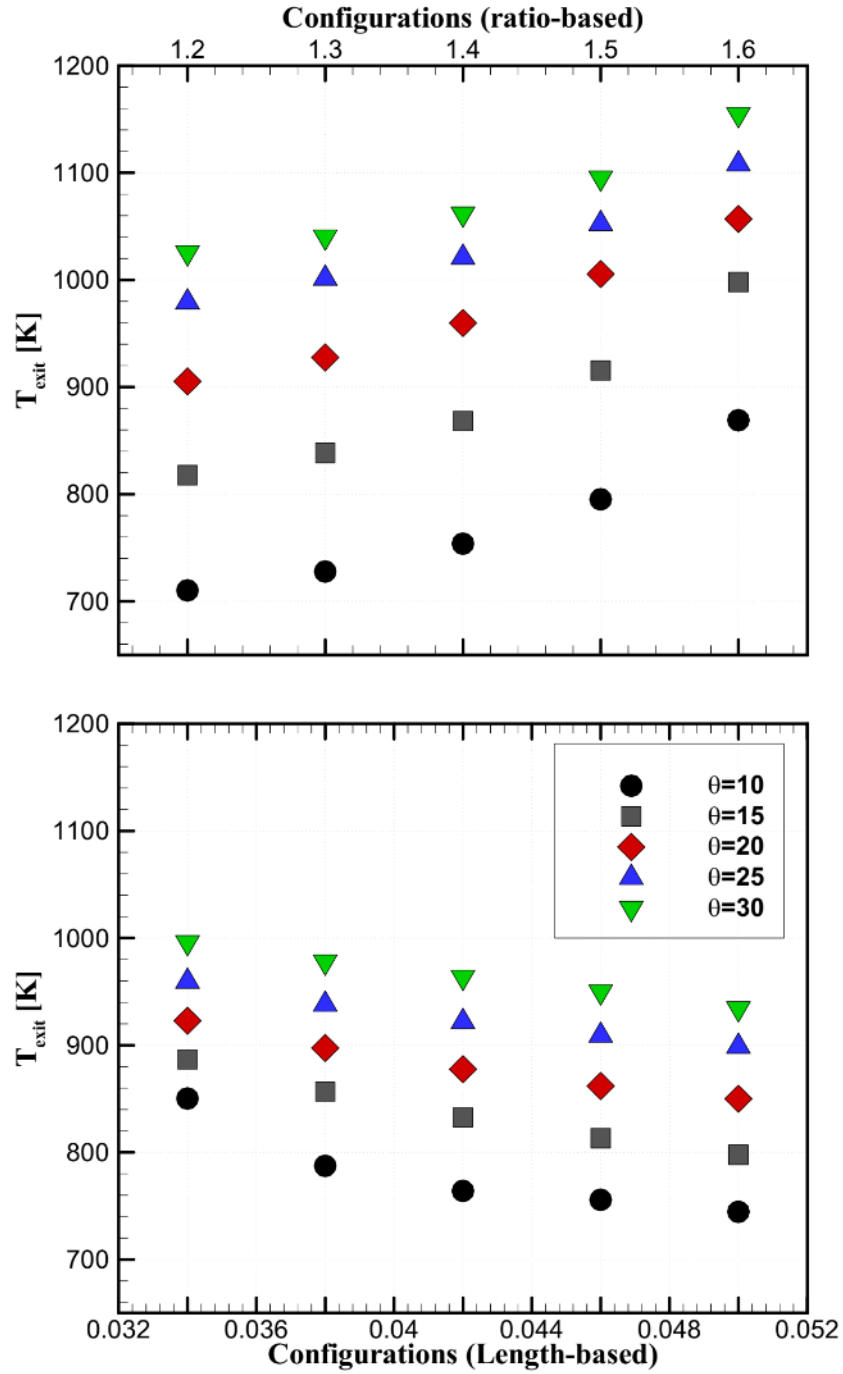


Fig. 5.34: Exit temperature (T_{exit}) taken as mass-averaged at air outlet of receiver for all proposed configurations.

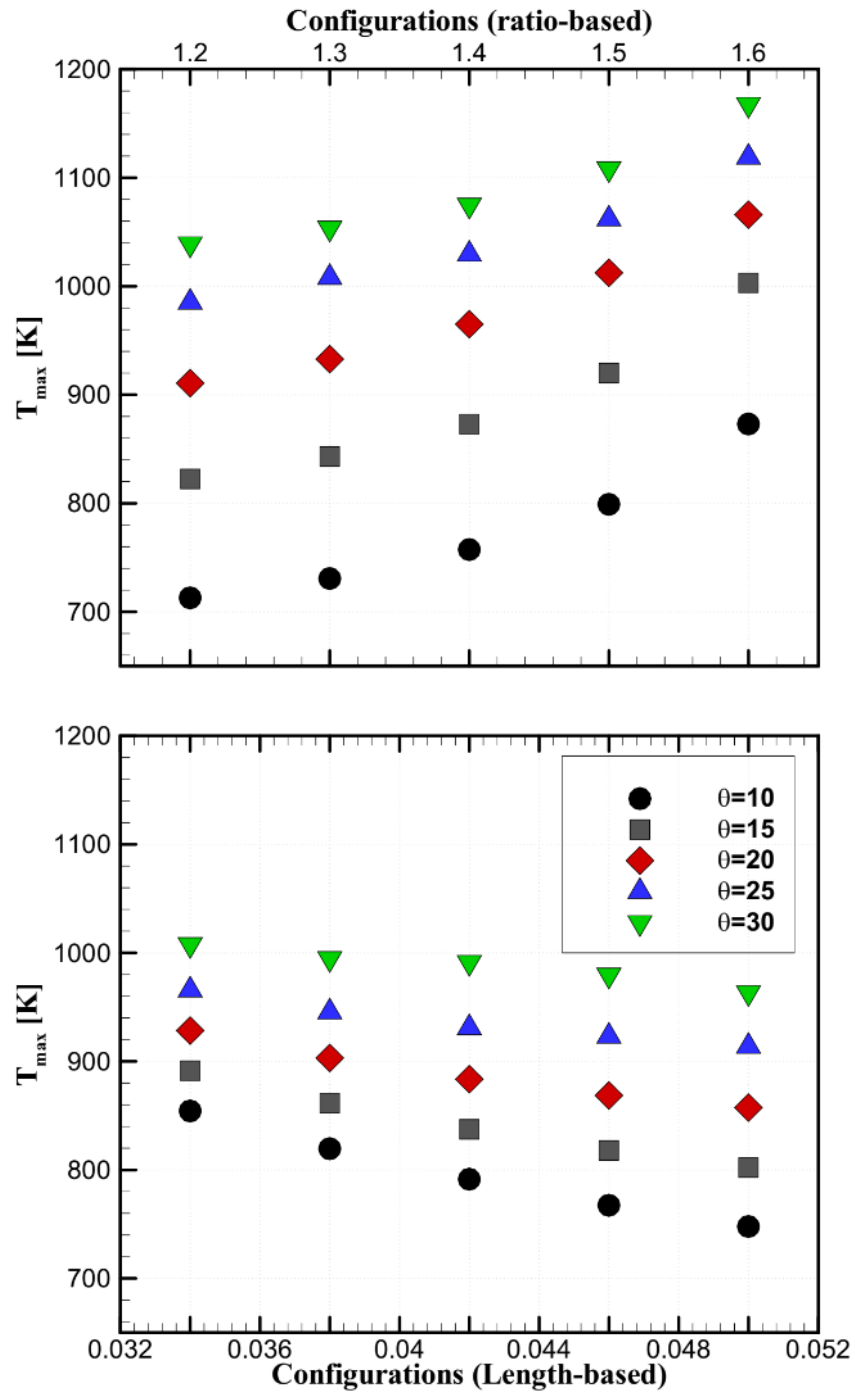


Fig. 5.35: Maximum temperature ($T_{\max}$) achieved within numerical domain for all proposed configurations.

While keeping volume as a constraint, only effective area are not complete to examine the optimized configuration. Consequently, effective area is plotted correspondingly to exit temperature in order to identify effective and enhanced volumetric receiver design for all proposed configurations (as shown in Fig. 5.36). In ratio-based configurations, for $\theta = 10$, increasing ratio drastically improved the exit temperature while no significant enhancement is found in effective area. In addition, as cone angle increases, trend of thermal enhancement is shifted towards higher effective area. For example, $\theta = 30$ shows dramatically increment in effective area while the exit temperature slightly increase. For length-based configurations, effective area increases among same cone angle while no significant improvement in exit temperature. Consequently, it concludes that configuration having $\theta = 30$ and $r = 1.2$ seems to perform better when the size of receiver tends to be lower with higher exit temperature. Similarly, the configuration having $\theta = 30$ and $L = 0.05\text{m}$ is reasonable among length-based configurations to achieve higher exit temperature and lower receiver size.

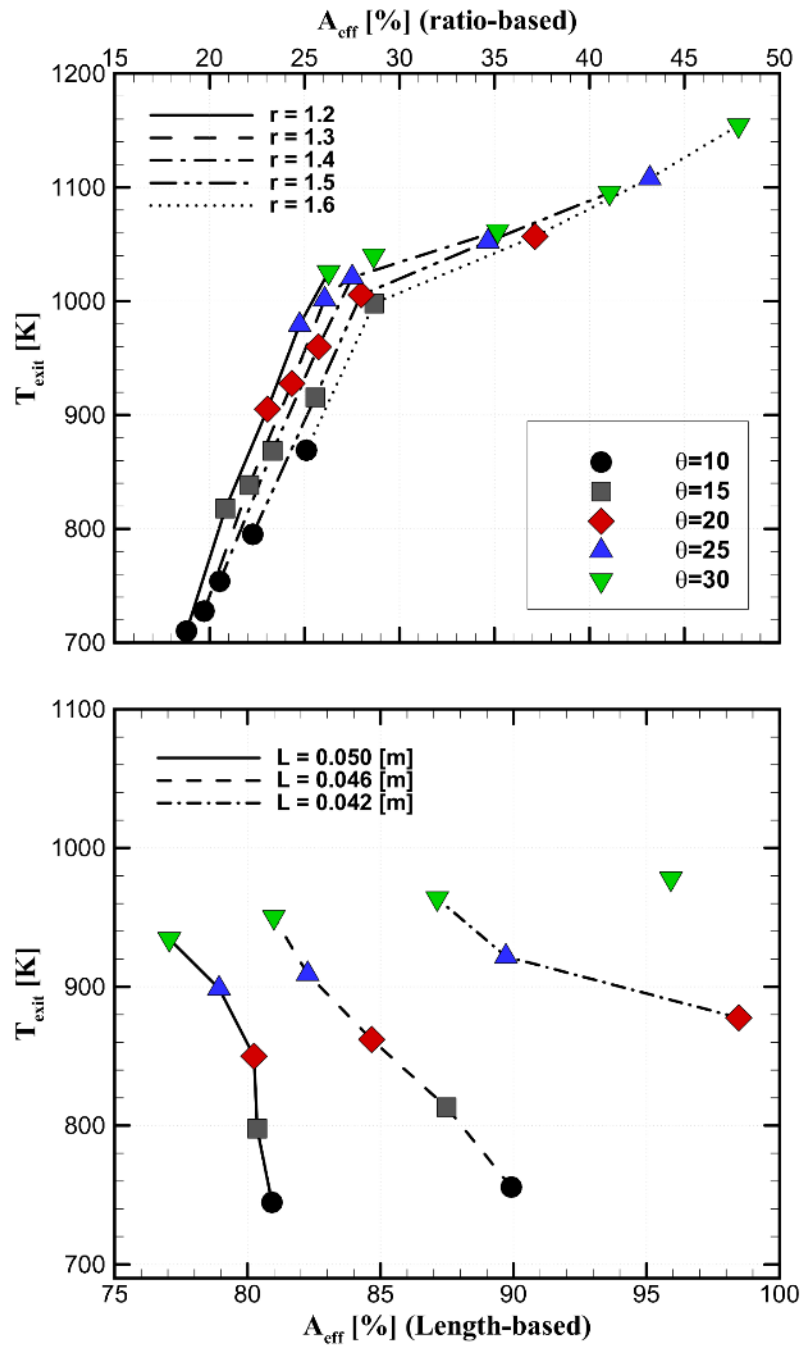


Fig. 5.36: Exit temperature (T_{exit}) corresponding to its effective area for all proposed configurations.

5.4.2. Flow Analysis

The hydrodynamic analysis is essential to examine with thermal performance because fluid pumping power requirement is directly related design of receiver which plays crucial role in fluid power loss due to pressure drop. Pores and converging shape of volumetric receiver leads to increased pressure drop occurs under drag as well as fluid friction within the pores and the flow channel. Pumping power is one of the challenges while employing a porous structure for harvesting solar energy. Figure 5.37 shows the pressure drop across the flow path for proposed configurations. In these cases, porosity and mass-flow rate are kept same, therefore higher pressure drop mainly associated with the cone angle and receiver length which creates complexity of flow field inside the porous media. Temperature-dependent properties of fluid (e.g., viscosity) can cause high rate of shear which leads to more turbulence behavior of fluid. However, the increase in receiver temperature seemed not to have a notable effect on the hydrodynamic performance. The higher temperature was achieved by larger cone angles of ratio-based configurations, yet, these configurations are associated with low pressure drop. Length-based configurations show more pressure drop with drastic increment for configurations having shortest length moving towards larger cone angle. Hence, the configurations with optimal thermal performance and receiver size hold acceptable pressure drop value compared the competitors.

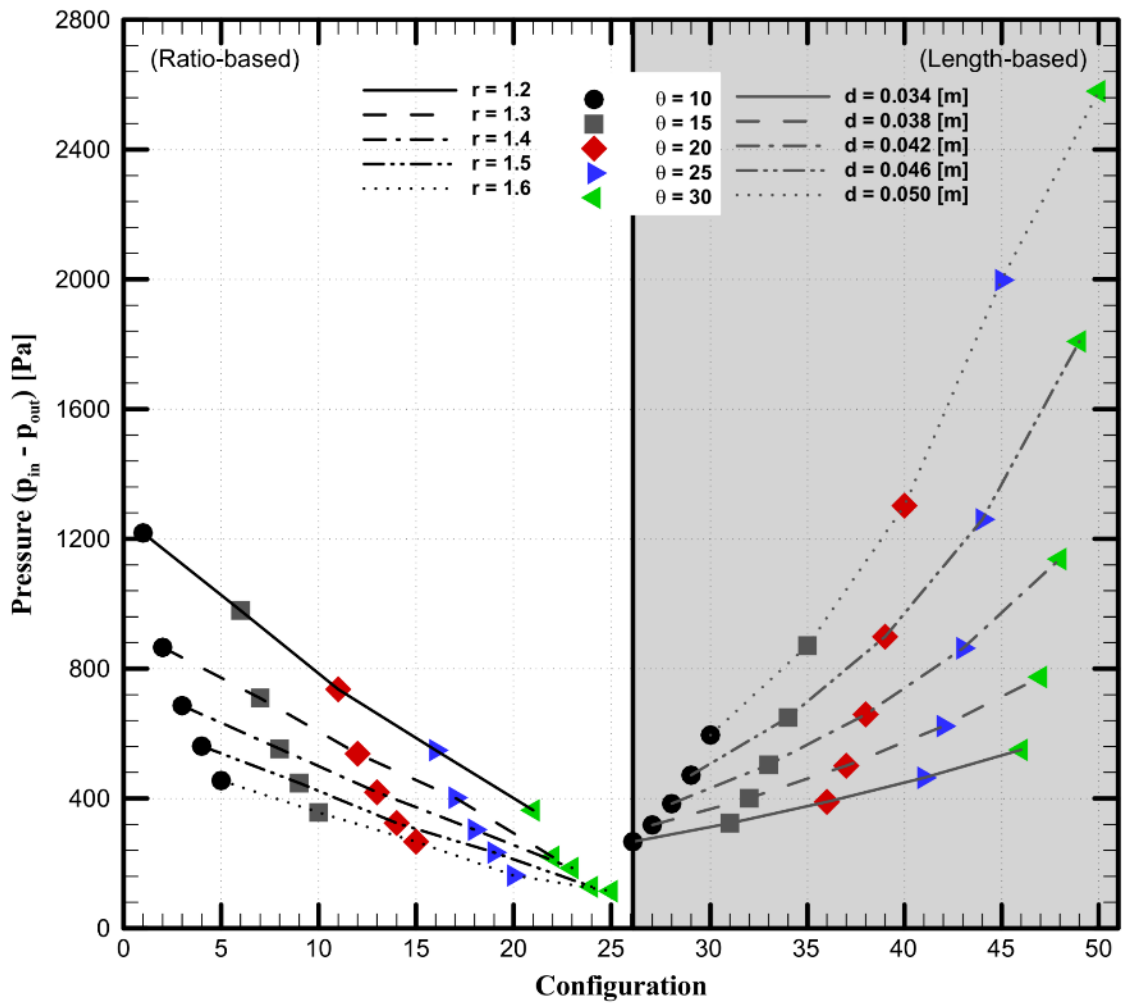


Fig. 5.37: Pressure drop across over the complete length or pressure difference inlet to outlet for all proposed configurations.

CHAPTER 6

CONCLUSION AND RECOMMENDATIONS

6. 1. Conclusions

Heat transfer enhancement and solar radiation absorption remain critical in volumetric solar receivers in terms of the thermal performance of the system. In this case, SiC foam having a high solar absorption coefficient and low thermal emissivity is considered as a solar volumetric receiver located in a flow channel of a square cross-section and optically transparent from the top boundary. The pore morphology of the foam is obtained through CT-scanning data and later transferred into the numerical computational domain for 3D simulations. The numerical study is performed towards analyzing the thermal and flow behaviors of the receiver for different foam porosities, solar concentrations, and various flow Reynolds numbers, including laminar and turbulent flow regimes. Air and water are considered as the heat transfer fluid for the volumetric receiver. The Discrete Ordinate radiation model is employed to compute the intensity of radiation due to scattering, reflection, and diffusion within the receiver. Furthermore, to obtain the best hydrodynamic and thermal performances, 50 different channel's configurations are also numerically examined. For square cross-sectional channel configuration, the thermal performance of the solar volumetric receiver is evaluated through the overall heat transfer coefficient and the normalized Nusselt number. Moreover, the normalized temperature based on the mass average fluid temperature and the normalized maximum temperature in the flow system is evaluated for various Reynolds numbers, solar concentrations, and foam porosities. The

model was validated with an experimental work of the same conditions. The findings reveal that predictions of temperature agreed well with its counterpart obtained experimentally.

For laminar flow where the Reynolds numbers are 50, 150, 300, and 500 at the inlet of the square cross-sectional channel while air is the working fluid, the following outcomes were obtained:

1. The maximum enhancement in heat transfer coefficient occurs for low values of foam porosity and solar concentration. Normalized Nusselt number is considerably influenced by Reynolds number; in which case, increasing Reynolds number results in a sharp reduction of normalized Nusselt number.
2. Low porosity of the solar receiver gives rise to high values of normalized Nusselt number and normalized temperature ratios, i.e. heat transfer rates improve and mass average fluid temperature increases.
3. Increasing solar concentration enhances the local temperature rise in the fluid; however, increasing Reynolds number reduces the normalized maximum temperature in the flow system.
4. Pressure drop is highly influenced by the Reynolds number; in which case, increasing Reynolds number increase pressure drop through the channel. However, reducing the porosity of foam lowers the pressure drop of the working fluid.
5. Solar concentration has a minor effect on the pressure drop; hence, increasing fluid temperature, under high solar concentration, enhances the viscosity of the working fluid which contribute to the pressure drop in the fluid.

For the case of turbulent flow, additional Reynolds stresses in governing equation are computed using the $k-\varepsilon$ turbulent model while convection, conduction, and radiation heat transfer modes in the channel are considered incorporating the energy equation and DO method. Channel geometry for turbulent flow also consists of square cross-sectional with the top boundary of optical transparent for radiation entrance. Moreover, water as a heat transfer fluid is also included in turbulent flow thermal and hydrodynamic assessment.

1. Water as a heat transfer fluid in this thermal system shows a decreasing trend for overall normalized Nusselt number, and thermal efficiencies.
2. Air as a heat transfer fluid demonstrates a negligible effect on the heat flux with increasing Reynolds numbers. However, the normalized Nusselt number and thermal efficiencies increased at high Reynolds numbers.
3. In the case of air, the highest values of normalized Nusselt number are achieved at low porosity, low solar concentration, and low Reynolds number while thermal efficiencies exhibit opposite behavior.
4. In the case of water, the highest value of normalized Nusselt number is obtained at low Reynolds numbers and high porosities, but at low solar concentration. The thermal efficiencies are enhanced at high porosities and high solar concentrations.
5. Temperature variation within the porous structure changes with the Reynolds numbers and porosity, which is considerably true for the air.

In case of varying the channel's configuration, the fluid flow is turbulent and computed using the $k-\varepsilon$ /STM model while temperature profiles are predicted with energy equation and LTNE. Similarly, solar radiation modeling is dealt with using the Discrete Ordinate

radiation model with solar radiation tracing (scattering, absorption, emission, and diffusion). A porous zone is designed by permeability coefficient and inertial resistance. Mass-flow rate of 0.003221 kg/s for air, porosity of 0.8 for the SiC, solar concentrated radiation of 600 kW/m², and volume of the receiver are kept as a constraint to evaluate the influence of various configurations of volumetric receivers. In this regard, two configuration approaches of cone frustum shaped receiver (Fig. 3.3) are adopted including 25 configurations which were constructed by varying the cone angle, ratio of inlet/outlet diameters, and other 25 configurations possess variation of receiver's length using different cone angle. The optimal designs among these configurations are chosen based on thermal performance, size of the receiver, and hydrodynamic performance. Thermal performance is assessed using exit temperature while effective area corresponds to the size of receiver and pressure drop is defined for hydrodynamic performance. Following conclusions were obtained from the findings of the numerical solution:

1. The ratio-based approach for developing optimum design performs better than the length-based approach in all studies.
2. Absorption of radiation heat flux increased the temperature of the solid and fluid phases which contributes to the heating effect. Lower heat absorption is observed near to the channel walls due to the no-slip boundary and volumetric effect.
3. The converging shape of the receiver cause increment in temperature of both phases in the flow direction compared with the reference design.
4. The highest exit temperature (thermal efficiency) is found while using larger cone angles and higher ratio for ratio-based configurations. In the case of length-based

consideration, higher exit temperature was obtained when the cone angle is maximized while the length is minimized.

5. Effective length and effective area are used for optimum size of the receiver with exit temperature which identifies the higher ratio and smaller cone angle.
6. Pressure drop results also support the configurations which was found optimum according to the effective area and exit temperature.

6. 2. Recommendations and future considerations

Every technology suffered various challenges to place itself in the commercial market or better solution for fulfilling the specific gap in human society. The optimization of such systems proves them the best candidate for this specific gap. Similarly, solar volumetric receivers working on the principle of CSP and one of a green, renewable, and sustainable system of power generation, confronted many obstacles and rapidly claiming its worth to be viable among competitors. Despite, this incredible progress in the field of the solar volumetric receiver, few gaps still need attention for urgent solutions from research point of view. Most of these challenges are not just associated with the volumetric solar receiver, yet they are also related to other CSP systems especially tubular receivers, small particle receivers and falling-film receivers. Following areas require further attention and can be taken as future work to be investigated:

- 1) The study of absorption property of volumetric receivers, for both numerically and experimentally, was rarely investigated. Absorption quality plays a role to mitigate radiation losses. This can be extended to include other optical properties such as scattering phase function.

- 2) As most of the results revealed that the outlet temperature of receivers is between 800 - 1200°C, which significantly disturbs the durability of systems. Less focus has been paid towards the analysis of the lifespan of solar receiver systems which is one of the most crucial assessments for market evolution of these devices.
- 3) The turbulent streams of fluids and the complex shape of the receivers promote thermal instability, through creating a considerable difference between the maximum and the outlet temperatures. This undesired difference is not only excess heat loading, but it also reduces the thermal efficiency. Thus, optimization of the pore design and input conditions of working fluids should be more considered by maintaining the value of maximum temperature closer to outlet temperature.
- 4) The volumetric solar systems were claimed to be environmentally friendly. Fabrication of complex porous receiver is associated with higher cost and expensive technology. So, its manufacturing should be eco-friendly and free from brown processes.
- 5) Studies on influence of coatings, protection films or paints on receivers' optical properties and durability are not explored in detail. Materials used in coatings and thickness of these paints should be optimized in order to improve the overall performance of system.
- 6) Reusability and regeneration of receivers are equally important to their ecofriendly manufacturing process. Small losses can become a source of complete shutdown of the system. Therefore, receivers should be manufactured in such a way that small damages can be replaceable and also capable of regeneration.

Although, these solar harvesting technologies with CSP approach and volumetric receiver are encouraging and reliable to adopt on large scale. Seeking more advantages of the solar volumetric receiver are infancy and still challenging. This thesis provided a brief overview of advancement in the field of the solar receiver and ended with new findings by investigating various input and design parameters. Additionally, future perspective of this study predicts outstanding enhancement.

REFERENCE

- [1] M. Ali, M. Rady, M.A.A. Attia, E.M.M. Ewais, Consistent coupled optical and thermal analysis of volumetric solar receivers with honeycomb absorbers, *Renew. Energy*. 145 (2020) 1849–1861. <https://doi.org/10.1016/j.renene.2019.07.082>.
- [2] S. Du, T. Xia, Y.-L. He, Z.-Y. Li, D. Li, X.-Q. Xie, Experiment and optimization study on the radial graded porous volumetric solar receiver matching non-uniform solar flux distribution, *Appl. Energy*. 275 (2020) 115343. <https://doi.org/10.1016/j.apenergy.2020.115343>.
- [3] O. Singh, K. Bhatwadekar, N.G. Kartheek, S.B. Kedare, S. Singh, A Novel Design of Liquid Volumetric Plated Cavity Receiver for Central Tower Systems, *J. Sol. Energy Eng.* 143 (2020). <https://doi.org/10.1115/1.4047669>.
- [4] A. Raza, J.-Y. Lu, S. Alzaim, H. Li, T. Zhang, Novel Receiver-Enhanced Solar Vapor Generation: Review and Perspectives, *Energies* . 11 (2018). <https://doi.org/10.3390/en11010253>.
- [5] G. Domingues, A.M. Monthe, S. Guévelou, B. Rousseau, Study by molecular dynamics of the influence of temperature and pressure on the optical properties of undoped 3C-SiC structures, *J. Quant. Spectrosc. Radiat. Transf.* 205 (2018) 220–229. <https://doi.org/https://doi.org/10.1016/j.jqsrt.2017.10.018>.
- [6] M. Podowski, C. Break, O. Remarks, S. Carbon, D. Brayton, D. Priorities, C. Solar, P. Systems, T.P. Plant, F. Fuel, H. Sources, M. Podowski, M. Podowski, P. Pickard, A. Moiseyev, P. Title, M. Podowski, N. Alpy, P. Hejzlar, T. Gallaway, C. Turchi,

- D. Chapman, A. Solar, K. Kimball, M. Mcdowell, Supercritical CO₂ Power Cycle Symposium, (2009).
- [7] G. Glatzmaier, C. Turchi, Supercritical CO₂ as a Heat Transfer and Power Cycle Fluid for CSP Systems, 2009. <https://doi.org/10.1115/ES2009-90332>.
- [8] AALBORD CSP, History of CSP, (n.d.). <https://www.aalborgcsp.com/business-areas/solar-district-heating/csp-parabolic-troughs/history-of-csp/> (accessed September 25, 2021).
- [9] U.S. Department of Energy, The history of solar, 2011. https://www1.eere.energy.gov/solar/pdfs/solar_timeline.pdf.
- [10] K. Lovergrove, W. Stein, Concentrating Solar Power Technology: Principles, Developments and Applications, Woodheas Publising Limited, Phiiladelphia, USA, 2012. <https://doi.org/10.1016/B978-1-84569-769-3.50020-0>.
- [11] World Bank, CONCENTRATING SOLAR POWER CLEAN POWER ON DEMAND 24/7, 2011. www.worldbank.org (accessed September 25, 2021).
- [12] J.F.P. Pitot de la Beaujardiere, H.C.R. Reuter, A review of performance modelling studies associated with open volumetric receiver CSP plant technology, *Renew. Sustain. Energy Rev.* 82 (2018) 3848–3862. <https://doi.org/10.1016/j.rser.2017.10.086>.
- [13] G. Domingues, A.M. Monthe, S. Guévelou, B. Rousseau, Study by molecular dynamics of the influence of temperature and pressure on the optical properties of undoped 3C-SiC structures, *J. Quant. Spectrosc. Radiat. Transf.* 205 (2018) 220–229. <https://doi.org/10.1016/j.jqsrt.2017.10.018>.

- [14] S. Erdle, The DESERTEC initiative: powering the development perspectives of Southern Mediterranean countries?, Deutsches Institut für Entwicklungspolitik (DIE), Bonn, 2010. <http://hdl.handle.net/10419/199347>.
- [15] World Energy Outlook – Topics - IEA, (n.d.). <https://www.iea.org/topics/world-energy-outlook> (accessed May 18, 2021).
- [16] J. Kauffman, K.M. Lee, Handbook of Sustainable Engineering, Springer Netherlands, 2013. <https://doi.org/10.1007/978-1-4020-8939-8>.
- [17] G.A. Heath, J.J. Burkhardt III, Meta-Analysis of Estimates of Life Cycle Greenhouse Gas Emissions from Concentrating Solar Power Preprint (Report number: DE-AC36-08GO28308), Pix. 16560 (2011). <http://www.osti.gov/bridge>.
- [18] V. Bhalla, H. Tyagi, Parameters influencing the performance of nanoparticles-laden fluid-based solar thermal collectors: A review on optical properties, Renew. Sustain. Energy Rev. 84 (2018) 12–42. <https://doi.org/10.1016/j.rser.2017.12.007>.
- [19] A.L. Ávila-Marín, Volumetric receivers in Solar Thermal Power Plants with Central Receiver System technology: A review, Sol. Energy. 85 (2011) 891–910. <https://doi.org/10.1016/j.solener.2011.02.002>.
- [20] K. Hennecke, P. Schwarzbözl, B. Hoffschmidt, J. Götsche, G. Koll, M. Beuter, T. Hartz, The solar power tower jülich - A solar thermal power plant for test and demonstration of air receiver technology, ISES Sol. World Congr. 2007, ISES 2007. 3 (2007) 1749–1753. https://doi.org/10.1007/978-3-540-75997-3_358.
- [21] German Aerospace Center, DLR - Institute of Solar Research - CIEMAT Plataforma Solar de Almería - Europe's biggest Test Center for Concentrating Solar Power

- (CSP), (n.d.). https://www.dlr.de/sf/en/desktopdefault.aspx/tabid-7176/11942_read-28189/ (accessed September 12, 2021).
- [22] M. Faizan, S.Z. Shuja, B.S. Yilbas, M. Khan, H. Al-Qahtani, Flow analysis of a rectangular channel with triangular and semi-spherical protrusions, *Int. J. Therm. Sci.* 162 (2021) 106793. <https://doi.org/https://doi.org/10.1016/j.ijthermalsci.2020.106793>.
- [23] Z. Wu, C. Caliot, G. Flamant, Z. Wang, Coupled radiation and flow modeling in ceramic foam volumetric solar air receivers, *Sol. Energy.* 85 (2011) 2374–2385. <https://doi.org/10.1016/j.solener.2011.06.030>.
- [24] T. Fend, B. Hoffschmidt, R. Pitz-Paal, O. Reutter, P. Rietbrock, Porous materials as open volumetric solar receivers: Experimental determination of thermophysical and heat transfer properties, *Energy.* 29 (2004) 823–833. [https://doi.org/10.1016/S0360-5442\(03\)00188-9](https://doi.org/10.1016/S0360-5442(03)00188-9).
- [25] J.-F.P. Pitot de la Beaujardiere, H.C.R. Reuter, A review of performance modelling studies associated with open volumetric receiver CSP plant technology, *Renew. Sustain. Energy Rev.* 82 (2018) 3848–3862. <https://doi.org/https://doi.org/10.1016/j.rser.2017.10.086>.
- [26] R. Pitz-Paal, B. Hoffschmidt, M. Böhmer, M. Becker, Experimental and numerical evaluation of the performance and flow stability of different types of open volumetric absorbers under non-homogeneous irradiation, *Sol. Energy.* 60 (1997) 135–150. [https://doi.org/https://doi.org/10.1016/S0038-092X\(97\)00007-8](https://doi.org/https://doi.org/10.1016/S0038-092X(97)00007-8).
- [27] C. Albanakis, D. Missirlis, N. Michailidis, K. Yakinthos, A. Goulas, H. Omar, D.

- Tsipas, B. Granier, Experimental analysis of the pressure drop and heat transfer through metal foams used as volumetric receivers under concentrated solar radiation, *Exp. Therm. Fluid Sci.* 33 (2009) 246–252.
<https://doi.org/https://doi.org/10.1016/j.expthermflusci.2008.08.007>.
- [28] A.D. Rao, Combined Cycle Systems for Near-Zero Emission Power Generation, Woodhead Publishing, 2012.
<https://www.sciencedirect.com/book/9780857090133/combined-cycle-systems-for-near-zero-emission-power-generation>.
- [29] C.K. Ho, B.D. Iverson, Review of high-temperature central receiver designs for concentrating solar power, *Renew. Sustain. Energy Rev.* 29 (2014) 835–846.
<https://doi.org/10.1016/j.rser.2013.08.099>.
- [30] C.C. Agrafiotis, I. Mavroidis, A.G. Konstandopoulos, B. Hoffschmidt, P. Stobbe, M. Romero, V. Fernandez-Quero, Evaluation of porous silicon carbide monolithic honeycombs as volumetric receivers/collectors of concentrated solar radiation, *Sol. Energy Mater. Sol. Cells.* 91 (2007) 474–488.
<https://doi.org/10.1016/j.solmat.2006.10.021>.
- [31] V. Bhalla, H. Tyagi, Parameters influencing the performance of nanoparticles-laden fluid-based solar thermal collectors: A review on optical properties, *Renew. Sustain. Energy Rev.* 84 (2018) 12–42.
<https://doi.org/https://doi.org/10.1016/j.rser.2017.12.007>.
- [32] V.R. Patil, F. Kiener, A. Grylka, A. Steinfeld, Experimental testing of a solar air cavity-receiver with reticulated porous ceramic absorbers for thermal processing at

- above 1000 °C, Sol. Energy. 214 (2021) 72–85.
<https://doi.org/https://doi.org/10.1016/j.solener.2020.11.045>.
- [33] G. Barreto, P. Canhoto, M. Collares-Pereira, Three-dimensional modelling and analysis of solar radiation absorption in porous volumetric receivers, Appl. Energy. 215 (2018) 602–614. <https://doi.org/10.1016/j.apenergy.2018.02.065>.
- [34] G. Barreto, P. Canhoto, M. Collares-Pereira, Three-dimensional CFD modelling and thermal performance analysis of porous volumetric receivers coupled to solar concentration systems, Appl. Energy. 252 (2019) 113433. <https://doi.org/10.1016/j.apenergy.2019.113433>.
- [35] O.K. Siddiqui, B.S. Yilbas, Solar absorption heating in horizontal channel: Influence of absorbing plate location on thermal performance, Energy Convers. Manag. 74 (2013) 140–148. <https://doi.org/10.1016/j.enconman.2013.05.009>.
- [36] S. Du, Q. Ren, Y.-L. He, Optical and radiative properties analysis and optimization study of the gradually-varied volumetric solar receiver, Appl. Energy. 207 (2017) 27–35. <https://doi.org/https://doi.org/10.1016/j.apenergy.2017.05.165>.
- [37] M. Sedighi, R.V. Padilla, R.A. Taylor, M. Lake, I. Izadgoshasb, A. Rose, High-temperature, point-focus, pressurised gas-phase solar receivers: A comprehensive review, Energy Convers. Manag. 185 (2019) 678–717. <https://doi.org/10.1016/j.enconman.2019.02.020>.
- [38] F. Gomez-Garcia, J. González-Aguilar, G. Olalde, M. Romero, Thermal and hydrodynamic behavior of ceramic volumetric absorbers for central receiver solar power plants: A review, Renew. Sustain. Energy Rev. 57 (2016) 648–658.

<https://doi.org/https://doi.org/10.1016/j.rser.2015.12.106>.

- [39] R. Capuano, T. Fend, P. Schwarzbözl, O. Smirnova, H. Stadler, B. Hoffschmidt, R. Pitz-Paal, Numerical models of advanced ceramic absorbers for volumetric solar receivers, *Renew. Sustain. Energy Rev.* 58 (2016) 656–665. <https://doi.org/https://doi.org/10.1016/j.rser.2015.12.068>.
- [40] F. Gomez-Garcia, J. González-Aguilar, G. Olalde, M. Romero, Thermal and hydrodynamic behavior of ceramic volumetric absorbers for central receiver solar power plants: A review, *Renew. Sustain. Energy Rev.* 57 (2016) 648–658. <https://doi.org/10.1016/j.rser.2015.12.106>.
- [41] B.S. Yilbas, S.Z. Shuja, A Solar Volumetric Receiver: Influence of Absorbing Cells Configuration on Device Thermal Performance, *Int. J. Thermophys.* 38 (2017). <https://doi.org/10.1007/s10765-016-2132-3>.
- [42] A. Kribus, H. Ries, W. Spirk, Inherent Limitations of Volumetric Solar Receivers, *J. Sol. Energy Eng. Asme - J SOL ENERGY ENG.* 118 (1996) 151–155. <https://doi.org/10.1115/1.2870891>.
- [43] B. Hoffschmidt, F. Tellez, A. Valverde, J. Fernández, V. Fernández, Performance Evaluation of the 200kW[sub th] HiTRec-II Open Volumetric Air Receiver, *J. Sol. Energy Eng. Asme - J SOL ENERGY ENG.* 125 (2003). <https://doi.org/10.1115/1.1530627>.
- [44] L. Antonio Marin, Volumetric Receivers in Solar Thermal Power Plants with Central Receiver System Technology: A Review, *Sol. Energy - Sol. ENERG.* 85 (2011) 891–910. <https://doi.org/10.1016/j.solener.2011.02.002>.

- [45] B. Hoffschmidt, S. Alexopoulos, C. Rau, J. Sattler, A. Anthrakidis, C. Boura, B. O'Connor, P. Hilger, 3.18 - Concentrating Solar Power, in: A.B.T.-C.R.E. Sayigh (Ed.), Elsevier, Oxford, 2012: pp. 595–636. <https://doi.org/10.1016/B978-0-08-087872-0.00319-X>.
- [46] Y.-L. He, K. Wang, Y. Qiu, B.-C. Du, Q. Liang, S. Du, Review of the solar flux distribution in concentrated solar power: Non-uniform features, challenges, and solutions, *Appl. Therm. Eng.* 149 (2019) 448–474. <https://doi.org/10.1016/j.applthermaleng.2018.12.006>.
- [47] S. Zunft, M. Hänel, M. Krüger, V. Dreißigacker, F. Göhring, E. Wahl, Jülich Solar Power Tower—Experimental Evaluation of the Storage Subsystem and Performance Calculation, *J. Sol. Energy Eng.* 133 (2011). <https://doi.org/10.1115/1.4004358>.
- [48] J.M. Chavez, C. Chaza, Testing of a porous ceramic absorber for a volumetric air receiver, *Sol. Energy Mater.* 24 (1991) 172–181. [https://doi.org/https://doi.org/10.1016/0165-1633\(91\)90057-R](https://doi.org/https://doi.org/10.1016/0165-1633(91)90057-R).
- [49] B. Variot, T. Menigault, G. Flamant, Modelling and optimization of a two-slab selective volumetric solar receiver, *Sol. Energy.* 53 (1994) 359–368. [https://doi.org/https://doi.org/10.1016/0038-092X\(94\)90039-6](https://doi.org/https://doi.org/10.1016/0038-092X(94)90039-6).
- [50] T. Menigault, G. Flamant, B. Rivoire, Advanced high-temperature two-slab selective volumetric receiver, *Sol. Energy Mater.* 24 (1991) 192–203. [https://doi.org/https://doi.org/10.1016/0165-1633\(91\)90059-T](https://doi.org/https://doi.org/10.1016/0165-1633(91)90059-T).
- [51] R. Pitz-Paal, J. Morhenne, M. Fiebig, A new concept of a selective solar receiver for

- high temperature applications, *Sol. Energy Mater.* 24 (1991) 293–306.
[https://doi.org/https://doi.org/10.1016/0165-1633\(91\)90070-2](https://doi.org/https://doi.org/10.1016/0165-1633(91)90070-2).
- [52] M.J. Marcos, M. Romero, S. Palero, Analysis of air return alternatives for CRS-type open volumetric receiver, *Energy*. 29 (2004) 677–686.
[https://doi.org/https://doi.org/10.1016/S0360-5442\(03\)00176-2](https://doi.org/https://doi.org/10.1016/S0360-5442(03)00176-2).
- [53] A. Godini, S. Kheradmand, Optimization of volumetric solar receiver geometry and porous media specifications, *Renew. Energy*. 172 (2021) 574–581.
<https://doi.org/10.1016/j.renene.2021.03.040>.
- [54] S. Du, M.-J. Li, Y.-L. He, S. Shen, Conceptual design of porous volumetric solar receiver using molten salt as heat transfer fluid, *Appl. Energy*. 301 (2021) 117400.
<https://doi.org/10.1016/j.apenergy.2021.117400>.
- [55] J. Karni, A. Kribus, R. Rubin, P. Doron, The “Porcupine”: A Novel High-Flux Absorber for Volumetric Solar Receivers, *J. Sol. Energy Eng.* 120 (1998) 85–95.
<https://doi.org/10.1115/1.2888060>.
- [56] M. Abdelrahman, P. Fumeaux, P. Suter, Study of solid-gas-suspensions used for direct absorption of concentrated solar radiation, *Sol. Energy*. 22 (1979) 45–48.
[https://doi.org/https://doi.org/10.1016/0038-092X\(79\)90058-6](https://doi.org/https://doi.org/10.1016/0038-092X(79)90058-6).
- [57] A.J. Hunt, Small Particle Heat Exchangers, Lawrence Berkeley Natl. Lab. Pap. LBL-7841. (1978) 1–26. <http://escholarship.org/uc/item/65k6n507>.
- [58] F. Miller, R. Koenigsdorff, Theoretical analysis of a high-temperature small-particle solar receiver, *Sol. Energy Mater.* 24 (1991) 210–221.
[https://doi.org/https://doi.org/10.1016/0165-1633\(91\)90061-O](https://doi.org/https://doi.org/10.1016/0165-1633(91)90061-O).

- [59] M. FJ, K. RW, Thermal modeling of a small-particle solar central receiver, *J. Sol. Energy Eng. ASME*. 122 (2000) 23–29. <https://doi.org/10.1115/1.556277>.
- [60] P.K. Falcone, J.E. Noring, J.M. Hruby, Assessment of a solid particle receiver for a high temperature solar central receiver system, *Energy Procedia*. 49 (1985) 398–407.
- [61] C.K. Ho, B.D. Iverson, Review of high-temperature central receiver designs for concentrating solar power, *Renew. Sustain. Energy Rev.* 29 (2014) 835–846. <https://doi.org/10.1016/j.rser.2013.08.099>.
- [62] C.K. Ho, Advances in central receivers for concentrating solar applications, *Sol. Energy*. 152 (2017) 38–56. <https://doi.org/10.1016/j.solener.2017.03.048>.
- [63] C. Giovanni, Concept Design of a Heat Pipe Solar Receiver Gas Turbine, 2 (1978).
- [64] L. Amsbeck, R. Buck, P. Heller, J. Jedamski, R. Uhlig, Development of a tube receiver for a solar-hybrid microturbine system, *Proc. 14th SolarPACES Conf. Las Vegas, NV, March*. (2008) 4–7.
- [65] P. Heller, M. Pfänder, T. Denk, F. Tellez, A. Valverde, J. Fernandez, A. Ring, Test and evaluation of a solar powered gas turbine system, *Sol. Energy*. 80 (2006) 1225–1230. <https://doi.org/https://doi.org/10.1016/j.solener.2005.04.020>.
- [66] A. Gide, Heat Pipe Test Plan Transmittal, *Angew. Chemie Int. Ed.* 6(11), 951–952. (1967) 5–24.
- [67] L. Amsbeck, T. Denk, M. Ebert, C. Gertig, P. Heller, P. Herrmann, J. Jedamski, J. John, R. Pitz-Paal, T. Prosinečki, J. Rehn, W. Reinalter, R. Uhlig, Test of a solar-

- hybrid microturbine system and evaluation of storage deployment, SolarPaces Conf. (2010) 1–8.
- [68] I. Hischier, D. Hess, W. Lipiński, M. Modest, A. Steinfeld, Heat Transfer Analysis of a Novel Pressurized Air Receiver for Concentrated Solar Power via Combined Cycles, *J. Therm. Sci. Eng. Appl.* 1 (2010). <https://doi.org/10.1115/1.4001259>.
- [69] R.W. Bradshaw, D.B. Dawson, W. De la Rosa, R. Gilbert, S.H. Goods, M.J. Hale, P. Jacobs, S. a. Jones, G.J. Kolb, J.E. Pacheco, M.R. Prairie, H.E. Reilly, S.K. Showalter, L.L. Vant-Hull, Final Test and Evaluation Results from the Solar Two Project, Contract. (2002) 294. <https://doi.org/10.2172/793226>.
- [70] M.B. Structures, Final report on the Power Production Phase of the 10 MWe Solar Thermal Central Pilot Plant, (n.d.).
- [71] T.W. Neises, M.J. Wagner, A.K. Gray, Structural design considerations for tubular power tower receivers operating at 650°C, *ASME 2014 8th Int. Conf. Energy Sustain. ES 2014 Collocated with ASME 2014 12th Int. Conf. Fuel Cell Sci. Eng. Technol.* 1 (2014). <https://doi.org/10.1115/ES2014-6603>.
- [72] P. Schöttl, G. Bern, D.W. van Rooyen, J.A. Fernández Pretel, T. Fluri, P. Nitz, Optimization of Solar Tower molten salt cavity receivers for maximum yield based on annual performance assessment, *Sol. Energy.* 199 (2020) 278–294. <https://doi.org/https://doi.org/10.1016/j.solener.2020.02.007>.
- [73] J.M. Chavez, D.C. Smith, A final report on the phase 1 testing of a molten-salt cavity receiver. Volume II, the main report, United States, 1992. <https://doi.org/10.2172/5042358>.

- [74] L.P. Drouot, M.J. Hillairet, The Themis Program and the 2500-KW Themis Solar Power Station at Targassonne, *J. Sol. Energy Eng.* 106 (1984) 83–89. <https://doi.org/10.1115/1.3267567>.
- [75] W.J.C. Schiel, M.A. Geyer, Testing an external sodium receiver up to heat fluxes of 2.5 MW/m²: Results and conclusions from the IEA-SSPS high flux experiment conducted at the central receiver system of the Plataforma Solar de Almeria (Spain), *Sol. Energy.* 41 (1988) 255–265. [https://doi.org/https://doi.org/10.1016/0038-092X\(88\)90143-0](https://doi.org/10.1016/0038-092X(88)90143-0).
- [76] P.K. Falcone, A handbook for solar central receiver design, United States, 1986. <https://doi.org/10.2172/6545992>.
- [77] C.W. Forsberg, P.F. Peterson, H. Zhao, High-Temperature Liquid-Fluoride-Salt Closed-Brayton-Cycle Solar Power Towers, *J. Sol. Energy Eng.* 129 (2006) 141–146. <https://doi.org/10.1115/1.2710245>.
- [78] M. Sedighi, R. Vasquez Padilla, M. Lake, A. Rose, Y.Y. Lim, J.P. Novak, R.A. Taylor, Design of high-temperature atmospheric and pressurised gas-phase solar receivers: A comprehensive review on numerical modelling and performance parameters, *Sol. Energy.* 201 (2020) 701–723. <https://doi.org/10.1016/j.solener.2020.03.025>.
- [79] C. Singer, R. Buck, R. Pitz-Paal, H. Müller-Steinhagen, Assessment of Solar Power Tower Driven Ultrasupercritical Steam Cycles Applying Tubular Central Receivers With Varied Heat Transfer Media, *J. Sol. Energy Eng.* 132 (2010). <https://doi.org/10.1115/1.4002137>.

- [80] V.D. Kumar, V.K. Upadhyay, G. Singh, S. Mukhopadhyay, L. Chandra, Open volumetric air receiver: An innovative application and a major challenge, *WIREs Energy Environ.* n/a (2021) e404. <https://doi.org/https://doi.org/10.1002/wene.404>.
- [81] E. Stechel, A. Ambrosini, A. Hall, T. Lambert, C. Staiger, M. Bencomo, Improved High Temperature Solar Absorbers for Use in Concentrating Solar Power Central Receiver Applications, *ASME 2011 5th Int. Conf. Energy Sustain. ES 2011.* (2011). <https://doi.org/10.2172/1008136>.
- [82] C. Forsberg, P. Peterson, H. Zhao, High-Temperature Liquid-Fluoride-Salt Closed-Brayton-Cycle Solar Power Towers, *J. Sol. Energy Eng.* 129 (2007). <https://doi.org/10.1115/1.2710245>.
- [83] G. Kolb, S. Power, T. Systems, An Evaluation of Possible Next-Generation High-Temperature Molten-Salt Power Towers, (2016). <https://doi.org/10.2172/1035342>.
- [84] M. Epstein, D. Liebermann, M. Rosh, A.J. Shor, Solar testing of 2 MWth water/steam receiver at the Weizmann Institute solar tower, *Sol. Energy Mater.* 24 (1991) 265–278. [https://doi.org/https://doi.org/10.1016/0165-1633\(91\)90067-U](https://doi.org/https://doi.org/10.1016/0165-1633(91)90067-U).
- [85] D.L. Siebers, J.S. Kraabel, Estimating convective energy losses from solar central receivers, United States, 1984. <https://doi.org/10.2172/6906848>.
- [86] X. Li, W. Kong, Z. Wang, C. Chang, F. Bai, Thermal model and thermodynamic performance of molten salt cavity receiver, *Renew. Energy.* 35 (2010) 981–988. <https://doi.org/https://doi.org/10.1016/j.renene.2009.11.017>.
- [87] N.E. Bergan, Testing of the Molten Salt Electric Experiment solar central receiver in an external configuration, United States, 1986. <https://doi.org/10.2172/6972655>.

- [88] M.J. Persky, M. Szczesniak, Infrared, spectral, directional-hemispherical reflectance of fused silica, Teflon polytetrafluoroethylene polymer, chrome oxide ceramic particle surface, Pyromark 2500 paint, Krylon 1602 paint, and Duraflect coating, *Appl. Opt.* 47 (2008) 1389–1396. <https://doi.org/10.1364/AO.47.001389>.
- [89] A. Ambrosini, T.N. Lambert, M. Bencomo, A. Hall, K. vanEvery, N.P. Siegel, C.K. Ho, Improved High Temperature Solar Absorbers for Use in Concentrating Solar Power Central Receiver Applications, *ASME 2011 5th Int. Conf. Energy Sustain.* (2011) 587–594. <https://doi.org/10.1115/ES2011-54241>.
- [90] A. Hall, A. Ambrosini, C. Ho, Solar selective coatings for concentrating, *Adv. Mater. Process.* 170 (2012) 28–32. <https://www.scopus.com/inward/record.uri?eid=2-s2.0-84856345379&partnerID=40&md5=edb5744af9ffeea8775424629e62926b>.
- [91] Q. Ma, L. Luo, R.Z. Wang, G. Sauce, A review on transportation of heat energy over long distance: Exploratory development, *Renew. Sustain. Energy Rev.* 13 (2009) 1532–1540. <https://doi.org/https://doi.org/10.1016/j.rser.2008.10.004>.
- [92] J.M. Chavez, C.E. Tyner, W.A. Couch, Direct absorption receiver flow testing and evaluation, in: United States, 1988. <https://www.osti.gov/biblio/7165521>.
- [93] B.W. Webb, R. Viskanta, Analysis of Heat Transfer and Solar Radiation Absorption in an Irradiated Thin, Falling Molten Salt Film, *J. Sol. Energy Eng.* 107 (1985) 113–119. <https://doi.org/10.1115/1.3267663>.
- [94] W.D. Drotning, Solar absorption properties of a high temperature direct-absorbing heat transfer fluid, in: United States, 1977. <https://www.osti.gov/biblio/7310810>.

- [95] W.D. Drotning, Optical properties of solar-absorbing oxide particles suspended in a molten salt heat transfer fluid, *Sol. Energy.* 20 (1978) 313–319. [https://doi.org/https://doi.org/10.1016/0038-092X\(78\)90123-8](https://doi.org/https://doi.org/10.1016/0038-092X(78)90123-8).
- [96] G. Jorgensen, P. Schissel, R. Burrows, Optical properties of high-temperature materials for direct absorption receivers, *Sol. Energy Mater.* 14 (1986) 385–394. [https://doi.org/https://doi.org/10.1016/0165-1633\(86\)90061-4](https://doi.org/https://doi.org/10.1016/0165-1633(86)90061-4).
- [97] M.S. Bohn, H.J. Green, Heat transfer in molten salt direct absorption receivers, *Sol. Energy.* 42 (1989) 57–66. [https://doi.org/https://doi.org/10.1016/0038-092X\(89\)90130-8](https://doi.org/https://doi.org/10.1016/0038-092X(89)90130-8).
- [98] A. Veeraragavan, A. Lenert, B. Yilbas, S. Al-Dini, E.N. Wang, Analytical model for the design of volumetric solar flow receivers, *Int. J. Heat Mass Transf.* 55 (2012) 556–564. <https://doi.org/10.1016/j.ijheatmasstransfer.2011.11.001>.
- [99] E. Bilgen, K.G.T. Hollands, INTERSOL 85 proceedings of the Ninth Biennial Congress of the International Solar Energy Society, in: Montreal Canada, 1986. https://www.si.edu/object/siris_sil_470855 (accessed October 31, 2021).
- [100] T.A. Newell, K.Y. Wang, R.J. Copeland, Film flow characteristics for direct absorption solar receiver surfaces, *Present. Am. Soc. Mech. Eng. Winter Annu. Meet.* (1985). <https://ui.adsabs.harvard.edu/abs/1985asme.meetR....N>.
- [101] C.E. Tyner, Status of the direct absorption receiver panel research experiment: Salt flow and solar test requirements and plans, United States, 1989. <https://doi.org/10.2172/6128380>.
- [102] S. -M. Yih, R.C. Seagrave, Hydrodynamic stability of thin liquid films flowing

- down an inclined plane with accompanying heat transfer and interfacial shear, *AIChE J.* 24 (1978) 803–810. <https://doi.org/10.1002/aic.690240506>.
- [103] A. Faghri, R.A. Seban, Heat transfer in wavy liquid films, *Int. J. Heat Mass Transf.* 28 (1985) 506–508. [https://doi.org/https://doi.org/10.1016/0017-9310\(85\)90086-9](https://doi.org/https://doi.org/10.1016/0017-9310(85)90086-9).
- [104] A. Faghri, R.A. Seban, Heat and mass transfer to a turbulent liquid film, *Int. J. Heat Mass Transf.* 31 (1988) 891–894. [https://doi.org/https://doi.org/10.1016/0017-9310\(88\)90145-7](https://doi.org/https://doi.org/10.1016/0017-9310(88)90145-7).
- [105] M.S. Bohn, S.H. Davis, Thermocapillary breakdown of falling liquid films at high Reynolds numbers, *Int. J. Heat Mass Transf.* 36 (1993) 1875–1881. [https://doi.org/https://doi.org/10.1016/S0017-9310\(05\)80175-9](https://doi.org/https://doi.org/10.1016/S0017-9310(05)80175-9).
- [106] G. Angelino, Carbon Dioxide Condensation Cycles For Power Production, *J. Eng. Power.* 90 (1968) 287–295. <https://doi.org/10.1115/1.3609190>.
- [107] M.S. Bohn, Experimental investigation of the direct absorption receiver concept, *Energy.* 12 (1987) 227–233. [https://doi.org/https://doi.org/10.1016/0360-5442\(87\)90081-8](https://doi.org/https://doi.org/10.1016/0360-5442(87)90081-8).
- [108] W. Wu, B. Gobereit, C.. Singer, L. Amsbeck, R. Pitz-Paal, Direct absorption receivers for high temperatures, 2011.
- [109] B.D. Kelly, Advanced Thermal Storage for Central Receivers with Supercritical Coolants, United States, 2010. <https://doi.org/10.2172/981926>.
- [110] D.M. Gruen, Fused-salt spectrophotometry, *Q. Rev. Chem. Soc.* 19 (1965) 349–368. <https://doi.org/10.1039/qr9651900349>.

- [111] S.F. Wu, T. V Narayanan, Commercial direct absorption receiver design studies, United States, 1988. <https://doi.org/10.2172/6514065>.
- [112] S. Paitoonsurikarn, K. Lovegrove, G. Hughes, J. Pye, Numerical Investigation of Natural Convection Loss From Cavity Receivers in Solar Dish Applications, *J. Sol. Energy Eng.* 133 (2011). <https://doi.org/10.1115/1.4003582>.
- [113] T.R. Tracey, C.E. Tyner, E.R. Weber, Potential of advanced-design solar central receiver power systems, in: *Sol. Eng. - 1989; Proc. Elev. Annu. ASME Sol. Energy Conf.*, AA(Sandia National Laboratories, Albuquerque, NM), AB(Sandia National Laboratories, Albuquerque, NM), AC(Sandia National Laboratories, Albuquerque, NM), 1989: pp. 83–88. <https://ui.adsabs.harvard.edu/abs/1989soen.proc...83T>.
- [114] J.I. Ortega, J.I. Burgaleta, F.M. Téllez, Central Receiver System Solar Power Plant Using Molten Salt as Heat Transfer Fluid, *J. Sol. Energy Eng.* 130 (2008). <https://doi.org/10.1115/1.2807210>.
- [115] X. Xu, K. Vignarooban, B. Xu, K. Hsu, A.M. Kannan, Prospects and problems of concentrating solar power technologies for power generation in the desert regions, *Renew. Sustain. Energy Rev.* 53 (2016) 1106–1131. <https://doi.org/10.1016/j.rser.2015.09.015>.
- [116] J. León, M. Sánchez, J. Pacheco E., Internal film receiver possibilities for the third generation of central receiver technology, *J. Phys. IV Fr.* 09 (1999) Pr3-525-Pr3-530. <https://doi.org/10.1051/jp4:1999383>.
- [117] M. Romero, J. González-Aguilar, Solar thermal CSP technology, *WIREs Energy Environ.* 3 (2014) 42–59. <https://doi.org/https://doi.org/10.1002/wene.79>.

- [118] S. Mey-Cloutier, C. Caliot, A. Kribus, Y. Gray, G. Flamant, Experimental study of ceramic foams used as high temperature volumetric solar absorber, *Sol. Energy*. 136 (2016) 226–235. <https://doi.org/10.1016/j.solener.2016.06.066>.
- [119] X. Chen, X.-L. Xia, X.-W. Yan, C. Sun, Heat transfer analysis of a volumetric solar receiver with composite porous structure, *Energy Convers. Manag.* 136 (2017) 262–269. <https://doi.org/10.1016/j.enconman.2017.01.018>.
- [120] U.M. Noebauer, S.A. Sim, Monte Carlo radiative transfer, (n.d.). <https://doi.org/10.1007/s41115-019-0004-9>.
- [121] J. Spanier, Gelbard, Ely M., Monte Carlo principles and neutron transport problems, Addison-Wesley Pub. Co., Reading, Mass., 1969.
- [122] Y. Gu, Y. Wu, T. Liu, L. Chen, Q. Zhang, Heat Transfer Performance of a Solar LiBr Absorption Refrigeration System, in: 5th Int. Energy Convers. Eng. Conf. Exhib., American Institute of Aeronautics and Astronautics, 2007. <https://doi.org/10.2514/6.2007-4777>.
- [123] S. Mey-Cloutier, C. Caliot, A. Kribus, Y. Gray, G. Flamant, Experimental study of ceramic foams used as high temperature volumetric solar absorber, *Sol. Energy*. 136 (2016) 226–235. <https://doi.org/10.1016/j.solener.2016.06.066>.
- [124] A. Andreozzi, N. Bianco, M. Iasiello, V. Naso, Thermo-Fluid-Dynamics of a Ceramic Foam Solar Receiver: A Parametric Analysis, *Heat Transf. Eng.* 41 (2020) 1085–1099. <https://doi.org/10.1080/01457632.2019.1600876>.
- [125] A. Carotenuto, G. Ruocco, F. Reale, Heat exchange in a multi-cavity volumetric solar receiver, *Sol. Energy*. 46 (1991) 241–248. <https://doi.org/10.1016/0038->

092X(91)90069-9.

- [126] Z.-D. Cheng, Y.L. He, F.Q. Cui, Numerical investigations on coupled heat transfer and synthetical performance of a pressurized volumetric receiver with MCRT–FVM method, *Appl. Therm. Eng.* 50 (2013) 1044–1054. <https://doi.org/10.1016/j.applthermaleng.2012.08.045>.
- [127] X. Liu, G. Gong, H. Cheng, Combined Natural Convection and Radiation Heat Transfer of Various Absorbing-Emitting-Scattering Media in a Square Cavity, *Adv. Mech. Eng.* 6 (2014) 403690. <https://doi.org/10.1155/2014/403690>.
- [128] P. Sharma, R. Sarma, L. Chandra, R. Shekhar, P.S.Ghoshdastidar, On the Design and Evaluation of Open Volumetric Air Receiver for Process Heat Applications, *Energy Procedia.* 57 (2014) 2994–3003. <https://doi.org/10.1016/j.egypro.2014.10.335>.
- [129] B. Hoffschmidt, S. Alexopoulos, J. Götsche, M. Sauerborn, O. Kaufhold, 3.06 - High Concentration Solar Collectors, in: A.B.T.-C.R.E. Sayigh (Ed.), Elsevier, Oxford, 2012: pp. 165–209. <https://doi.org/10.1016/B978-0-08-087872-0.00306-1>.
- [130] J.A. Harris, T.G. Lenz, Thermal performance of solar concentrator/cavity receiver systems, *Sol. Energy.* 34 (1985) 135–142. [https://doi.org/10.1016/0038-092X\(85\)90170-7](https://doi.org/10.1016/0038-092X(85)90170-7).
- [131] C. Zou, H. Feng, Y. Zhang, Q. Falcoz, C. Zhang, W. Gao, Geometric optimization model for the solar cavity receiver with helical pipe at different solar radiation, *Front. Energy.* 13 (2019) 284–295. <https://doi.org/10.1007/s11708-019-0613-3>.
- [132] A. Almerbati, Hexagonal and mixed arrays of flow channel design in counterflow

- heat exchanger, *Int. Commun. Heat Mass Transf.* 124 (2021) 105268.
<https://doi.org/https://doi.org/10.1016/j.icheatmasstransfer.2021.105268>.
- [133] S. Lim, Y. Kang, H. Lee, S. Shin, Design optimization of a tubular solar receiver with a porous medium, *Appl. Therm. Eng.* 62 (2014) 566–572.
<https://doi.org/10.1016/j.applthermaleng.2013.10.025>.
- [134] X. Guo, X. Cai, L. Zhu, H. Yang, Preparation and pore structure characteristics of SiC honeycomb ceramics with macroporous walls, *Ceram. Int.* 40 (2014) 6339–6343. <https://doi.org/https://doi.org/10.1016/j.ceramint.2013.10.126>.
- [135] T. Fend, W. Völker, R. Miebach, O. Smirnova, D. Gonsior, D. Schöllgen, P. Rietbrock, Experimental investigation of compact silicon carbide heat exchangers for high temperatures, *Int. J. Heat Mass Transf.* 54 (2011) 4175–4181.
<https://doi.org/https://doi.org/10.1016/j.ijheatmasstransfer.2011.05.028>.
- [136] G. Barreto, P. Canhoto, M. Collares-Pereira, Three-dimensional modelling and analysis of solar radiation absorption in porous volumetric receivers, *Appl. Energy.* 215 (2018) 602–614. <https://doi.org/10.1016/j.apenergy.2018.02.065>.
- [137] M. Melot, J.-Y. Trépanier, R. Camarero, E. Petro, Comparison of Two Models for Radiative Heat Transfer in High Temperature Thermal Plasmas, *Model. Simul. Eng.* 2011 (2011) 285108. <https://doi.org/10.1155/2011/285108>.
- [138] M.B. Patil, Radiation Heat Transfer in a Particulate Medium Using a Ray Tracing Method, (2016).
- [139] M. Becker, T. Fend, B. Hoffschmidt, R. Pitz-Paal, O. Reutter, V. Stamatov, D. Trimis, Theoretical and Numerical Investigation of Flow Stability in Porous

- Materials Applied as Volumetric Solar Receivers, *Sol. Energy*. 80 (2006) 1241–1248. <https://doi.org/10.1016/j.solener.2005.11.006>.
- [140] A. Alvarez, O. Cabeza, M.C. Muñiz, L.M. Varela, Experimental and numerical investigation of a flat-plate solar collector, *Energy*. 35 (2010) 3707–3716. <https://doi.org/10.1016/j.energy.2010.05.016>.
- [141] P. Wang, J.B. Li, L. Zhou, K. Vafai, Experimental characterization on pore parameter and the irradiation absorption efficiency of a series SiC foam specimens, *Energy Convers. Manag.* 212 (2020) 1–13. <https://doi.org/10.1016/j.enconman.2020.112795>.
- [142] G. Barreto, P. Canhoto, M. Collares-Pereira, Parametric analysis and optimisation of porous volumetric solar receivers made of open-cell SiC ceramic foam, *Energy*. 200 (2020) 117476. <https://doi.org/10.1016/j.energy.2020.117476>.
- [143] S. De Schampheleire, K. De Kerpel, B. Ameer, P. De Jaeger, O. Bagci, M. De Paepe, A discussion on the interpretation of the darcy equation in case of open-cell metal foam based on numerical simulations, *Materials (Basel)*. 9 (2016). <https://doi.org/10.3390/ma9060409>.
- [144] A.J. Organ, New correlations for pressure loss, in: A.J.B.T.-T.A.E. Organ (Ed.), *Woodhead Publishing*, 2007: pp. 135–146. <https://doi.org/https://doi.org/10.1533/9781845693602.3.135>.
- [145] Z. Wu, C. Caliot, G. Flamant, Z. Wang, Coupled radiation and flow modeling in ceramic foam volumetric solar air receivers, *Sol. Energy*. 85 (2011) 2374–2385. <https://doi.org/https://doi.org/10.1016/j.solener.2011.06.030>.

- [146] ANSYS FLUENT, ANSYS FLUENT 12.0 Theory Guide - 4.4.3 Realizable - Model, (n.d.).
<https://www.afs.enea.it/project/neptunius/docs/fluent/html/th/node60.htm>
 (accessed August 19, 2021).
- [147] ANSYS FLUENT, ANSYS FLUENT 12.0 Theory Guide - 4.4 Standard, RNG, and Realizable - Models, (n.d.).
<https://www.afs.enea.it/project/neptunius/docs/fluent/html/th/node57.htm>
 (accessed August 19, 2021).
- [148] M.A. Schuetz, L.R. Glicksman, A Basic Study of Heat Transfer Through Foam Insulation, J. Cell. Plast. 20 (1984) 114–121.
<https://doi.org/10.1177/0021955X8402000203>.
- [149] C. Tseng, K. Kuo, Thermal radiative properties of phenolic foam insulation, J. Quant. Spectrosc. Radiat. Transf. 72 (2002) 349–359.
- [150] ANSYS FLUENT, 5 . 3 . 6 Discrete Ordinates (DO) Radiation Model Theory, (2009) 1–19.
- [151] E.E. Shpilrain, AIR (PROPERTIES OF), in: A-to-Z Guid. to Thermodyn. Heat Mass Transf. Fluids Eng., Begellhouse, 2008.
https://doi.org/10.1615/atoz.a.air_properties_of.
- [152] Japan Society of Thermophysical Properties (2008) Thermophysical Properties Handbook. Yokendo Co. Ltd., Tokyo, 469. - References - Scientific Research Publishing, (2008).
[https://www.scirp.org/\(S\(351jmbntvnsjt1aadkposzje\)\)/reference/ReferencesPapers](https://www.scirp.org/(S(351jmbntvnsjt1aadkposzje))/reference/ReferencesPapers).

aspx?ReferenceID=1374765 (accessed April 8, 2021).

- [153] M. Nakakura, S. Bellan, K. Matsubara, T. Kodama, Conjugate radiation-convection-conduction simulation of volumetric solar receivers with cut-back inlets, *Sol. Energy*. 170 (2018) 606–617. <https://doi.org/10.1016/j.solener.2018.06.006>.
- [154] Y. Jin, A. V. Kuznetsov, Turbulence modeling for flows in wall bounded porous media: An analysis based on direct numerical simulations, *Phys. Fluids*. 29 (2017). <https://doi.org/10.1063/1.4979062>.

CURRICULUM VITAE

Name : Muhammad Faizan

Nationality : Pakistani

Date of Birth : September 15, 1994

Email : fx.faizan@yahoo.com

Address : Near Ali Homes, Hayat Street, Street # 15, Shalimar Town,
Multan, 60000

Academic Background : Master of Science (M.Sc.) in Mechanical Engineering,
KFUPM, Dhahran, KSA

Bachelor of Science (B.Sc.) in Mechanical Engineering,
UET Lahore, Pakistan

Publications **M. Faizan**, A. Almerbati, B. S. Yilbas. “A Novel Approach
for Volumetric Receiver Performance Assessments”
Applied Thermal Engineering (Submitted - Required
Review Completed)

M. Faizan, A. Almerbati, B. S. Yilbas. “Thermal and Flow
Characteristics of a Volumetric Solar Receiver; including

Turbulent Flow” Case Studies in Thermal Engineering.

(Submitted – Under Review)

M. Faizan, A. Almerbati, B. S. Yilbas. “Numerical assessment on various design’s configurations of volumetric receiver for solar harvesting system” (Ready to Submit)

Software : ANSYS, Solidworks, Solid edge, Gambit, Blender
Geomagic, EES, Microsoft office, Minitab, Tecplot,
MATLAB, Python