Research and Development of Printed Circuit Heat exchanger

Submitted By

Seone Ousmane

200011155

Traore Selikouma

2000111160

Aliyu Shuaibu Abdul Rahman

200011164

Supervised By

Prof Dr. Mohammad Monjurul Eshan

A Thesis submitted in partial fulfillment of the requirement for the degree of Bachelor of Science in Mechanical Engineering

Department of Mechanical and Production Engineering (MPE)

Islamic University of Technology (IUT)

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CERTIFICATE OF RESEARCH

This thesis titled "Research and Development of Printed Circuit Heat Exchanger" submitted by Seone Ousmane (200011155), Traore Selikouma (200011160), Aliyu Shuaibu Abdul Rahman (200011164) has been acknowledged as adequate as part of the requirements for a Bachelor of Science in Mechanical Engineering degree.

Supervisor

Dr. Mohammad Monjurul Eshan

Professor

Department of Mechanical and Production Engineering (MPE)

Head of the Department

Dr. M. Ahsan Habib

Professor

Department of Mechanical and Production Engineering (MPE)

Islamic University of Technology (IUT)

Candidate's Declaration

This is to certify that the work presented in this thesis, titled, "Research and Development Printed circuit Heat Exchanger" is the outcome of the investigation and research carried out by me under the supervision of Dr. Mohammad Monjurul Eshan Professor, MPE, IUT.

It is also declared that neither this thesis nor any part of it has been submitted elsewhere for the award of any degree or diploma.

Seone Ousmane 200011155

Traore Selikouma 200011160

Aliyu Shuaibu Abdul
Rahman 200011164

Recommendation of the Thesis Supervisors

The thesis titled "Research and Development of Printed Circuit Heat Exchangers" Submitted by Seone Ousmane, Student No: 200011155, Traore Selikouma, Student No: 200011160, Aliyu Shuaibu Abdul Rahman, Student No:200011164 has been accepted as satisfactory in partial fulfillment of the requirements for the degree of B Sc. in Mechanical Engineering **on 18th October, 2025.**

Dr. Mohammad Monjurul Eshan (Supervisor)
Professor
MPE Dept., IUT, Board Bazar, Gazipur-1704, Bangladesh.

CO-PO Mapping of ME 4800 -Theis and Project

COs	Course Outcomes (CO) Statement	(PO)	Addressed by	
CO1	Discover and Locate research problems and illustrate them via figures/tables or projections/ideas through field visit and literature review and determine/Setting aim and objectives of the project/work/research in specific, measurable, achievable, realistic and timeframe manner.	**PO2 Problem analysis**	Thesis Book	
			Performance by research	
			Presentation and soft skill	
CO2	Design research solutions of the problems towards achieving the objectives and its application. Design systems, components or processes that meets related needs in the field of mechanical engineering	**PO3 Design/development of solutions**	Thesis Book	
			Performance by research	
			Presentation and soft skill	
CO3	Review, debate, compare and contrast the relevant literature contents. Relevance of this research/study. Methods, tools, and techniques used by past researchers and justification of use of them in this work.	**PO4 Investigation**	Thesis Book	
			Performance by research	
			Presentation and soft skill	
CO4	Analyse data and exhibit results using tables, diagrams, graphs with their interpretation. Investigate the designed solutions to solve the problems through case study/survey study/experimentation/simulation using modern tools and techniques.	**PO5 Modern tool usage**	Thesis Book	
			Performance by research	
			Presentation and soft skill	
CO5	Apply moral values and research/professional ethics throughout the work, and justify genuine referencing on sources, and demonstration of own contribution.	**PO8 Ethics**	Thesis Book	
			Performance by research	
			Presentation and soft skill	
CO6	Perform own self and manage group activities from the beginning to the end of the research/work as a quality work.	**PO9 Individual work and teamwork**	Thesis Book	
			Performance by research	
			Presentation and soft skill	

CO7	<u>Compile and arrange</u> the work outputs, write the report/thesis, a sample journal paper, and present the work to wider audience using modern communication tools and techniques.	**PO10 Communication**	Thesis Book	
			Performance by research	
			Presentation and soft skill	
CO8	<u>Recognize</u> the necessity of life-long learning in career development in dynamic real-world situations from the experience of completing this project.	**PO12 Life-long learning**	Thesis Book	
			Performance by research	
			Presentation and soft skill	

Student Name /ID: Signature of the Supervisor: ....

1. Seone Ousman/200011155

Name of the Supervisor: Prof.Dr. M. Monjurul Eshan

2. Traore Selikouma/200011160

3. Aliyu Shuaibu Abdul Rahman/200011164

K-P-A Mapping of ME 4800 -Theis and Project

COs	POs	Related Ks								Related Ps							Related As				
		K1	K2	K3	K4	K5	K6	K7	K8	P1	P2	P3	P4	P5	P6	P7	A1	A2	A3	A4	A5
CO1	PO2	√	√	√	√					√											
CO2	PO3					√				√	√	√				√	√	√	√		
CO3	PO4								√	√	√	√					√	√			
CO4	PO5						√			√				√							
CO5	PO8							√													
CO6	PO9																				
CO7	PO10																				
CO8	PO12																				

Student Name /ID: Signature of the Supervisor: ....

1. Seone Ousman/200011155 Name of the Supervisor: Prof.Dr. M. Monjurul Eshan

2. Traore Selikouma/200011160

3. Aliyu Shuaibu Abdul Rahman/200011164

List of Sustainable Development Goals (SDGs) Addressed in this Project

SDG No	Goals	Targets	Relevance to the Thesis	Remarks
1	No Poverty	1.1 Eradicate extreme poverty (people living on less than $1.25/day).		
		1.2 Reduce poverty in all its forms by at least half.		
		1.3 Implement nationally appropriate social protection systems.		
		1.4 Ensure equal rights to economic resources, services, property, inheritance, technology, and financial services		
		1.5 Build resilience of the poor and reduce exposure to climate-related and other shocks.		
		1.a Mobilize resources to end poverty.		
		1.b Create pro-poor policy frameworks.		
2	Zero Hunger	2.1 End hunger and ensure access to safe, nutritious food year-round.		
		2.2 End all forms of malnutrition.		
		2.3 Double agricultural productivity and incomes of small-scale producers.	√	Supports energy-efficient processes in agro-industries, improving productivity and food preservation.
		2.4 Ensure sustainable food production systems and resilient agricultural practices.	√	Contributes to sustainable processing and energy-efficient food production
		2.5 Maintain genetic diversity of seeds, plants, and animals.		

		2.a Increase investment in rural infrastructure, research, and technology.		
		2.b Correct and prevent trade restrictions/distortions in global food markets		
		2.c Adopt measures to ensure proper functioning of food commodity markets.		
3	Good Health and Well Being	3.1 Reduce global maternal mortality ratio.		
		3.2 End preventable deaths of newborns and under-5 children.		
		3.3 End epidemics of AIDS, tuberculosis, malaria, and neglected tropical diseases.		
		3.4 Reduce premature mortality from NCDs and promote mental health.		
		3.5 Strengthen prevention and treatment of substance abuse.		
		3.6 Halve global deaths/injuries from road traffic accidents.		
		3.7 Ensure universal access to sexual and reproductive healthcare.		
		3.8 Achieve universal health coverage.	√	Printed Circuit Heat Exchangers (PCHEs) are able to be applied to medical equipment (e.g. sterilization systems, MRI machines, and cooling devices), and this will help to make healthcare system more efficient and reliable.
		3.9 Reduce deaths from hazardous chemicals, pollution, and contamination.		
		3.a Strengthen tobacco control (WHO FCTC)		
		3.b Support R&D of vaccines and medicines.		

		3.c Increase health financing and workforce.		
		3.d Strengthen capacity for early warning and risk management.		
4	Quality Education	4.1 Ensure all complete free, equitable, quality primary and secondary education.		
		4.2 Ensure access to quality early childhood development and pre-primary education.		
		4.3 Ensure equal access to affordable technical, vocational, and higher education.	√	The project may be a learning and research resource towards training in thermal engineering and technological innovation.
		4.4 Increase skills for employment and entrepreneurship.	√	The study instills high-ranking skills in the field of research and development, production, and thermal modelling, which can be useful in the industrial labour market.
		4.5 Eliminate gender disparities in education.		
		4.6 Ensure literacy and numeracy for youth and adults.		
		4.7 Ensure learners acquire knowledge/skills for sustainable development.	√	PCHE technology is sustainable in terms of energy usage, and this technology is in line with the concept of sustainable development which is taught in engineering schools.

		4.a Build and upgrade education facilities that are inclusive and safe.		
		4.b Expand scholarships for developing countries.		
		4.c Increase supply of qualified teachers.		
5	Gender Equality	5.1 End all forms of discrimination against women and girls.		
		5.2 Eliminate violence against women and girls.		
		5.3 Eliminate harmful practices (child, early, forced marriage, FGM).		
		5.4 Recognize and value unpaid care and domestic work.		
		5.5 Ensure women's participation in leadership and decision-making.		
		5.6 Ensure universal access to reproductive health and rights.		
		5.a Undertake reforms to give women equal rights to resources.		
		5.b Enhance use of enabling technology to empower women.		
		5.c Adopt and strengthen policies and laws for gender equality.		
6	Clean Water and Sanitation	6.1 Achieve universal and equitable access to safe drinking water.		
		6.2 Achieve access to adequate sanitation and hygiene.		
		6.3 Improve water quality by reducing pollution.		
		6.4 Increase water-use efficiency and sustainable withdrawals.		
		6.5 Implement integrated water resources management.		
		6.6 Protect and restore water-related ecosystems.		
		6.a Expand international cooperation in water and sanitation.		
		6.b Support participation of local communities.		

7	Affordable and Clean Energy	7.1 Ensure universal access to affordable, reliable, modern energy services.	√	Increment in the development of Printed Circuit Heat Exchangers (PCHEs) helps in the enhancement of accessibility to modern, efficient and reliable energy systems due to their thermal management.
		7.2 Increase substantially the share of renewable energy.	√	An improved technology of the heat exchanger encourages the integration and maximization of renewable energy resources through enhancement of performance and efficiency of thermal systems.
		7.3 Double global rate of improvement in energy efficiency.		
		7.a Enhance international cooperation on clean energy research/technology.		
		7.b Expand infrastructure and upgrade technology for sustainable energy.		
8	Decent Work and Economic Growth	8.1 Sustain per capita economic growth.	√	Your system helps in the improvement of efficiency and operator productivity hence contributing to the growth of the economy in a sustainable manner.

		8.2 Achieve higher levels of productivity through diversification, tech, and innovation.	√	The device enhances productivity based on technological innovation and diversification of the business modes of operation.
		8.3 Promote policies for decent job creation and entrepreneurship.	√	Your activities improve the quality of work and ergonomics, which promote the development of decent and sustainable jobs.
		8.4 Improve resource efficiency in production and consumption.	√	Your idea is the most efficient in terms of resources, as it is based on the improvement of human-system interaction and thermal control.
		8.5 Achieve full and productive employment for all.		
		8.6 Substantially reduce youth not in employment/education/training.		
		8.7 Eradicate forced labour, modern slavery, and child labour.		
		8.8 Protect labour rights and safe working environments.	√	The project helps to create safer and more controlled thermal systems which contribute to the protection of people and their working environments.
		8.9 Promote sustainable tourism.		

		8.a Increase aid for trade support.		
		8.b Develop a global youth employment strategy.		
9	Industry, Innovation, and Infrastructure	9.1 Develop quality, reliable, sustainable infrastructure.	√	Through the innovation of Printed Circuit Heat Exchangers (PCHEs) your work has been involved in the development of innovative and sustainable technological infrastructure.
		9.2 Promote inclusive and sustainable industrialization.	√	Human-centered technological innovation facilitates the advancement of inclusive and sustainable industrialization and allows creating more efficient and less harmful industrial systems.
		9.3 Increase access of SMEs to financial services and integration into value chains.		
		9.4 Upgrade infrastructure for sustainability and resource efficiency.		
		9.5 Enhance scientific research and technology development.	√	The research helps to enhance scientific research and technological innovations in the area of heat transfer and advanced thermal systems, which contributes to the industrial and academic research and development.

		9.a Facilitate sustainable infrastructure in developing countries.		
		9.b Support domestic tech development and value addition.		
		9.c Increase access to ICT and internet.		
10	Reduced Inequalities	10.1 Achieve income growth of bottom 40%.		
		10.2 Empower and promote inclusion regardless of status.		
		10.3 Ensure equal opportunity and reduce inequalities of outcome.		
		10.4 Adopt policies for fiscal, wage, and social protection equality.		
		10.5 Improve regulation of global financial markets.		
		10.6 Ensure enhanced representation in global institutions.		
		10.7 Facilitate safe, regular, and responsible migration.		
		10.a Implement special treatment for developing countries.		
		10.b Encourage development assistance and investment in least developed areas.		
		10.c Reduce remittance costs.		
11	Sustainable Cities and Communities	11.1 Ensure access to adequate, safe, and affordable housing.		
		11.2 Provide sustainable transport systems.		
		11.3 Enhance inclusive urbanization and capacity for planning.		
		11.4 Protect cultural and natural heritage.		
		11.5 Reduce disaster impact and losses.		
		11.6 Reduce environmental impact of cities (air quality, waste).		
		11.7 Provide access to safe, inclusive green/public spaces.		

		11.a Support positive links between urban, peri-urban, rural.		
		11.b Increase disaster risk reduction strategies.		
		11.c Support least developed countries in sustainable building.		
12	Responsible Consumption and Production	12.1 Implement 10-Year Framework on sustainable consumption/production.		
		12.2 Achieve sustainable management and use of resources.	√	sustainable resource management is essential for optimizing the performance and efficiency of the technologies developed in this research.
		12.3 Halve per capita global food waste.		
		12.4 Manage chemicals and waste sustainably.		
		12.5 Substantially reduce waste generation.		
		12.6 Encourage companies to adopt sustainable practices.		
		12.7 Promote sustainable public procurement.		
		12.8 Ensure people have relevant information for sustainable development.		
		12.a Support developing countries' scientific and technological capacity.	√	this target aligns with strengthening scientific and technological capacities in developing countries, directly connected to the objectives of this research.
		12.b Develop tools to monitor sustainable tourism impacts.		
		12.c Rationalize inefficient fossil-fuel subsidies.		

13	Climate Action	13.1 Strengthen resilience and adaptive capacity to climate-related hazards.		
		13.2 Integrate climate measures into national policies.		
		13.3 Improve education and awareness on climate change.		
		13.a Implement UNFCCC commitments (mobilize $100 billion annually).		
		13.b Promote mechanisms for capacity building in least developed countries.		
14	Life Below Water	14.1 Reduce marine pollution.		
		14.2 Sustainably manage and protect marine ecosystems.		
		14.3 Minimize and address ocean acidification.		
		14.4 Regulate harvesting and end overfishing.		
		14.5 Conserve at least 10% of coastal and marine areas.		
		14.6 Prohibit harmful fisheries subsidies.		
		14.7 Increase economic benefits from sustainable marine resources.		
		14.a Increase scientific knowledge and marine technology transfer.		
		14.b Provide access for small-scale artisanal fishers.		
		14.c Implement international law for oceans.		
15	Peace, Justice and Strong Institutions	15.1 Conserve terrestrial and freshwater ecosystems.	√	The PCHE technology promotes efficient energy use, reducing industrial energy demand and limiting harmful emissions that can affect terrestrial and freshwater ecosystems.

		15.2 Promote sustainable management of forests.	√	By enhancing thermal efficiency, PCHEs reduce dependence on fossil fuels, indirectly decreasing deforestation for energy needs.
		15.3 Combat desertification and restore degraded land.		
		15.4 Ensure conservation of mountain ecosystems.	√	The reduction of greenhouse gas emissions and waste heat through PCHE innovation supports environmental sustainability and biodiversity conservation.
		15.5 Take urgent action to reduce biodiversity loss.	√	Indirectly supports this by contributing to a cleaner industrial environment that benefits sustainable natural resource management.
		15.6 Promote fair benefit-sharing from genetic resources.		
		15.7 End poaching and trafficking of protected species.		
		15.8 Prevent introduction of invasive alien species.		
		15.9 Integrate ecosystem values into policies/planning.		

		15.a Mobilize resources for biodiversity.	√	PCHE research exemplifies innovation that can attract green funding and promote sustainable industrial practices benefiting ecosystems.
		15.b Finance sustainable forest management.		
		15.c Support local communities for forest 12 and wildlife.		
16	Peace, Justice and Strong Institutions	16.1 Reduce violence and related death rates.		
		16.2 End abuse, trafficking, and violence against children.		
		16.3 Promote rules of law and equal access to justice.		
		16.4 Reduce illicit financial/arms flows, organized crime.		
		16.5 Reduce corruption and bribery.		
		16.6 Develop effective, accountable institutions.		
		16.7 Ensure inclusive, participatory decision-making.		
		16.8 Broaden participation of developing countries in global governance.		
		16.9 Provide legal identity for all (including birth registration).		
		16.10 Ensure access to information and protect freedoms.		
		16.a Strengthen national institutions for prevention of violence.		
		16.b Promote/enforce non-discriminatory laws and policies.		
17	Partnerships for the Goals	17.1 Strengthen domestic resource mobilization.		
		17.2 Developed countries to implement ODA commitments.		
		17.3 Mobilize additional financial resources.		
		17.4 Assist developing countries with debt sustainability.		

		17.5 Invest in least developed countries.		
		17.6 Enhance access to science, technology, innovation.	√	this research directly contributes to this objective by developing an innovative heat exchanger technology applicable across various industrial sectors.
		17.7 Promote environmentally sound technologies.	√	Printed Circuit Heat Exchangers (PCHEs) are more compact, efficient, and eco-friendly compared to conventional heat exchangers.
		17.8 Fully operationalize technology bank for LDCs.		
		17.9 Enhance international support for capacity-building.		
		17.10 Promote a universal, rules-based trading system (WTO).		
		17.11 Increase exports of developing countries.		
		17.12 Timely implementation of duty-free, quota-free market access.		
		17.13 Enhance global macroeconomic stability.		
		17.14 Enhance policy coherence for sustainable development.		
		17.15 Respect national policy space.		
		17.16 Enhance global partnerships.		
		17.17 Encourage multi-stakeholder partnerships.		
		17.18 Enhance data capacity of developing countries.		
		17.19 Support capacity-building sustainable development indicators.		

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Abstract

The present report is a summary of modern, experimental, numerical, and analytical studies on the printed circuit board heat exchanger geometries and will include zigzag, serpentine, straight, wavy, and cellular channel designs to arrive at practical conclusions on the efficiency of high-performance compact recuperators with and without the use of He-Xe and supercritical sCO_2 environments. Printed Circuit Heat Exchangers (PCHEs) are known to have a high ratio of heat-transfer area to volume and demonstrate desirable traits in respect of their pressure and temperature performance and have been proposed to be used in micro transport reactors, in sCO_2 or Brayton systems. In this context, geometry is a critical variable that represents the trade-off between thermal effectiveness and pressure drop: zigzag and cellular zigzag designs can considerably improve the effectiveness and the heat-transfer coefficient compared to straight or serpentine channels, but at the price of a larger pressure drop; but with possible compensations across modified zigzag designs, with cellular designs or with channel designs made of straight channels, or simply three-dimensional wavy or sinusoidal geometries. The secondary effects of interest encompass: axial conduction through thin plates, fin performance, the sensitivity of working fluid properties (such as the He-Xe mole fraction or sCO_2 operation just above pseudo-critical conditions), and mechanical/manufacturing constraints (including brazing, tolerances, and stresses), all of which exert a profound influence on the performance and durability that can be attained. It is recommended that multi-objective geometry optimization (considering performance, pressure drop, and mass) be undertaken, experimental validation should occur under conditions relevant to the application (such as He-Xe mixtures and sCO_2 transcritical operations), and enhancements in materials and joining techniques should be pursued to mitigate thermal stresses and leakage risks; furthermore, additional research on fouling, transient response, and manufacturability is warranted.

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Nomenclature

c_p	Specific heat at constant pressure($W/kg.k$)
D_h	hydraulic diameter (m)
A	heat transfer area (m^2)
h	thermal convective coefficient (W.$m^{-2}.K^{-1}$)
s_i	Displacement [m]
h_s	plate thickness, [mm]
ΔP	pressure drop (Pa)
$\dot{m}$	mass flow rate (kg/s)
Re	Reynolds number
$\vec{u}$	velocity ($m.s^{-1}$)
ΔT	temperature difference, [K]

Q	heat transfer rate (W)
G_r	relative error of effectiveness (−)
W	Width (mm)
$\vec{K}$	Viscous force (N.m^{-2})
Tb	Bulk temperature of the fluid [K]
Nu	Nusselt number (−)
f	friction factor
L	length (mm)
P	static pressure (Pa)
s-CO_2	Supercritical Carbon dioxide

Greek Letters

α	Zigzag angles[degree]
θ	Teta
µ	viscosity (Pa·s)
μ_t	turbulent viscosity (Pa·s)
ρ	density (kg.m^{-3})
σ_k	constant (—)
σ_w	constant (—)
ε_{ij}	Shear strain
ν	kinematic viscosity, [$m^2 s^{-1}$]

Subscripts

in	Inlet
out	Outlet

c	cold side
h	hot side
f	Fluid
b	Bulk mean

Abbreviations

PCHE	Printed Circuit Heat Exchangers
FLNG	floating liquefied natural gas
CFD	Computational Fluid Dynamics
CSP	Concentrated Solar Power
CHE	compact heat exchangers
PHE	plate heat exchanger
FEA	finite element analysis

PFHE	Plate-fin heat exchanger
PCM	phase-change material
PEC	performance evaluation criteria
LNG	Liquefied Natural Gas
PFD	probability density function

Chapter 1

Introduction

One new direction that has emerged as the solution of high efficiency thermal management in small and high-pressure systems is the Printed Circuit Heat Exchangers (PCHEs). Initially, PCHEs were invented by Heatric in the 1980s, which is built on chemically etched microchannels and a diffusion bonding approach: they have a high potential of heat transfer and resistance to harsh environments [1]. The shape of the flow channels is also a determining factor and its shape influences performance of the heat exchanger. Various channel configurations that have attracted considerable consideration are straight, zigzag and S-shaped channel configurations with regards to their combined influence on the enhancement of the heat transfer and pressure drop. The straight channels are not so hard to model and produce a fixed flow profile yet with low generated turbulence levels. The former channels, as compared to the zigzag and S-shaped ones, produce secondary flows and mixing and hence enhance the thermal functionality at the expense of increased-pressure losses [2] These trade-offs should be noted to develop the best system in designing the optimum system particularly in systems such as supercritical CO2 cycles and nuclear power plants. With recent advances in the sphere of Computational Fluid Dynamics (CFD), it is now possible to conduct comprehensive research into these microchannel geometry's in which researchers have been capable of observing the local flow behavior, thermal and flow maldistribution [3], [4]. Mathematical models with the use of CFD tools have been identified to be effective in the steady-state and the transient analysis of PCHEs with different geometries. This paper is a comparative analysis of PCHEs with straight, zigzag and S-shaped channel design based on the validated mathematical models as well as simulated data. This aims at enhancing channel design to enhance thermal performance at lower pressure loss with the information available in the existing literatures and numeric experiments.

1.1. Research scope and problem statement

With the increasing demands for energy around the world and increasing challenges with the environment, heat exchange systems are necessities in the energy, aerospace, chemical, and nuclear industries. Among all these systems, the Printed Circuit Heat Exchanger (PCHE) has been the latest technology in the shape of its ultimate compactness, strong pressure, and thermal functionality. The design of inner flow channels in straight, S-shaped, and zigzag forms, however, is significant in the fundamental role to define the thermal-hydraulic performance of PCHEs [4]. Such arrangements in geometrics contribute significant contribution to the flow distribution, the coefficient of heat transfer, pressure loss, and general system effectiveness. Although the experimental and computational analysis has been done extensively, there is no perfect balance of heat transferred and the pressure drop occurs. Consequently, the applications of mathematical modeling and CFD-based simulations are growing in an attempt to consider these flow paths, compare them and optimize their application. This research aims at a comparative examination of S-shaped, straight, and zigzag microchannels in PCHEs through the means of mathematical and numerical modeling in order to fill up the gaps related to prediction of heat transfer, pressure trends, and the highest energy efficiency.

1.2. Historical Background of PCHE

Originally, PCHEs were developed by Heatric in the 1980s for meeting the compactness and high efficiency and for operating in high parameter conditions. The manufacturing of their microchannel plates involves chemical etching and diffusion bonding that will result to intricate patterns of microchannels on metal plates. PCHEs have developed and been enhanced in their geometry over time in an effort to enhance the uniformity of flow and maximize the S/V ratio [5]. The newly introduced serpentine (S-shaped), zigzag, and straight channels allowed the engineers to possess an extended variety of choices of arranging the flow patterns and the convective heat transfer. The first few years of research involved experimental research on thermal performance, whereas in the later year's computational visualization, finite volume modeling, and ANN were more utilized for performance and design prediction

1.3 Goals and Objectives of the Study

1. To see the distinct peculiarities of the straight, zigzag and S-shaped patterns of channels in plate clad heat exchangers (PCHE's).
2. To correlate their thermal and hydraulic performance (heat transfer and pressure drop) with those of the previous studies in terms of being straight, zigzag and S-shape channels.

3. To study the thermal and hydraulic performance behavior of the configurations under a number of operating conditions.
4. In order to point out the main flaws and problems that are applicable to each of the specific channel arrangement.
5. To justify potential areas of more research and to provide strategies for improving PCHEs.

1.4. Review of Methodology

This paper will bring a detailed review of the existing literature in comparison to PCHEs with a specific focus on the role of cove geometry in determining the thermal performance and hydraulic performance of the PCHEs. The research conducted by the study gives a critical examination of the three most popular geometrical shapes of the channel: S-shaped, straight and zigzag. This review identifies the different modeling and analysis approaches developed in the literature, such as CFD, FEA, usage of AI and optimal control techniques. Analysis of 112 references enabled a cohesive summary of the major drivers of PCHE performance with the aim of informing the optimal design strategy for the future [6].

i Section 2 describes important geometric principles and standard equations of the research literature used to study heat transfer and fluid dynamics characteristics of PHEs. It gives a comparative study on the three main flow channel geometries, straight, S-shaped, and zigzag channels.

ii The performance data collected from studies involving straight and compound channel geometries in laminar, transitional and turbulent flow regimes is summarized in Section 3. A literature review of available research outlining local distributions of the Nusselt number and friction factors is given and categorized.

iii Section 4 integrates studies that address thermal efficiency and optimization in the operation of PCHEs, focusing on the application of CFD methods in high-pressure and high-temperature operation. Notwithstanding working lack of original simulations, this review pools and analyzes results from parametric studies investigating the relationship between the channel shape height, width, curvature radius and heat transfer performance.

iv In section 5, mathematical models of both compressible and incompressible fluid flows in plate-channel heat exchangers (PCHEs) are reviewed based on past work. The review emphasizes methods used by previous studies to handle the boundary conditions, grid sensitivity, and simulation approaches investigated in two-dimensional and even three-dimensional PCHE designs.

v In Section 6, investigations into zigzag channel designs and their capacity to stimulate secondary flows, vorticity, and turbulence, thereby increasing thermally driven convection, are central.

vi This part represents an analysis of optimization approaches from the literature to pursue trade-offs between thermal efficiency, pressure loss, and fabrication feasibility. The section provides a view of the different type of modeling that are done by researchers to optimize the design of PCHEs.

vii Also, the literature review identifies a few gaps, including a lack of numerical work on highly curved channels, absence of machine-learning approaches in PCHE design, and relative absence of studies of flow maldistribution in transient operation.

To make this literature review organization and thematic content clearer, an outline of its structure and methods is presented by a flowchart on Figure 1.

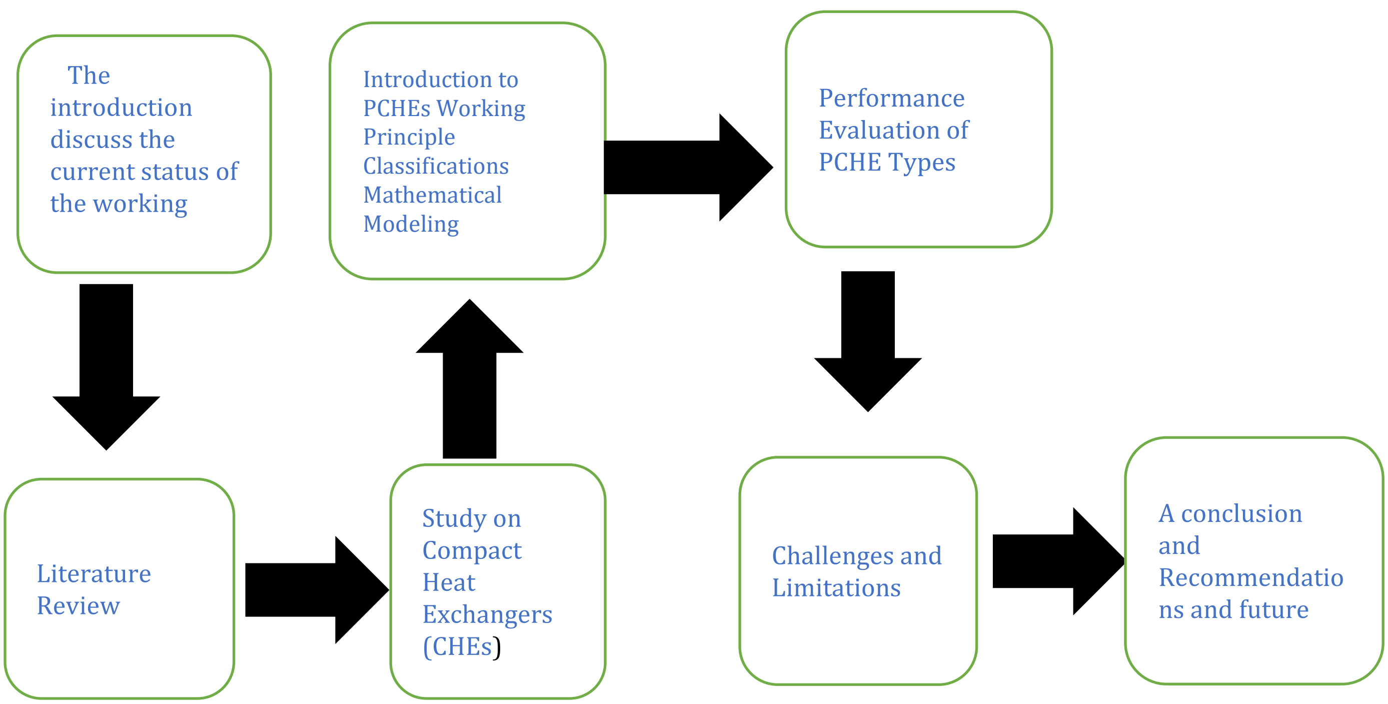


Figure 1: Flowchart of the research work

Chapter 2

LITERATURE REVIEW

Printed Circuit Heat Exchangers (PCHEs) can be described as an advanced type of high-performance heat exchangers designed to meet the challenging needs of high-pressure activities, reduction of space footprint and high-performance thermal work. The design of these machines is carefully designed to work under extreme conditions of operations and stability of the mechanisms and high efficiency. The development of PCHEs dates back to the field of nuclear energy where high thermal efficiency and mechanical strength and resistance to high tempos were in high demand and imparted to the components; important to the development of reactor technology. Their use has over the years been extended to many industries other than the nuclear industry, including the aerospace, hydrogen energy systems, cryogenics, and waste heat recovery [7]. The fundamental architecture of a Printed Circuit Heat Exchanger (PCHE) is characterized by an arrangement of microchannels that are meticulously chemically etched into metallic substrates and subsequently diffusion bonded to fabricate a compact and durable structure that is both miniature and robust. This pioneering methodology of diffusion bonding obviates the requirement for mechanical connections or welds, thereby resulting in a monolithic assembly capable of enduring considerable internal pressures and thermal variations. The incorporation of microchannels significantly amplifies the surface-area-to-volume ratio, thereby facilitating effective heat transfer between the working fluids with minimal complications. As a result, PCHEs demonstrate the capacity to achieve significantly elevated rates of heat transfer when juxtaposed with comparably sized shell-and-tube or plate-fin heat exchangers.

The seminal research conducted by Ishizuka and Oh constituted a critical advancement in the exploration of PCHEs. Their investigations concentrated on evaluating the thermal and mechanical performance of PCHEs functioning within high-temperature nuclear contexts, affirming that these exchangers display exceptional resistance to mechanical deformation and thermal stress. Their results substantiated the structural integrity of PCHEs for application in nuclear reactors, thereby establishing them as a feasible component for the forthcoming generation of high-temperature gas-cooled reactors (HTGRs). In the wake of this transformative research, Mylavarapu et al [8]. propelled the investigation further by utilizing Computational Fluid Dynamics (CFD) simulations to scrutinize fluid flow and heat transfer characteristics within PCHEs exhibiting diverse channel configurations. Their inquiry illuminated the impact of channel geometry on the thermal-hydraulic performance of PCHEs and proposed strategies for optimizing channel geometry to achieve an equilibrium between pressure drop and heat transfer efficiency.

These simulations enhanced understanding regarding flow distributions, turbulence effects, and local temperature gradients within microchannels.

Significant advancements in the manufacturing processes have notably contributed to the enhancement of PCHE performance. Kim et al. [1] [9]. explored the applications of additive manufacturing (AM) and laser machining in the fabrication of PCHEs with intricate and unconventional channel designs. These innovative solutions facilitate the realization of complex geometries which are otherwise challenging to produce through conventional chemical etching methods. Furthermore, additive manufacturing permits precise control over dimensional tolerances, ensuring uniformity in channel spacing and wall thickness.

This augmented accuracy is directly correlated with enhanced heat transfer efficiency and improved mechanical reliability. Furthermore, additive manufacturing (AM) offers opportunities for the creation of multi-material printed compact heat exchangers (PCHEs), wherein various alloys can be combined to achieve superior thermal resistance and mechanical strength. In this context, emerging manufacturing technologies are poised to revolutionize the fabrication processes of next-generation PCHEs. Quantitatively, significant advancements have been achieved in the modeling of the behavior of printed circuit heat exchangers (PCHEs) across a range of operational conditions. In an investigation carried out by a cohort of researchers, turbulence models, including the k-epsilon and Shear Stress Transport (SST) models, were utilized to accurately characterize the flow and thermal distributions within the microchannels. These computational simulations are pivotal in predicting temperature fluctuations, pressure drop occurrences, and localized heat transfer coefficients, thus enhancing the optimization process of PCHE designs. to satisfy specific working fluid and thermal load requirements. These computational investigations have been complemented by experimental research conducted by Huminic [10].who assessed performance data under laboratory conditions. Huminic's experiments evaluated PCHEs utilizing various working fluids, including supercritical carbon dioxide (SCO_2), and demonstrated that the heat exchangers exhibited remarkable efficiency and performance under supercritical conditions due to the advantageous thermophysical properties of CO_2. This finding holds considerable significance for modern power generation systems utilizing SCO_2 Brayton cycles, which demand the presence of compact and efficient heat exchangers capable of operating at elevated pressures and temperatures. Beyond nuclear and supercritical CO_2 systems, PCHEs have also found applications in cryogenic systems and hydrogen refueling infrastructures. Lee et al. additionally conducted investigations into the performance of PCHEs in hydrocarbon liquefied natural gas (LNG) applications at extremely low temperatures [11]. Their research revealed that PCHEs were adept at providing substantial thermal resistance and elevated heat transfer rates, even at cryogenic temperatures. Diffusion bonding is particularly crucial in cryogenic applications as PCHEs offer robust structural integrity, thus mitigating fatigue issues associated with thermal cycling, which are prevalent in traditional heat exchangers. Likewise, Yoo et al [12]. compared the experimental Nusselt number of a PCHE prototype with theoretical predictions and established a strong correlation across a range of Reynolds numbers. These validations confirm that PCHEs consistently operate predictably under varied operational conditions, rendering them exceptionally versatile components in both low-temperature and high-temperature applications. The ongoing development of PCHE design has necessitated extensive research focused on the optimization of channel geometries that maximize thermal efficiency while minimizing pressure losses. Numerous studies have explored innovative channel configurations, including zigzag, sinusoidal, wavy, and trapezoidal shapes. For instance, Xie and Chueh compared straight, zigzag, and serpentine channels, discovering that the zigzag configuration yielded superior heat transfer performance due to its elevated turbulence levels albeit with a slight escalation in pressure differential [13].

The zigzag-type printed circuit heat exchanger (PCHE) was subsequently advanced by Shenghui Liu et al. who systematically optimized its bending angles and curvature to enhance the overall efficiency of supercritical carbon dioxide Brayton cycles. Their configuration demonstrated an improvement of 2530 percent in heat transfer efficiency when compared to conventional designs, thereby highlighting the critical role of geometric optimization. In a similar vein, Ling-Hong Tang et al [14]. conducted an investigation into the thermal performance of PCHE under diverse operational conditions.

, discovering that optimal performance was achieved with zigzag channels operating under moderate flow regimes, wherein secondary vortices facilitated mixing with minimal energy loss. Recent investigations have also proposed alternative design geometries, such as wavy and trapezoidal channels, which have demonstrated favorable outcomes near the pseudo-critical point of natural gas. These configurations significantly improve turbulence and mixing, as well as augmenting local heat transfer coefficients by up to 53 percent compared to linear channels. This enhancement arises from the introduction of flow disturbances via the non-uniform wall structure, which amplifies convective heat transfer. Furthermore, these channel designs can achieve improved thermal performance with negligible compromises on pressure drop, making them advantageous in scenarios where both efficiency and compactness are requisite attributes of the channel.

Chapter 3

Study on Compact Heat Exchangers

Compact Heat Exchangers (CHEs) are special thermal management systems that are meant to receive very high quantity of heat in a small space. In comparison to the other standard heat exchangers, the CHEs are designed in such a way that they take advantage of the ratio of surface area to volume, which is effective in maximizing the heat transfer and the area required to install the loop is intact in a small low compared to the conventional [15]. For this reason, CHEs become the perfect solutions for the applications where the limited space, weight, and high requirements for thermal efficiency are the constraints. Typical CHEs contain a packed array of tiny channels, fins, or plates that increases the surface area available for heat exchange at a minimal volume requirement. Consequently, they facilitate improved heat transfer performance and lower thermal resistance, thus providing a good-working case for small/modular units. There is a need for tough performance in extreme conditions that has resulted in a high rate of importance of CHEs such as in high-pressure and high-temperature environments. Concentrated solar power (CSP) systems, supercritical co_2 Brayton cycles, aircraft thermal control systems and high-performance electronics cooling are just some of the key examples where CHEs succeed. In these contexts, CHEs contribute to greater thermal efficiency simultaneously making it easier for the system to be smaller, lighter in weight, and provide better mechanical reliability.

Further, their size allows them to have rapid thermal response times and fit easily into complex system designs with reduced space and installation problems. They can also be built in a geometry-specific form for different fluid properties and operating conditions, providing versatility for emerging energy and aerospace technology.

3.1. Types of compact heat exchangers

The compact heat exchangers (CHE) characterized by elevated area density represent advantageous alternatives [16]. Conventional shell and tube heat exchangers exhibit an area density of approximately 100 m^2/m^3 [19], whereas the CHE demonstrate values exceeding 700 m^2/m^3 [2]. This metric, derived as the ratio of the heat transfer area to the volume of the apparatus, is referred to as compactness. Additional criteria for classifying a heat exchanger as compact include a hydraulic diameter of less than 6 mm for gas fluids or an area density of 400 m^2/m^3 for liquid or multiphasic fluids [19]. An augmented area density enhances the heat transfer per unit volume, as can be inferred from Equation 1.3, albeit this also leads to an increase in pressure drop. The reduced hydraulic diameters of the CHE frequently induce flow regimes that reside within the laminar region. Among the various designs of CHE are the plate heat exchanger (PHE), the plate fin heat exchanger (PFHE), and the printed circuit heat exchanger (PCHE), which are categorized as plate types distinguished by diverse manufacturing technologies and flow passage configurations..The study of CHEs and their usability in various aspects like power plants, nuclear reactions, automobiles, aerospace among others has grown rapidly due to the necessity of energy minimization, capital expenditure reduction, and component flexibility. The statement is especially true in case of high-temperature applications.

3.2. Plate heat exchanger (PHE)

The Plate Heat Exchanger (PHE) employs plates to facilitate the transfer of thermal energy between fluids, which dictate the configuration of the flow channels. These plates are affixed to a supporting framework and are compressed by two clamped plates that provide structural integrity. Each plate is equipped with nozzles through which the fluids traverse; one fluid is propelled by the even-numbered plates, while the other is propelled by the odd-numbered plates. Figure 1.3 elucidates the configuration and components of a PHE. Its area density spans from 120 to 660 m^2/m^3, with hydraulic diameters ranging from 2 mm to 10 mm [19]. A notable advantage of the PHE is its diminished spatial requirements. For an equivalent effective heat transfer area, the mass and volume of the PHE constitute approximately 30% and 20%, respectively, of those associated with shell and tube heat exchangers [20].

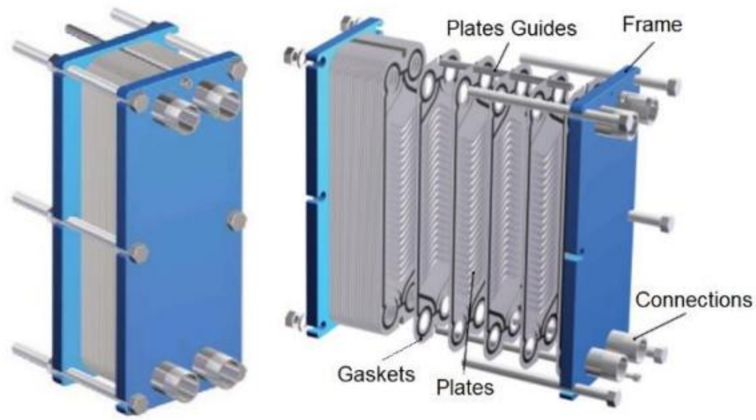


Figure 2:Plate heat exchanger (courtesy of Comeval) *[17]*

The Plate Fin Heat Exchanger (PFHE) features a compact architecture characterized by alternating layers of corrugated sheets or fins, interspersed with parting sheets and bordered by side bars, thereby engendering a series of finned chambers. The assembly of the fins and parting sheets is accomplished through brazing, resulting in a singular, rigid core structure. The plates and fins serve to segregate the fluids and establish distinct flow channels, as delineated in Figure 3. The fins perform multiple functions, including augmenting the effective heat transfer surface area and enduring the design pressure at the specified temperature, thereby acting as an integral structural component. The compactness of a standard PFHE typically ranges from 700 to 800 m^2/m^3; however, contemporary designs have achieved densities as high as 5900 m^2/m^3 [19]. Its volumetric capacity can approximate only 10% of that found in a shell and tube heat exchanger possessing a hydraulic diameter of 19 mm [14]. Among the various PFHE configurations, the offset strip-fin heat exchanger (OSFHE) is recognized for its superior thermal-hydraulic performance [18]. While the PFHE design exhibits certain constraints, the most notable limitation pertains to the presence of brazed joints, where the likelihood of failure is markedly increased. For applications subjected to exceedingly high temperatures, it is imperative that the fin thickness be relatively substantial to accommodate the requisite structural loading conditions.

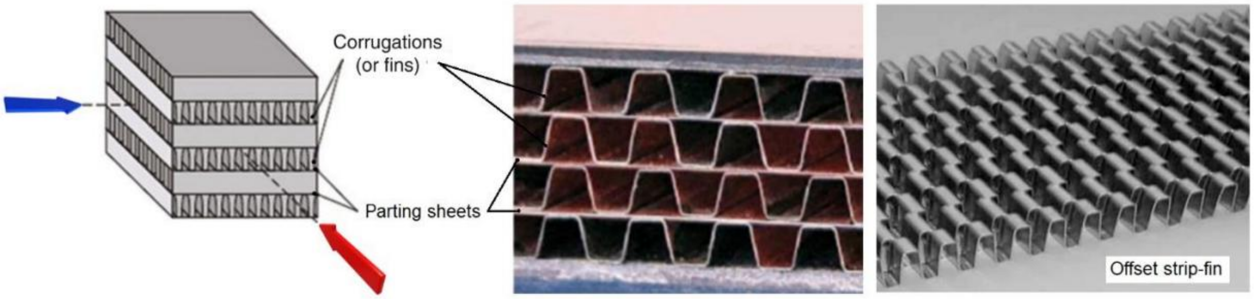


Figure 3:Plate fin heat exchanger functioning and offset strip-fin structure *[19]*

The first application of the double-wall PHE in a published work took place in 1986, intended to use the amalgamation of the fluids very carefully since the risk of contamination or the possibility of unwanted chemical interactions were possible [20]. This new design will replace the single plates with two plates of similar kind that in case of a failure where there would be any leakage it will be directed to the interstitial space between the two plates and then it will be discharged to the external environment so that there will be no leakage that is not conspicuously detectable to the heat exchange.

3.3. Printed Circuit Heat Exchanger (PCHE)

The construction process which is applied in production of Printed Circuit Heat Exchangers (PCHE) differs fundamentally with that of PFHE. The important element of PCHE is a metallic plate, which is characterized by flow channels, [8]. which are generated through the process of photochemical etching. Then, these etched metallic plates are in large quantity installed and stacked in a pre-established layout, and linked to each other by means of diffusion bonding at the essence of the PCHE is created in the circumstances of high temperature and pressure. Under the diffusion bonding technique, any melting or other forms of related imperfections are prevented and the bonding pressure is significantly weaker than the yield strength of the material. This method entirely eliminates plastic deformation of bonded materials. Due to diffusion bonding, PCHEs are strong and stable in extreme conditions in terms of thermal and pressure, which makes them applicable in a broad spectrum of applications. In most cases, PCHEs are classified into two groups namely those having continuous flow channels and others having discontinuous flow channels. The straight, zigzag, and wavy channels are the most widely used continuous forms and S-shaped and airfoil fin of a discontinuous form is common. As the attention to the Supercritical Carbon Dioxide (SCO_2) Brayton cycle is also increasing, the utilisation of PCHEs is becoming a more significant topic of investigation as well, [11]. the compact heat exchanger, the concern of paramount significance to the above-mentioned process has also started to be studied. PCHE is among the numerous variations of compact heat exchangers and this exchanger type has significant potential, and thus, this paper would meticulously examine and collect the developments made in the study pertaining to PCHE.

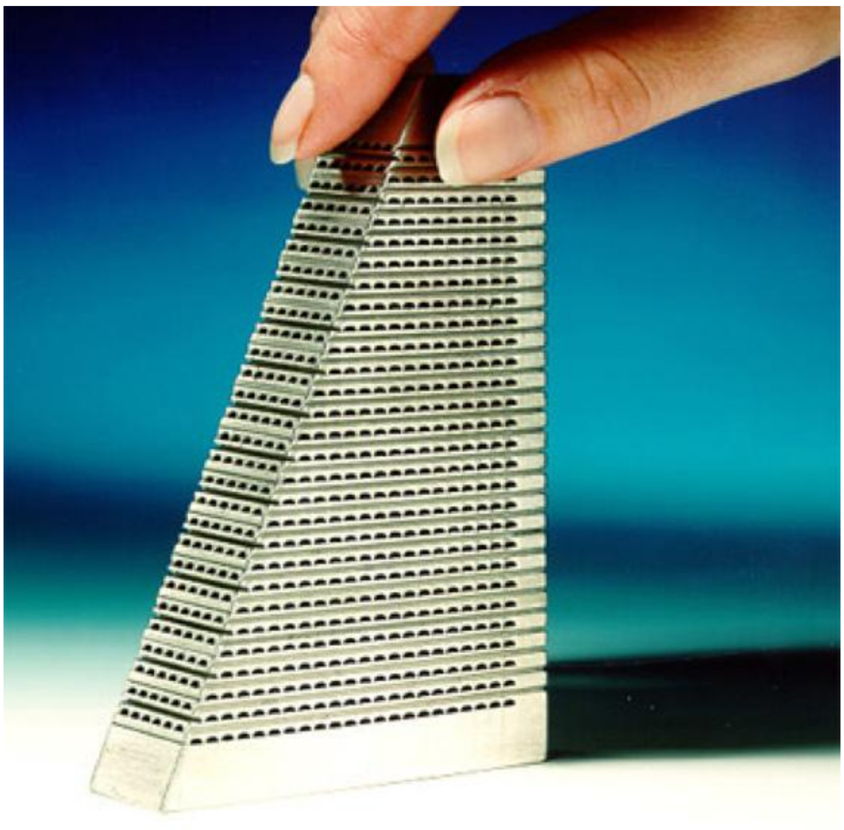


Figure 4: A printed circuit heat exchanger created by Heatric in the shape of an industrial prototype *[18]*.

3.3.1. PCHEs with Straight Channels

The channel geometry in printed circuit heat exchangers (PCHE) that is the easiest is straight channel, and the channel geometry paradigm in that context. Due to the constant flow of fluid in the direct channel, PCHEs using straight channels can attain very small pressure drops, hence, straight channel configuration allows the optimum hydraulic performance. PCHE geometry and development are not separable with empirical data concerning thermal and hydraulic performance of this type of heat exchanger. On cooling the conditions, it was found by experiment that the net heat removal performance is rather efficient around the pseudo-critical region. A work done by Baek et al [21], observed that the heat exchange of PCHEs used in low temperature activities is primarily influenced by the thermal conductivity of the axial path heat exchange in low Reynolds number regimes. Besides this study, other studies have been conducted in an attempt to examine the performance of PCHEs in high-temperature and high-pressure conditions. The dimensionless parameters j and Nu were measured at a large range of operating temperatures, pressures and flow rates using a high temperature helium testing facility [22]. These experiments showed that PCHEs show an improved overall performance with increase in the operating pressure. In addition, thermophysical properties of carbon dioxide in PCHEs comprising of straight channels were studied in three different operating regimes: trans-critical (cooling of supercritical to create a sub-cooled liquid sample), near-critical (cooling a supercritical gaseous state to a supercritical liquid), and far-critical (cooling a supercritical gaseous state within the supercritical liquid). Numerical simulations have a number of benefits as compared to experimental methods, such as reduced cost, flexibility to a variety of conditions, high-speed generation of solutions, and capability to generate large volumes of data. As a result, in cases where the experimental setups cannot be conducted, numerical simulation can be considered the most prevalent methodology of examining the thermal design and economic assessment of the cross-flow printed circuit heat exchangers (PCHEs). Convective heat transfer of supercritical carbon dioxide (SCO_2) through horizontal tubes in a cooling condition has been studied theoretically; the heat flux has been shown to be an important factor in the spatial positioning of the fluid characteristics. It is also noted that localized heat transfer decreases in the conditions of low mass flux or large heat flux which is explained by the effects of buoyancy which change the distributions of cross-sectional parameters and increase the impact of secondary flows. Even in the case of ignoring buoyancy, the specific heat change of the working fluid remains a significant factor in the definition of heat transfer behavior. Notably, the derivation of effective correlations has been a major aspect of the evaluation of the thermal-hydraulic performance of PCHEs, which has been well covered by many scholars using both numerical modelling and experimental studies. The review of obtained data and their calculation have given rise to several applicable correlations, which positively influence the further evolution of PCHE. The numerical studies of the hydraulic characteristics of PCHE at lower Reynolds number regimes ($Re < 150$) have been carried out and the simulations findings confirmed by experimental findings. Eventually, a numerical correlation was established between the friction factor (j), The forced convective heat transfer properties of supercritical SCO_2 were derived by developing a correlation that used the time-averaged properties of a probability density functions (PDFs). It was constructed to have a mathematical model in which the mathematical investigations were done over a long period. This model took into consideration various PCHE settings, such as crossed, parallel and counter current settings. In solving the heat conduction equation of straight-channel PCHE fins, [11]. Used numerical calculations to find the longitudinal temperature distribution across the fins, and the equations used to calculate fin effectiveness.

Combined simulation and experimental findings led to the development of cooling correlations of the SCO_2 taking into account the buoyancy effect and inner tube diameters. Also, the mean Nusselt number and friction factor of single channels in PCHEs at supercritical nitrogen were presented in numerical form, which gave accurate prediction properties. The semi-empirical relationship that was made possible by the application of the characteristic correction methods which are built on PDF methods helped to better understand SCO_2 forced convection due to instantaneous changes in turbulent temperatures.

One of the factors that impact PCHE performance is the flow pattern, so there is need to introduce correlations in correcting non-uniformity in both thermal and hydraulic performance. Moreover, correlation was formulated to encompass the influence of buoyancy and thermophysical property variations. At the current time, PCHEs have been regarded as a possible alternative to the cooling systems of the new generation nuclear reactors. Thermal efficiency of PCHEs at steady-state condition is important in the overall performance of the primary cooling system and thus is of major concern to researchers who are investigating the dynamic response of these heat exchangers in transient operating conditions.

Both experimental measurements and numerical simulations have been used to develop a dynamic model that is useful in predicting the equilibrium and dynamic performance characterization of straight-channel PCHEs. The model however fails to explain the deviations between the helium outlet temperatures predicted numerically and experimental values completely which are credited to the fact that thermal losses exist inside the heat exchanger. More so, embedding the built-in PCHE models into the overall design of power units enables complex numerical simulations [23]. The outcomes based on dynamic simulations revealed the fact that the thermal expansion, which occurs as the density rapidly subsides along with the growth of system pressure, results in sudden changes of temperature and thermal stress, which can pose a threat to the operation integrity of the system. The original circumstances under which heat exchangers undergo in different working situations show inconsistency, one of them being divergent heat flux and inlet temperature which always influence the effectiveness of PCHEs; hence, there are numerous researchers that have conducted corresponding studies. The consequences of simulation were calculated and examined in various heat flux conditions, the results of which demonstrate that an increased heat flux significantly limits the ability of heat transfer during heating operations, but has little impact on cooling operations. In addition, there were consistent findings in another study which focused on Experimental and numerical studies have been performed to a great extent to understand the forced convection heat transfer of PCHEs. Experiments investigating the cooling flow characteristics of supercritical carbon dioxide (SCO_2) in tubular designs showed that the pressure, mass flux and internal diameter have a different level of influence on the cooling behavior of SCO_2 in terms of heat transfer and the pressure drop. Numerical study of flow of SCO_2 in a straight channel in fully turbulent flow was also carried out and compared with the results of zigzag channel. A detailed three-dimensional numerical model was modified including the inlet effects, the conjugate heat transfer effects, the thermophysical properties of real gas depending on the NIST data, and the effect of buoyancy. Findings showed that the region of inlet leads to a steep decrease in the local heat transfer at the onset of the flow which is stabilized gradually as the length of the channel is increased. Also, the pressure gradient was found to have a positive correlation with the SCO_2 temperature. Based on the particular role of different boundary conditions in the dynamic response behavior and equilibrium time of the thermodynamic parameters in PCHEs, predictive models were developed with the help of neural network algorithms. The models have played a major role in the ongoing formulation of the complete SCO_2 power system dynamic modelling.

The other significant addition technique is that of surface interruption which is commonly applied to increase the heat exchange. This process periodically breaks down the building of the boundary layer, thus causing fluid mixing and continual eradication of the thermal boundary layer, which increases the total efficiency of heat exchanger. As a direct result, the introduction of turbulence-imposing geometries into the channels has taken the center-stage in the optimization strategy of the performance of linear-type Printed Circuit Heat Exchangers (PCHEs). The numerical simulation findings have validated the fact that the thermal performance of straight channels with different numbers of hemispherical pits evenly distributed along their length is enhanced with a corresponding rise in pressure loss. Based on these observations, scientists have also been encouraged to further optimize the geometrical arrangement of straight channels by changing the shape, size and location of the placed structures with an aim to ensure a higher thermal efficiency at minimum pressure drop.

Researchers have primarily focused on the thermal transfer properties of the core architecture of printed circuit heat exchangers (PCHE); nevertheless, the increase in pressure loss resulting from non-uniform flow rates can detrimentally affect the overall performance of the PCHE, thus causing the heat exchanger to deviate from an optimal operational condition and leading to diminished efficiency. The inlet manifold of the PCHE plays a pivotal role in influencing flow distribution; consequently, a select cohort of researchers has embarked on studies aimed at refining the design of the PCHE inlet manifold. By employing principles of streamline flow, they have introduced an innovative hyperbolic inlet configuration that significantly improves the uniformity of flow distribution and the overall effectiveness of the PCHE. Moreover, the irregularity of flow can be alleviated through modifications to the core length.

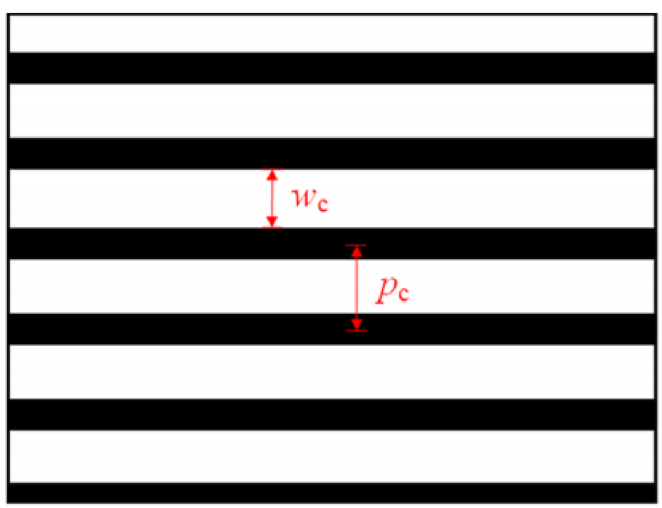


Figure 5: straight channels *[24]*

3.3.2. PCHE with Zigzag Channels

In contrast to Printed Circuit Heat Exchangers (PCHEs) that incorporate discontinuous fins, those that utilize rectilinear and zigzag channel configurations display a more streamlined methodology in the chemical etching of flow channels and exhibit superior structural integrity following their assembly into the core through diffusion bonding. Presently, a substantial group of scientists is busy with the efforts directed to investigate and optimize the performance qualities of PCHEs with zigzag flow lines [25]. The engineering paradigms are the amalgamation of the research and development practices of heat exchangers to achieve improved energy efficiency conversion. Considering the numerous operational environments that heat exchangers have to face in the context of engineering practice, it is necessary as well as vital to experimentally and numerically evaluate the performance of PCHEs in a range of working conditions. Nikitin et al. also performed an empirical study using the supercritical carbon dioxide (SCO_2) To obtain an empirical correlation that predicts the local coefficient of heat transfer and pressure drop as a function of Reynolds number (Re), an experimental loop circuit was employed as well as taking into account the changes in inlet temperature, pressure, and mass flow rate. Similar to the research done by experimented on the laminar flow regime of helium at the range of $350 < Re < 1200$ using the KAIST helium test loop. Based on their experiments involving PCHEs at different conditions of inlet, they came up with a universal friction factor correlation and a generalized correlation of the Nusselt number [12]. At the same time, a numerical simulation in three dimensions was also performed and this demonstrated excellent consistency with the experimental results, causing additional introduction of a local correlation of pitch-averaged Nusselt number. They also showed that the derived correlations using system analysis software are quite effective in system level analysis. Later researchers enlarged this study through introducing more auxiliary correlations in the context of computational fluid dynamics (CFD). They compared CFD results with experimental measurements and developed correlations over a wide range of Reynolds number, 2,000 - 58,000, conducted a detailed analysis of the correlations of the Nusselt number and the friction factor of cold and hot channels synthesizing the influence of channel geometry and inlet conditions on PCHEs with zigzag structures. Although these developments have been made, the experimental studies of the use of Printed Circuit Heat Exchangers (PCHEs) as precoolers in the Supercritical Carbon Dioxide (SCO_2) Brayton cycle are limited because of the experimental limitations and technical difficulties. In single research which examined the effect of inlet Reynolds number and inlet temperature on the thermal-hydraulic properties and the overall efficiency of the heat exchanger, researchers used empirical techniques to investigate the effect of the inlet Reynolds number and the inlet temperature on the heat exchanger. The findings showed that an increment in the inlet water temperature would cause a decrease in pressure losses but at the same time, reduce the efficiency of the exchanger in heat transfer and performance [26]. On the other hand, the larger water inlet Reynolds numbers or smaller SCO_2 inlet Reynolds numbers may improve the performance of the heat exchanger. The numerical simulations also indicated that the flow and thermal fields inside PCHEs are not fully developed when their thermal conditions are at high levels. However, with these high conditions the profile of dimensionless velocity and temperature generally stabilizes at the second pitch like the flow conditions of low thermal conditions. Based on the obtained empirical data, a novel correlation was developed that related the pressure drop and heat transfer behavior in zigzag channel with rounded bends. This was established to be a similar trend in performance in linear configurations, circular, and conduit, the zigzag channels depict significant benefits in the transitional flow conditions. Using detailed examination of localized hydrothermal features, it was determined that fully established flow conditions were not achieved inside the PCHE caused by periodic perturbations of flow at every bend of the zigzag channel, which explains the important differences. The correlation between the local and global heat transfer coefficient of the PCHE was created. Moreover, [27] it has been noted that fluid temperature and heat flux distribution are non-uniform across the longitudinal direction of the channel flow and the profile temperature of the flow has specific undulating properties. The predictability of the computational fluid dynamics models is highly reliant on how valid the assumptions used in them are, since to a large extent, the validity of the assumptions used is what defines the level of agreement between the results obtained in the numerical simulations and the experiment. Here, the horizontal and vertical arrangements of Printed Circuit Heat Exchanger were fully examined using the KAIST helium-water testing loop which was supplemented by thorough numerical modelling. It is interesting to note that the numerical data of the pressure drop in the vertical setup had a very good correlation with experimental values. On the basis of the assessment of PCHE performance, both the friction factor (f) and Nusselt number (Nu) correlations were suggested. Further studies were carried out to formulate and check correlations of Nu and f under laminar flow regimes in PCHEs with semi-circular zigzag channels. [28] study aimed at modelling and studying the dynamic behavior of

PCHEs with zigzag channels at different helium inlet temperatures and changes in helium mass flow rate (steps). It was done to show how the dynamic model can forecast steady-state and transient responses of PCHEs depending on experimental validation. Just like linear structures, the structural geometry of the heat exchanger such as the cross-sectional shape and pattern of channels can also perturb the flow of the working fluid to a certain level, thus, influencing the thermal behaviour of PCHEs that uses zigzag channels. Comparative study of PCHEs using zigzag channels in various configurations and cross-sectional geometries showed that the effectiveness of the exchanger and the coefficient of friction is in positive co-relation with the heat transfer surface area of the channel. In addition, out of four layouts of proposed flow channels, some of the layouts offered the best performance as they offered a better balance between the improved heat transfer and the low pressure drop as illustrated in the comparative studies of the simulations. The effects of the geometric parameters on the performance metrics of PCHEs were the systematic study of a different study. The results showed that the optimal performance was at a relative cold channel angle of about 110 degrees and the dimensionless pressure drop declined steadily with an increase in the channel angle. Additional studies were also done on various design configurations of PCHE architectures that can act as secondary heat exchangers with FLiNaK-helium as a working fluid. Optimization techniques were used to evaluate the performance of PCHEs at a series of Reynolds numbers (Re) and in a series of geometrical configurations. The findings revealed a very sensitive serrated structural design of PCHEs that causes a strong impact on performance parameters with a broad range of Re values [8] .Therefore, the empirical correlations that were already available were discovered to be of limited applicability, which led to the creation of specific sets of correlations that can be applied to particular ranges of Reynolds number. The case of bending angle affecting the thermal and flow peculiarities of zigzag channels was also examined using field synergy and entropy generation principles. The numerical simulations gave solid support that the increase in the heat transfer performance with the increase in the bending angle is present, but the increase is accompanied by an increase in the pressure drop under some angular dimensions. Moreover, it was discovered that reverse and secondary flow effects have pronounced effects on the localized operational behavior of serrated channels. In addition to this preliminary research on inlet conditions, channel geometry and arrangement effects in PCHEs, it is still necessary to improve the PCHEs further to reach high thermal and hydraulic efficiency. In this respect, the zigzag-channel PCHEs, elliptical aspect ratio and cold channel angle optimization was achieved using Response Surface

Methodology surrogate modelling technique coupled with a genetic algorithm, and offered a fine-tuning way of making any design improvement in future. Both high-temperature and low-temperature regenerators were designed and simulated in terms of zigzag channels in both Brayton cycle systems of 100 kWe SCO_2 recompression and the validity of these two models supported by comparisons to the experimental data of small exchangers used in systems of 100 kWe. Finally, the findings of the optimized design indicated that the cold plates had fewer metallic mass than the two hot plates and the high-angle channel designs, which made them a more favorable solution to large-scale implementation. In the particular example of zigzag channels, it was found that there was a significant pressure drop as a result of separation of flow and reverse flow at the bend points which means that the hydraulic efficiency of the zigzag configuration is significantly low. In this regard, a new printed circuit heat exchanger (PCHE) was proposed by the introduction of a new type of zigzag structure with straight channels to overcome the significant effect of bend points on pressure loss. It was also found using computational simulations that the pressure drop could be reduced by inserting straight channels of length 0.5 mm and 1 mm, and the heat transfer performance was comparable to that of the traditional zigzag channel. The implication of this observation is that the addition of straight channel segments in the zigzag design has the potential to reduce the separation of flows and reverse flow around the bends and hence enhance hydraulic performance without affecting thermal performance. The paper therefore offers a much-needed conceptual framework towards improvement of the performance of PCHEs with zigzag channels.

The further concept of design involved further incorporation of straight channels in zigzags PCHEs, which led to the creation of a new elliptically etched zigzag design using elliptical slots on both sides. Numerical computations performed later using this proposed design showed that in contrary to the traditional zigzag channel with a semi-elliptical cross-section, the new design had enhanced heat transfer capacity with a corresponding increment in pressure loss as the height of the elliptical slot went up. The literature dedicated to PCHEs that have zigzag flow channels is extensive. In addition to the investigations mentioned above, numerous scholarly publications have covered the complementary concerns like evaluation method and sensitivity tests. The evaluation method that was used to measure the overall heat transfer performance in different operating temperatures and pressures was based on operational points. Sensitivity analysis then was done by taking into consideration primary factors as well as the interaction of two factors. The outcome of simulations showed that parameters of hydrothermal performance of zigzag-channel PCHEs were the most sensitive to changes in channel bending angle, the curvature radius of bends, mass flow rate, channel width. Also, zigzag-channel PCHEs were analyzed by fluid-structure interaction (FSI) through the finite element analysis (FEA), providing a more profound understanding of structural and thermal behavior of such complicated systems.).

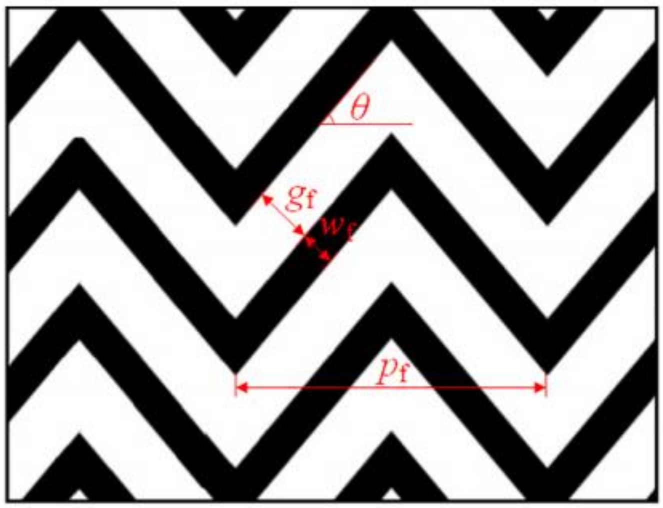


Figure 6:zigzag channels *[29]*

3.3.3. PCHE with S-Shaped fin channel

A popular arrangement, used in printed circuit heat exchangers (PCHEs), is the S-shaped shape with its discontinuous flow path. It is commonly thought that this design was developed out of the sinuoidal design in order to reduce the occurrence of reverse flow and to reduce the creation of low-momentum zones at the ends of the channel. To study the thermo-hydraulic performance of a PCHE with an S-shaped fin channel, its original idea was an experiment and numerical analysis of its behavior. Experiments have shown that the redesigned PCHE was 3.3 times smaller in volume, its pressure loss at the CO 2 side was reduced by 37 percent, and its pressure loss at the water side was reduced by ten-fold compared with conventional heat exchangers that are utilized in domestic hot water systems [30]. Response surface methodology was used in optimization of S-shaped fin geometry with the help of genetic algorithm.

According to the results of the simulation, a correlation among the heat transfer, as well as pressure drop, was determined regarding the optimized channel. It was observed that the pressure drops across the optimised S-shaped structure was 2.4 times less than that of a conventional zigzag channel which indicates the presence of better thermal-hydraulic performance, particularly at low Reynolds number. By varying the structural layout and the inclination angle of the fin, the channel arrangement giving the highest performance was obtained, maximizing the total PCHE efficiency. The optimized PCHE only experienced a tenth of the decrease in pressure in the zigzag channel structure mainly because of the fact that the fluid movement was smooth as opposed to reverse flow and formation of vortices which appeared at bends. Later massive studies examined the effect of different geometric parameters of S-shaped fins on the performance of PCHE. It was shown that the guide wing angle has the most decisive influence on the thermal and hydraulic performance of exchanger. Conversely, curvature at the head and tail of the fin had minimal effects on heat transfer and had a major influence on losses in pressure. Taking into account total thermal efficiency, pressure drop and mechanical stability, the best setting of guide wing, fin width and fin length was finally determined.

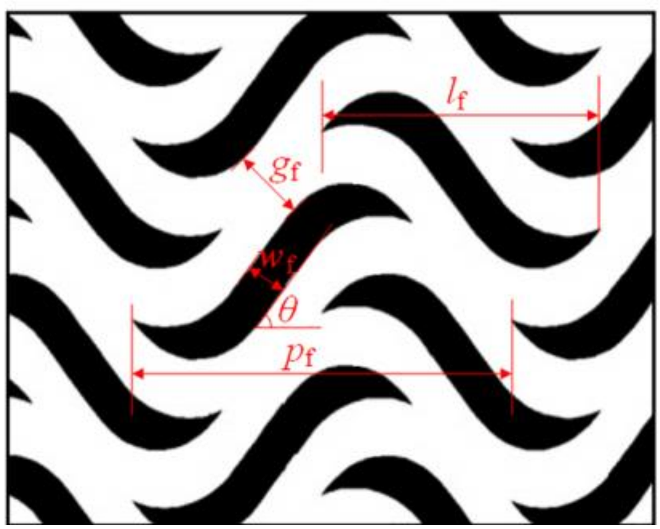


Figure 7:S-shaped fins *[31]*

3.4. Plate-fin heat exchanger (PFHE)

The concept of plate-fin heat exchanger (PFHE) is not novel: it has been used more than a century, originally in the automotive industry, and was fabricated using copper fins and brass pipes. They were used in the aerospace industry by the 1940s and since the 1950s have become part of the natural gas liquefaction processes- since the implementation of aluminum; which is much stronger and resists low temperatures. Unlike standard compact heat exchangers, PFHEs are constructed by a stacking of corrugated (fin) sheets and flat parting sheets which are bounded by side bars to form a system of finned flow channels (shown in Fig. 8). Joining of the fins and the plates is done by way of vacuum brazing to create a solid which is mechanically strong core [5] . The exchanger can be consisting of one or more cores depending on the design requirements. Important design aspects such as fin configuration, plate spacing, fin height and fin type are pre-optimized to achieve maximum thermal behavior. Also, PFHEs may be modified with ease to support a wide range of flow configurations, including cross-flow, counter-flow, cross-counterflow, or co-current configurations. Simple cross-flow designs are most often used in this category and are more efficient in moderate heat load situations and in gases with low pressure. Conversely, however, counter-flow designs are said to be more efficient in more challenging thermal use cases [32] and especially in cryogenic systems where a high temperature difference is essential.

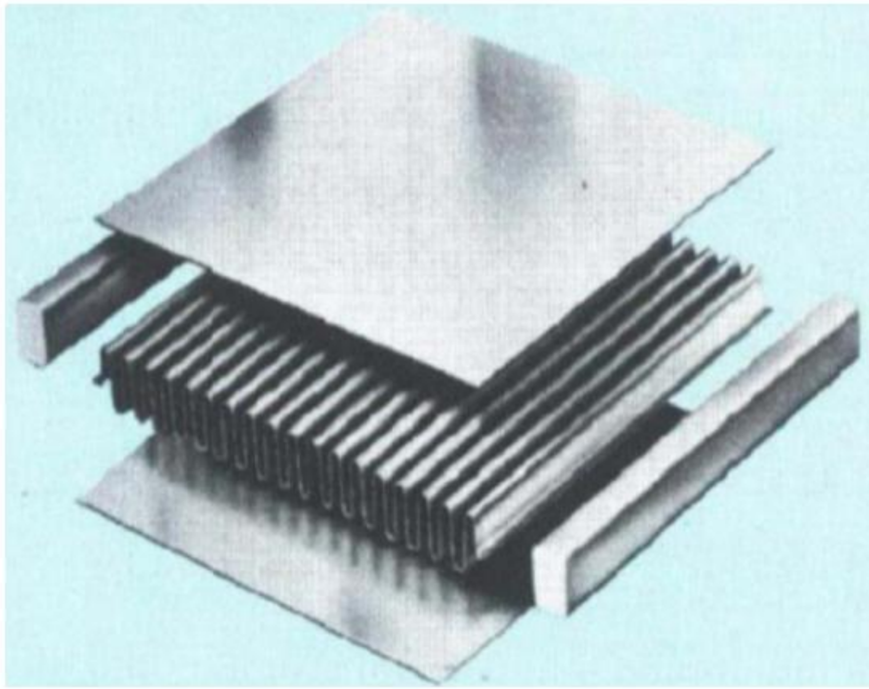


Figure 8:PFHE layer and additional building components in detail *[33]*

The heat transfer in PFHEs is conducted in the same way as in the previous heat exchangers the warm fluid exchanges the energy via the separating plate to the cold fluid in the next channel (refer to Fig. 8). The internal scheme, depicted with the help of Chart, reveals that the plate and the base of the fins are the major exchange area *[20]*. The flow paths are characterized by the fins as well as serve as structural elements to support pressure loads. The fins have two functions in terms of functioning: to increase the surface area, which exchanges heat, reduces the hydraulic diameter and thermal resistance, and they provide structural support against operating pressure. There are different fin geometries depending on the applications as summarized in Table 2. In the case of plain fins, the improvement of heat transfer is mainly due to the increased surface area as opposed to the visibly improved heat transfer coefficient. On the other hand, wavy, corrugated and herringbone fin designs are those that significantly enhance the rate of heat transfer because of the disturbances that they induce to the flow which enhance mixing and break the boundary layer. Interrupted fins e.g. serrated or perforated are also effective as they dislodge the flow occasionally by disruption. These distortions cause the thermal boundary layer to not keep on growing thick and thus retain a thin layer that has lesser resistance to heat transfer. This mechanism is useful even in the laminar flow conditions and low Reynolds numbers. By increasing Reynolds numbers, vortex shedding due to the surface interruptions also increases heat transfer. These vortices form at the leading edge of the fin and travel downstream and occasionally cause von Karmann vortices which extend into the wake between the fins. These types of flow structures support large scales mixing due to the constant supply of the fin surface with the cool fluid and the removal of the warm fluid. Nonetheless, pressure drop and pumping power demand increase is also a result of this process, mainly caused by the high level of skin friction and the reconstruction of the hydrodynamic boundary layer. Vortex shedding is also unsteady, which further increases energy loss due to dissipation of Stokes layer and form drag. This in turn influences the heat transfer ability and hydraulic losses simultaneously. Thus, to obtain an optimal PFHE design there must be a careful balance between these conflicting effects - the increased heat transfer with the reduced increase in the extra pumping power required because of flow resistance.

Table 1:Different fin configuration applications(general).

Corrugation	Description	Application	Features	
			Relative pressure drop	Relative heat transfer
Plain	Straight, triangular or rectangular	For general use	Lowest	Lowest
Perforated	Straight, with small hole	Most frequently used for any purpose	low	low
Serrated	Straight, offset half a Pitch-usually about every 3–4 mm	Loose fitting, and commonly used in air separation plants on	Highest	Highest

		low pressure gas streams.		
Herringbone or long-lanced serrated	Smooth but in waves of about 10 mm pitch to give a zigzag, or with length serrated. serration pitch	Usually used with low allowable pressure drop gas streams.	High	High

Chapter 4

INTRODUCTION OF PRINTED HEAT EXCHANGER

A printed circuit heat exchanger (PCHE) is currently recognized as one of the most prevalent varieties of heat exchangers due to its remarkable efficiency and compact design. PCHE finds application across various sectors, including waste heat recovery, power generation cycles, and energy storage systems that integrate phase-change material (PCM) with PCHE technology. The fabrication of PCHE can involve diffusion bonding plates that incorporate chemically etched micro-channels. In general, the flow channels of PCHE are significantly smaller in comparison to those found in traditional shell-and-tube heat exchangers. The manufacturing technique referred to as diffusion bonding entails the assembly of multiple metallic layers, which are subsequently subjected to a chamber environment characterized by elevated mechanical pressure, high temperatures, and a vacuum or an oxygen-free atmosphere. This process induces a phase transition at the atomic level, facilitating the fusion of the metallic layers. The adoption of diffusion bonding as a fabrication approach results in reduced plastic deformation since it negates the necessity for welding materials or heating metallic components to a molten state. Consequently, enhanced mechanical strength can be attained through the construction of multiple layers of metal plates within flow channels via diffusion bonding, as will be elucidated in subsequent sections. The production methodology for the microchannels, characterized by diameters ranging from two to three millimeters, similarly employs chemical etching techniques. In the context of multilayer configurations, the optimal manufacturing technique employed is chemical etching. This process is typically executed in a sequential fashion, encompassing stages such as cleaning, laminating, exposure, development, etching, and stripping. Such a methodology guarantees a considerable degree of precision in the formation of sharp edges, circular apertures, and linear boundaries [18]. This particular manufacturing process has contributed to the fact that the volume of the printed circuit heat exchanger (PCHE) is about ten times less than the conventional heat exchangers. Its small size allows reducing the size considerably thus reducing the flow channel distribution and increasing the surface area volume ratio thus simultaneously increasing the operational efficiency of the PCHE.

The area that is used to conduct thermal exchange has a significant impact on the effectiveness of a PCHE. The expansion of the area of the heat exchange is a strategy of vital performance and a feasible approach. The improvement of this is achievable by using overlapping channels, which is used to increase the heat flux by a factor that is proportional to the decrease in thermal resistance. The reduction of the distances between the hot surface and the cold surface is an absolute must to increase the thermal conductivity, maximize heat transfer per unit area, and eventually maximize efficiency of heat exchange. At the same time, heat exchanger design can be optimized by methods. In a study by Rao et al [25], different optimization strategies are investigated using different range of heat exchangers which consist of fin-and-tube, plate-fin, shell-and-tube configuration, where the objective-function is in the form of effectiveness, pressure drop, volume, uniformity, or entropy. Regarding the behavior of specific channel geometries in heat exchangers, e.g., linear or zigzag channels, and design variables, such as working fluids and temperature ranges, the behavior of many scholarly articles have been able to discuss relevant experimental and calculated problems. Individually, Annesh Metal showed that trapezoidal channels performed better as far as their thermo-hydraulic performance was concerned than sinusoidal or triangular channels, though at the price of greater pressure drop. It has been determined that the cross-sectional area of a rectangular configuration has a better effectiveness than the cross-sectional areas of other geometric shapes, trapezoids, circles and semicircles. The latter caused the decrease of the efficiency of the systems because of the rise of the energy consumption accompanied by the decrease of pressure. The performance was improved through the use of truncation of the right-angle channel bends using straight diagonal segments and curved bends in accordance with the performance evaluation criteria (PEC) of heat exchangers that is ratio of the Nusselt number to the friction factor Evaluations of exergy and entropy were used to determine the performance of the heat transfer [6]. Numerous investigations have explored the application of various working fluids, including supercritical carbon dioxide (SCO2) and helium, with the aim of augmenting heat transfer efficiency.

4.1. Classifications And Working Principle

4.1.1. Straight PCHE

PCHE in the present study to be researched on will consist of three predominant groups of elements, namely the four straight flow plates of the straight flow channels and the stream inlet and outlet tubes of the hot and cold streams on the other side of the flow channels as illustrated in Figure 9. The two hot flow plates plus two cold flow plates that will be vertical to each other (as in figure 2) are the four straight flow plates that are situated in the core of the PCHE. Hot and cold streams inlet temperature is set at 90 ℃ C and 30 ℃ respectively and the outlet temperature is set at one standard atmosphere pressure. No-slips and adiabatic have a solid boundary that is prescribed. The incoming stream of hot liquid at the inlet side passes through the hallway L to the flow channels 13 channels on one side and then through the hallway R to the other one to exit the PCHE as it can be seen in Figure 9 [34].The flow of cold fluid is in a completely opposite direction of the flow of hot. The flow channels of the hot and cold streams are to be made alternated so as to realize effective heat transfer of the plates of these channels. The mass flow rate size may affect the heat transfer performance. Figure 8 indicates that the fluid at the rate of mass flow less than 0.0282kg/s flow is uniformly entering the flow channels of 13. This has a beneficial effect on the development of a significant increase in the heat transfer in favor.40,41 In Figure 9, the has been shown to have a positive effect patterns of streamlines, the respective velocity flow contours of the inlet to the hallway to are presented the 13 flow channels, in which the flow division occurs on the opposite sides of the hall, as a result of unfavorable pressure gradient.42,43 Channel flow 9 and channel flow13 channel are relatively smaller compared to other channels This also implies increased non-uniformity of velocity distribution. Besides, the average velocity of all flow channel expansion and flow channel 13 to 9 and the variation of flow distributions along all flow passages augments with the mass flow rate increment. The less than 0.0282 kg / s of mass flow rate cases are the mass flow rate of the medium within the tube (a pipe) that conduit that is smaller than the configuration of the tube (pipe) itself. The less than 0.0282 kg/s mass flow rate instances are the mass flow rate of the medium in the tube (a pipe) conduit which is smaller than the arrangement of the tube (pipe) itself analogous level of mean velocities of each stream channels. Conversely, under some circumstances in which the mass flow rate exceeds 0.0282 kg/s, there are some severe flow separations at the edges of channels of flow No. 9 to 11 which cause losses of friction. The extreme negative pressure gradient is present on the left-hand side of the inlet and this makes the fluid move to the right-hand side, such that the distribution of pressure in the channels No. 1-6 will be greater than the distribution of pressure in channels flow 7-13. The high-pressure drop is the con chain of the frictional loses (see Figure 9 that reduces the heat transfer efficiency). Extreme decrease in pressure also implies that a lot of power is required in the fluid pumping.

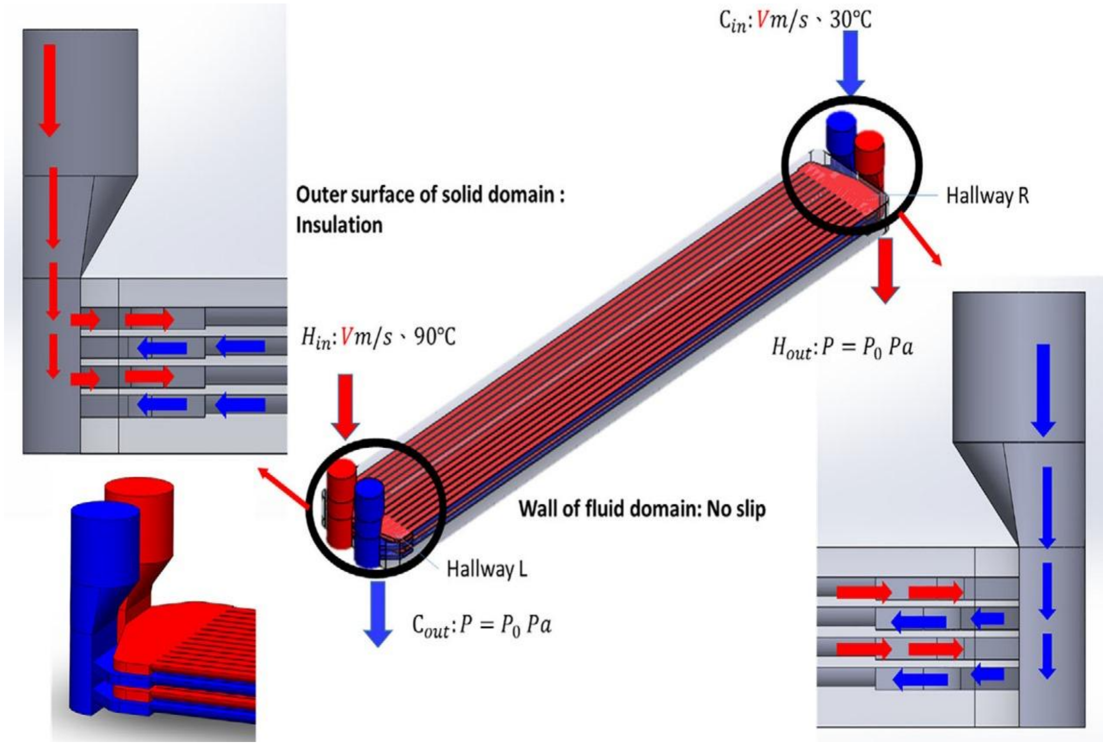


Figure 9: one-way channel heat exchanger functional diagram that shows the hot and cold part with flow direction *[34]*.

4.1.2. Zigzag PCHE

The behavior of the flow in a PCHE with straight microchannels is usually tending to a steady-state regime, also known as a fully developed flow, where the velocity and temperature profiles are almost unchanged in the direction along the axis. The pressure distribution on the flow and the wall shear stress become balanced in this state. Although this balance gives stability to the flow, it may also stop the mixing of fluids leading to reduced heat transfer enhancement. In order to eliminate this shortcoming and actively mix the flow, a zigzag channel arrangement is taken into consideration in this subsection as seen in Figure 10. Zigzag PCHE It has 13 semicircular flow channels with circular cross-section of 2.5 mm diameter and a bending angle of 150deg with the same boundary conditions as those of the straight PCHE. The four hot and cold plates are stacked at right angles and in the same alternating order as is the case with the straight-channel design mentioned above. Introduction of bends into the flow path is used to change the direction of fluid flow and create centrifugal forces. Although these directional variations cause the increase of frictional losses, as the extra resistance is introduced at the bends, they are also important in improving the capacity of heat transfer, as the fluid is stirred and the mixture by convection. This improvement can be seen especially in the first and third rows of the flow cross-section where velocity streamlines in the semicircular channels flow opposite to each other and this results in better heat transfer [35].The alternating flow motion causes the thinning of the thermal boundary layer and consequently an amplification of the local heat flux along the channel [46]. Moreover, the flow direction in the channels with the shape of a zigzag influences the curvature, which in turn develops unfavorable pressure gradients and the following pressure drops due to the interplay between centrifugal forces and the pressure field. This flow effect is commonly known as secondary flow or the Dean vortex that helps in the increased mixing of the fluid and heat transfer in the channel. In contrast to straight-channel PCHEs, in which the heat transfer is mainly done through conduction and convective flow, zigzag configurations develop such vortical flows, which basically produce localized heat sources or sinks, which do a huge job in enhancing the thermal effectiveness of the exchanger.

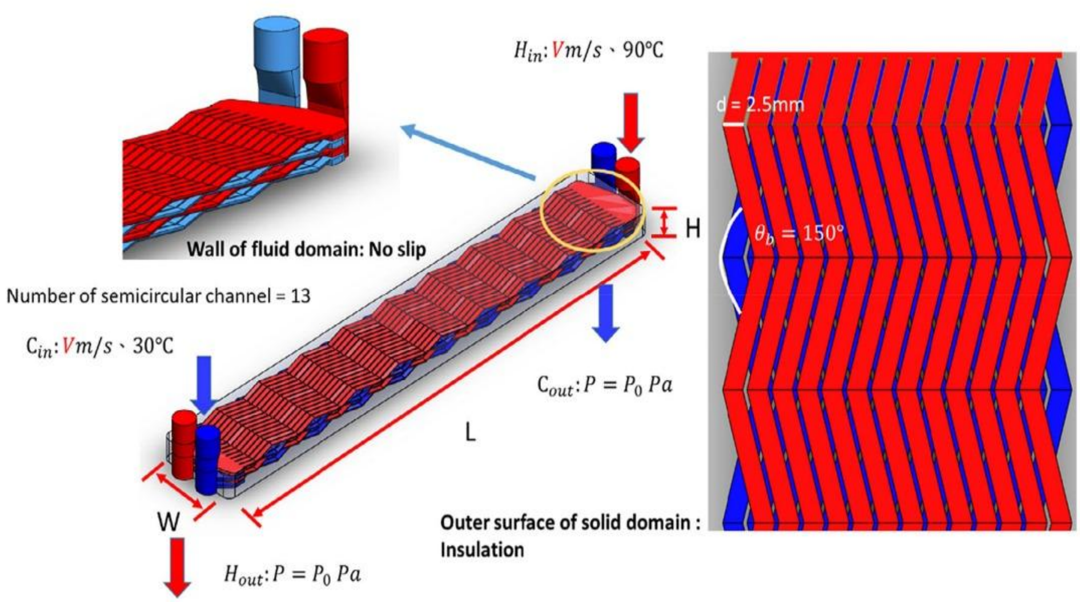


Figure 10:The several wavy flow channels of the heat exchanger with the hot fluid being in the red space and the cold in the blue space that the blue and red arrows are symbolized by the direction of the cold and hot liquid flows *[34]*.

4.1.3. S- shaped PCHE

A fluid path can be extended by a PCHE which could be beneficial in augmenting effective heat transfer area. In this regard, a serpentine PCHE is designed to attain this effect. In contrast to the already mentioned PCHE configurations, S-shaped PCHE has serpentine flow channels, as shown in Figure 11. The operating conditions that are employed in the current simulation however are identical to those employed in the other PCHE designs. The distribution of flow in the semicircular channels in S-shaped PCHEs is usually not uniform and Figure 7 shows a non-uniform distribution of flow in the semicircular channels. The flow slows down in some areas as a result of unevenly distributed velocity by the establishment of velocity boundary layers that forces the fluid to flow towards the outer channels preferentially. In other instances, the velocity of the flow tends to zero (Figure 11) resulting in inhibited heat flux transport and localized deteriorations in thermal performance. This is in line with previous studies by Ke [34], where sudden changes in the inlet diameter to semicircular channels led to separation and heterogeneity of the velocity in the channels. The complicated geometry of the S-shaped channels has a lot of contribution towards the frictional losses and as a consequence, causes a high pressure drop [51, 52] which is not favorable to the overall PCHE performance. The extension of channels into S-shaped therefore increases the pressure loss and uneven flow, so S-shaped channels are not so advantageous in terms of thermal efficiency, which is the reason why zigzag or straight-channel PCHE designs should be more preferable in practical cases.

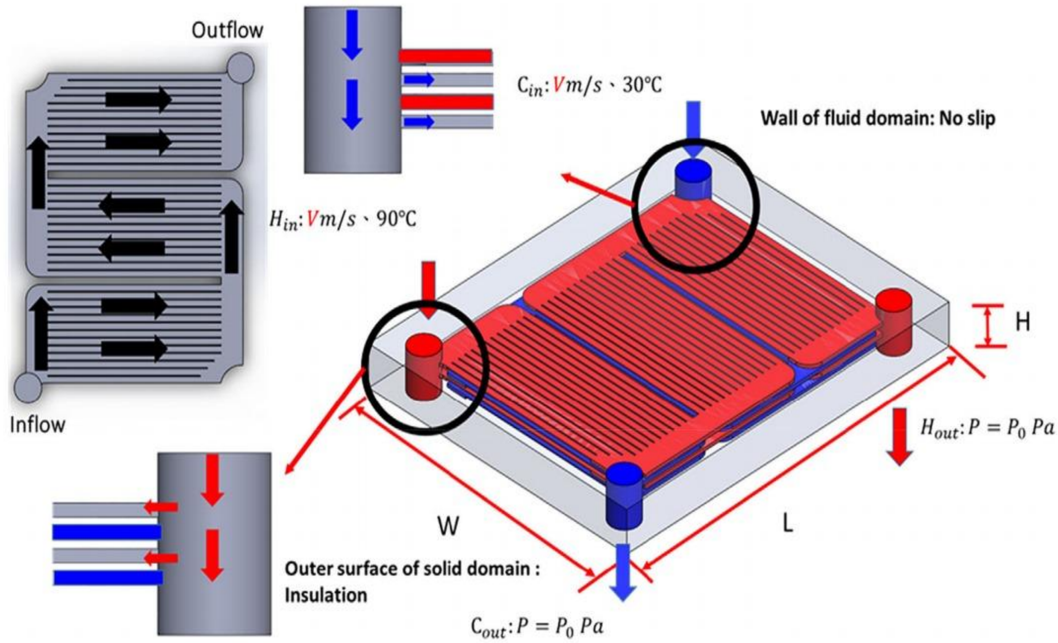


Figure 11:Schematic illustration of an exchanger containing serpentine conduits of S-shape revealing the reverse flow of fluids *[34]*.

4.2. MATHEMATICAL MODELS

4.2.1. Comptutational Fluid Dynamics

The computational fluid dynamics (CFD) commercial software Ansys CFX [107] was employed to model the straight, S-shaped and zigzag channel printed circuit heat exchanger (PCHE). This software facilitates the resolution of the governing CFD equations within three-dimensional geometric domains, as delineated in Equations 1-3 under steady-state conditions [107]. These equations correspond to the principles of mass conservation, momentum conservation, and energy conservation, respectively.

$$\nabla \cdot (\rho \vec{u}) = 0 \tag{1}$$

$$\nabla \cdot (\rho \vec{u}\ \vec{u}) = -\nabla P + \nabla \cdot (\tau) \tag{2}$$

$$\nabla\left(\rho\vec{u}\left(H+\frac{1}{2}u^2\right)\right)=\nabla\cdot(k\nabla T)+\nabla\cdot(\tau\cdot\vec{u}) \tag{3}$$

where u represents the velocity vector, H denotes the enthalpy, and T signifies the temperature. The computation of the stress tensor (τ) is performed in accordance with Equation 4

$$\tau=\mu\left[(\nabla\vec{u}\ +\nabla\vec{u}^T)+\frac{2}{3}I\nabla\cdot\vec{u}\right] \tag{4}$$

where I represent the identity tensor and the symbol T signifies the operation of transposition.
When fluid flow transitions into turbulence To calculate the current values, Shear Stress Transport k- -k-omega (SST k-omega) turbulence model is used since it integrates the advantages of two others namely the k- and k- models [107] using a blending function. Namely, k- o formulation is used in fluid components close to the wall areas and k- e formulation is in fluid elements close to the external boundaries of the layer and in the free-shear layers [108]. In turbulent boundary layers, the maximum eddy viscosity is controlled by making sure that, the turbulent shear stress is not greater than the product of the turbulent kinetic energy [108]. In this study, the heat transfer is determined by the use of the Computational Fluid Dynamics (CFD) results by means of the final results, given in terms of averages in the whole channel. Those mean values take into consideration density, velocity and viscosity averaged at the fluid volume in the channel. Moreover, the rate of heat transfer and surface temperature is obtained as averages quantities in the heat transfer field, and the bulk temperature is obtained by integrating the thermal energy transport rate across the cross-sectional area of a channel (A cs) as stipulated in Equation 5.

$$T_b\int_{A_{cs}}u\rho\ c_p\ dA_{cs}=\int_{A_{cs}}u\rho c_p\ TdA_{cs} \tag{5}$$

The attainment of Eq. 6 can be accomplished by integrating both sides of Eq. 5 over the entire real channel length (L_R).

$$\int_{L_R}T_b\int_{A_{cs}}u\rho\ c_p\ dA_{cs}dL_R=\int_V u\rho\ c_p\ TdV \tag{6}$$

The average value of the bulk temperature of each channel was required to be obtained to determine the average value of the Nusselt number. Eq. 7 can be used to find the in all the cross-sectional areas of a channel.

$$\overline{T}_b = \frac{\int_V u\rho c_p T dV}{\int_V u\rho c_p T dV} \tag{7}$$

The mean Nusselt in a channel is then determined by use of the equation 8.

$$Nu = \frac{qD_h}{kA(T_s - \overline{T}_b)} \tag{8}$$

where thermal conductivity of the fluid is also averaged in the fluid volume.

Apparent Fanning friction factor is determined by considering averaged values of the density in the equation and the velocity in fluid domain. The pressure drops that results because of the friction losses is obtained by multiplying the pressure drop by the square of the pipe length divided by the square of the pipe length. since the fluid accelerates at the inlet and outlet of the channel should be removed, and therefore can. approximately arrive at the real Fanning friction factor. The acceleration pressure drops (ΔP_{ac}) can be determined as. Eq. 9.

$$\Delta P_{ac} = (\rho u^2)_o - (\rho u^2)_i \tag{9}$$

Then,

$$\Delta P_{fric} = \Delta P - \Delta P_{ac} \tag{10}$$

where ΔP_{fric} is the pressure loss that occurs because of the friction losses.

4.2.2. Finite Element Analysis

The Finite Element Analysis (FEA) of the Printed Circuit Heat Exchanger (PCHE) was conducted utilizing Ansys Mechanical software. The fundamental equations governing solid mechanics are adhered to within the framework of the FEA. These equations ensure the equilibrium of forces between surface forces and body forces as articulated in Eq. 11 [109]. They facilitate the computations of normal strain and shear strain as delineated in Eq. 12 and Eq. 13 [109], respectively, alongside the implementation of the generalized Hooke's law as represented in Eq.15 and Eq. 16 [102]. All aforementioned equations are articulated within the Cartesian coordinate system.

$$\frac{\partial \sigma_x}{\partial x} + \frac{\partial \sigma_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} + b_x = 0;\ \frac{\partial \sigma_y}{\partial y} + \frac{\partial \tau_{yx}}{\partial x} + \frac{\partial \tau_{yz}}{\partial z} + b_y = 0;\ \frac{\partial \sigma_z}{\partial z} + \frac{\partial \tau_{zx}}{\partial x} + \frac{\partial \tau_{zy}}{\partial y} + b_z = 0 \tag{11}$$

$$\varepsilon_x = \frac{\partial s_x}{\partial x};\ \varepsilon_y = \frac{\partial s_y}{\partial y};\ \varepsilon_z = \frac{\partial s_z}{\partial z} \tag{12}$$

$$\varepsilon_{xy} = \frac{1}{2}\left(\frac{\partial s_x}{\partial y} + \frac{\partial s_y}{\partial x}\right);\ \varepsilon_{yz} = \frac{1}{2}\left(\frac{\partial s_z}{\partial y} + \frac{\partial s_y}{\partial z}\right);\ \varepsilon_{zx} = \frac{1}{2}\left(\frac{\partial s_z}{\partial x} + \frac{\partial s_x}{\partial z}\right) \tag{13}$$

$$\varepsilon_x = \frac{\sigma_x}{E} - v\frac{\sigma_y}{E} - v\frac{\sigma_z}{E}\alpha_L \Delta T;\ \varepsilon_y = \frac{\sigma_y}{E} - v\frac{\sigma_x}{E} - v\frac{\sigma_z}{E}\alpha_L \Delta T;\ \varepsilon_z = \frac{\sigma_z}{E} - v\frac{\sigma_x}{E} - v\frac{\sigma_y}{E}\alpha_L \Delta T \tag{14}$$

$$\varepsilon_{xy} = \frac{\tau_{xy}}{G};\ \varepsilon_{yz} = \frac{\tau_{yz}}{G};\ \varepsilon_{zx} = \frac{\tau_{zx}}{G} \tag{15}$$

In the aforementioned equations, σ_i denotes the normal stress measured in Pascals, τij signifies the shear stress also quantified in Pascals, bi represents the body force expressed in Newtons per cubic meter (N/m³), ε_i illustrates the normal strain, s_i indicates the displacement, ε_{ij} characterizes the shear strain, ν symbolizes the Poisson's ratio, and G corresponds to the shear modulus. The indices i and j denote the three Cartesian coordinates utilized within the context of these equations.

Chapter 5

PRFORMANCE EVALUATION OF VARIOUS TYPES OF PRINTED CIRCUIT HEAT EXCHANGER

The impact of the new channel design which is known as the two-way corrugated channel on the hydrothermal performance of PCHEs is studied in comparison to the conventional PCHEs. Nusselt number, friction factor, effectiveness, and performance improvement factor are the parameters that were used to evaluate the hydrothermal performance of the modified PCHE against the different wave amplitudes applied to sinuoidal plates and channels. In order to include the laminar flow regime, the cooling medium that is used on the hot and cold sides in the Reynolds number regime (300-1900) is used. The temperatures of the hot and cold water at the inlet are adjusted to 323 K and 293 K respectively. Figure 13 shows the velocity contours of hot and cold water in the transverse direction at 6 locations along the longitudinal direction of the PCHE at a Reynolds number of 1900, with differing axial positions on yz-plane of the conventional PCHE and the modified PCHE model No. 6. The longitudinal velocity distributions on both cool and hot sides are also almost uniform after the inlet of the hot and cold water as seen in the case of the conventional PCHE in Figure 12. The velocity profile does not change dramatically throughout the five sections after the entry section and it can be explained by the fact that the geometry of the straight channel has a constant cross-sectional area. The fact that the longitudinal velocity is stable is a benefit because it enables the formation of the further boundary layer [36]. The fluid velocity will be slowed down in the transverse direction to a maximum at the centre of the channel and slowly slowed down to zero at the channel walls. Since the Reynolds number is the same and the channels are also the same geometries, the velocity of the cold water will be the same as the velocity of the hot water. Figure 12 shows that the modified PCHE model No. 6 has a considerably different velocity distribution. The velocity pattern of modified PCHE differs with that of the traditional design in four major ways. To start with, the longitudinal and transverse velocity distributions of the cold water ceases to be the same as that of the hot water. Second, both the hot and cold water have transverse velocity changing along each longitudinal direction position. Third, there are no zero points of velocity, which implies that there is no stagnation. Fourth, the maximum velocities exist at the edges of the channel and not in the middle leading to a lower boundary layer thickness. These variations are due to the fact that the unceasing directional variations in the four axes (right, left, up, and down) in the corrugated channels favor unending mixing of the flow [37] . In general, the boundary layer in the modified PCHE is discontinuous, and it develops along the channel. It is worth noting that the velocity distributions of all transverse sections in the case of hot and cold water are different. The size and geographical size of the high-velocity areas vary and are discontinued in some parts. The resulting variability in velocity distribution is due to the unceasing horizontal and vertical deviations of the channel path, that make the flow change direction. As a result, they consist of areas of moving flow and slack ones, creating recirculation areas of different sizes in the channel.

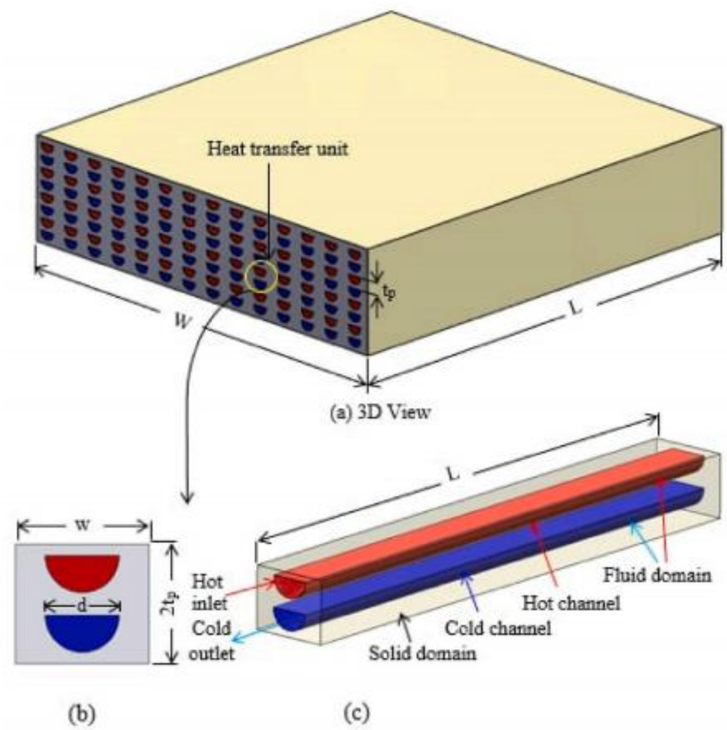


Figure 12:3D views, cross section, and computer aided design of a PCHE heat transfer unit. *[38]*.

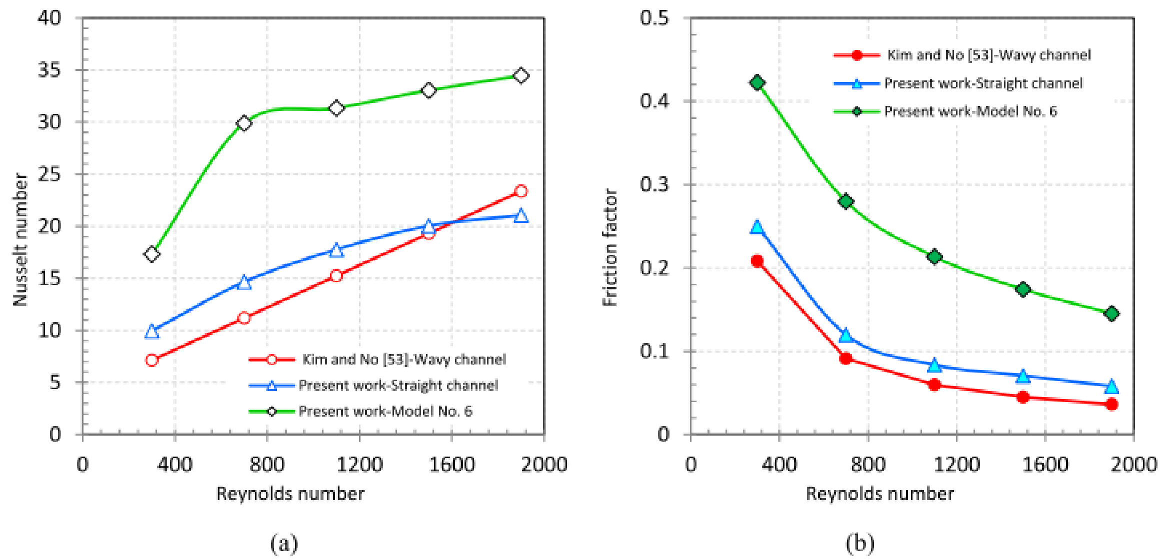


Figure 13: Comparison graph between experimental results as well as the results of this modeler with regard to the Nusselt number and the friction factor *[39]*.

The conventional PCHE and the modified PCHE model No. 6 are compared in terms of the temperature distribution of the hot and cold water across cross-sections along six positions of the longitudinal direction of the PCHE, Reynolds number of 1900 and different axial distances on the plane of yz, as shown in Figure 13. Figure 13 below illustrates that in the case of the traditional PCHE, the temperature of the hot water will decline progressively between the inlet and the outlet by the exchange of heat with the aluminum walls and the cold water. In any transverse section, the hottest water is that in the middle of the channel, and the colder water is found near the edges of the channel. On the other hand, when the heat loss is minimal the cold water undergoes an increase in temperature along the inlet-outlet channel due to heating of the hot water through the aluminum. Cold water has its lowest temperatures at the transverse sections which lie at the center of the core of the channel and the highest temperatures are at the outer sides of the core.

Figure 14b shows the distribution of temperatures of hot and cold water in the modified PCHE model No. 6. It reveals that the similarity is that the temperature of the hot water decreases steadily as we move down the channel to the outlet, and the temperature of the cold water increases as we move down the channel. It is worth noting that the temperature drop in the hot water is much greater in the modified PCHE model No. 6 than in the conventional PCHE, and a correspondingly great temperature increase in the cold water.

This means that the modified PCHE is better in terms of heat transfer performance because the two-way corrugated channels allow the mixing of fluid to be encouraged. In these channels, the enhanced mixing results in increased velocity gradients across transverse sections and heightened turbulence that in combination all increase the heat transfer process.

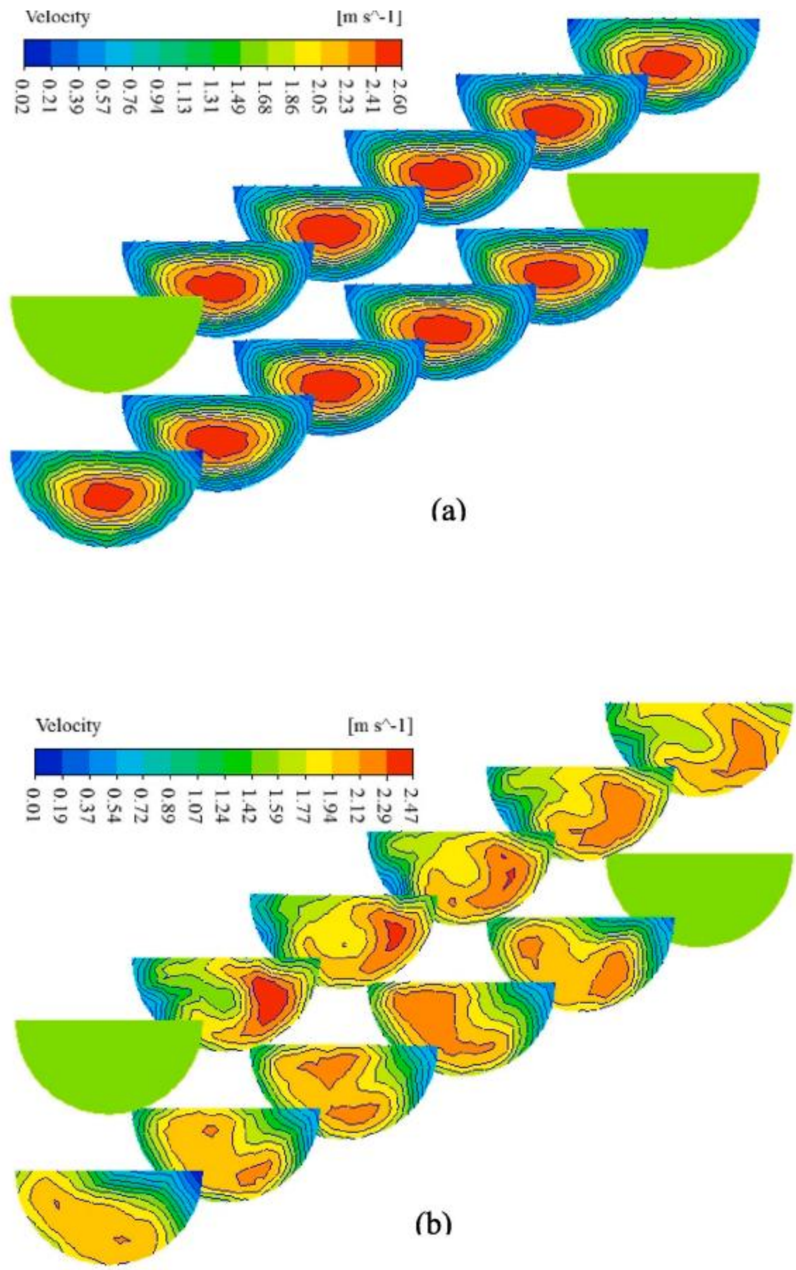


Figure 14:Velocity field of both hot and cold fluids in the conventional and modified PCHE *[40]*.

5.1. Thermal Hydraulic performance evaluation of the PCHEs

The complexity of the geometric structure of the PCHE and more or less low experimental data concerning that the numerical studies are fundamental, is due to its performance. Within the past decades, the rise in magnitude of computational capacity has provided the potential of solving difficult tasks by use of computational fluid dynamics (CFD). The CFD methods are the solutions to the Navier-Stokes equations that ensure the conservation of mass, energy and momentum compliance, as will be observed in Chapter 3. Then, the heat transfer and the energy loss in have been calculated by the use of simulations and experimental tests. The PCHE operation in a number of conditions. A number of methods have been used in an attempt to evaluate the best. hydraulic process combination of benefits by the heat transfer and losses by the hydraulic process. The attainment of an optimum balance may vary, being dependent on the weights that are assigned to each thermal-hydraulic parameter in the various applications. Various types have been developed using optimization methods of the heat exchangers depending on their costs as in the case of Caputo et al. [41]. of the shell and tube heat exchangers. In the case of the PCHE type, a methodology is given according to the overall costs analysis as proposed by Kim et al. [57], the heat transfer is linked the capital cost and the friction losses are connected with the operating cost. That is, the size of the heat exchanger core value of heat transfer value and the cost per year of capital is determined as a purpose of the cost of structural material and time of service. Subsequently, the energy used in the value of pressure drop is used to generate compression or pumping work of the fluid and the operating cost. Knowing the cost of energy per year is calculated. The cost and the results of the simulation were used by Kim and No [58]. optimal means to fill a zigzag channel PCHE on IHX conditions in the presence of helium fluids. Yoon et al. [44] applied this methodology to maximize the PCHE performance using helium fluids and using a mixture of sodium and S-CO2 as fluids. Kim et al. [40] took into account the total cost criterion in order to optimize a PCHE applied as SHX in SFR plant consisting of a mixture of primary and secondary fluids of FLiNaK and S-CO2, respectively. The total cost is also an objective function that Gyu et al. [41] used to measure the performance of S-CO2 fluids on an airfoil PCHE. In Figley et al. [26], power plant based on a IHX can be found minimization of the total cost was maximized which was the aim of small modular reactor (SMR). In general, the cost weights have been defined by the methodology that were used to maximize the thermal-hydraulic performance of the PCHE to parameters of other kind such as the heat transfer and the friction losses. However, there are values that can be different in each of the authors such as the energy cost, the structural material cost, and the time of service of the device. Thus, authors have a different weight and optimization criterion.

Table 2:Key structural and design attributes of the PCHE

Variables	Dimensions
inner diameter, (meter)	0.0021
Channel spacing, (meter)	0.00303
Plate wall thickness t_p , (meter)	0.00117
Fin sections t_f, (meter)	0.000313

Table 3:Reference temperature and pressure settings of heat exchangers in advanced small modular reactors

Coolant Type	Circulating fluid	System operating		
	Main/Auxiliary	In/Out T (°C)		Primary/Secondary Pressure (MPa)
		Main	Auxiliary	
Water-cooled	H_2O /H_2O	323.0/295.0	200.0/293.0	15.00/5
Gas-cooled	He/He	950.0/637.0	351.0/925.0	7.00/7
Metallic liquid cooled	Sodium/Sodium	450/390.0	320.0/526.0	0.1/0.10
Molten Salt-cooled	FLiBe/FLiNaK	700.0/650.0	600.0/690.0	0.1/0.10

There have been other criteria that have been taken into consideration in determining the thermal-hydraulic performance of the PCHE. Lee et al [42] . A multi-objective evolutionary algorithm was suggested by [53] to arrive at designs that possess an exergy saving advantage over the reference design. The improved designs depicted varying features: the former features a great first one reduction of the pressure drop, but the effectiveness is reduced too; and the second one is an increment of the thermal power with increased pressure drop. The flow mass rate distribution and the pressure drop Koo et al. [59] have also used as multi-objective functions to optimize the inlet part of a PCHE. In summary, the heat transfer and the energy loss obtained as two optimization functions have been obtained a good procedure. The Nusselt number can be maximized whilst the friction factor can be minimized then used as non-dimensional numbers in order to get optimized designs. Thus, correlations were developed one of the most significant objectives of recent studies is to compute these values of the PCHE.

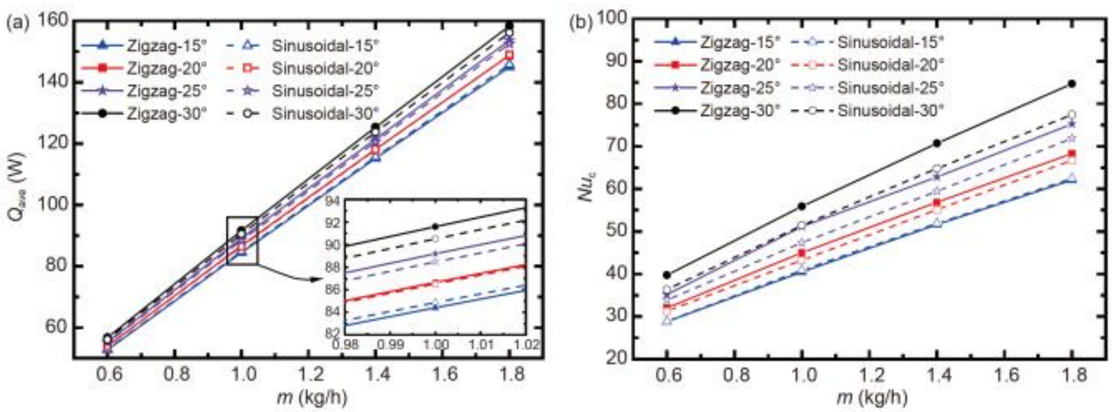


Figure 15:Change of the mean heat transfer rate and Nusselt number on the cold surface *[43]*.

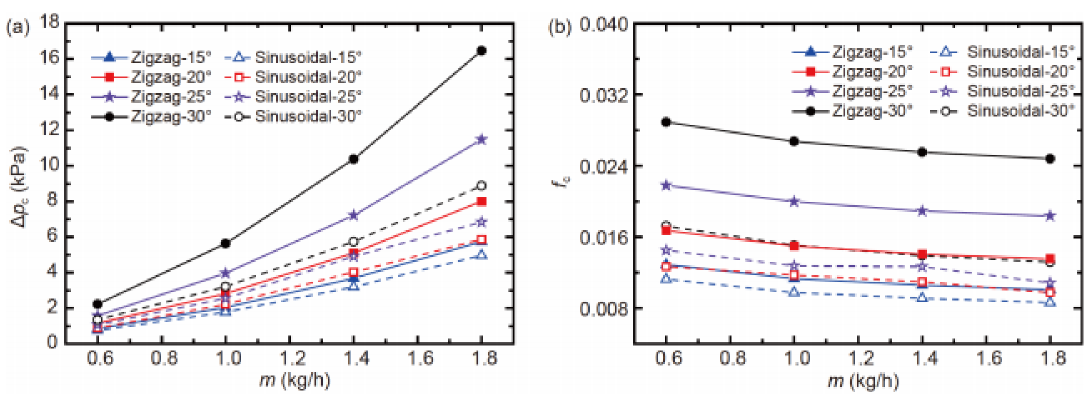


Figure 16:Comparison between pressure drops and friction factor of the various cold runners' structures. *[44]*.

The ratio of the Colburn factor to the Fanning friction factor [36,37] is chosen as a measure of the hydraulic performance of sinuoidal and zigzag channels in this research as the area goodness factor. As shown in Figure 15, the area goodness factors (j/f) of all the sinusoidal channels are evidently bigger than zigzag channels at the same bend angles. Smaller bending angles will always lead to improved overall thermal-hydraulic performance of sinuoidal and zigzag channel designs at a given mass flow rate. It can be noted also that there is variation in the difference in j/f between sinuoidal and zigzag channels with reference to the bend angle. A higher bend angle causes the number of channels to increase, but the thermal-hydraulic performance reduces. Thus, the best performance of the two channel types is that at less bend angles. The additional optimization must consider the following factors: heat transfer load, pumping power, compactness, thermal stress, and costs. As an example, Kim et al. [45]. Optimized the design of an PCHE using zigzag channels of different bend angle combinations, using a cost analysis approach of intermediate heat exchangers in high-temperature gas-cooled reactors. As demonstrated in Figure 7, there is a correlation between the heat transfer enhancement ratio and the channel enlargement ratio. In this case, Qo and Ao are the heat transfer rate and the heat transfer surface area of a straight channel with a length of 246 mm that may be considered a sinuoidal channel or a zigzag channel with a bend angle of 0. It should be mentioned that serpentine nature of its structure may offer sinuoidal channels slightly more heat transfer area as compared to zigzag channels with the same bend angle. Figure 16 shows how is growing in the sinuoidal and zigzag channel.

Table 4:Variation of pressure drop and heat transfer rate with respect to the zigzag channel

masse (kg/h)	Teta (°)	$(Q_{zig} - Q_{sin})/Q_{zig}(\%)$		$(\Delta p_{zig} - \Delta p_{sin})\Delta p_{zig}(\%)$	
		Cold-fluid side	Hot-fluid side	Cold-fluid side	Hot-fluid side
0.60	15.0	0.4	-0.4	11.8	12.3
	20	0	0	23.3	25.2
	25	0.3	0.3	32.3	35.3
	30	1.1	1.1	39.3	41.6
1.8	15	-0.6	-0.4	13.4	16.8
	20	-0.1	-0.1	26.8	29.3
	25	0.7	0.7	40.3	43.1
	30	1.4	1.4	46.1	48.4

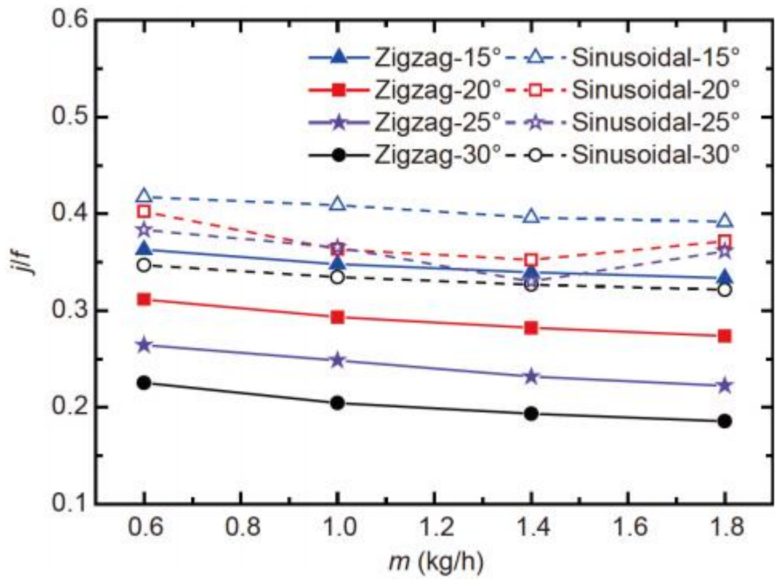


Figure 17:The way in which the j/f ratio varies with the channel geometry *[35]*.

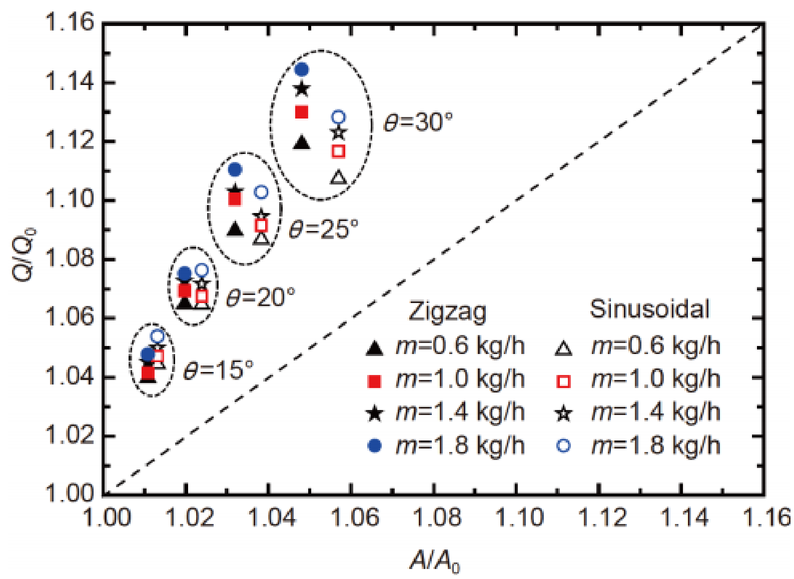


Figure 18:Relationship between the rise of the heat transfer and the increase in the size of the exchange surface *[46]*.

Thermal-hydraulic performance in the use of a recuperator in a fusion-based experimental reactor at 3 MPa pressure reveals better performance of wavy channels over straight-channel PCHEs. A correlation between the friction factor and Nusselt number is put forward as the Reynolds number (350 1 2100) of wavy channels with a 5° , 10° and 15° bend angle. Spatially periodic separation or tripping of the boundary layer at the bends and the resulting wringing effect cause the flow not to become fully-developed, which causes variations in Reynolds number throughout the channel as well as spatially periodic changes in the Nusselt number and local heat flux of the wavy-channel PCHE. Future research will be done on increased design changes and how operating conditions affect straight-channel PCHEs. In an attempt to compare the thermal-hydraulic behavior of sinuoidal and zigzag channel PCHEs, both the heat transfer and the flow properties are tested in varying inlet mass flow rate and various bend angles. Figure 4 shows the variations of the average heat transfer rate and the Nusselt number at the cold side at the various mass flow rates. It is noted that the Nusselt number and the heat transfer rate rise with the increase in mass flow rate.

5.1.1. Average Thermal Flux for the Distinct Wavy Segments of the Printed Circuit Heat Exchanger Model

The surface-averaged heat flux of different wavy sections is calculated as the difference of heat flux across each wavy section of the wavy channel and averaging it across all wavy sections. Figure 19 shows the heat flux distribution throughout the various wavy sections of the channel of wavy at a range of Reynolds numbers, and with a wavy-section angle of $\theta = 5$deg, of the wavy-channel. The curves reveal that the heat flux of every wavy part is higher, this is mainly attributed to increased speed of the local flow and the increase in the mixing of the fluid downstream of the bends [47].
Also, the total rate of heat transfer increases notably with the bend angle, as confirmed in Figure 18, which is by far larger than the change in the heat flux of an individual section.

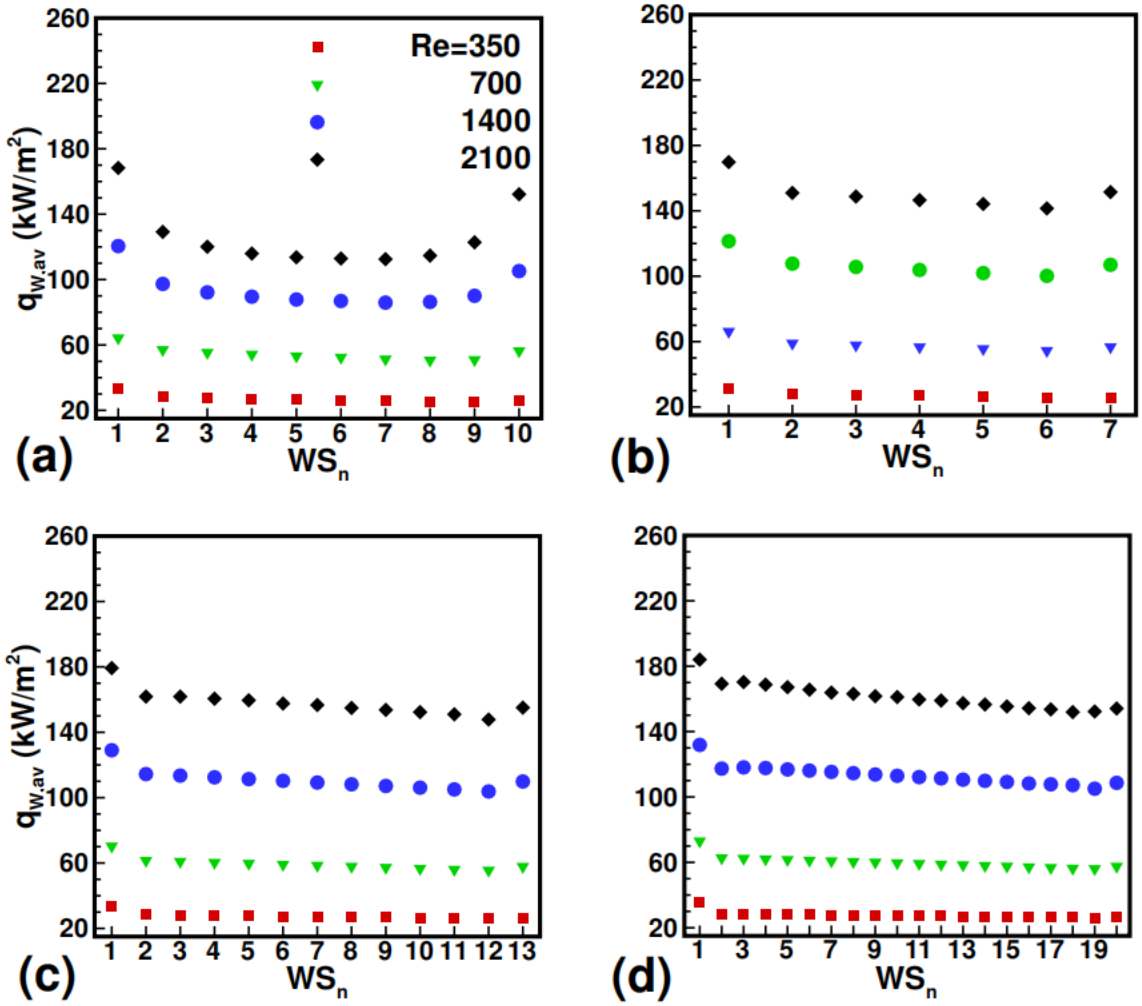


Figure 19: mean heat flux distribution as control variables are curvature angles and the Reynolds number in a corrugated channel [48]

Figure 19 indicates that the heat flux of a wavy section is normally less in lower sections compared to the upper ones with the only exception of the last wavy section which is linked to the inlet or the first section of the cold channel. This fact is in line with the results of Kim and others [49].It is worth noting that the area with the highest heat flux is the first wavy part as the boundary layer is much thinner at this point. Besides variations in the flow and heat transfer patterns, variations in the thermo-physical properties, especially, the thermal conductivity, modify the heat flux, as well. This tendency toward a lower thermal conductivity flowing along the hot channel with a decrease in temperature is one of the reasons why the heat flux is reduced in downstream wavy sections. The influence of bends in a section of wavy section on heat flux is further discussed by breaking down a section into four equal straight sections marked A-D as in Figure 18(a), Figure 18(b), (c) and (d) display results of other Reynolds numbers and angle of bend. Figure 8(a) represents the heat flux distribution across the four parts in Figure 18(a) in the form of a shaded bar chart, covering the second wavy section. It is noted that part B and D of each wavy segment are extremely higher in heat flux compared to part A and C. This is because the direction of flow at the inlets of part B and D is switched but parts B-C and D-A are a straight line. The heat flux increases at once just below the turns in the flow of the bends because the flow is separated and mixed more intensely than before as observed by Kim et al. [50] and Tsuzuki et al. [10]. As can be seen in Figures 18(b). As the bend angle changes 0 to 0 and Re to the Reynolds number, the intensity of a heat flux in different parts of the wavy sections also increases (Figures 18(b, d, and e)). Specifically, element B and D always have high heat flux compared to element A and C because of the above-mentioned reasons. Interestingly, in low Re, the second and fourth maxima (parts B and D) and minima (parts A and C) are almost equal and as the Re increases, the difference in the two becomes more pronounced. This means that the heat flux in part D with part B as compared to part A with part C is considerably dissimilar at high Re. On the same note, the effects of the first bend (part A to part B) and the second bend (part C to part D) on the local heat transfer rate are nearly similar when at low Re but differ when at high Re. Lastly, Figure 19 illustrates the streamwise vorticity distribution of the ninth wavy section at Re = 1400 and θ = 15 degree at cross sectional views of each of the four parts and top views of three radial sections, which further demonstrates the effects of the bends on the flow structure and heat flux at a local scale.

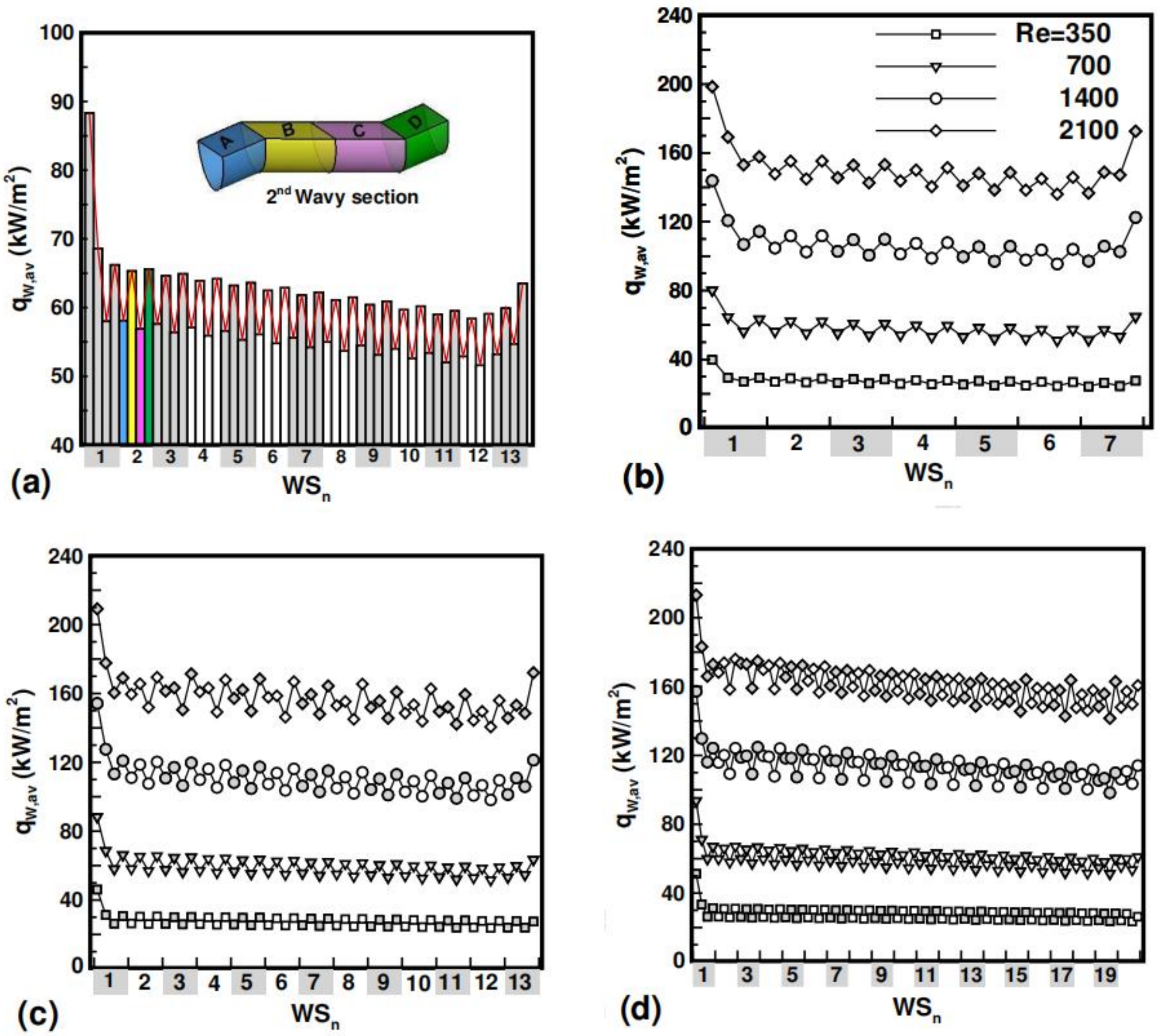


Figure 20: the heat flux across various parts of a corrugated channel at varying conditions, mean [51].

The perspectives presented exhibit a distinction in both the spatial positioning and the nature of vortex formation within sections A and C, which subsequently results in a higher mean heat flux from section A in contrast to section C.

5.1.2. Flow and Heat Transfer of the Wavy and Straight Channel in PCHE.

It should be mentioned that the findings presented in this section are applicable to the hot channel, but the same analysis can be applied to the cold channel with minor variation based on the density of power. As shown in figure 10, the ratio between power density and pressure drop between the wavy and straight hot channel increments with the Reynolds number (Re) and the bend angle θ. The results however show that the influence of Re on the rise of power density is stiffer than the influence of θ, but the vice versa is true with the rise of pressure drop. The extra cost caused by the pressure drop is not much as compared to the expenses incurred in maintaining the high operating pressure. Consequently, the power density increase is enormous when compared to the penalty that is associated with the pressure drop in the present application. The rise in Re causes the power density to rise because it is accompanied by an equal rise in the logarithmic mean temperature difference in counter-flow heat exchange of PCHE model. To achieve enhanced precision in the prediction of PCHE performance, the Nusselt number (Nu) as well as the friction factor (f) are assessed on a wavy section-by-section basis and then averaged, as opposed to using calculations at a global level only [7]. This method is used to give a better estimate of the thermal-hydraulic performance by capturing local differences in the flow as well as heat transfer characteristics within the wavy sections.

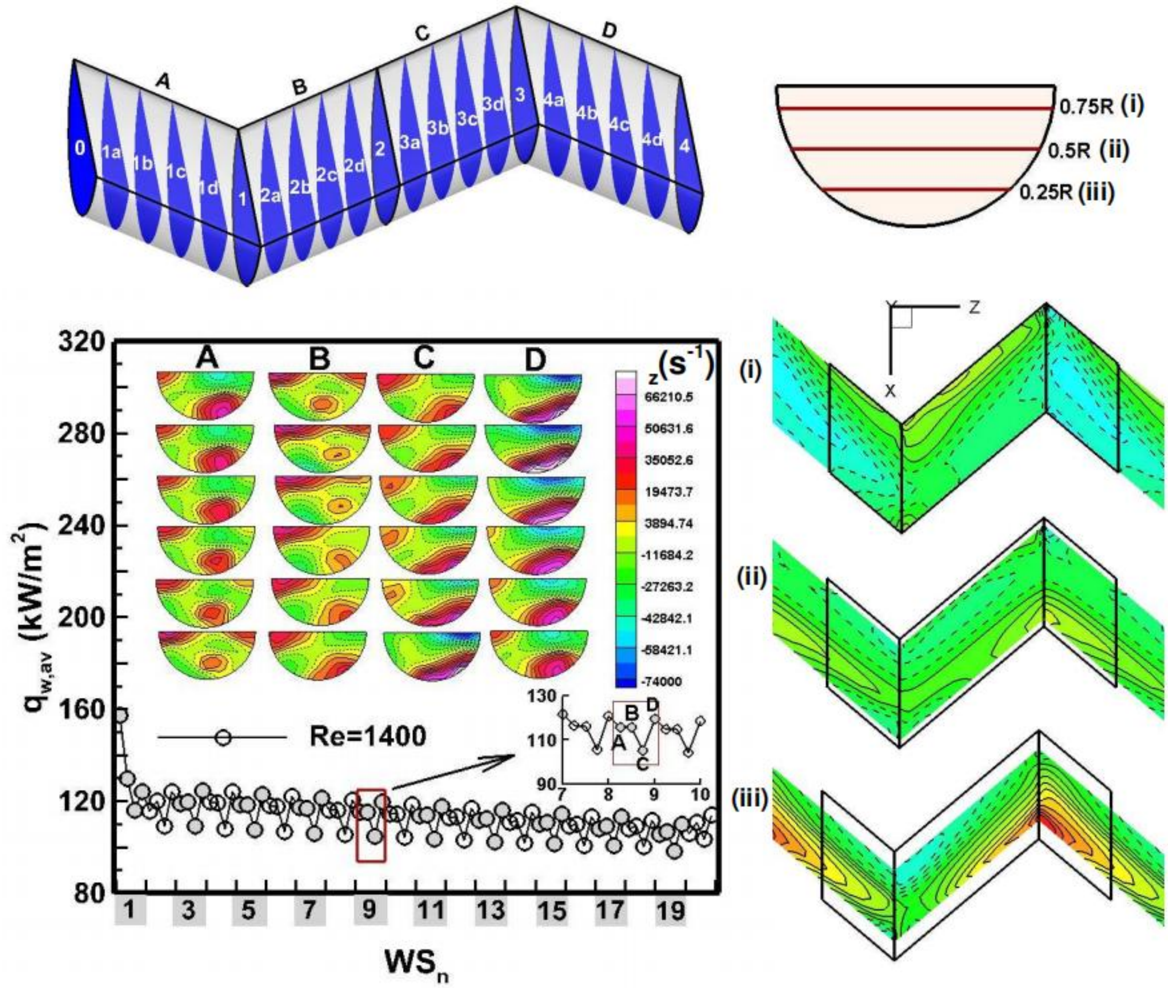


Figure 21: The profile of the heat flux and vortex in the semicircular and zigzag regions in a channel at Re = 1400 [45]

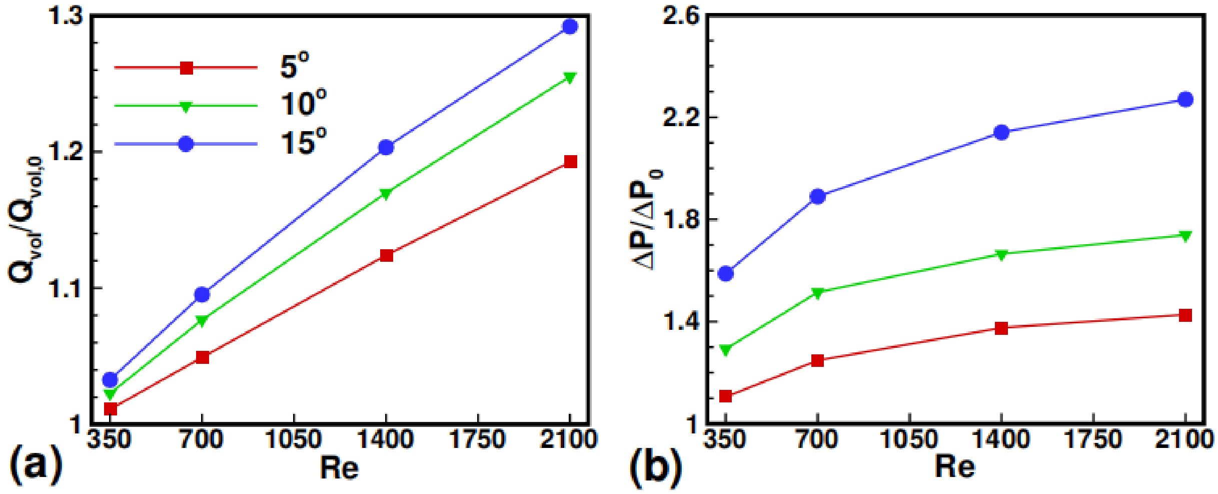


Figure 22: Comparisons between straight and corrugated geometry in terms of power density, and power loss through the pressure *[52]*.

In the international approach to the performance assessment of the channel, the local Nusselt number of each wavy section is determined with a heat transfer coefficient that is based on the logarithmic mean temperature difference. The difference between the temperature of the hot channel surface and the temperature of the bulk mean fluid at both the inlet and outlet is obtained using methods used by Metwally and Manglik [14] in wavy-plate channels and by Sui et al. [53] in 3D wavy microchannels. Local Fanning friction factor of every wavy section is calculated as $f_w = \Delta P_w / 2l_p$, where: ΔP_w is the pressure drop across the section, m is the mass flux and LP is the pitch of the section. As shown in Figure 21 both local averages of f and Nu in the hot wavy channel rise with Reynolds number (Re) and bend angle θ. To conduct future research, a functional relationship proposed by Kim et al. [54] is taken and a correlation between f and Nu is suggested. Figure 11 shows that there is a strong correlation of the present numerical results with this correlation with a maximum deviation of 4.3. Figures 22(a-b) indicate that the values of Nu/Nu0, f/f0 and thermal performance factor (TPF = Nu1/3/f 1/3) of wavy channel versus θ increases with Re and θ as compared to the planar channel. It is important to note that the increase in heat exchange is higher as compared to the pressure penalty as measured in Figure 22(c). The figure also shows that the TPF is greater than one at different values of Re and θ, and tends to increase in proportion to both Re and θ, proving that wavy channels can be better thermal-hydraulic.

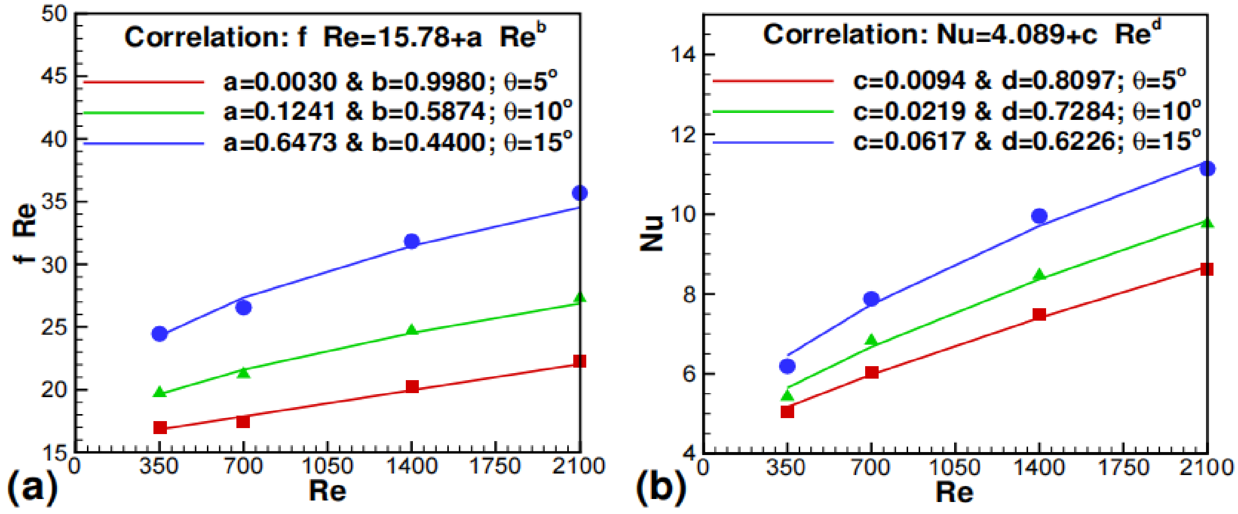


Figure 23:Comparison between suggested correlations and numerical values of the friction factor and Nusselt number *[55]*

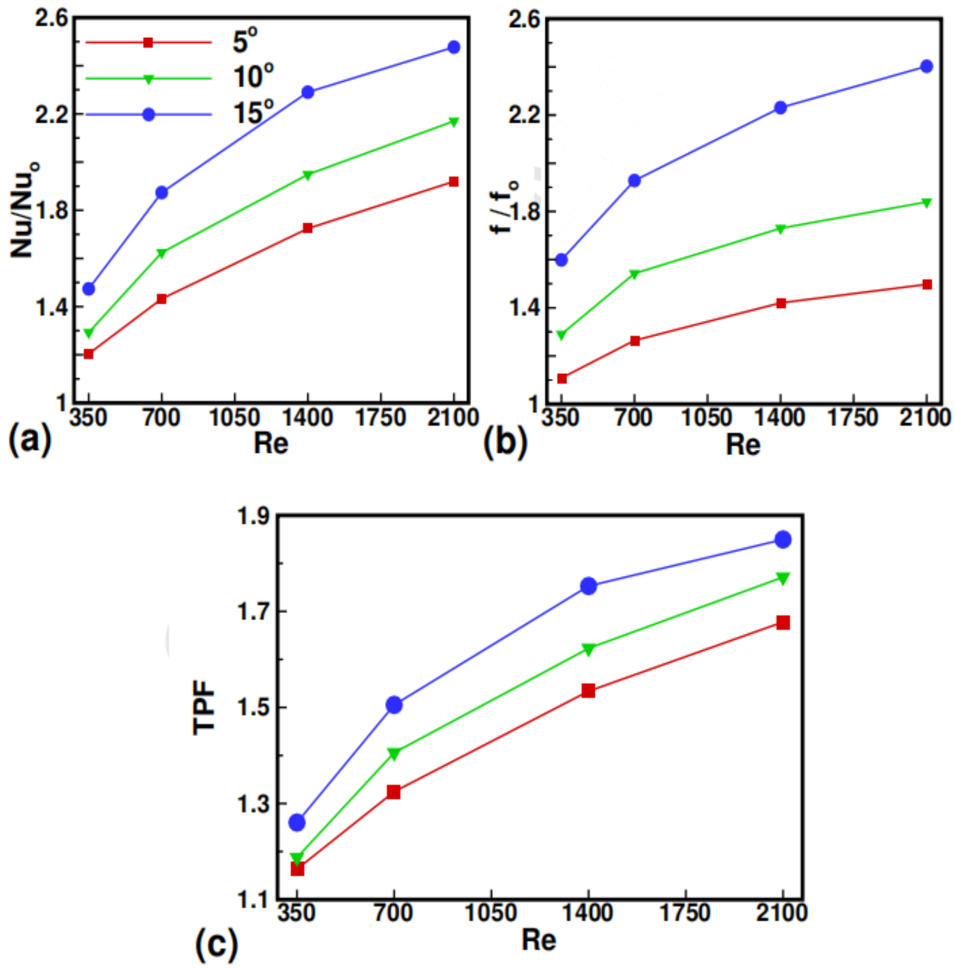


Figure 24: The thermal performance of straight and corrugated channel PCHEs with respect to Reynolds number and curvature angle *[56]*.

In this paper, a comparative study of the thermal-hydraulic performance of PCHEs when the channels are zigzag and sinusoidal is made. These are the variations of the rate of heat transfer, Nusselt number, pressure drop, and friction factor analyzed at various bend angles and mass flow rates. The distribution of flow fields and heat flux are also analyzed further, and one can make several important conclusions. The sinuoidal channel can minimize the pressure drop to an extent of up to 48.4 percent, but with virtually the same heat transfer performance at the same flow conditions as in the zigzag channel. On the whole, sinuoidal channels have better extensive thermal-hydraulic performance at constant bend angle. They also offer further advantages of decreased pressure drop with increment of the bend angle. The enhancement of the heat transfer of sinuoidal channels to straight channels is not only due to the increase of the heat transfer area but also the increase in the heat transfer coefficient. The discrepancies between the literature and the current study could be caused by the differences in the flow regimes in the channels. The greatest part of the total pressure drop is the first pitch of the channel that is maximal 17 percent in zigzag channels and 31 percent in sinuoidal channels [54]. Thus, the design of the inlet parts is very important to reduce pressure drop especially on sinusoidal channels. Density and heat flux distributions the complex flows in the channel typically result in nonuniform distributions. The flow and heat transfer processes of both zigzag and sinuoidal channels are based on the periodic reversal of the direction of flow and centrifugal forces they produce. In order to further improve the performance of PCHE in this type of configurations, it is suggested that some future work be directed to optimize current designs and to come up with new channel structures.

5.2. Thermal Performance evaluation of PCHE

The thermodynamic cycle has its own inherent limitations. Nonetheless, since gas turbines make use of gas as the working fluid, which has a low thermal capacity, inefficiencies are created especially when heat exchangers are included in the process, like recuperators and precoolers, because of the bulk of the quantities to work with. Several researches have been done in the last several decades aimed at resolving this issue, and compact heat exchangers with high thermal efficiency were developed despite their limited size. A notable example of a miniature device of this type is the printed circuit heat exchanger (PCHE), which was first invented by HEATRIC [57] PCHE is produced through both chemical etching and diffusion bonding which gives it a great thermal performance and thus highly applicable in the gas turbine system integration. Computational fluid dynamics (CFD) and Reynolds-Averaged Navier-Stokes (RANS) analysis have become a common practice to determine thermal-hydraulic performance of PCHEs, as computational techniques have been used in other fields of thermal engineering [58]. As an example, zigzag channel PCHEs were experimentally investigated to compare the thermal performance of supercritical carbon dioxide (CO2) and pressure drop values under different flow rates, pressures and temperatures. Ngo and associates analyzed the behavior of an S-shaped fin PCHE channel to supply hot water through a three-dimensional (3-D) analysis of the RANS. In a different study, Kim et al. [9] performed 3-D numerical simulations to determine heat transfer and pressure drop in both zigzag and airfoil-shaped fin PCHEs, and it was found that the airfoil-shaped fin PCHEs transfer the same amount of heat per unit volume and have a lower pressure drop in comparison with the zigzag designed PCHE. Subsequent studies by Nikitin et al. [10] examined the thermal and hydraulic characteristics of PCHEs with the help of a supercritical carbon dioxide loop and were explained by the corresponding equations. [58] created a numerical model of PCHEs and found the best operating conditions simulating various accident cases. Furthermore, [12] suggested calculating the heat transfer and fluid flow in PCHEs based on the GAMMA code [13] and studied the thermal and hydraulic properties of such equipment at the same time. Oh et al. [14] researched the heat transfer and tritium penetration in PCHEs with standard and offset channel configuration where the spacing between channels (horizontal and vertical) were determined to affect overall performance. There was also an attempt to optimize the performance of PCHE with the 3-D RANS analysis done by considering two important design parameters; cold channel angle and the aspect ratio of ellipticity of the cold channel [15]. In the present analysis, it is proposed to use a new PCHE design using a thin plate to increase the contact surface area between hot and cold flow streams, which improves the thermal efficiency of the heat exchanger. The flow and heat transfer analysis in the channels, that is, in the solid material of this PCHE, were analyzed using three-dimensional RANS equations in order to examine the operational performance of this PCHE. The cold channel simulation of Reynolds number was between 67,000 and 280,000. Some of the major thermal performance indicators used in the proposed PCHE (such as Nusselt number, Colburn j-factor, thermal effectiveness, and net exergy gain) were evaluated against a standard PCHE. The efficiency of zigzag PCHE channels was used as a proportion of the real transfer rate of heat to the highest theoretically achievable rate of heat transfer, which is the practical physically realistic operation of the heat exchanger. Effectiveness is mathematically defined as P_r is the hydraulic diameter of the flow channel. The active flow lengthand the number of thermal units (NTU), which is the ratio of total thermal conductance to the minimum heat capacity rate was also taken. The j-factor of Colburn is generally the feature of PCHEs, thermal properties, and compactness are the design advantages. According to the average surface heat flux.

$$\mathrm{Nu} = \frac{\mathrm{q}''\mathrm{D_h}}{\mathrm{k}(\mathrm{T_b} - \mathrm{T_s})} \qquad (16)$$

Exergy analysis is a basic tool in the design and optimization of Printed Circuit Heat Exchangers (PCHEs) in terms of energy saving. Exergy is somehow related to the quality of energy and it is the possibility of useful work. Exergy may be produced in the course of operation by the heat transfer process and be simultaneously lost to irreversible processes like friction and heat losses. Therefore, the net exergy gain is a significant measure of the performance of the system, as it represents the same as the energy efficiency and quality of the energy used. An example of this is a dimpled cooling channel optimization study [19] done by Lee and Kim using exergy analysis to optimize the performance of the system. The net exergy gain in a PCHE which is defined as the difference in exergy between the hot and cold channel [20] can be mathematically represented as shown below.

$$E = E_{cold} + E_{hot}$$

$$E_{cold} = \left(1 - \frac{T_0}{T_{cold,s}}\right) q''_{cold} A_{cold,s} - \left(1 - \frac{T_0}{T_{cold,outlet}}\right) q''_{cold} A_{cold,s} - \Delta P_{cold} \frac{\dot{m}_{cold}}{\rho_{cold,Avg}} \frac{T_0}{T_{cold,outlet}} \qquad (17)$$

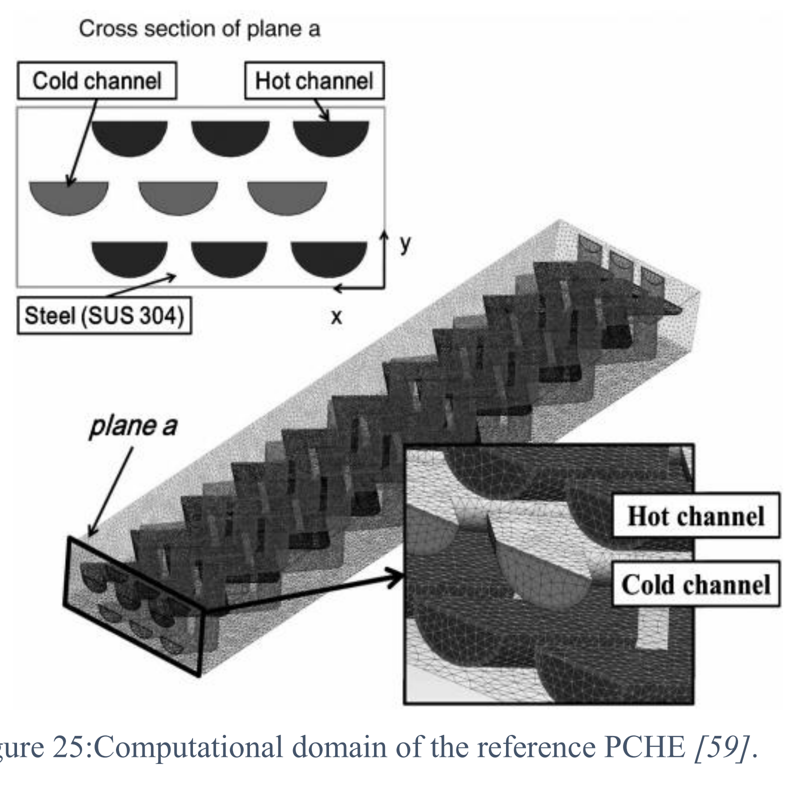


Figure 25:Computational domain of the reference PCHE *[59]*.

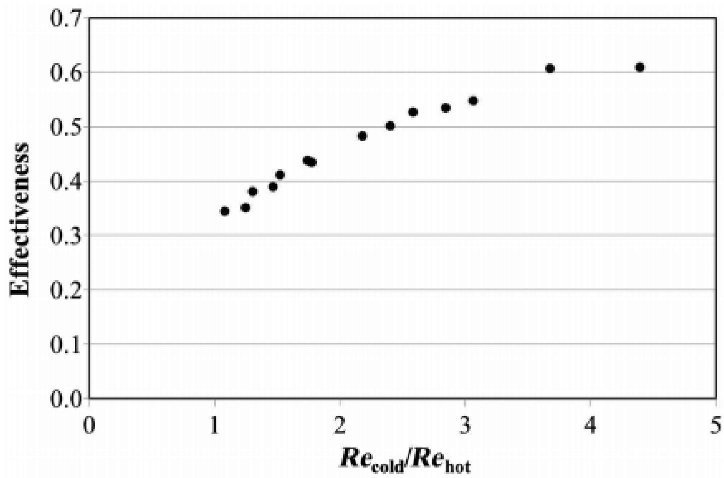


Figure 26:Variations in effectiveness with Re_{cold}/Re_{hot} [60].

Table 5:Channel geometry specifications of the reference PCHEs

Variables	Data
Cold channel angle of orientation.	100.0 deg
Height of cold channel upwards	0.900 mm
Lateral dimension of cold channel.	1.900 mm
Thickness of cold channel at the wall.	0.500 mm
Centre to centre distance (pitch) between cold channels	7.250 mm
Lateral offset towards the x-direction, x_{off}	0.0 mm
Vertical lateral in z-direction, z_{off}	0.0 mm
The angle of inclination of the hot channel	116.0 deg
Lateral width of hot channel	1.70.0 mm
Wall Thickness of hot channel	0.600 mm

Inter channel hot channel pitch (pitch)	8.00 mm
Height of hot channel through vertical arrangement	0.950 mm

$$E_{hot} = \left(1 - \frac{T_0}{T_{hot,s}}\right) q''_{hot} A_{hot,s} - \left(1 - \frac{T_0}{T_{hot,outlet}}\right) q''_{hot} A_{hot,s}$$

$$-\Delta P_{hot} \frac{\dot{m}_{hot}}{\rho_{hot,Avg}} \frac{T_0}{T_{hot,outlet}} \qquad (18)$$

And here signify the exergy increments in the cold channel and the hot channel, respectively. The three values in the right-hand side of the equations. The exergy input and output in a heat transfer have the following representations [61] is the exergy gained through heat transfer, is the exergy lost through thermal transfer and the exergy lost through frictional losses in each channel. Here, denotes the difference in pressure in the static condition of the channel, are the ambient temperature, and are the spatial averaged temperatures of both cold and hot channel walls. Additionally, represent the surface areas of cold and hot channels respectively, and represent the cold and hot channel mass flow rates. Lastly, are the means of the densities of the working fluids in the respective channels.

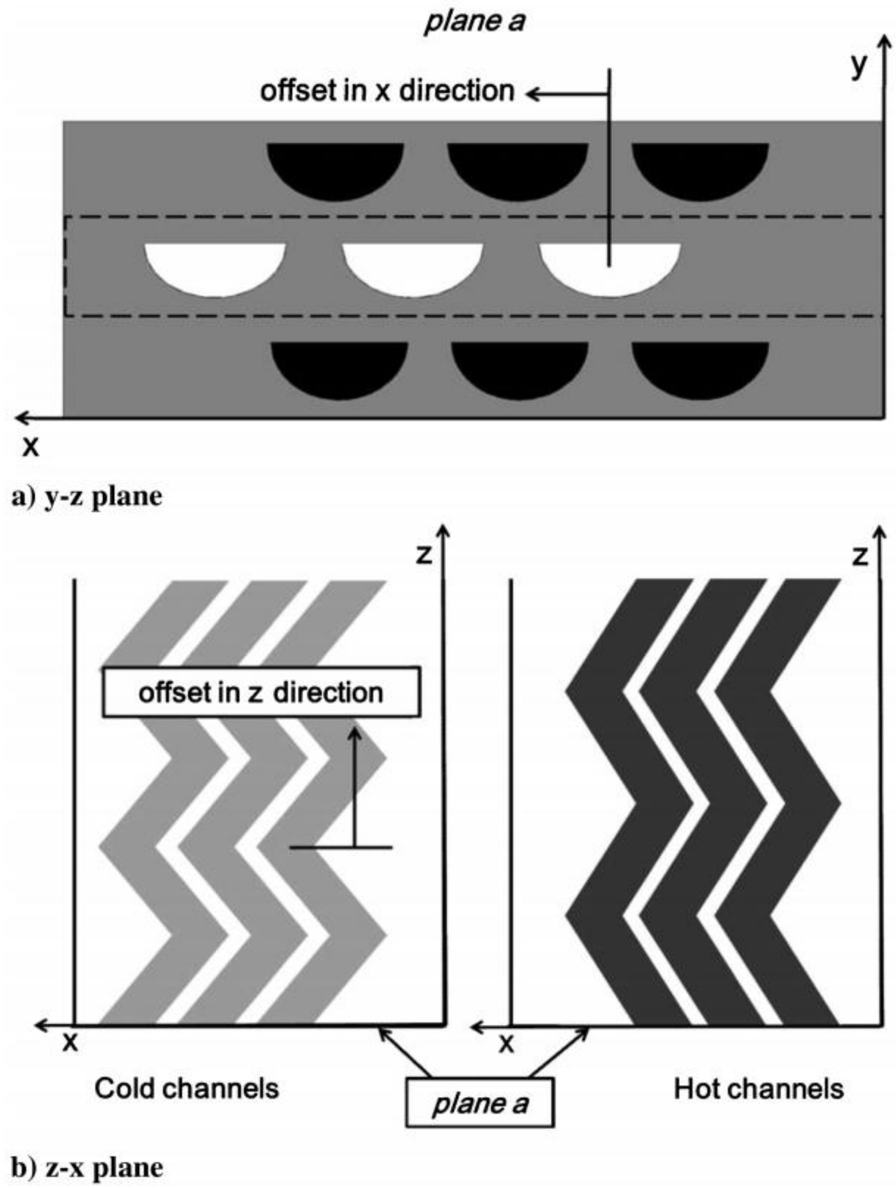
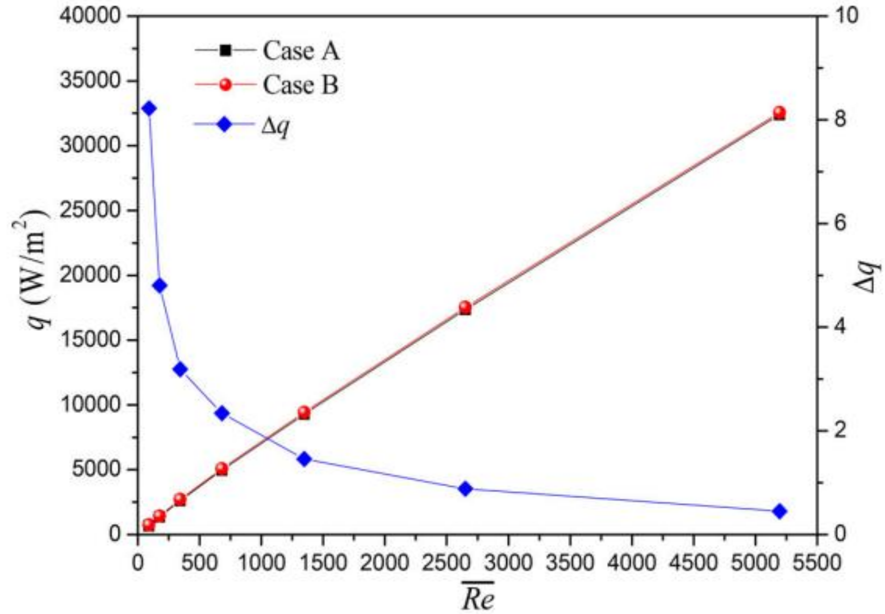
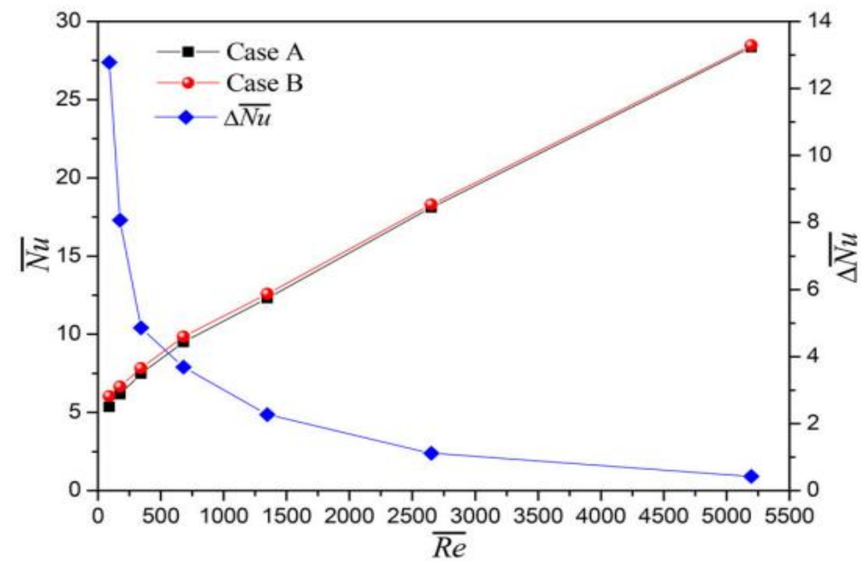


Figure 27:Zero-offset conditions in the reference PCHE *[62]*

The main aim of the research is to explore the thermal characteristic of a zigzag channel printed circuit heat exchanger when heating a load. In this regard a full evaluation of the thermal performance on the cool surface is also carried out. The difference in performance as calculated based on the equations is shown in Figure 26 where a comparison was made between the thermal performance of Case A and Case B



(a) q comparisons



(b) $\overline{Nu}$ comparisons

Figure 28: Since Reynolds is increasing, the average thermal performance improves, Edwin *[63]*.

$$\Delta_q = \frac{q_1 - q_2}{q_2} * 100 \qquad (19)$$

$$\Delta\overline{Nu} = \frac{\overline{Nu}_1 - \overline{Nu}_2}{Nu_2} * 100 \quad (20)$$

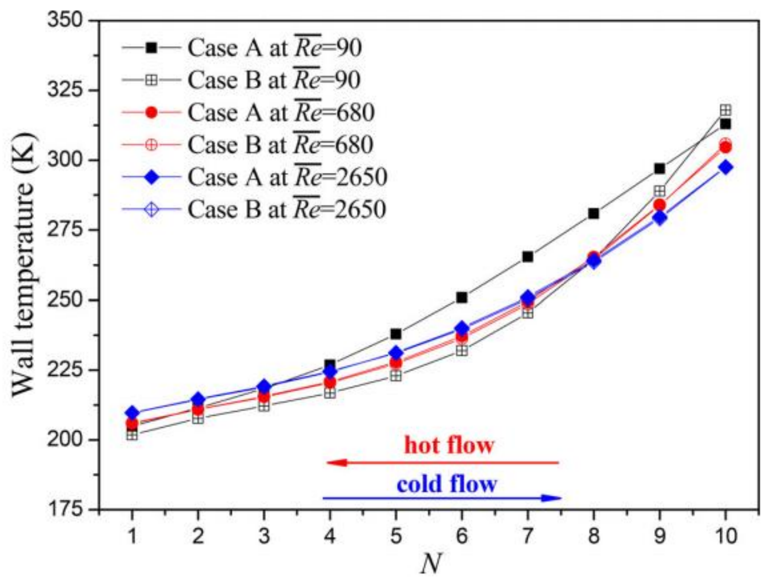


Figure 29: Comparison of cold side wall temperature of the various cases of operating *[64]*.

Figure 29 shows that the Reynolds number increases are associated with an increase in the heat flux and the average Nusselt number, and the differences in thermal performance of Case A and Case B also decrease. The differences in the average Nusselt number and the heat flux in the two cases namely; the average Reynolds numbers of 10, 12, 14, and 16 are 8.2, 2.3, 0.9 and 1.1 respectively. This observation shows that axial heat conduction is more influential at low Reynolds numbers, but its effects become less influential with increase in Reynolds number. Figures 26 and 27 give a comparative analysis of the wall and bulk temperature on cold side at various Reynolds numbers. As can be seen, the differences in wall and bulk fluid temperature in Case A and Case B are larger at lower Reynolds numbers all the way along the PCHE channel [65] . The disparities in the wall and bulk fluid temperatures diminish with the increase in the average Reynolds number, making the impact of the axial heat conduction insignificant with an increase in the Reynolds number.

5.2.1. Average Heat Flux Pertaining to the Influence of Wall Thermal Conductivity on Axial Heat Conduction in the Distinct Wavy Sections of the Plate-Fin Compact Heat Exchanger Model

It is not a secret that one of the most significant issues, which affect the thermal conduction on the axial level, is the thermal conductivity of the wall. The section is devoted to the discussion of the effect of thermal conductivity of walls on the performance of zigzag channel Printed Circuit Heat Exchanger (PCHE). The numerical simulations are carried out in conditions of operation that are the same as Environment No. 4 in Table 7.

Table 6:Performance settings under varying conditions

Parameter	Symbol/unit	Obs. 1	Obs. 2	Obs. 3	Obs. 4
System pressure	P/MPa	8.00	7.0,8.0,9.00	8.00	8.00
unit area	$G/kg/(m^2.s)$	1.4-89.4	1.4-22.4	1.4-22.4	1.4-22.4
Temperature of hot fluid at entry	$T_{in\ hot}/K$	350	350	350	350
Temperature of cold fluid at entry.	$T_{in\ cold}/K$	196	196	180,195,220	196
wall thermal conductivity	$\lambda/W/(m.K)$	15	15.0	15.0	5,15.0,45.0

Figure 30 shows that the wall thermal conductivity has an upward trend on the thermal performance of the PCHE, whereby average Reynolds number increase with mass flux and the average Nusselt number decrease with increase in wall thermal conductivity [66]. Such a behavior is expounded in the following way, though increasing thermal conductivity of solid wall minimizes thermal resistance and maximizes heat flux across the separating wall, it also raises the axial heat conduction into the wall. The sum of these two opposite effects is a net reduction in heat flux as shown in Figure 30. Also, the mean value of the

Prandtl number of the hot fluid decreases with an increase in the wall thermal conductivity. Because Nusselt number has a positive correlation with the Prandtl number, the findings are that the thermal conductivity of a wall is lower, the better thermal performance of the PCHE is in the present circumstances of the mass flux.

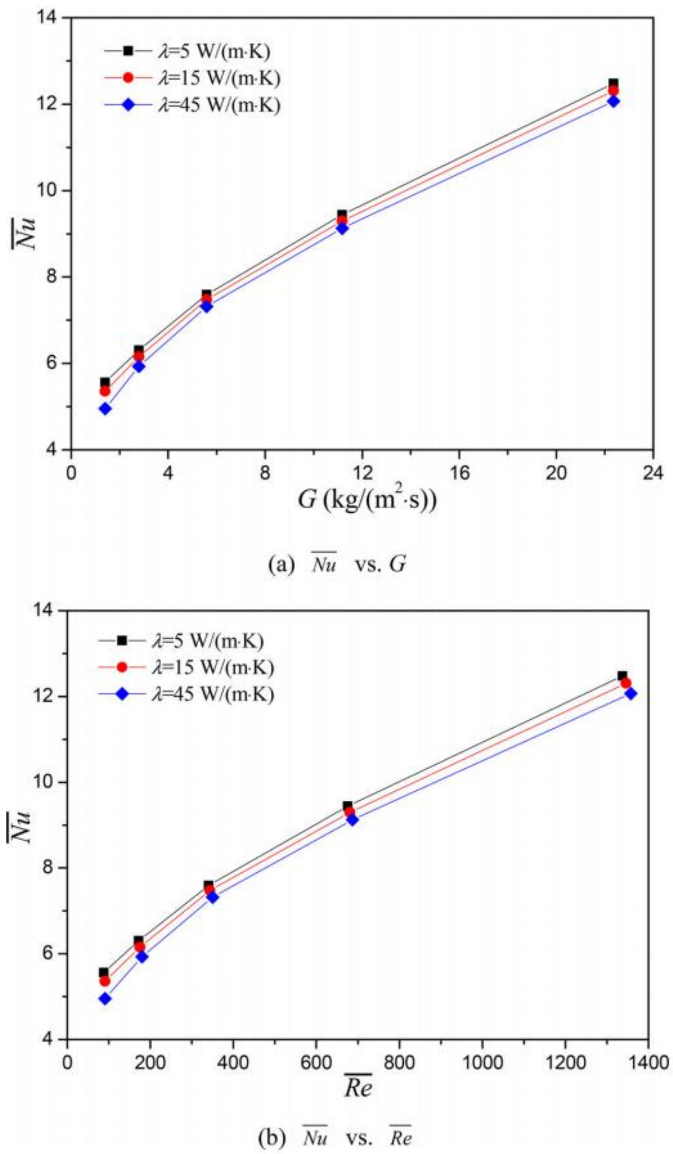
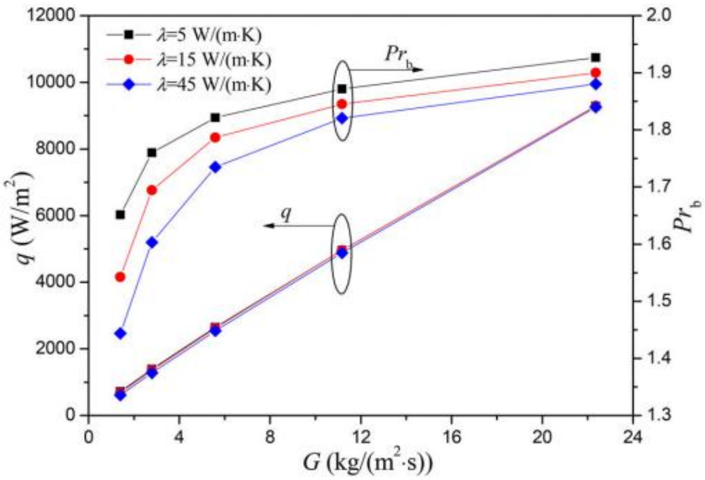
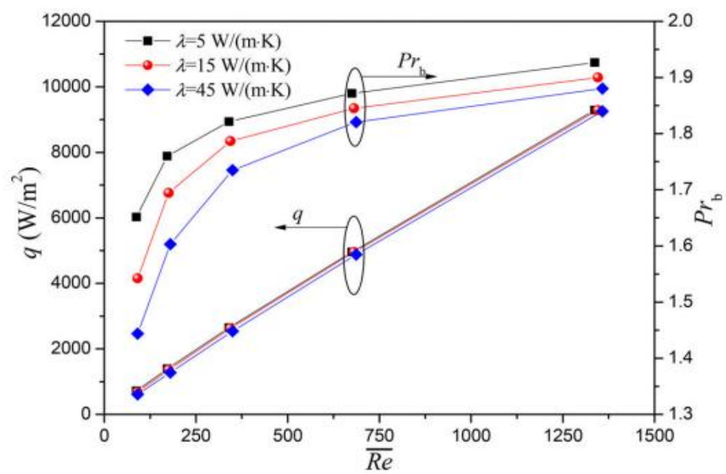


Figure 30:The effect of thermal conductivity of a wall material on thermal-hydraulic performance *[67]*

Figure 30 represents the comparative study of heat flux and average Prandtl number in zigzag channel PCHE at different wall thermal conductivities with the result that both the heat flux and average Prandtl number show a downward trend with the increase of wall thermal conductivity at the same mass flux or average Reynolds number.



(a) q and Pr_b vs. G

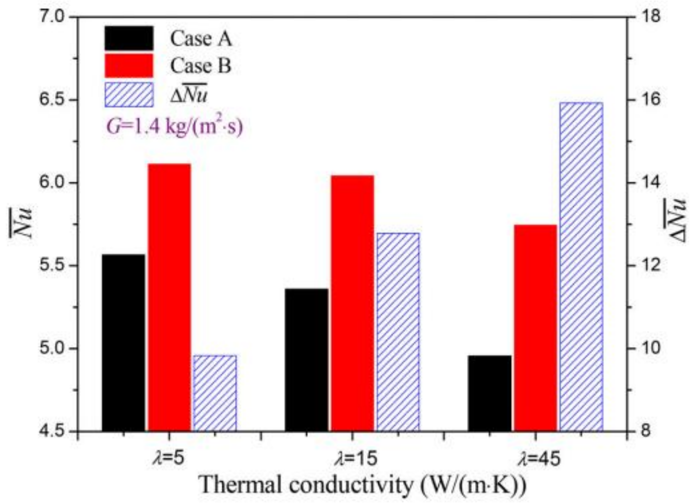


(b) q and Pr_b vs. $\overline{Re}$

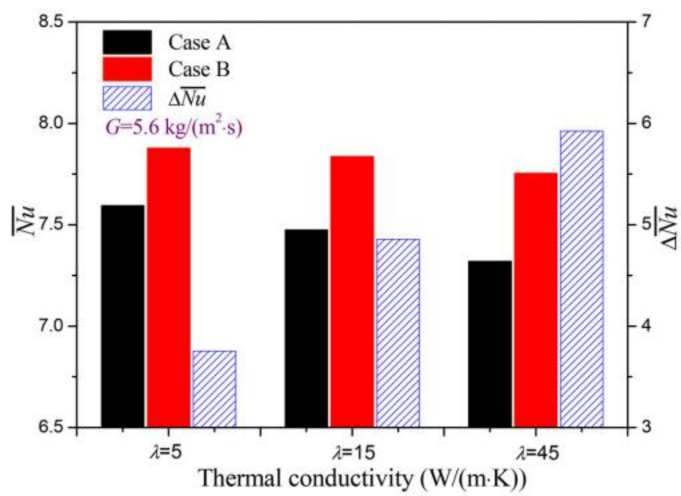
Figure 31:The variation of the heat flux and Pr_bparameter jointly on the wall conductivity *[39]*.

Judging by the numerical data, it can be noted that the heat flux at the walls with the conductivity of 5 and 15 W/m. K grows substantially in comparison with the cases of wall conductivity of 45 W/m.K. At low mass flow the increases are 16.7% and 11.6, respectively and at moderate mass flow, the increases are

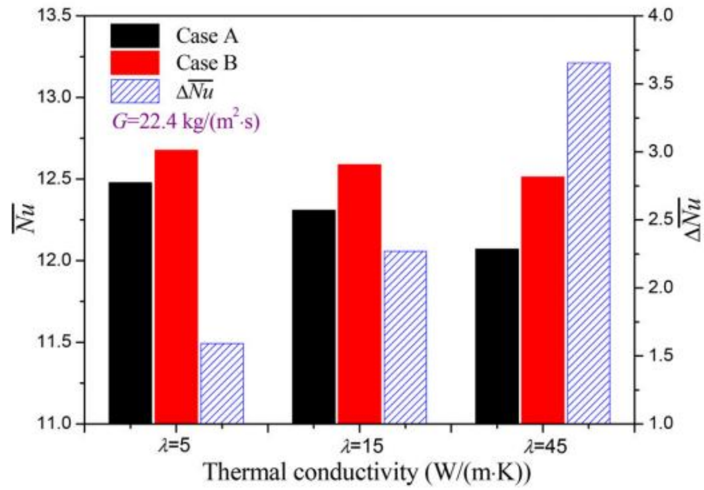
4.0% and 3.2. The gains are lesser in the condition of higher mass flow, at 2.5 and 1%. These results imply that walls with low thermal conductivity may be used with the view of enhancing the thermal performance of the PCHE. The improvement is primarily given by the influence of the axial heat conduction within the separating walls



(a) G=1.4 kg/(m^2·s)



(b) G=5.6 kg/(m^2·s)



(c) G=22.4 kg/(m^2·s)

Figure 32: Comparative evaluation of thermal performance of different wall material *[68]*.

Figure 32 shows the effect of the wall thermal conductivity on the thermal performance of Cases A and B. It is noted that the mean Nusselt number of Case B shows a rising tendency whereas the thermal efficiency variance between Case A and Case B reduces. This shows that the effect of wall thermal conductivity on the thermal performance of the Printed Circuit Heat Exchanger (PCHE) will be insignificant in high mass flux conditions.

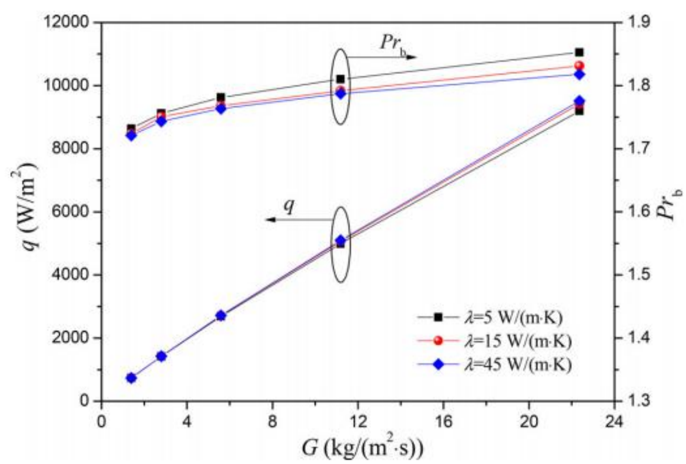


Figure 33: q *vs* Pr_b of Case B, at a number of thermal conductivities *[69]*.

Analytical comparison between the heat flux and the forced flow is made in figure 33 and Prandtl number of Case B with different values of wall thermal conductivity. Findings indicate that the heat flux is elevated as the wall thermal conductivity is elevated, but average Prandtl number is reduced as the wall thermal conductivity is elevated. The effect of the Prandtl number on increasing the Nusselt number is greater in this case that leads to increased Nusselt numbers at lower wall thermal conductivity values. Nevertheless, the comparative differences in Nusselt numbers between Case B and Case A are not as significant as shown in Figure 33.

5.2.2. Influence of Operating Pressure on Axial Heat Conduction

Operational pressure and inlet temperature are recognized as key parameters influencing the thermal efficiency of Printed Circuit Heat Exchangers (PCHEs). This section presents numerical simulations conducted to investigate the effect of axial heat conduction in the zigzag channel of a PCHE under various operating pressures. The simulations are carried out under the working conditions specified as Condition No. 2 in Table 1. During the heating process in the cold-side channel of the PCHE, supercritical liquefied natural gas (LNG) gradually experiences a temperature rise, accompanied by significant changes in its thermophysical properties. In this study, the Prandtl number is evaluated at the central cross-section of the cold side along the z-direction, perpendicular to the main flow direction, for a mass flux of $G =$ 5.6 kg/m² as illustrated in Figure 33 [38]. It is observed that the Prandtl number initially exhibits a high value and subsequently decreases along the flow direction at different operating pressures.

The thermal performance of the zigzag channel PCHE is assessed under varying operating pressures, as depicted in Figure 11, with mass flux plotted along the abscissa. It is evident that the mean Nusselt number increases with rising mass flux, while it decreases as operating pressure increases. This behavior can be explained as follows: lower operating pressures reduce the viscosity of the supercritical LNG within the channel, as shown in Figure 33, which in turn increases the Reynolds number under these conditions. Additionally, Figure 34 demonstrates that the Prandtl number of supercritical LNG decreases with increasing operational pressure. Since the Nusselt number is known to increase with both the Reynolds and Prandtl numbers, these two factors collectively result in an improvement of the thermal performance of the PCHE through the zigzag channel as the operating pressure decreases.

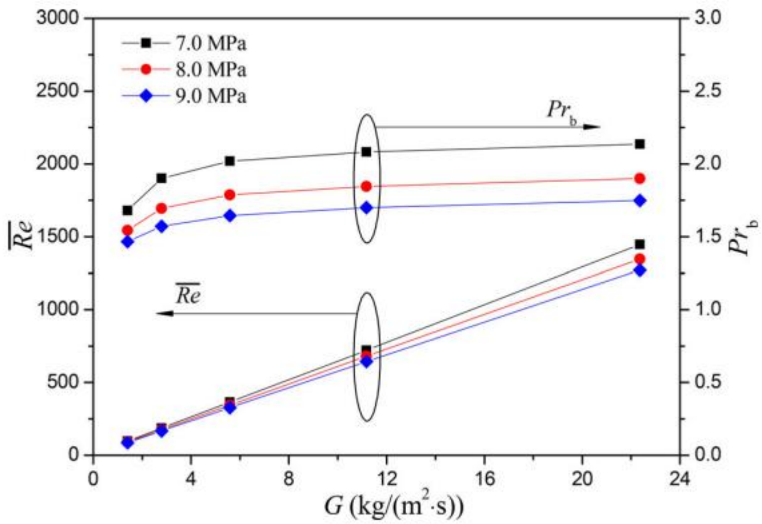
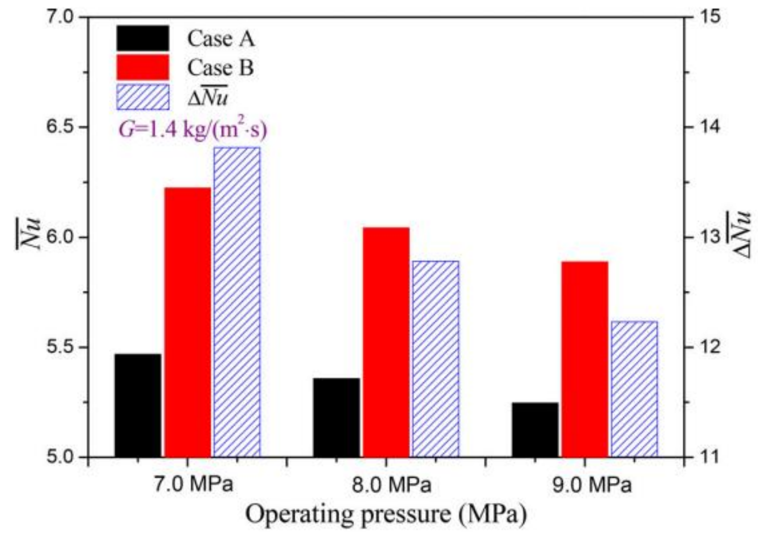
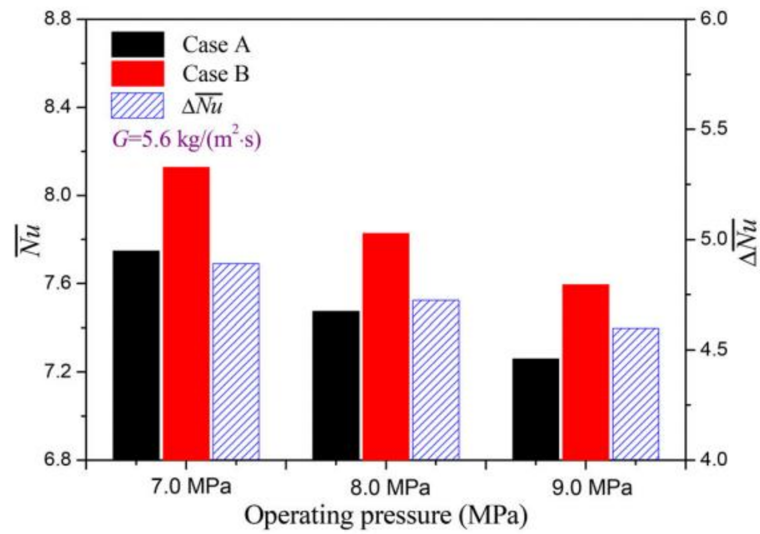


Figure 34: Effect of the operating pressure on the relationship between *Re and* Pr_b *[68]*.

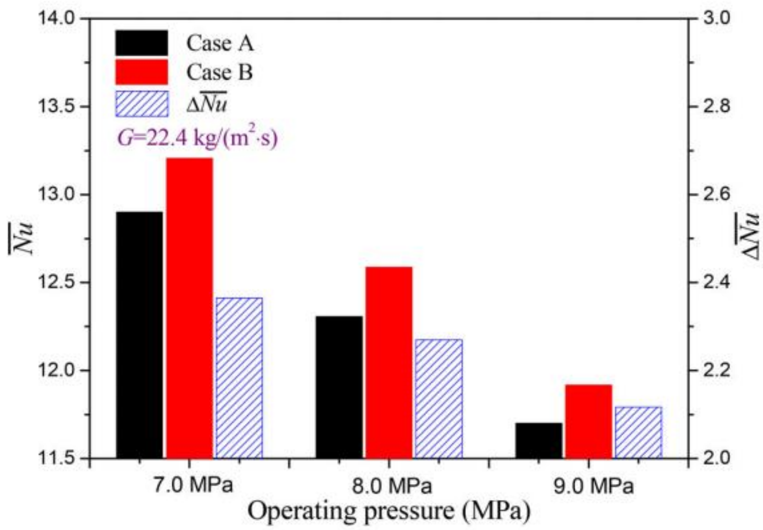
The effect of the operating pressure on the thermal performance of Case A and B is shown by Figure 33, the analysis of the figure reveals that the average Nusselt number of Case B increases gradually with the mass flux and the thermal performance difference between Case A and Case B also decreases slowly.



(a) G=1.4 kg/(m^2·s)



(b) G=5.6 kg/(m^2·s)



(c) G=22.4 kg/(m^2·s)

Figure 35: Response to near thermal performance at various operating pressure *[70]*.

At mass flux of, the differences between Case A and Case B are found to be 13.80%, 12.80 and 12.20 percent when the operating pressures are variable to 7.0 MPa, 8.0 MPa and 9.0 MPa respectively. G=1.40kg/m^2, and 4.9, 4.7 and 4.6 at greater mass flux of respectively. Such findings show that the axial heat conduction influences a great deal on the thermal performance of the Printed Circuit Heat Exchanger (PCHE) with low mass fluxes, and its effect is less prominent with high mass fluxes [66]. Figure 36 gives a comparison of the average Prandtl number in each row of the channel PCHE under varying operating pressure and mass flux of less than 0.01. It is noted that Prandtl number initially increases and subsequently decreases along the stream of the main flow with Case B having a higher Prandtl number as compared to Case A when mass flux conditions are held constant. The findings of the simulation process also indicate that the mean Prandtl number of Case A in the zigzag channel PCHE reduce by 11.5 and 18.6 percent at the operating pressure of 8.0 MPa and 9.0 MPa, respectively, compared to the condition of 7.0 MPa. This implies the operating pressure drives the average Prandtl number of Case B more significantly than the Case A and hence the rationale why the thermal performance difference between Case A and Case B decreases with the operating pressure [38].

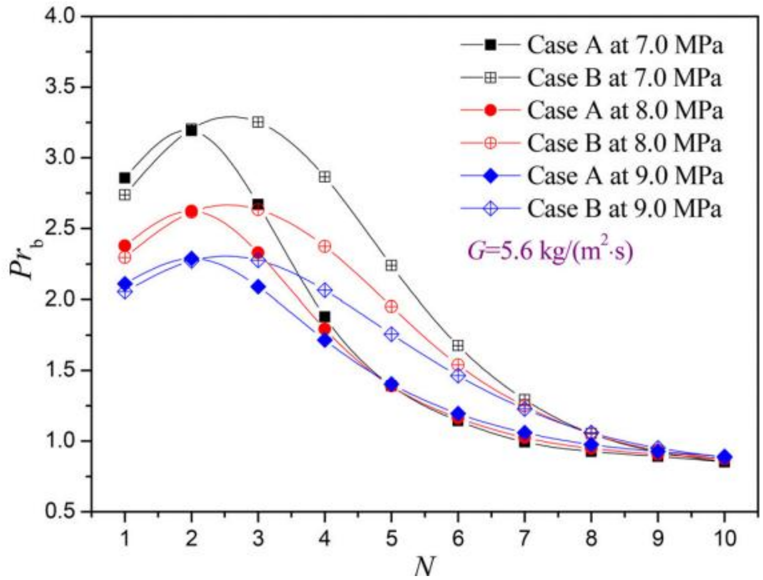


Figure 36: Pr_b profile when mass flow rate is set to 5.6 kg/m2 s *[69].*

5.3. Heat transfer performance evaluation of PCHE

Additional assessment of the heat transfer performance was done by showing the temperature difference between the cold and hot fluids and the ratio of flow rate in Figure 6b. Besides, Figure 6b shows the profiles of heat exchanger effectiveness as stipulated in Equation (25). Both the effectiveness (ε) and the values of D are monotonically decreasing with the flow rate ratio (Chang et al., 2021). Physically, effectiveness is the ratio of actual heat transfer rate to the maximum rate of heat transfer rate that is theoretically possible. Therefore, the higher the effectiveness, the higher the heat transfer performance. Figure 6b shows the highest efficiency of 0.979 when the fluid inlet temperature is 95 degC in hot fluid and the proportion of the flow rate at 0.1. Conversely, the lowest value is 0.428, which is realized at a hot fluid inlet temperature of 75 degC and flow rate ratio of 1. This shows that the PCHE gives maximum thermal performance at a high flow rate ratio which is a relatively low rate of cold fluid flow and a high rate of hot fluid flow. In general, the distributions of D and ε show a high correlation and prove the connection between the parameters of performance [34].

$$\varepsilon = \frac{\mathrm{NTU}}{1 + \mathrm{NTU}} \qquad (21)$$

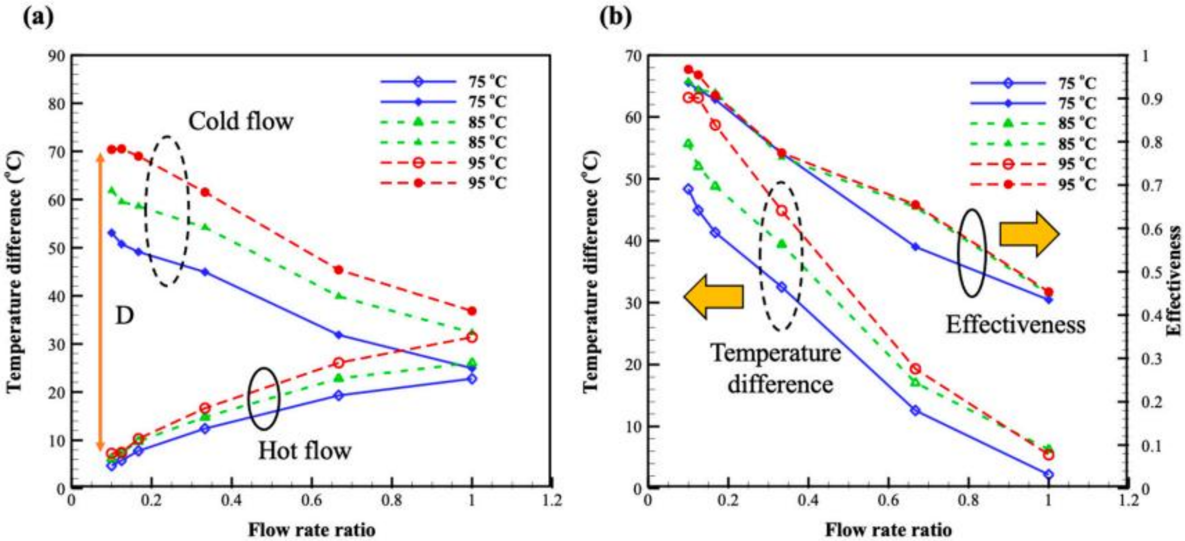


Figure 37: The temperature difference plot for different flow rate ratios and effectiveness (a) the temperature difference inlet and outlet of hot (solid line) and cold (dash line) fluid. (b) the temperature difference (solid line) between hot and cold fluid and effectiveness ess (dash line)

Figure 37 shows the trends of the convective heat transfer coefficient of the PCHE with different Reynolds numbers of the cold fluid flows. It is clear that the growth in Reynolds number has a positive effect on convective heat transfer process. The trend is in line with the results obtained in earlier studies [38]. As a result, the convective heat transfer coefficient will show a significant increment with the increase in the Reynolds number. To be more precise, the heat transfer coefficient is increased by an impressive factor when the inlet temperature of the hot fluid is altered, an increase of 67.8 per cent when the Reynolds number is altered between 50 and 300. At certain conditions of the other hot fluid inlet temperature, the increase is at least 64%. In the lower Reynolds numbers as in 50 and 100, the change in the hot fluid inlet temperature on the convective heat transfer coefficient is insignificant, and the change is not much. However, when the hot fluid inlet temperature is 95degC the convective heat transfer coefficient at the Inlet temperature of 300 Re is about 5 times larger than the coefficient at 75degC inlet temperature. Figure 37 is good enough in illustrating the variation between the convective heat transfer coefficient and the Reynolds numbers of the cold fluid flows, and that increased Reynolds numbers favor convective heat transfer. The same was observed in previous research [38].

The heat transfer coefficient also rises by 67.8 percent between Re = 50 and Re = 300, when the inlet temperature of the hot fluid is 95degC, whilst under other inlet temperature scenarios of the hot fluid, the rise is at least 64 percent. At low Reynolds numbers, that is Re = 50 and Re = 100, the change of the heat transfer coefficient in terms of hot fluid inlet temperature is not very sensitive, and the change so obtained is not significant. Nevertheless, the convective heat transfer coefficient when the fluid inlet temperature is hot, that is, 95degC, is nearly 5 percent higher than the one at 75degC. The correlation of the effectiveness and Reynolds number is indicated by figure 37. Unlike the convective heat transfer coefficient, the overall effectiveness of the heat exchanger reduces with an increase in the Reynolds number. Although the convective coefficient of heat transfer is positively correlated to the Reynolds number, the fluid moving at higher velocity decreases the residence time in the channel and, thus, it lowers the effectiveness of the heat exchanger. The PCHE flow plates have an S-shaped shape, which enhances the residence time of the working fluid showing that the residence time effect is more eminent than the flow rate. Thus, the heat exchanger is more effective when Reynolds numbers are low. Yan et al. [39] compared effectiveness of heat exchangers at different flow rate configurations and also, they found a similar case of decreasing trend of effectiveness with the increase in flow rates.

Figure 39 also serves to further support the argument that the effectiveness decreases with increasing Reynolds numbers as opposed to the convective heat transfer coefficient. The convective heat transfer coefficient has a positive correlation with the Reynolds number, meaning that the greater the amount of heat transfer in the cold fluid the greater will be the coefficient. However, the increased velocity of the fluid reduces dwell time of the cold fluid of the channel, and thus decreases the performance of the heat exchanger. The S-shaped flow plates of the PCHE increase the residence time of the working fluid and this means that the effect of residence time is greater than the influence flow rate. Consequently, the effectiveness reduces as the Reynolds numbers increase. The effectiveness also was investigated by Yan et al. at different flow rate regimes and showed the same downward trend in the performance of the heat exchanger with the rise in flow rate.

Figure 40 also analyses the correlation of Nusselt number (Nu) and the Reynolds number. Nusselt number is directly proportional to the convective heat transfer coefficient and it is dependent on the flow rate or Reynolds number. Consequently, the Nusselt number increases with a rise in the Reynolds number and the general pattern of the Nusselt number curves resembles the convective heat transfer coefficient curves.

The same patterns were expressed in research conducted by Yang et al. [80] on the performance of heat transfer in mini channels, which were designed with hexagonal fins under laminar flow regime. Figure 38 shows an impact of Nusselt number and Reynolds number in the channel. Nusselt number is dependent on the convective heat transfer coefficient and consequently, its dependency is on the flow rate or Reynolds number. The influence of this relationship is that the Nusselt number rises with Reynolds number and the trend showed is similar to the one of the convective heat transfer coefficient. The same performance of the heat transfer was found in a study [88] conducted on laminar flow through mini channels using hexagonal fins. When the Reynolds number increases by 68 percent between 50 and 300, the Nusselt number increases by 68 percent at a fixed rate of hot fluid inlet temperature of 95degC. Just like the convective heat transfer coefficient, the Nusselt number is also not that sensitive to changes in the inlet temperature of the hot fluid. Reynolds number dominates over the convective heat transfer coefficient and Nusselt number with the inlet temperature of hot fluid having little effect. The fundamental methodology needed to determine a quantitative relationship between the Nusselt number and the Reynolds number is a use of a logarithmic formulation:

$$\mathrm{Nu} = 0.084280\mathrm{Re}^{0.6135} \qquad (22)$$

Table 3 shows the performance analysis of various PCHE designs with regard to fluid flow properties and heat transfer. As can be obtained in the table, the flow passage of molten salt and the associated heat transfer resistance are the main factors that cause the pressure drop in PCHEs, especially when considering the flow configuration of supercritical (SCO_2). It follows that increasing the size of the flow passages of the molten salt can considerably decrease the pressure drop, and the overall heat transfer coefficient is not much affected. This means that the general performance of the PCHE is capable of being enhanced significantly.

As PCHE Model 1 is replaced by Model 3, the pressure drops of the molten salt will reduce by a small factor, as the total heat transfer coefficient decreases to 345 W/m 2 K, instead of 364 W/m 2 K. Consequently, the total performance optimization variable whereby Model 1 is the reference system grows by 1.0 to 1.20 indicating that the efficiency and effectiveness of the PCHE design improved.

Table 7:Flow and heat transfer performance for different PCHEs

	Design A	Design B	Design C
Overall HTC/$W. m^2.°C^{-1}$	364	369	457
HTC of molten sal/$W. m^2.°C^{-1}$	2202	2088	1890
HTC of SCO_2 /$W. m^2.°C^{-1}$	420	430	450
Molten salt pressure drop/Pa	1642	1202	1073
SCO_2 Pressure drop/Pa	510	505	400
Comprehensive performance optimization criterion	1.0	1.34	130

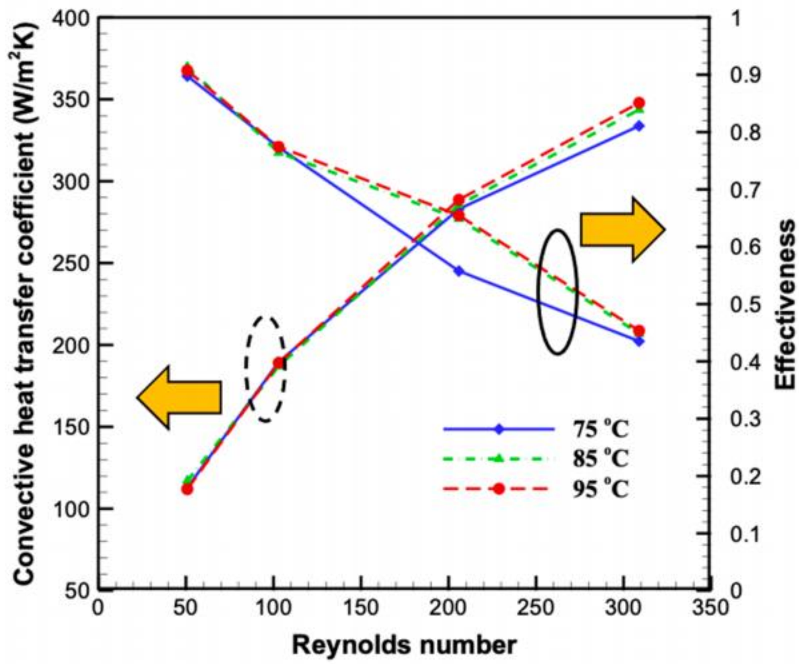


Figure 38:Convective heat transfer coefficient and effectiveness versus Reynolds number for different hot inlet temperatures *[42]*

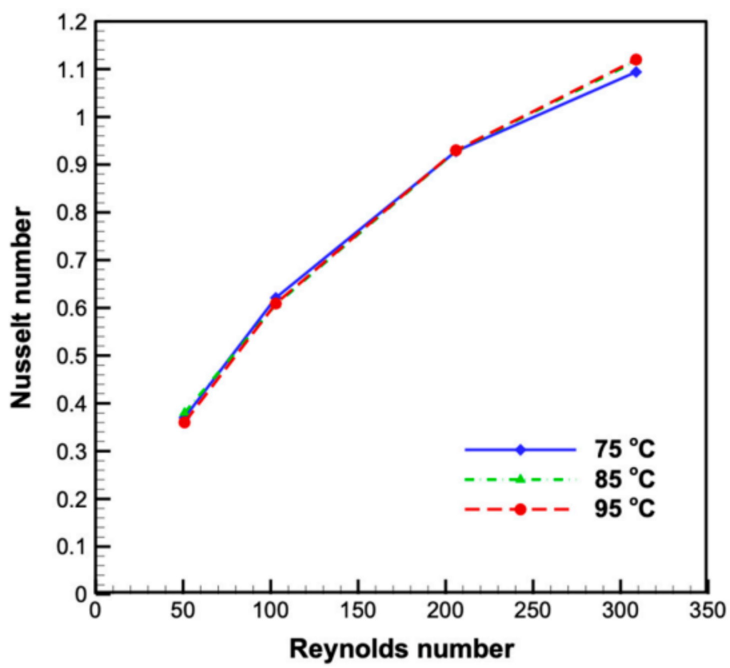


Figure 39:Nusselt number with different inlet temperatures for different Reynold numbers *[44]*

The density gradient of supercritical SCO_2 produces large buoyancy effects that make the high-temperature fluid rise along the wall, increasing heat transfer. Patterns of streamlines are well formed in the inlet region where some of the vortices are created mainly because of the variations in the spiral flow. The flow in the middle and outlet areas is also likely to approach eight streams of parallel flow as the buoyancy force becomes weak. The Natural convection decreases along the flow direction and the Richardson number impact reduces as a result of this which leads to gradual decrease of the local heat transfer coefficient of SCO_2. The heat transfer coefficient is however amplified in the outlet region because the turbulent heat transfer is enhanced [34]. The heat transfer and fluid behavior of the SCO_2 is largely influenced by the inlet temperature which influences the thermophysical properties and local heat transfer behavior of the fluid. The Richardson number of the super critical process is smaller than the trans critical process in the inlet region resulting in the local heat transfer coefficient being smaller. On the other hand, the supercritical process in the middle region and outlet region has more kinetic energy of turbulent flow than the transcritical process, and therefore, the local heat transfer coefficient is relatively higher.

The heat transfer in the SCO2 channels and pressure drop in the molten salt passages are the major factors that determine the overall flow and heat transfer performance of the PCHE. The use of molten salt flow passages in increasing numbers decreases the pressure drop by a large margin, and the overall heat transfer coefficient varies only slightly. As a result, the overall effectiveness of the PCHE is increased, which proves the efficiency of the optimization of the flow passage dimensions in molten salt.

Chapter 6

CHALLANGES AND LIMITATIONS

1. Trade-off between efficiency and pressure drop. Zigzag and other geometrically complex channels significantly enhance convective heat transfer and overall efficiency; however, this improvement is accompanied by a notable increase in pressure drop. Both numerical simulations and experimental observations indicate that effectiveness can improve by considerable percentages while concurrently necessitating a substantial increase in pumping power. This dynamic encapsulates the essential trade-off inherent in the design of printed circuit heat exchangers (PCHE) for Brayton cycles, wherein the net efficiency of the cycle is substantially influenced by the pressure losses incurred by the recuperator.

2. Axial conduction coupled with thin plate interactions. In the context of zigzag or serpentine PCHEs, the localized effectiveness can be significantly influenced by axial heat conduction occurring between the solid plates and the fins, potentially resulting in diminished temperature gradients available for convective heat transfer. Numerous scholarly investigations have highlighted that the exclusion of axial conduction can yield excessively optimistic performance outcomes. This phenomenon is particularly critical in the case of compact cores composed of materials with high thermal conductivity.

3. Constraints imposed by mechanical design. The manufacturability of these devices is constrained by the techniques employed, such as photochemical etching combined with diffusion brazing, which restrict the minimum and maximum feature sizes, the thickness of the plates, and the quality of the brazing process. Various mechanical challenges, including substantial thermal stresses due to large temperature differentials, fatigue from cyclic loading, integrity of brazed joints, and sealing issues between channels, may contribute to a reduction in lifespan or necessitate conservative design approaches that adversely affect thermal performance. Additionally, considerations surrounding nontrivial damage tolerance and the need for rigorous inspection protocols are paramount.

4. Challenges posed by scale and contamination. The impacts of fouling, particulate matter, or erosion caused by entrained solids (or the long-term deposition of such materials) may not be accurately represented in laboratory or small-scale experimental setups. The accumulation of fouling within small PCHE channels can precipitate a rapid decline in performance and result in uneven flow distribution. There exists a paucity of long-duration experimental data conducted under conditions reflective of realistic contaminants and operational cycling. This assertion is indirectly supported by performance-related publications, which acknowledge the necessity for long-term testing in general.

5. Economic considerations of costs and trade-offs. The pursuit of high-precision manufacturing processes (involving thin plates, intricate cellular configurations, and exotic alloys capable of withstanding high-temperature supercritical carbon dioxide) invariably escalates unit costs and increases fabrication complexity. The existing literature on economic optimization, particularly concerning the interplay between capital expenditures and operational costs including pumping power and maintenance is relatively underdeveloped and requires further application within commercial contexts.

Chapter 7
CONCLUSION AND RECOMMENDATIONS

7.1 Conclusion

Printed circuit heat exchangers (PCHEs) [13] have been identified as one of the best potential technologies in the universe of recuperators in the miniature high-efficiency Brayton-type systems; these technologies exhibit remarkable surface-area density and possess the capability to withstand environments characterized by elevated pressure and temperature. Available scholarly research indicates that the geometric design is paramount: configurations employing zigzag arrangements and cellular layouts demonstrate the potential to significantly enhance heat transfer efficiency; nevertheless, it is imperative for designers to judiciously evaluate the associated costs of pressure drop and secondary phenomena, including axial conduction, fin efficiency, and mechanical or joining limitations [71]. To actualize application-ready PCHEs in reaction systems, such as He-Xe microreactors or supercritical carbon dioxide (sCO_2) Brayton cycles, the trajectory towards practical application of this technology necessitates (1) the optimization of geometry through multi-objective optimization methodologies, (2) comprehensive experimental evaluation of performance across diverse operating conditions and fluid combinations, (3) meticulous selection of materials and brazing techniques to mitigate thermal loads and the risk of leakage, and (4) consideration of manufacturability along with (5) the long-term behavior concerning fouling and maintenance. Adhering to these measures will facilitate the attainment of the highest possible thermal performance without jeopardizing reliability and economic viability, thereby enabling the transition from appealing numerical and academic findings to sustainable industrial outcomes [72].

7.2 Recommendations

- Establish lower order models and correlations to be utilized in engineering applications. Nusselt number (Nu), friction factor (f) from computational fluid dynamics (CFD) and experimental results are transformed into design-level correlations (Nu, f against Reynolds number, geometric parameters, and axial conduction correction factors) to facilitate system designers in promptly evaluating the trade-offs without necessitating comprehensive CFD analyses. Existing literature already explores straight, zigzag, and serpentine channel configurations, providing a foundational dataset for comparative analysis.

- Investigate the complexity that can be achieved: cellular and three-dimensional channels through advanced processes. Cellular zigzag and three-dimensional wavy or sinusoidal channels exhibit favorable thermal performance; the feasibility of economically scaling these configurations to larger dimensions remains contingent upon techniques such as photochemical etching combined with brazing or additive manufacturing derived from cut sheets. Within the framework of comparative studies, it is imperative to incorporate life-cycle assessments and cost modeling.

- Early adoption of multi-objective optimization concerning effectiveness, pressure drop (Δp), mass flow, and mechanical stress. Channel geometries (for instance, cellular-zigzag, modified wavy, and straight segments inserted) should be optimized through the integration of coupled CFD and structural solvers alongside Pareto optimization, thereby enabling the attainment of a balance between heat transfer enhancement, pressure loss, and mechanical stress constraints. Numerous studies indicate that cellular zigzag or inserted straight channels have the potential to augment the overall heat transfer coefficient (U) and effectiveness while maintaining a moderate Δp, thereby positioning them as prime candidates for optimization endeavors.

- Conduct experiments on representative thermodynamic states and compositions. For the systems of helium-xenon (He-Xe) and supercritical carbon dioxide (sCO_2), it is essential to conduct experiments and validations across various mole fractions and within pseudo-critical ranges, specifically in the context of sCO_2. Reliance on single-point numerical validation is inadvisable; instead, a broader mapping of fluid property sensitivity is required

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