

DESIGN AND CHARACTERIZATION OF AN ADDITIVELY
MANUFACTURED HEAT EXCHANGER WITH TRIPLY PERIODIC MINIMAL
SURFACE CORE

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**DESIGN AND CHARACTERIZATION OF AN ADDITIVELY
MANUFACTURED HEAT EXCHANGER WITH TRIPLY PERIODIC
MINIMAL SURFACE CORE**

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ABSTRACT

DESIGN AND CHARACTERIZATION OF AN ADDITIVELY MANUFACTURED HEAT EXCHANGER WITH TRIPLY PERIODIC MINIMAL SURFACE CORE

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Triply periodic minimal surface (TPMS) structures are being used more frequently as a result of growing interest in additive manufacturing (AM), as intricate shapes like TPMS structures cannot be manufactured using conventional fabrication methods. Having tailorable properties such as relative density, wall thickness and cell size, mathematically regulated TPMS based structures can have intriguing usage in engineering applications. Naturally dividing the volume that they cover into two makes TPMS based structures very handful in heat transfer applications, whereas, having high stiffness values while having less material makes TPMS structures also applicable for light-weighting designs for space applications. This study focuses on a compact counterflow heat exchanger (CCHEx) designed by utilizing the high surface area to volume ratio of different TPMS structures. The heat transfer areas for three different TPMS structures are calculated inside a prescribed heat exchanger volume. Among many other design options, a Gyroid based heat exchanger is manufactured using 17-4 PH. stainless steel powder. The manufactured prototype is designed considering the AM design criteria so that no support structures are needed. Furthermore, a test setup capable of measuring the pressure, temperature, and flow rate data for the

fluid sides at the inlet and outlet sections is constructed. Test results are validated using computational fluid dynamics simulations so that many other heat exchanger designs can be evaluated without manufacturing and testing. Using the workflow generated, various types of TPMS structures can be analyzed and compared for their thermal performance and volume requirements in laminar flow region.

Keywords: TPMS Structures, Heat Exchangers, Additive Manufacturing



ÖZ

EKLEMELİ İMALAT İLE ÜRETİLEN ÜÇLÜ PERİYODİK MİNİMAL YÜZEY YAPILI EŞANJÖR TASARIMI VE KARAKTERİZASYONU

Cevahir, Burak

Yüksek Lisans, Makina Mühendisliği Bölümü

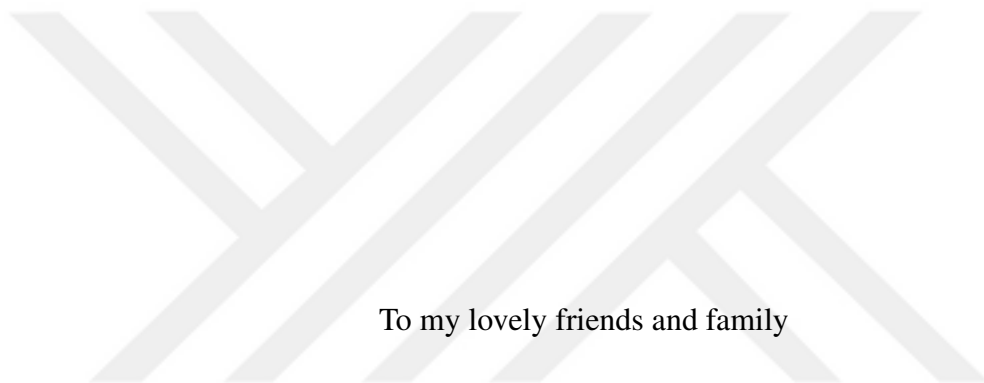
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Üçlü periyodik minimal yüzey (TPMS) yapıları, geleneksel imalat yöntemleriyle üretilemeyen karmaşık şekillerin üretilmesini mümkün kılan eklemeli imalat alanındaki artan ilgi nedeniyle daha sık kullanılmaktadır. Bağlı yoğunluk, duvar kalınlığı ve hücre boyutu gibi uyarlanabilir özellikleri olan matematiksel olarak düzenlenmiş TPMS bazlı yapılar, mühendislik uygulamalarında ilginç kullanım alanlarına sahip olabilmektedir. Tanımları gereği kapladıkları hacmi iki bölgeye ayırmaları, TPMS bazlı yapıları ısı transferi uygulamalarında oldukça kullanışlı hale getirirken, daha az malzeme ile daha benzer veya daha yüksek parça dayanımları elde edilebilmesi TPMS yapılarını uzay uygulamalarına yönelik hafif tasarımlarda da uygulanabilir kılmaktadır. Bu çalışma, farklı TPMS yapılarının yüksek yüzey alanı/hacim oranından yararlanarak tasarlanan ters akışlı kompakt ısı değiştirgeci (CCHEx) tasarımlarına odaklanmaktadır. Bu çalışmada kullanılan TPMS yapılarının ısı transferi yüzey alanları, çeşitli kısıtlarla belirlenen bir ısı değiştirgeci (eşanjör) hacminde hesaplanmıştır. Birçok farklı tasarım seçeneğinden biri olan Gyroid bazlı bir ısı değiştirgeci, 17-4PH paslanmaz çelik malzeme ile eklemeli imalat ile üretilmiştir. Üretilen ısı değiştir-

geci, eklemeli imalat tasarım kriterleri gözetilerek, çıkartılması imkansız olacak olan destek yapılarına ihtiyaç duymayacak şekilde tasarlanmıştır. Ayrıca, soğuk ve sıcak akışkanların ısı değiştirgecine giriş ve çıkış kısımlarından basınç, sıcaklık ve debi değerlerini okuyabilen bir test düzeneği kurulmuştur. Yapılan testlerin sonuçları, hesaplamalı akışkanlar dinamiği simülasyonları ile doğrulanmıştır. Bu çalışma, yalnızca kullanılan TPMS yapıları içerisinde geçen laminar akışı kapsamaktadır. Oluşturulan iş akışı kullanılarak, çeşitli TPMS yapıları laminar akış bölgesinde, üretilmeye ve test edilmeye ihtiyaç duymadan termal başarımları ve hacim gereksinimleri açısından analiz edilebilir ve karşılaştırılabilir.

Anahtar Kelimeler: TPMS Yapıları, Isı Değiştirgeci, Eklemeli İmalat



To my lovely friends and family

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LIST OF ABBREVIATIONS

ABBREVIATIONS

AM	Additive Manufacturing
CCHEX	Compact Counterflow Heat Exchanger
CFD	Computational Fluid Dynamics
CHEX	Compact Heat Exchangers
CPU	Central Process Unit
DFAM	Design for Additive Manufacturing
EDM	Electrical Discharge Machining
FEA	Finite Element Analysis
FDM	Fused Deposition Modeling
FFF	Fused Filament Fabrication
HEX	Heat Exchanger
IPMS	Infinite Periodic Minimal Surfaces
LCD	Liquid Crystal Display
LMTD	Logarithmic Mean Temperature Difference
NI	Natural Instruments
NTU	Number of Transfer Units
PBF	Powder Bed Fusion
PHEX	Plate Heat Exchangers
SLA	Stereolithography
SLM	Selective Laser Melting
STL	Standard Tessellation Language
TPMS	Triply Periodic Minimal Surface
VDF	Volumetric Distance Fields

Nomenclature

Variables

x	Position or length dimension, m
D	Diameter, m
ρ	Fluid density, kg/m ³
v	Velocity, m/s
v_d	Darcian velocity, m/s
v_p	Pore velocity, m/s
d_h	Hydraulic diameter, m
μ	Dynamic viscosity, kg/(m·s)
Re	Reynolds number
Pr	Prandtl number
Ra_L	Rayleigh number, based on length L
Nu_L	Nusselt number, based on length L
c_p	Specific heat at constant pressure, J/(kg·K)
k	Thermal conductivity, W/(m·K)
ϕ	Porosity or function of state variables
V	Volume, m ³
q	Heat transfer rate, W
p	Pressure, Pa
Δp	Pressure drop, Pa
h	Heat transfer coefficient, W/(m ² ·K)
T	Temperature, K
ΔT	Temperature difference, K
A	Area, m ²
f	Friction factor
β	Thermal expansion coefficient, 1/K
g	Acceleration due to gravity, m/s ²
ν	Kinematic viscosity, m ² /s
L	Length, m
ϵ	Effectiveness

$\dot{m}$	Mass flow rate, kg/s
$\dot{V}$	Volumetric flow rate, m ³ /s
U	Overall heat transfer coefficient, W/(m ² ·K) or Uncertainty
NTU	Number of transfer units
C	Heat capacity rate, W/K
C^*	Heat capacity rate ratio
η_p	Pump efficiency
P	Power, W
t	Thickness, m
N	Number
A_{sf}	Total solid - fluid interaction area, m ²
a_{sf}	Specific solid - fluid interaction area, 1/m

Subscripts

lam	Laminar
lm	Logarithmic mean
fd, h	Hydrodynamic, fully developed
fd, t	Thermal, fully developed
in	Inlet
out	Outlet
$total$	Total quantity
$cold$	Cold fluid
hot	Hot fluid
sim	Simulation
max	Maximum
$conv$	Convection
$core$	Core area or volume
$wall$	Wall quantity
$pipe$	Pipe quantity
s	Surface quantity
∞	Ambient quantity





CHAPTER 1

INTRODUCTION

The focus of this research is to generate a parametric design workflow for a heat exchanger based on Triply Periodic Minimal Surfaces (TPMS). Furthermore, the manufacturing of a prototype heat exchanger was also a focal point, allowing the practical application and validation of theoretical models and simulations. With experimentation and Computational Fluid Dynamics (CFD) analysis, laminar flow and heat transfer characteristics within the TPMS-based heat exchangers are investigated.

1.1 Background

1.1.1 Overview of Conventional Heat Exchanger Designs

Heat exchangers play an important role in optimizing thermal energy in a range of industrial and residential applications. They are essential components for exchanging heat between two or more fluids while keeping the fluids mixed or unmixed, depending on the application. From the climate control systems of any building to manufacturing processes in the industry, it is almost impossible not to run into a heat exchanger. Heat exchangers also ensure the proper functioning of all daily technological devices, such as the cooling of Central Process Units (CPUs) in our computers or the keeping the engines of our cars from overheating [2].

Heat exchangers are classified based on a variety of characteristics. The most widely used classifications are on the basis of their flow arrangements and building processes. In terms of flow arrangements, there are three main categories for a single-pass heat exchanger. These are parallel flow, counter flow, and cross flow configurations. In

parallel flow configurations, the entrance of the cold and hot side fluids is on the same side of the heat exchanger, meaning that their flow directions are parallel to each other. In contrast, in counter flow configurations, cold and hot fluids enter the heat exchanger on the opposite sides, flowing in reverse directions. Furthermore, in cross flow configurations, fluid flow directions are perpendicular to each other. Each of the three configurations has its own benefits and drawbacks. The counter flow arrangement provides the most temperature difference between two fluids at any given section inside the heat exchanger, while the parallel flow configuration provides the least. Cross flow configuration is at the midst of these flow arrangements.

Although other types of constructions are available for heat exchangers, there are four main construction types and they are tubular, plate type, extended surface and regenerative heat exchangers shown in Figure 1.1. A typical example of a tubular heat exchanger is a shell and tube heat exchanger, whereas plate-type heat exchangers are mostly represented by gasketed plate heat exchangers (PHEX). For extended surface heat exchangers, plate-fin type heat exchangers can be a common example, while rotary regenerators fall into the category of regenerative type heat exchangers [1].

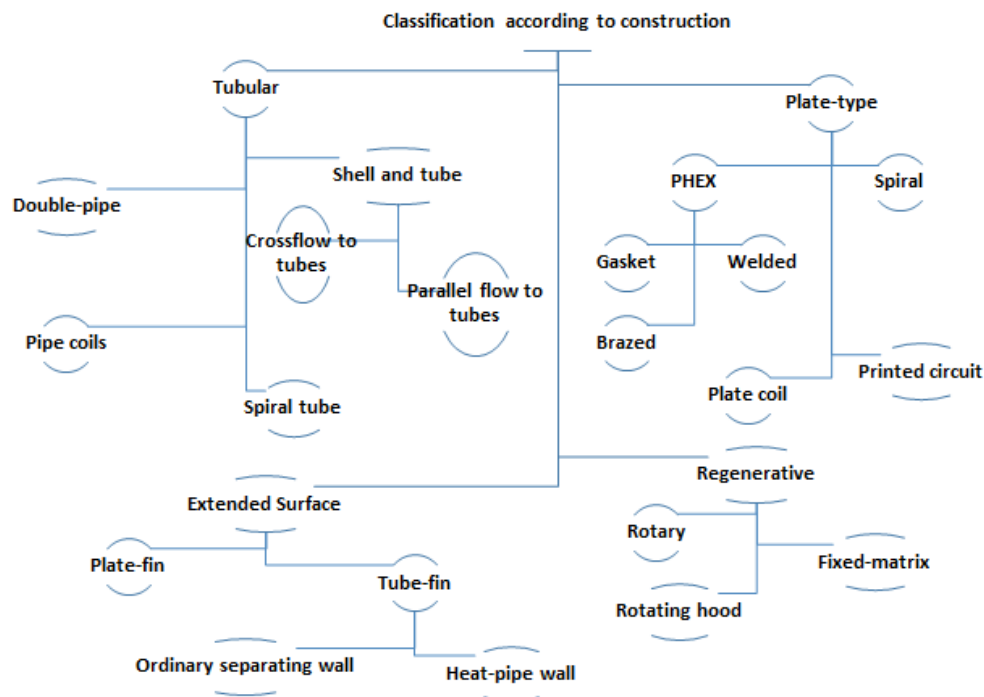


Figure 1.1: Classification of Heat Exchangers According to Construction Type [1]

1.1.2 Motivation for Using Additive Manufacturing in Heat Exchanger Applications

As in any other engineering applications, heat transfer applications need to be as efficient as possible to minimize wasted energy during the application. By minimizing wasted energy, efficient heat transfer prevents the application equipment from overheating. This leads to a reduction in operating and maintenance costs while extending the lifespan of the equipment. The minimum amount of wasted energy is also crucial for the least environmental impact. It can also help with the safety of the application in high temperature industries.

As technology advances, the design requirements for heat transfer applications also intensify. This pushes designers to design more compact, yet more efficient heat transfer geometries or alter the commercially available heat exchanger designs to meet higher performance requirements. For applications that require case-specific heat exchangers, AM can be utilized to optimize the heat exchanger designs summarized in Section 1.1.1 as it offers the possibility of manufacturing complex geometries that are impossible to manufacture with traditional manufacturing techniques.

For example, El Achkar et al. [3] altered a plate-fin type heat exchanger design using lattice structures inside the channels rather than conventionally manufactured fin structures and characterized the new design according to mass flow rates using experimentation and CFD analysis. Another example of the utilization of AM in conventional heat exchanger designs is the study carried out by Korobiichuk et al. [4]. Using CFD analysis, they compared different types of plate geometry that are easier to manufacture with AM. Furthermore, the study by Kaur et al. [5] focuses on metal lattice fabrication with AM and explores the potential of AM to replace traditional methods in heat transfer applications.

1.1.3 Development of TPMS Structures and Potential Applications

The first mentioned TPMS are Primitive and Diamond surfaces, and they were discovered by German mathematician Schwarz in 1865. With his student Neovius contributing by the discovery of the Neovius surface, no other TPMS type was discovered

until 1970. Alan H. Schoen made important contributions by introducing numerous more TPMS such as Gyroid and I-WP surfaces [6]. In earlier studies, these surfaces were called Infinite Periodic Minimal Surfaces (IPMS). In 1987, Fischer and Koch made further contributions to the TPMS field, such as the discovery of Fischer-Koch S type surface [7].

TPMS surfaces have the unique ability to minimize the surface area within their boundaries. The mean curvature at any point on the TPMS is also zero. These surfaces periodically extend in three dimensions without sharp edges or corners [8]. All types of TPMS are smooth surfaces since they are based on trigonometric functions. By adapting the parameters in the trigonometric equations that define each TPMS, designers can adjust properties such as the relative density and wall thickness of the resulting TPMS [9].

Table 1.1: TPMS Types and Their Trigonometric Surface Equations

TPMS Type	Corresponding Trigonometric Surface Equation
Schoen Gyroid	$\sin X \cos Y + \sin Y \cos Z + \sin Z \cos X = c$
Schwarz Diamond	$\cos X \cos Y \cos Z - \sin X \sin Y \sin Z = c$
Schwarz Primitive	$\cos X + \cos Y + \cos Z = c$
Fischer-Koch S	$\cos 2X \sin Y \cos Z + \cos X \cos 2Y \sin Z + \sin X \cos Y \cos 2Z = c$
Schoen I-WP	$2(\cos X \cos Y + \cos Y \cos Z + \cos Z \cos X) - (\cos 2X + \cos 2Y + \cos 2Z) = c$
Schoen F-RD	$4(\cos X \cos Y \cos Z) - (\cos 2X \cos 2Y + \cos 2Y \cos 2Z + \cos 2Z \cos 2X) = c$
Neovius	$3(\cos X + \cos Y + \cos Z) + 4(\cos X \cos Y \cos Z) = c$

In Table 1.1, trigonometric equations for seven different TPMS types are given and visualized in Figure 1.2. Visuals in Figure 1.2 are generated using nTopology. Unit cell sizes for all types of TPMS are determined by the period of the governing trigonometric function.

Multiplying the coordinate terms within the cosine and sine functions by the Period of the trigonometric function / Desired cell size, the unit cell size for a TPMS structure can be adjusted for each axis.

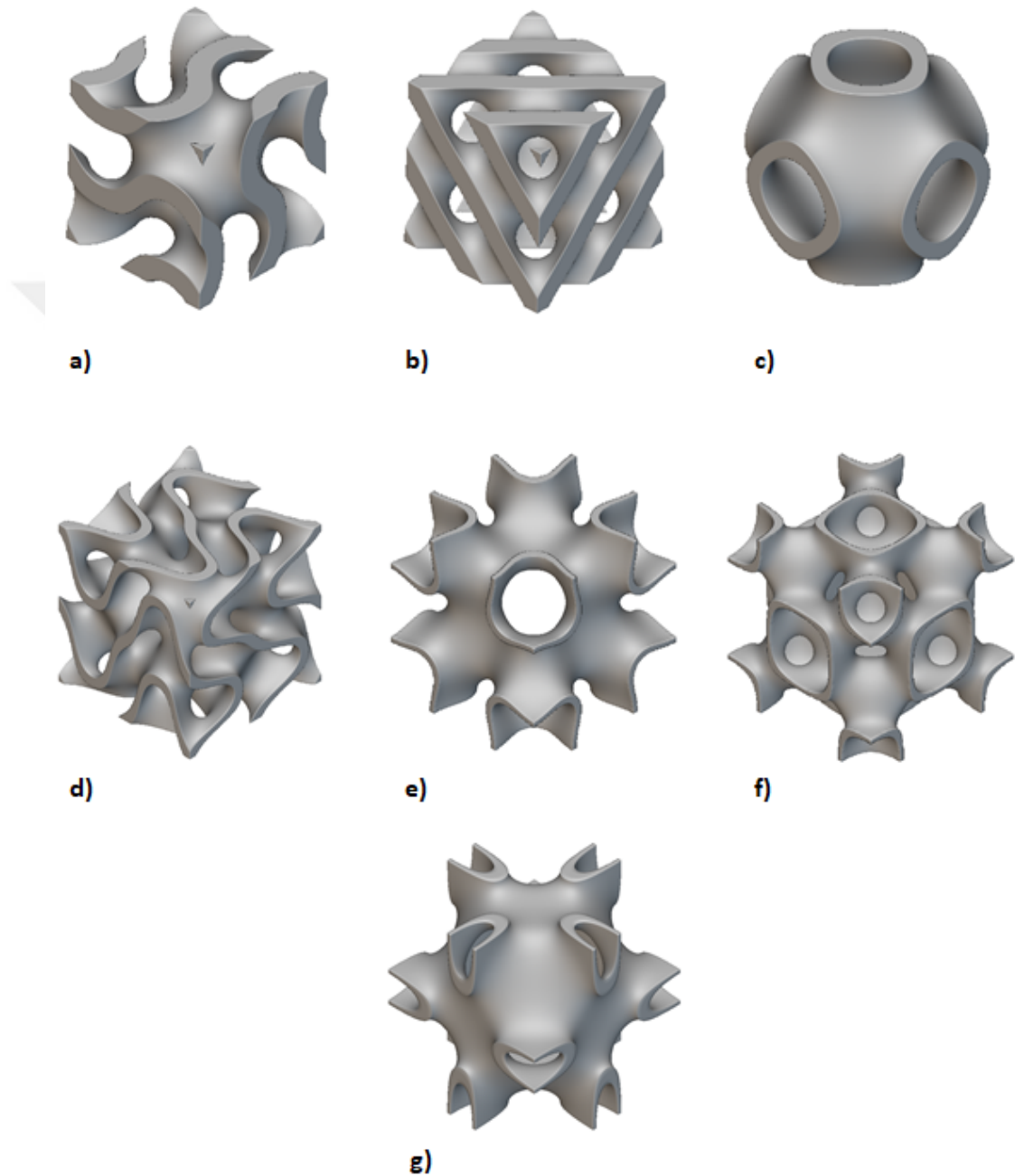


Figure 1.2: Examples of TPMS Structures: a) Schoen Gyroid, b) Schwarz Diamond, c) Schwarz Primitive, d) Fischer-Koch S, e) Schoen I-WP, f) Schoen F-RD g) Neovius

The applications of TPMS in thermal energy conversion are not limited to the heat exchanger applications mentioned in Section 1.1.2. Xu et al. [10] studied the solar energy conversion performance of TPMS structures and developed a numerical model for the correlation between the porous structure used and its properties for solar energy conversion. Having adjustable geometric and mass transfer properties gives designers the opportunity to easily characterize TPMS structures and use them in a vast variety of applications.

While being mathematically defined surfaces, TPMS are encountered in nature as well. Al-Ketan et al. [11] mentioned that surfaces resembling TPMS are found in soap films, while Han et al. [12] pointed out that TPMS structures are found in natural systems such as beetle shells and zeolite crystals. When all of these findings are taken into account, TPMS structures have the potential to be used in biomimicry applications. Ullah et al. [13] manufactured Gyroid based bone scaffolds with Stereolithography (SLA) and showed that TPMS-based scaffolds have the possibility of being used in bone repair applications. Dong et al. [14] state that, since conventional bone repair applications have high costs, the need for easy to manufacture bone replacements arises. By manufacturing and examining different Gyroid scaffolds, their study concluded that the mechanical properties and permeability of the scaffolds can be adjusted to meet specific requirements.

Moreover, TPMS structures can be used in lightweighting applications by replacing or filling parts with lattice structures. Zhao et al. [15] suggested a design approach for the optimization of TPMS structures used for direct replacement of parts. With their approach, lightweight and high stiffness parts can be achieved. A further lightweighting approach is to produce lighter TPMS structures with the same or better mechanical properties. Ormiston et al. [16] investigated the mechanical properties of the fiberglass reinforced lattice and TPMS structures. The mechanical properties of the low density fiberglass reinforced structures exceeded those of the higher density structures without reinforcement while remaining lighter. Furthermore, Weber et al. [17] produced TPMS foam structures and tested their mechanical properties and showed that better mechanical properties can be achieved by the foaming process. Both Ormiston et al. [16] and Weber et al. [17] suggested that their methods also increase the energy absorption performance of lattice structures, implying that

they can be used in automotive and aerospace applications where lightweight energy absorption means are required.

1.2 Problem Statement

In conventional heat exchangers, there are several limitations and downsides related to their working temperatures, the geometry of heat exchanging surfaces that can be manufactured with conventional methods, and their manufacturing and assembly costs [18].

The working temperature range for conventional heat exchangers is limited by the maximum temperatures that assembly means such as brazing, welding, or the gaskets used between plates can endure. Furthermore, the combination of many components often results in points of weakness that can lead to failures that may not be detectable until the heat exchanger is put into use.

In addition, the performance of heat exchangers is constrained by the manufacturing limitations of heat transfer surfaces produced by conventional manufacturing methods. Assuming that the heat transfer surface geometry is feasible, the heat exchanger performance is typically improved by physically including more of the same heat transfer surface geometry to the heat exchanger. This would lead to less compact heat exchangers. When custom heat transfer surfaces are required for an application, the incorporation of custom geometry into existing standard heat exchangers not only complicates the design but also increases the manufacturing and assembly costs.

With the advantages of AM, the surface area to volume ratio for heat transfer surfaces can be improved. AM can also lead to weight savings. Another benefit of AM is that heat exchangers can be manufactured in one piece. This mitigates the limitations of the assembly features and assembly related failure concerns [19]. However, replacement of defective parts is not an option. All AM benefits and drawbacks also apply to the utilization of TPMS structures as heat transfer surfaces in heat exchangers. In this study, a heat exchanger is manufactured via AM. For the heat transfer surfaces of the said heat exchanger, TPMS structures are selected as they separate the volume into two smooth interlocking channels with a high surface area to volume ratio.

Due to their inherent smoothness, stagnant flow areas are avoided in TPMS structures [20]. Moreover, with various types and customizable properties, flow characterization through TPMS structures is a compelling field for researchers.

1.3 Objectives

The primary objective of this thesis is to conduct a comparative analysis of fluid flow inside different TPMS structures while generating a parametric design workflow for single piece TPMS based heat exchangers. To achieve the primary objective, the following milestones are set.

- Manufacture a prototype heat exchanger with a Gyroid type TPMS core using AM
- Construct a flow test setup that is capable of supplying constant temperature and pressure hot and cold fluids inside the prototype
- Conduct flow tests with the prototype and validate the results through CFD analysis
- Design and analyze heat exchangers with different TPMS structures and cell sizes through CFD analysis
- Compare different TPMS structures as heat exchanger cores
- Compare analytical calculations for heat exchanger applications with the conducted experiments and CFD analysis
- Compare empirical relations regarding the flow characteristics inside TPMS structures with the data obtained through experiments and CFD analysis

1.4 Scope of the Study

The workflow generated in this work can be applied to design various other types of heat exchangers. The flow characteristics obtained in this work apply only for the

laminar flow inside the Schoen Gyroid, Schwarz Diamond, and Schwarz Primitive TPMS structures. The experimental test setup can also be a guide for other applications, such as the experimentation of cold plate heat exchangers.

1.5 Limitations of the Study

The flow regime considered in this work is limited to laminar flow within three types of TPMS structures, as the laminar flow regime inside TPMS structures is investigated more in the literature [21]. Furthermore, the most Gyroid, Primitive and Diamond structures are the most studied types of TPMS structures [22]. Using the most researched flow regime and TPMS types, the design approach and workflow in this study are aimed to be verified by comparing the results with those in the literature.

There are also manufacturing limitations encountered during the study. The powder material allotted for this study limits the dimensions and attempts to manufacture prototypes. With the given material, the outer dimensions of the prototype are limited to the values given in Table 3.1. Furthermore, the powder material is 17-4PH as mentioned in Chapter 3. Since only a single prototype could be manufactured, the TPMS type is chosen to be Gyroid, having the smallest bridge distances. Moreover, the unit cell size of the TPMS in the prototype is selected to be 5 mm in all directions to reduce the bridging distance.



CHAPTER 2

LITERATURE REVIEW

This chapter examines the development of heat exchanger designs. The influences of AM and TPMS structures in engineering applications are reviewed along with the previous studies on the TPMS based heat exchangers.

2.1 Additive Manufacturing in Heat Exchanger Applications

AM has become an important development in the field of heat exchanger applications. It provides more than just a replacement for conventional manufacturing techniques. This chapter examines the potential for AM to improve heat exchanger performance, materials, and design.

Traditionally, the constraints of subtractive manufacturing techniques have limited heat exchanger designs. But AM provides a change in perspective by making it possible to design unique heat exchanger designs. For instance, Thompson et al. [23] explored the potential of Selective Laser Melting (SLM) to create a novel flat plate heat exchanger with circular micro channels. Several design and manufacturing considerations were made to allow the SLM to produce the heat exchanger, such as including powder removal holes on the designed part. Compared to the solid heat exchanger alternative, much better performance was obtained. Likewise, Unger et al. [24] developed complex fin structures for tube fin heat exchangers using SLM technology, experimenting with different Reynolds numbers and fin geometries. A formula that estimates heat transfer performance based on design parameters and nondimensional data is the result of the study. According to Chekurov et al. [25], AM has the potential to outperform conventional manufacturing in heat exchanger applications even in

simpler structures. They designed a counter flow heat exchanger with 144 channels in a grid pattern. The design was manufactured using SLM technology, and the wall thicknesses of the channels were kept at a minimum for the best heat transfer performance. Since a simple failure in the channel wall manufacturing process would result in a defective part, they used double the value specified by the manufacturer as the minimum thickness of the wall. Through analytical calculations, the test results were validated and the performance of the heat exchanger designed in this study outperformed the conventionally manufactured heat exchangers.

By providing lightweight and corrosion resistant substitutes for metals such as polymers, AM offers possibilities to use novel materials for heat exchanger applications. Using thin polymer sheets that were laser welded together, Arie et al. [26] created an example of a polymer heat exchanger with a wall thickness of 150 μm . After flow tests were conducted, the polymer heat exchanger outperformed commercially available plate fin type heat exchangers at a much lower cost. Furthermore, the heat exchanger prototype was lightweight and resistant to corrosion. The Cross-Media Fiber concept where heat transfer between two working fluids is achieved through fins that have high conductivity was shown by Hymas et al. [27]. Using Fused Deposition Modeling (FDM) for assembly, their design combined low-conductivity composite tubes with high-conductivity fins. Cost effectiveness and high performance were achieved by incorporating metallic fibers into the polymer shell. Roper et al. [28] demonstrated the application of AM in the creation of periodic microlattice networks for cross-flow heat exchangers. For improved heat transfer, in addition to AM, the conformal coating method is also utilized. They manufactured periodic microlattice networks using SLA and photopolymer waveguide technology. The heat exchange networks produced are tested using hot and cold air. The approach used in this work can extend to biological heat and mass transfer applications, such as artificial organs.

Studies have also compared metal and polymer heat exchangers. Son et al. [29] investigated lattice frame structures as heat transfer surfaces in both polymer and metal forms through simulations and experiments with different flow conditions. When lattice structures manufactured by AM were compared with traditional cylindrical fins, their results showed that complex lattice structures offer superior heat transfer rates. It was concluded that lattice frame structures offer heat transfer rates that are compa-

rable to those of cylindrical fins. Because AM was employed, much higher porosity values were achieved than those of conventionally manufactured cylindrical fins.

2.2 Triply Periodic Minimal Surfaces in Engineering

Triply Periodic Minimal Surface (TPMS) structures, as discussed in Section 1.1.3, show versatility and have applications across different engineering fields. Due to their unique characteristics, TPMS structures have become more popular in a variety of fields, including tissue engineering, automobile engineering, and aerospace engineering. For example, Xiang-nan et al. [30] used SLM technology to evaluate the acoustic absorption performance of TPMS structures, especially the Gyroid and Diamond structures produced from Ti6AL4V. Their research showed that TPMS structures may satisfy a variety of sound absorption needs by varying the properties of the TPMS, such as porosity and wall thickness, using the impedance tube approach. Moreover, TPMS structures have shown useful in applications related to civil engineering, especially in areas like bridge bearings. Sengsri et al. [31] used Finite Element Analysis (FEA) to evaluate the free vibration characteristics of TPMS-based panel constructions designed for bridge bearings. Their results demonstrated the flexibility and advantages of TPMS structures in a structural environment, suggesting that they might improve civil engineering applications.

Because of their shape memory qualities, TPMS structures have also been investigated in structural engineering in addition to civil engineering. Sun et al. [32] examined TPMS structures consisting of NiTi that were produced using Powder Bed Fusion (PBF) technology. They performed direct and cycle compression testing on Gyroid, Diamond, and Primitive structures. These experiments demonstrated the suitability of NiTi-based TPMS structures for structural engineering, particularly for vibration isolation and crash softening applications. The usefulness of TPMS structures in resolving vibration-related issues in civil engineering applications has increased. Stereolithography (SLA) was used by Fu et al. [33] to study composite TPMS structures that were gyroid, primitive, and IWP. The usefulness of single freedom modal testing in attenuating vibrations has been established, indicating potential uses in civil engineering to enhance structural stability.

In addition to civil engineering, TPMS structures have also been explored in structural engineering for their shape memory properties. Sun et al. [32] investigated NiTi-made TPMS structures manufactured using PBF technology. They examined the Gyroid, Diamond, and Primitive structures, subjecting them to direct and cycling compression tests. These tests revealed the applicability of NiTi-based TPMS structures in structural engineering, especially in contexts such as crash softening and vibration isolation. TPMS structures further extend their utility in the solution of vibration related problems in engineering applications. Fu et al. [33] studied TPMS composite structures of Gyroid, Primitive, and IWP types, manufactured using SLA technology. Single freedom modal tests demonstrated their effectiveness in damping the vibrations, promising applications areas in engineering for improved structural stability.

Furthermore, TPMS structures have demonstrated potential uses in the area of strain sensors and structural health monitoring. The vibration behavior of Primitive TPMS structure used for strain sensors was studied by Sixt et al. [34]. Tests using uniaxial compression showed potential for minimal sensor delays and high frequency sensing capabilities, which can improve the accuracy of structural health monitoring. The energy absorption characteristics of the TPMS structures have also been studied. By utilizing SLM technology with AlSi10Mg material, Li et al. [35] investigated the energy absorption potential for hybrid TPMS structures that combined Gyroid and IWP surfaces. TPMS structures were shown to be suitable for energy absorption applications through compression testing. Furthermore, by using their high surface area-to-volume ratio, TPMS structures have proven their potential in hydrogen storage units. The potential of TPMS-based structures for hydrogen storage was established by Lesmana et al. [36] by applying mathematical models in conjunction with experimental data.

2.3 Previous Studies on TPMS Heat Exchangers

Various investigations have addressed the design and efficiency of TPMS-based heat exchangers, helping to better understand and improve this rapidly developing subject. Volumetric Distance Fields (VDFs) were used within the design method presented

by Kim et al. [37] to create complicated heat exchanger designs. They produced several prototypes and analyzed their flow characteristics. Moreover, Tang et al. [20] examined the convective heat transfer coefficients of the TPMS structures compared to conventional fin structures. Their findings showed that TPMS designs, particularly those containing through holes such as IWP and Gyroid structures, can improve heat transfer but may also hinder it because of the existence of low resistance flow paths. A similar study was carried out by Reynolds et al. [38] for enhancing the understanding of TPMS based heat exchangers. They developed a new correlation for the Nusselt number in micro channel heat exchangers, concluded that hydraulic diameter is an important parameter in thermal calculations for TPMS structures, and manufactured polymer heat exchangers using Liquid Crystal Display (LCD) technology.

Adding to these discoveries, Hawken et al. [39] examined pressure drops in Diamond type TPMS and observed that the pressure drops decrease with increasing hydraulic diameter and porosity. Additionally, they created a correlation for the pressure drop values in Diamond structures. When Röver et al. [40] compared a new TPMS-based heat exchanger to a traditional PHEX, they discovered that the TPMS design had better heat transfer and reduced pressure drops. After examining the geometric characteristics of the TPMS structures, Gao et al. [41] came to the conclusion that more uniform temperature distribution and improved heat transfer performance are caused by the smaller cell sizes of the TPMS structures. They designed and manufactured four different heat exchangers based on Diamond structures by using copper and SLM technology.

Wang et al. [42] conducted more research on flow characterization through TPMS structures. In this study, a correlation for Nusselt number prediction in flows through Fischer Koch S, IWP, Primitive and Neovius type TPMS structures was developed for the turbulent regime. Using CFD analysis, flows were simulated through TPMS arrays with different geometric parameters. For the characterization of laminar flow, Piedra et al. [43] carried out a research which is also based on CFD simulations. In this work, single unit cell Primitive and IWP surfaces were examined. A Darcy-Forchheimer model, which is used for flows through porous structures, was fitted to the CFD results. Additionally, using the fitted model, a correlation for pressure drops through the Primitive and IWP surfaces was developed based on the fluid velocity.



CHAPTER 3

DESIGN & CHARACTERIZATION OF THE HEAT EXCHANGER

In this work, characterization of flow within TPMS based heat exchangers is investigated. Different heat transfer surface geometries are generated by the parametric design workflow developed. Among the different generated geometries, a Gyroid type TPMS structure is selected as the reference geometry, and a prototype is manufactured. The prototype is tested under different flow conditions. Furthermore, CFD simulations for the experiment configuration and other generated geometries of heat exchangers are conducted. The results obtained from the simulations and experiments are compared with those in the literature.

The prototype is manufactured using Renishaw SLM AM-400 model 3D metal printer [44] with 17-4PH stainless steel powders. Due to material limitations, only one prototype could be manufactured. Although the metal 3D printer is capable of manufacturing larger unit cell sizes without support structures, in this work, a cubic unit cell size of 5 mm with a wall thickness of 1 mm is selected for the Gyroid type of TPMS structures inside the prototype to ensure production without any failures.

3.1 Designing and Manufacturing of Heat Exchangers with TPMS Cores

The parametric design workflow for the TPMS based heat exchangers is developed using nTopology [45]. nTopology is a software platform for computational design and engineering that enables the modeling and optimization of complex geometries for AM. It can be used for various engineering applications, such as topology optimization, field driven design, and engineering simulations such as static and modal analysis.

3.1.1 Modifying TPMS Structures

TPMS structures can be easily generated within nTopology. There are six built-in TPMS structures, Gyroid, Diamond, Primitive, Neovius, Split P, and Lidinoid. Other types of TPMS structures can also be generated by using their governing trigonometric equations, such as the equations given in Table 1.1. In Figure 3.1, built-in TPMS types are shown.

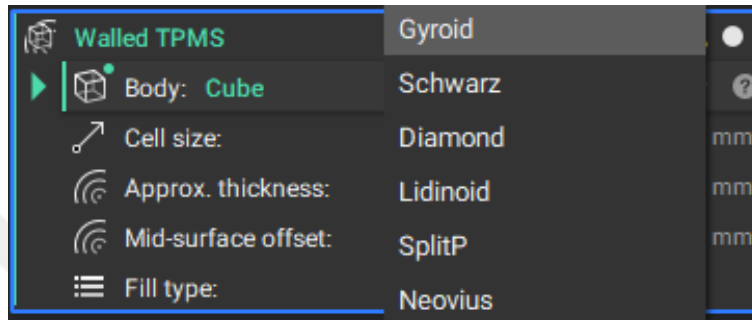


Figure 3.1: Walled TPMS Block and Built-in TPMS Types in nTopology

In order to generate TPMS structures that are not available within the nTopology, trigonometric equations for infinite surfaces are written in closed form using the Add, Subtract, Multiply, and Divide command blocks along with the Cosine and Sine blocks. An example of infinite surface generation for the Diamond type TPMS structures in nTopology is shown in Figure 3.2.

After generating the infinite surface, it is intersected with a unit cell block volume determined by the user. The dimensions of the unit cells can be adjusted by modifying the coordinate terms within the Cosine and Sine blocks as described in Section 1.1.3. The last operation to obtain a unit TPMS cell is to thicken the intersection body to the desired thickness value of the wall. Custom Unit Cell and Periodic Lattice blocks can be used to fill the design space with the developed TPMS unit cell to obtain the TPMS cores. nTopology is able to perform the above mentioned manipulation operations when creating built-in TPMS structures using the Walled TPMS block shown in 3.1. In nTopology, Boolean operations are performed with the blocks such as Boolean Intersection, Boolean Subtraction, and Boolean Union blocks.

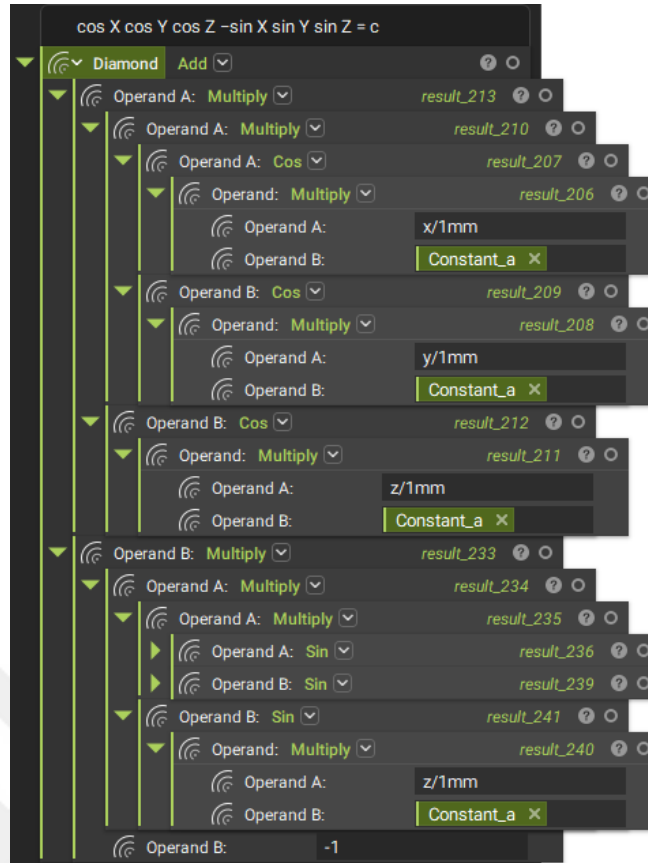


Figure 3.2: Closed Form Surface Equation for the Diamond Type of TPMS Structures in nTopology

Furthermore, the two non intersecting fluid channel volumes inside the TPMS structures can be obtained by using surface equations. In Table 1.1, the equations are equated to zero, defining the points on the surface. If, for a point, the left-hand side of the equation is smaller or greater than zero, then the point lies outside the surface and inside one of the fluid domains. When TPMS structures that are unavailable within the nTopology are used, fluid domains can be obtained by subtracting the TPMS core volume from the volume of the design space. The resulting domain contains both fluid channels and currently, the only method for separating these domains in nTopology is to first mesh and then use the Mesh Separate block. This adds computational costs to the design workflow. Again, nTopology is able to perform the above mentioned manipulation operations with simple blocks and parameters like the Walled TPMS block and the Mid Surface Offset parameter.

3.1.2 Generating Functional Interfaces

At the inlet and outlet sections of the heat exchanger, flow entering or exiting the TPMS core is prevented from mixing by creating baffles. Baffles are created by intersecting the fluid domain entering or exiting with the other fluid domain. Furthermore, to provide a larger area for flow, the pool volumes are removed from the TPMS core at the inlets and outlets. The resulting surface area for the flow provided by the cutout pool sections should be the same for the inlets and outlets, so that there is minimum blockage for the flow into or out of the TPMS structure.

In Figure 3.3, the generated baffles and cutout sections are shown with the reference geometry of Gyroid type TPMS core. In addition, the need for mechanical interfaces for hydraulic fittings, such as threads and counterbores, is satisfied by extending the geometry of the heat exchanger and creating pipe segments. The pipe segments at the inlet and outlet sections for both fluids can be seen in Figure 3.4.

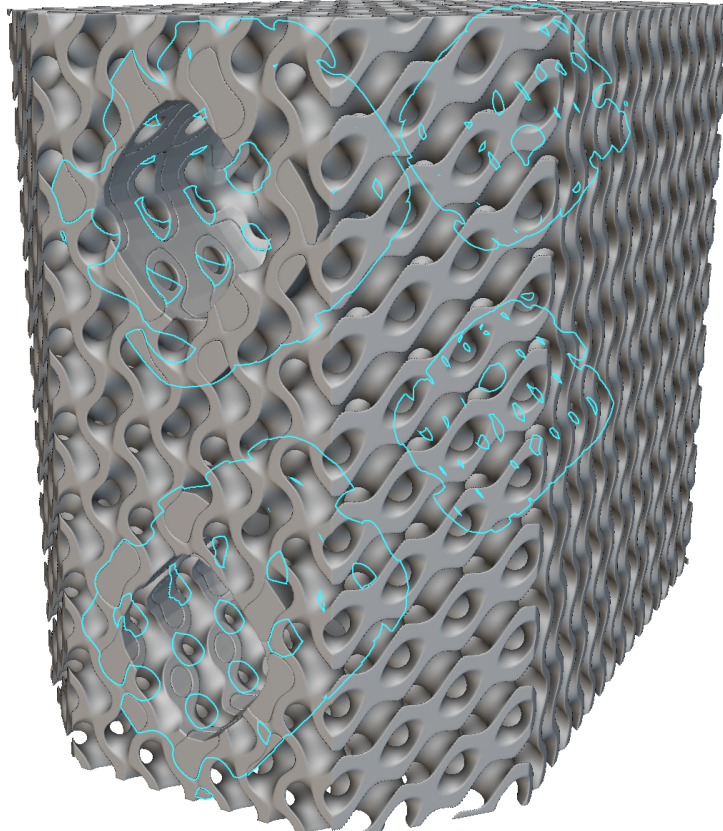


Figure 3.3: Generated Baffles and Pool Cutout Sections in a Gyroid TPMS Core

3.1.3 Design for Additive Manufacturing (DfAM) Considerations

To ensure successful manufacturing, DfAM considerations have to be taken into account during the design phase. In this work, prototype manufacturing is done using PBF technology.

In PBF, powders are bound together with heat input. Therefore, it is crucial to ensure that the wall thickness remains as constant as possible throughout the entire geometry. Since each layer of the part is created by adding heat input, the topmost layer has higher temperatures than the previous ones. Therefore, inconsistent wall thicknesses may lead to warping or separation of layers during the part production.

For metal PBF processes, the cross-sectional geometry of the first layer is fixed to the build plate. Therefore, it is necessary to have a 1 or 2 mm margin at the bottom of the designed part. After production, parts are separated from the building plate using the Wire-EDM method utilizing this margin. The margin left for cutting can disrupt the uniformly adjusted wall thicknesses during part design. Since the regions with cutting margin are thicker, cracks can start in these areas and propagate toward the upper layers. Therefore, the cutting margin should also be taken into account during part design.

Since the geometry is created by melting metal powder particles, the resulting part will have a porous structure. Therefore, the density and mechanical properties of the part will be different from those of the raw material used. Information about the estimated properties of the output parts can be found in the literature based on specific PBF processes, 3D printing machines, and materials used.

During part design, the bridging distance for the manufacturing process should also be considered. The unit cell size of the manufactured prototype was selected considering the bridge distance as 5x5x5 mm with 1 mm wall thickness.

Because specific regions in each powder layer are melted during production, there may be trapped powder particles inside the part depending on the design. The trapped powder can be removed using ultrasonic vibration.

After part production, it is critical to be cautious during processes such as threading,

assembly, and other operations to avoid clamping the part. The application of force perpendicular to the layers using clamps, wrenches, or other tools can lead to the separation of layers. Therefore, it is necessary to inform operators about the sensitivity of the part to ensure that they handle the part appropriately.

After production, machining operations that involve chip removal will introduce heat into the part. As mentioned above, unequal heat input to the part can lead to cracking between layers. Therefore, machining operations should be carried out as slowly as possible while using cooling fluids. Applying machining operations to the part before the powder removal process can lead to clogging of the heat exchanger prototype because the cooling fluids used can lump the fine particles inside the part. Care should also be taken to ensure that chips generated during machining operations do not become trapped inside the functional interfaces of the part.

Since removal of support structures is impossible inside the heat exchanger, the designed part should not have overhang angles greater than 45 degrees. For example, circular holes that are perpendicular to the manufacturing direction will require support structures. Therefore, the holes or pool volumes designed in the heat exchanger are in the shape of skewed rectangles with angles less than 45 degrees. In Figures 3.4 and 3.5, section view of the prototype is shown. The skewed rectangle inlet holes for hydraulic fittings, build plate margin and skewed rectangle pool sections can be observed. Due to unequal wall thicknesses at the inlet and outlet extensions of the prototype, cracks are expected. However, being far from the TPMS core structures, these cracks would not induce functional defects. Unequal wall thicknesses are generated at these specific regions for not to use support structures.

3.1.4 Exporting the Designed Geometry for Engineering Analysis and Manufacturing

The designed heat exchanger geometry can be exported using nTopology. Using surface meshing and mesh control blocks, the generated geometry for the prototype can be meshed according to the designer's needs. Then, using the export blocks present in nTopology, the STL file can be created for the prototype manufacturing.

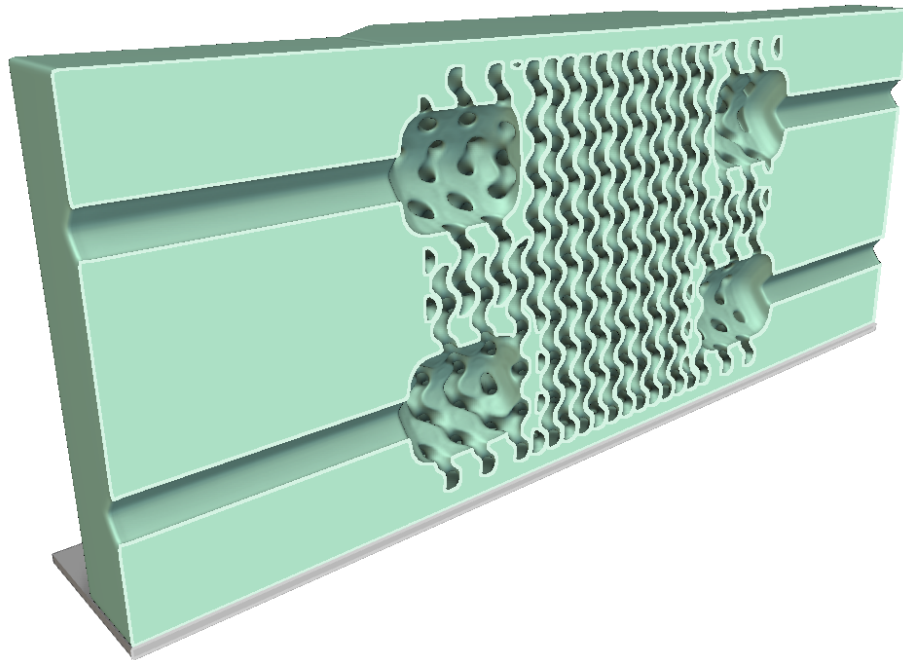


Figure 3.4: Section View of the Generated Gyroid based HEX Prototype

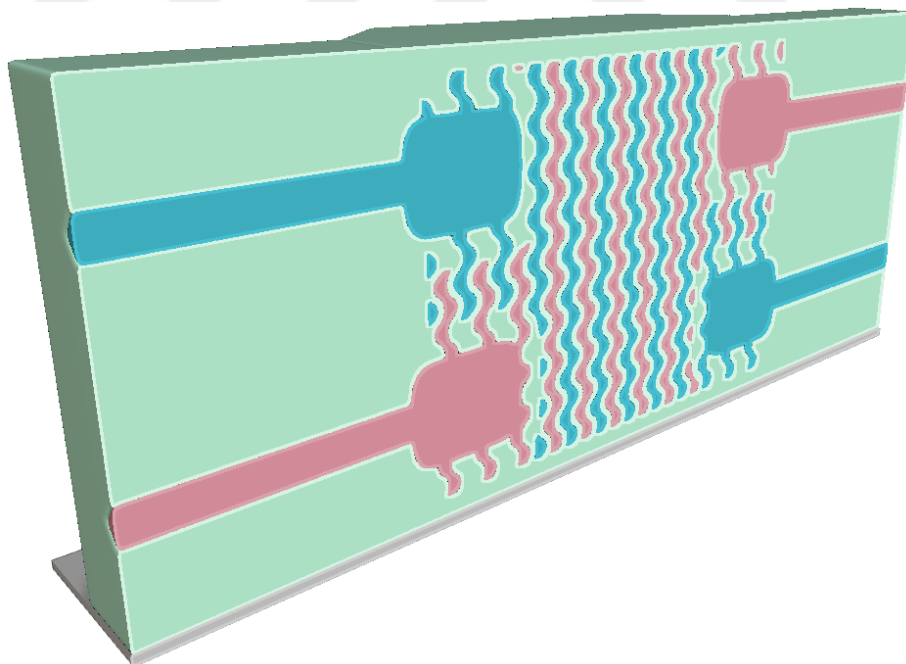


Figure 3.5: Section View of the Generated Gyroid based HEX Prototype with Cold (blue) and Hot (red) Fluid Channels

Using the STL file, the machine file is generated using Materialise Magics [46], a slicer software for AM. In the slicer software, regions with overhang angles greater than 45 degrees and bridging distance greater than 5 mm are investigated. As mentioned in Section 3.1.3, the characteristic length of the TPMS structure is selected considering the bridge distance requirement specified in the slicer software. The section view of the prototype in Materialise Magics software is shown in Figure 3.6.

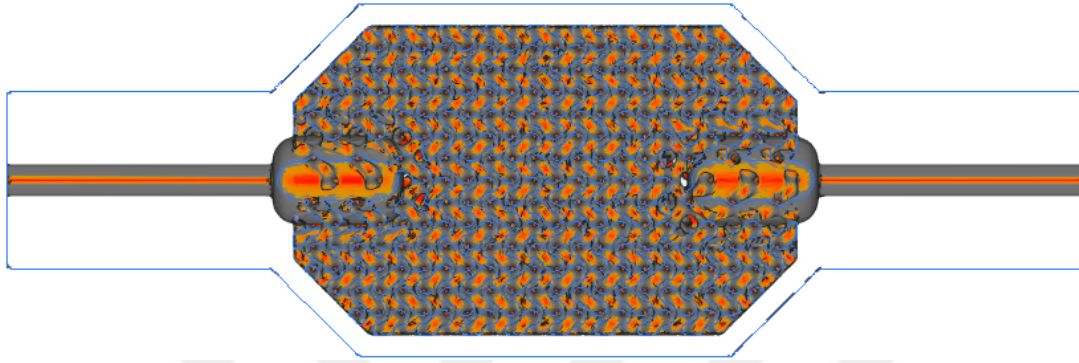


Figure 3.6: Section View Taken from the Materialise Magics Software

3.1.5 Quality Assurance in the Manufacturing Process

Before the heat exchanger prototype is manufactured, three sample parts are produced to examine any defects or failures. Sample 1 has the same Gyroid core structure as the actual prototype along with the inlet and outlet sections with pool volume removed in the shape of skewed rectangles. Sample 2 is composed of three cubic unit Diamond cells with a characteristic length of 5 mm, while Sample 3 is a larger Gyroid structure with a cubic unit cell size of 10 mm. No production defects are observed in these samples. In Figure 3.7, the samples and the manufactured prototype are shown.

To finalize the manufacturing process, the threads are machined in the inlet and outlet pipe sections as shown in Figure 3.8.

After the samples has been qualified, the actual Gyroid based heat exchanger prototype is manufactured. When the part is removed from the built plate with Wire-EDM, the inlet and outlet features are sealed to prevent the entry of chips and cooling fluids into the heat exchanger. Then, using an ultrasonic vibration table, the powder removal

is completed. Before the flow tests are applied on the prototype, it is ensured using X-ray imaging technology that there are no trapped powder particles within the heat exchanger as shown in Figure 3.9.

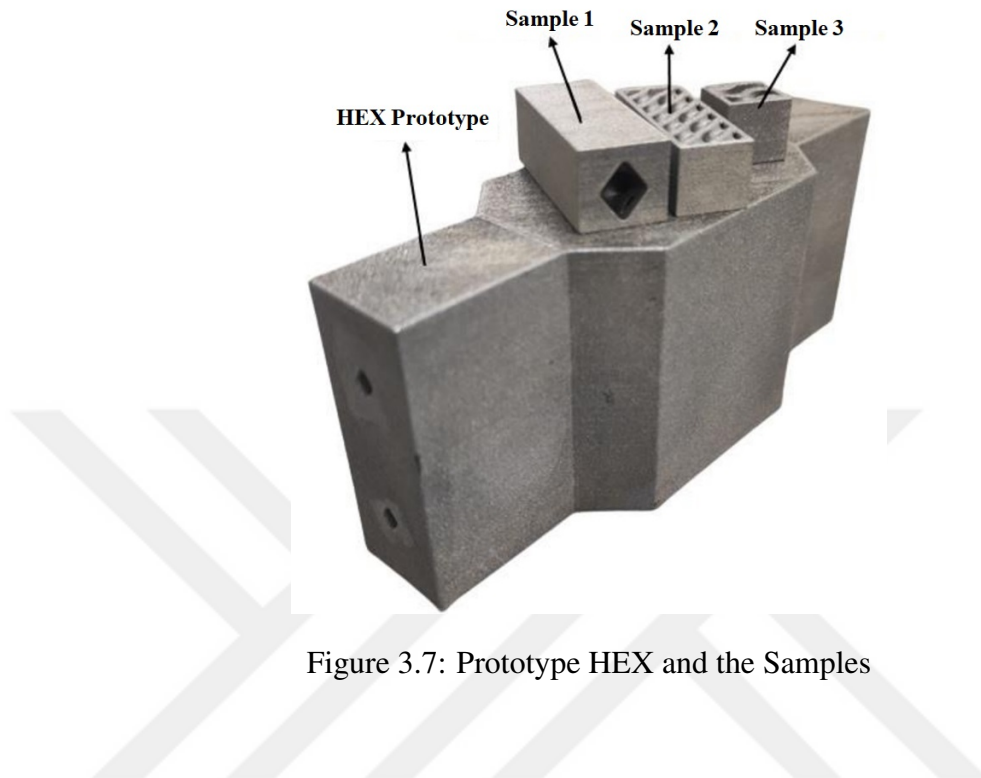


Figure 3.7: Prototype HEX and the Samples

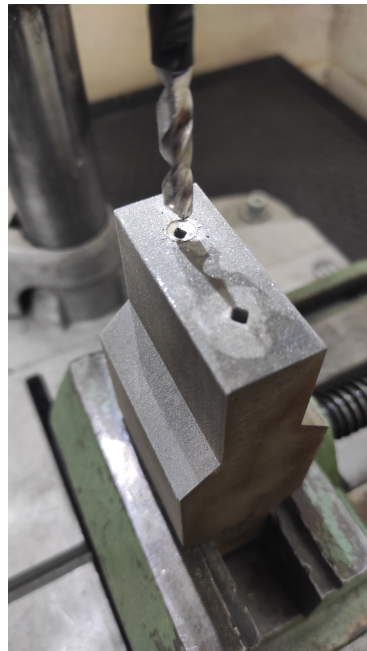


Figure 3.8: Machining Threads for the Hydraulic Fittings

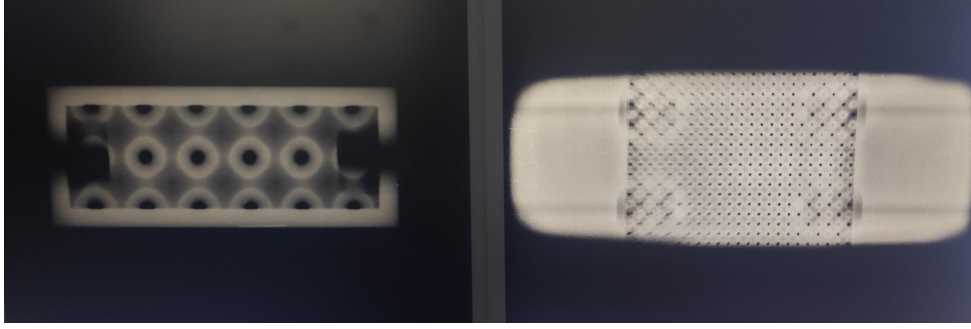


Figure 3.9: X-ray Images of Sample 1 and the HEX Prototype

The dimensions shown in Figure 3.10 for the manufactured prototype heat exchanger are measured and the results are shown in Table 3.1.

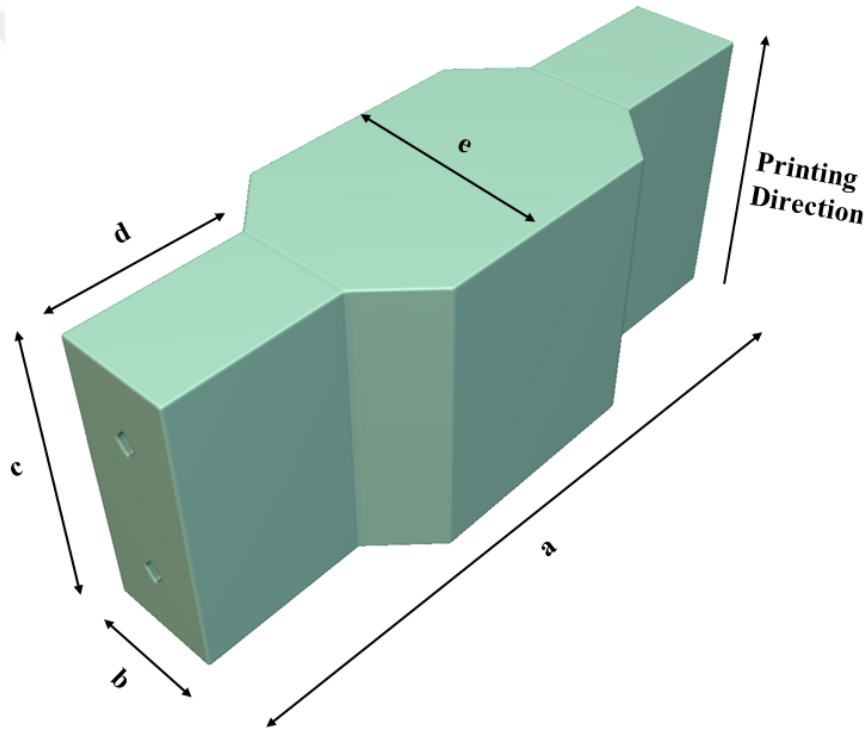


Figure 3.10: Dimensions of the Prototype

3.2 Construction of the Experimental Setup and Data Acquisition

To observe the flow characteristics through the manufactured Gyroid type TPMS based heat exchanger prototype, an experimental setup is constructed. The experi-

Table 3.1: Comparison of the Dimensions

Dimensions	Designed Value [mm]	Measured Value [mm]
a	122	121.90 ± 0.02
b	20	20.15 ± 0.02
c	50	50.55 ± 0.02
d	30	30.10 ± 0.02
e	40	40.60 ± 0.02

mental setup is capable of measuring and saving the data for mass flow rates, pressures, and temperatures in the inlet and outlet sections for the hot and cold sides.

3.2.1 Design and Assembly of the Experimental Apparatus

The scope of this study is limited to laminar flow inside the TPMS based heat exchangers, therefore, to ensure that the flow entering the heat exchanger prototype is laminar and hydrodynamically fully developed, external steel pipes are connected to the mechanical interfaces at the pipe segments of the heat exchanger prototype. Steel pipe lengths are chosen to meet the critical length for hydrodynamically fully developed internal flow. The hydrodynamic entry length for the pipes is calculated using Equation 3.1 and is found to be approximately 20 centimeters for all flow test conditions.

$$\left(\frac{x_{fd,h}}{D}\right)_{\text{lam}} \approx 0.0575 Re_D \quad (3.1)$$

$$Re = \frac{\rho v D}{\mu} \quad (3.2)$$

Re_D is the Reynolds number based on the diameter of the steel pipes and is calculated from Equation 3.2 for the flow inside the steel pipes. The flow rates used in the experiments are chosen such that the resulting Reynolds number is less than 2300 so that laminar flow is achieved within the steel pipes.

Moreover, using Equation 3.3 the thermal entry lengths for the flows entering the heat exchanger were calculated. The Prandtl number calculated using Equation 3.4 for the hydraulic oil used is around 190 which is very high. For instance, the Prandtl number for water under the same conditions is around 7. Due to the high Prandtl number, the thermal entrance length required for a thermally fully developed flow is around 30 meters for all test conditions. Therefore, the flow entering the heat exchanger is not thermally fully developed in the experiments.

$$\left(\frac{x_{fd,t}}{D}\right)_{\text{lam}} \approx 0.05 Re_D Pr \quad (3.3)$$

$$Pr = \frac{\mu C_p}{k} \quad (3.4)$$

Being hydrodynamically fully developed but not thermally fully developed, the flow entering the heat exchanger has a stable velocity profile but not a stable temperature profile. However, this does not affect the validity of the experiments, since the average inlet temperatures for the flows are measured, and the flow is laminar. [47].

For Reynolds number and hydraulic diameter calculations within the porous TPMS core, different equations are used in several works [21, 48, 49]. The Reynolds number is calculated using Equation 3.5. v_p , the pore velocity is the assumed average velocity inside the porous structure and is related to the Darcian velocity, v_d , at the inlet pipe segments of the heat exchanger through Equation 3.7. v_d is calculated from the measured mass flow rate data. The hydraulic diameter which is the characteristic length of the TPMS structure is calculated from Equation 3.8 where ϕ is the porosity of the TPMS structure and a_{sf} is the specific solid to fluid interface area. The calculated Reynolds numbers for the experimental conditions are presented in Table 3.2.

$$Re_p = \frac{\rho v_p d_h}{\mu} \quad (3.5)$$

$$\phi = \frac{V_{\text{fluid}}}{V_{\text{total}}} \quad (3.6)$$

$$v_d = \phi v_p \quad (3.7)$$

$$d_h = \frac{4\phi}{a_{sf}} \quad (3.8)$$

a_{sf} is calculated by dividing the solid to fluid interface area of the whole Gyroid core by the volume of the heat exchanger core, while porosity, ϕ , is calculated by dividing the total fluid volume inside the Gyroid core by the total volume of the heat exchanger core. In this work, all of the above mentioned calculations are performed by using nTopology since it is easier to mesh and measure the area and volume properties of the generated TPMS structures.

The fluid used in the experiments is a standard MIL-H-5606 hydraulic oil. Properties of the hydraulic oil used in the experiments are obtained from the work of Conte et al. [50]. Properties of the MIL-H-5606 hydraulic oil change with working temperature. Therefore, the values obtained are interpolated or extrapolated for different temperature values when needed. Flow properties such as ρ and μ are discussed in detail in Chapter 4.

Moreover, Dybbs et al. [51] conducted a detailed investigation into fluid flow through porous structures and showed that the transition from laminar to turbulent flow occurs at Reynolds number values which are different from those observed internal flow through pipes.

- $Re < 1$: Darcy regime
- $1 \leq Re < 150$: Moderate Forchheimer regime
- $150 \leq Re < 300$: Strong Forchheimer regime
- $Re \geq 300$: Turbulent regime

Using Equation 3.2 and Equation 3.5, the experimental flow rates are determined so that the Reynolds number through the pipes is less than 2300 while the Reynolds number for the TPMS core is less than 150. This ensures that the experiments are

within the scope of the study, which is laminar flow through heat exchangers with TPMS cores. Reynolds number can go up to 300 while staying in the laminar region for flow through porous structures, however, the flow becomes unstable for Reynolds numbers higher than 150 according to Dybbs et al. [51].

After the experimental flow rate limits and steel pipe lengths are determined, the hydraulic circuit is designed for the test setup. The first requirement of the experimental setup is to supply hot and cold side flows to the heat exchanger with constant flow rates and pressures. This requirement is met by an existing flow control device which is capable of supplying MIL-H-5606 hydraulic oil up to 24 L/min with a pressure of 200 bars. However, since the existing flow control device has only fluid supply tank, it can only supply hot or cold oil. Hence there were no in line heaters for the experiments, the oil tank inside the flow control device is heated before the experiments by circulating the oil inside the tank through another circuit through a relief valve with high pressure. By using Hydac OK-ELC5S/1.1/230V/1/S Oil Air Cooler, oil passing through the cold side of the heat exchanger is cooled before entering the heat exchanger. Thus, hot and cold oil sides are achieved.

Then, the test setup should be able to measure and save data for flow rates, pressures, and temperatures at the inlet and outlet sections for both fluids. Therefore, at least two flow rate sensors, four pressure and temperature sensors are needed. Furthermore, the outer surface temperature of the prototype heat exchanger will also be measured, hence a thermocouple is needed. Sensors used in the test setup are Hydac ETS 7246- B-010-000 for temperature measurements, Trafag NAT 8252 for pressure measurements and VSE Flow VS 0.1 GPO 12V 32N11/4 for flow rate measurements. Also, a K-type thermocouple is adhered to the bottom surface of the prototype. The measurement range of temperature sensors are $(-25, +100)$ °C and their accuracy is $\pm 2\%$. Pressure sensors can measure pressures between 0 to 40 bars with an accuracy of $\pm 0.5\%$ and the range of flow rate sensors is 0.01 to 10 L/min with an accuracy of $\pm 0.3\%$. Using the accuracy of all the sensors in the experimental setup, the uncertainty analysis of the experimental values is performed in Section 3.2.2.

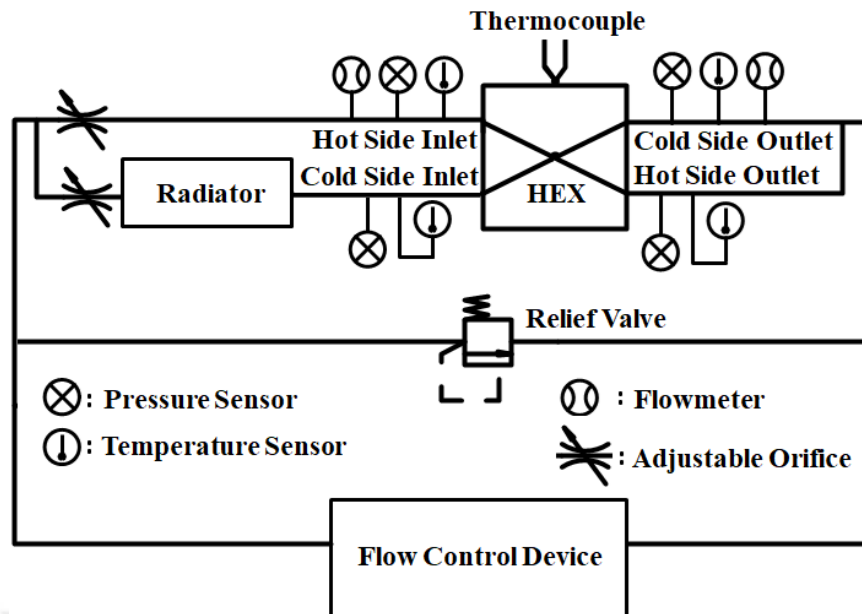


Figure 3.11: Schematic of the Test Setup

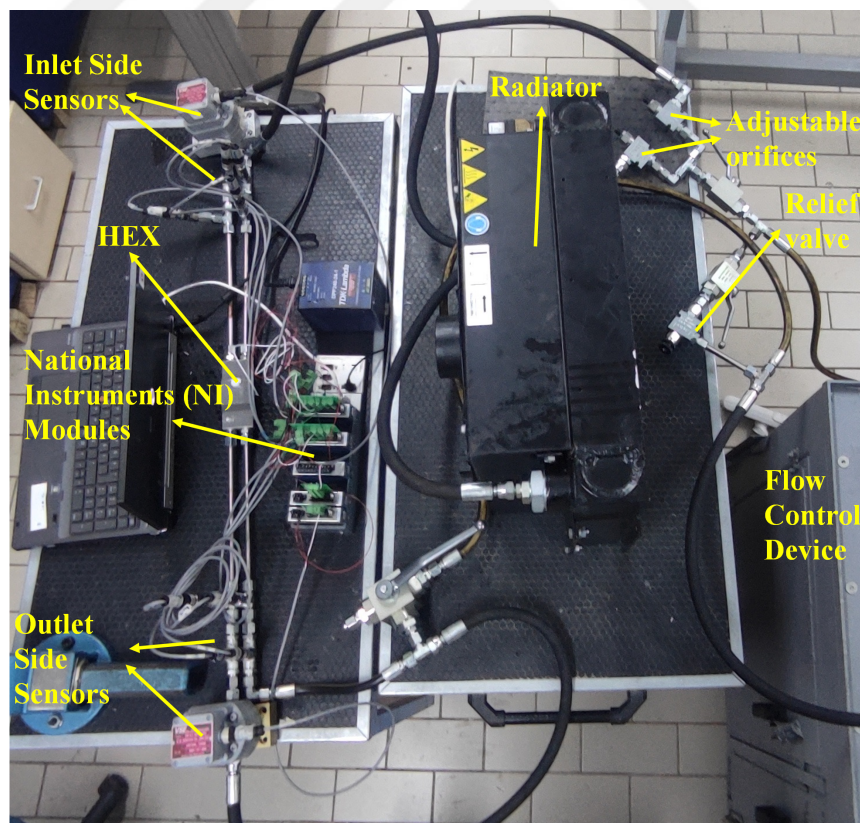


Figure 3.12: Picture of the Experimental Test Setup

With the above mentioned requirements met, the test setup is assembled using standard hydraulic fittings and fasteners. Lastly, a relief valve is added to the experimental setup to provide safety against high pressures. In Figure 3.11, a schematic of the test setup is shown. Furthermore, in Figure 3.12, the assembled test setup is shown in detail.

3.2.2 Methods for Data Collection and Uncertainty Analysis

All sensors used in the test setup are connected to different Natural Instruments (NI) modules that are able to read signals from each sensor. The data read by the NI modules are collected and stored using LabVIEW with a frequency of 10 Hz and exported in .csv format.

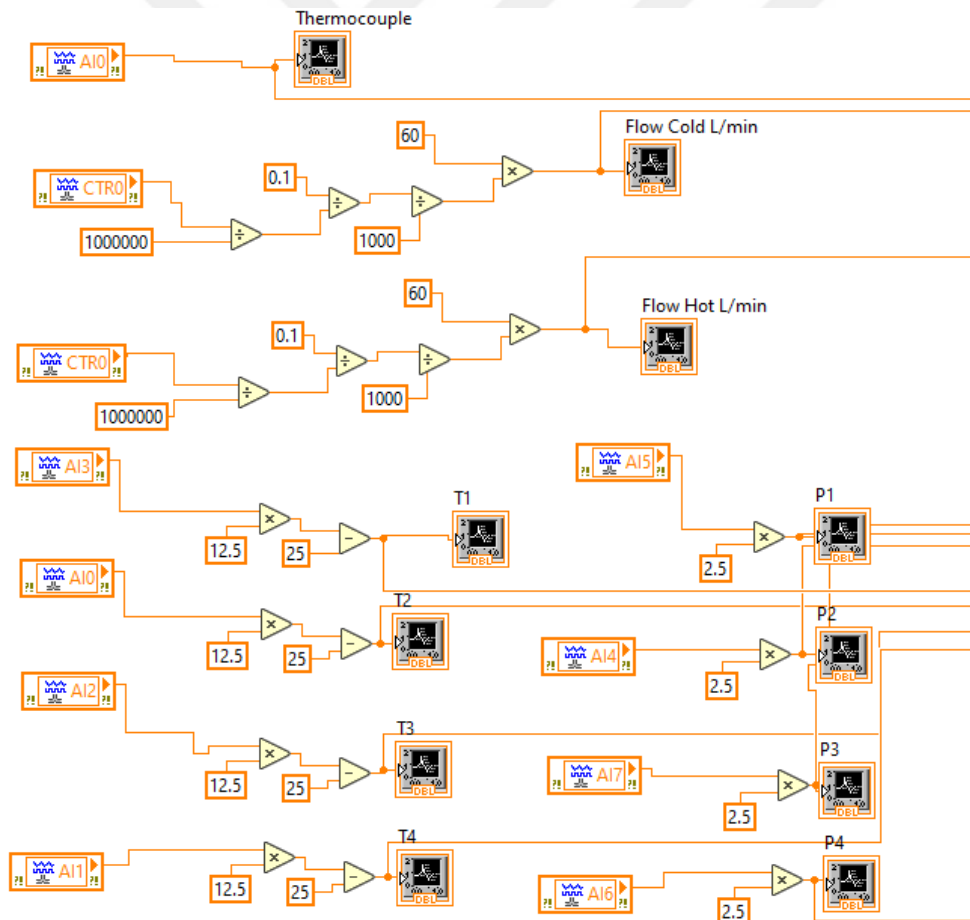


Figure 3.13: Schematic of the LabVIEW Sensor Readings

For temperature and pressure sensor readings, the NI-9205 unit is used, while for flow rate sensor readings, the NI-9411 unit is used. The thermocouple adhered to the prototype body is read by the NI-9211 unit. All NI modules are mounted on the NI-cRio-9068 board, and the board is connected to the computer via ethernet connection. Unit conversions for sensor readings are performed using LabVIEW and in Figure 3.13, a schematic of sensor readings and unit conversions is shown.

Using data collected through sensors, the heat transfer rates and pressure drops in Table 4.2 were calculated using Equation 3.9 and Equation 3.10, respectively.

$$q = \dot{m} c_p (T_{out} - T_{in}) \quad (3.9)$$

$$\Delta p = p_{in} - p_{out} \quad (3.10)$$

The uncertainty of individual variables measured in Equations 3.9 and 3.10 propagates to the results of calculated heat transfer rates and pressure drops. In general form, these experimental results can be described as functions of multiple variables as in Equation 3.11. The equation of the heat transfer rate has 3 variables, which are $\dot{m}$, T_{out} and T_{in} , therefore, $n = 3$ for the results of the heat transfer rate, while the equation of the pressure drop has P_{in} and P_{out} as variables, and therefore $n = 2$. The error for specific heat was neglected in the heat transfer rate uncertainty calculations since it is a material constant.

$$\phi = \phi(x_1, x_2, x_3, x_4, \dots, x_n) \quad (3.11)$$

Then, the percent uncertainty U of the calculated experimental result can be determined by Equation 3.12. Using the sensor accuracy information in Section 3.2.1, Equation 3.13 yields $\pm 2.84\%$ uncertainty for experimental heat transfer rate results and Equation 3.14 gives $\pm 0.71\%$ uncertainty for experimental pressure drop results [52].

$$\frac{U_\phi}{\phi} = \left[\left(\frac{\partial \phi}{\partial x_1} \frac{U_{x_1}}{\phi} \right)^2 + \left(\frac{\partial \phi}{\partial x_2} \frac{U_{x_2}}{\phi} \right)^2 + \cdots + \left(\frac{\partial \phi}{\partial x_n} \frac{U_{x_n}}{\phi} \right)^2 \right]^{1/2} \quad (3.12)$$

$$\left(\frac{U_q}{q} \right)^2 = \left(\frac{U_{\dot{m}}}{\dot{m}} \right)^2 + \left(\frac{U_{T_{\text{in}}}}{T_{\text{in}} - T_{\text{out}}} \right)^2 + \left(\frac{U_{T_{\text{out}}}}{T_{\text{in}} - T_{\text{out}}} \right)^2 \quad (3.13)$$

$$\left(\frac{U_{\Delta p}}{\Delta p} \right)^2 = \left(\frac{U_{p_{\text{in}}}}{p_{\text{in}} - p_{\text{out}}} \right)^2 + \left(\frac{U_{p_{\text{out}}}}{p_{\text{in}} - p_{\text{out}}} \right)^2 \quad (3.14)$$

3.3 Flow Experiments

The prototype of a heat exchanger manufactured with a Gyroid core is tested under five different flow conditions. Flow rates are selected so that the assumed average velocity within the Gyroid core, which is the pore velocity v_p , results in a Reynolds number smaller than 150 for the hot and cold sides. In Table 3.2, the volumetric flow rates and Reynolds numbers for the hot and cold sides of the experiments are provided. For Test #1 and Test #2, the Reynolds number on the hot side exceeds the unsteady laminar transition value of 150 as highlighted in Table 3.2. However, as discussed in Section 3.2.1, the laminar flow regime continues until a Reynolds number of 300. Therefore, Test #1 and Test #2 are considered as valid tests. The results of all experiments are discussed in detail in Chapter 4, together with the pressure, temperature, and flow rate data. The lowest flow rate that the flow control device can supply was 0.5 L/min. The flow rate is gradually increased in the other flow tests up to the Reynolds number of laminar - turbulent transition of. Then, to observe the flow behavior in the vicinity of the critical Reynolds number, the flow rate is increased to 3.25 L/min. Hence, 5 different flow tests were conducted.

Since the flow control device has only a single tank that contains hydraulic oil, the need for hot and cold sides is met by circulating the hydraulic oil inside the flow control device before feeding it through the heat exchanger as mentioned in Section 3.2.1. The heat exchanger circuit, the top part shown in Figure 3.11, was separated from the rest of the circuit by valves. When these operator-controlled valves were closed, hydraulic oil was fed through the relief valve in Figures 3.11 and 3.12 and

returned to the flow control device tank until the oil tank temperature increased to 55 °C. The heated hydraulic oil was fed through the heat exchanger circuit and halved for the hot and cold sides. One side of this separated flow goes into the heat exchanger prototype while the other first goes into a radiator to be cooled, and then it is fed into the heat exchanger prototype.

After both flows exit the heat exchanger, they were combined and fed to the flow control device oil tank, which cools the hydraulic oil inside the tank. However, since the oil tank is very large compared to the flow rates used in the experiments, constant inlet temperature assumptions were made for the hot and cold sides in the CFD analysis. The experimental data was collected after the flow through the heat exchanger reached steady state. The steady state assumption was also made by observing the temperature, pressure, and flow rate data while the experiments were being conducted. When data for all variables remained within a tolerance margin of $\pm 3\%$ for 60 seconds, the steady state assumption was made and experiments were continued for another 60 seconds. Average values for the last 120 seconds of each variable were used in the calculations and CFD analysis.

Furthermore, in Section 3.5, the geometric properties of the manufactured Gyroid based heat exchanger are discussed.

Table 3.2: Flow Rates and Reynolds Numbers Used in the Experiments

Flow Tests	$\dot{Q}_{\text{cold}}$ [L/min]	$\dot{Q}_{\text{hot}}$ [L/min]	Re_{cold}	Re_{hot}
Test #1	3.19	3.25	136.1	167.1
Test #2	2.93	3.02	124.2	152.1
Test #3	1.91	2.06	81.1	104
Test #4	0.93	1.08	39.3	53.4
Test #5	0.46	0.56	19.5	26.8

3.4 Model Construction for CFD Analysis

The experimental data obtained from the flow tests with the manufactured prototype were validated using CFD analysis. After the CFD model was validated, different

heat exchanger designs with different types of TPMS and cell sizes were generated using nTopology. In Chapter 4, the validation of the model is discussed in detail. Sections 3.4.1 and 3.5 explain the setup of the CFD model and various generated heat exchanger designs used in the simulations, respectively. In the simulations, the heat exchanger geometries are fully modeled. In this section, the geometry preparation and boundary conditions applied in the computational model will be described. Processing of the results and selection of the mesh size will be explained at the beginning of Chapter 4.

3.4.1 Geometry Preparation for CFD Analysis

The computational model setup was constructed and the CFD analysis was performed using *Simcenter STAR-CCM+* [53, 54], which will be referred to as Star CCM+ hereafter. In total 18 simulations were carried out. Five simulations were carried out for the selection of the mesh size with the flow properties of Test #2 in Table 3.2. Furthermore, four additional simulations were performed to simulate all flow tests in Table 3.2. Lastly, nine simulations were conducted with the generated heat exchanger designs. The geometric properties of the alternative heat exchanger geometries are explained in detail in Section 3.5.

For each CFD simulation, the complete heat exchanger geometry is generated and exported from nTopology. In nTopology, using the block "Export FE mesh to CFD", all boundary condition surfaces can be labeled and exported with the .stl extension. However, to use this method, two fluid domains and a solid domain must be meshed and exported, increasing the computational cost. The process of labeling surfaces with boundary conditions involves selecting nearby triangles in nTopology. If the mesh is defective, the surface labeling cannot be done correctly. In Figure 3.14 surface labeling is shown.

Therefore, by exporting only the surface mesh of the complete HEX body, the boundary condition surfaces are labeled in Star CCM+. By splitting the imported surface mesh from nTopology by angle, the regions of cold fluid, hot fluid, and solid body are created as shown in Figure 3.15.

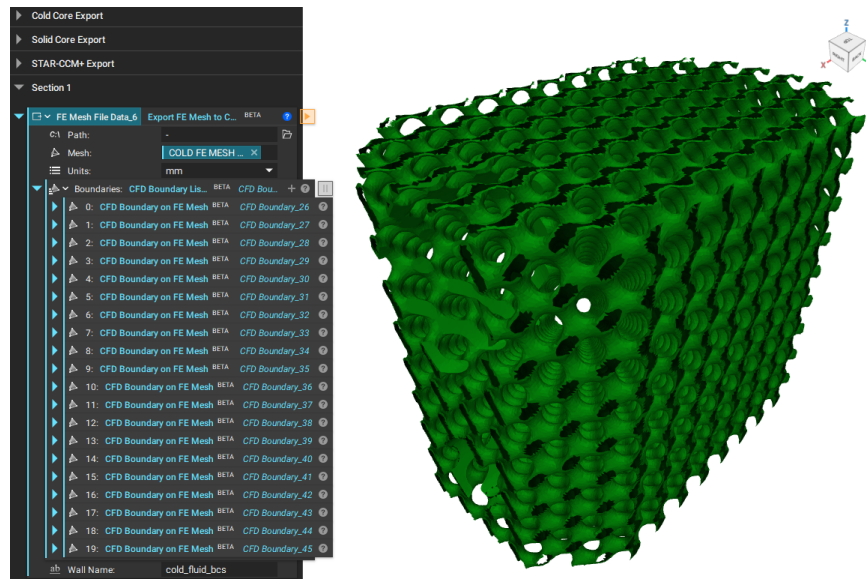


Figure 3.14: Boundary Condition Surface Labeling in nTopology

Simcenter STAR-CCM+

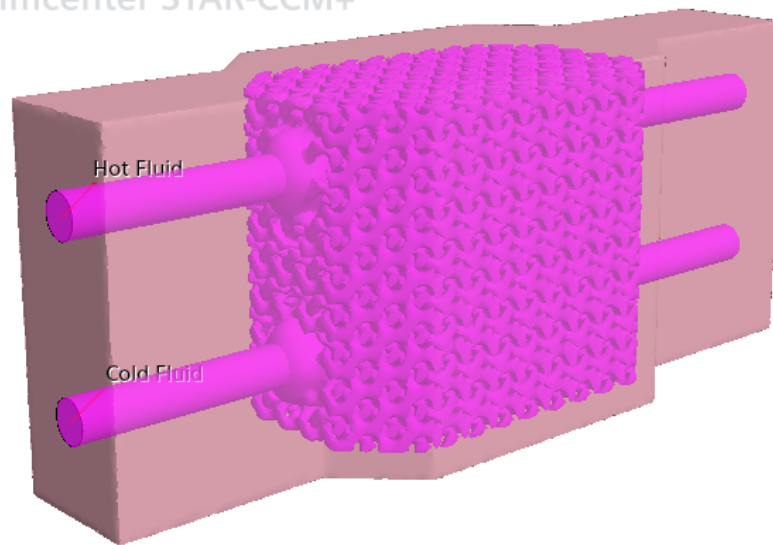


Figure 3.15: Splitting the Imported Surface Mesh into Regions

3.4.2 Geometry Check and Mesh Repair Before CFD Analysis

The exported surface meshes from nTopology are further inspected in Star CCM+. Using the surface repair tool, all pierced and duplicate faces, along with low qual-

ity and high skewness faces, can be detected as shown in Figure 3.16. Furthermore, the surface repair kit also allows for detection of duplicate vertices and edges in the STL files, however, none were found in the STL files exported from nTopology. All detected problematic faces in the exported HEX body surface meshes were repaired before any simulations were performed. The number of problematic surfaces detected was very low compared to the actual number of surfaces in the STL files for each simulation case, and the repair operations were automatically performed in Star CCM+. The final mesh of the manufactured heat exchanger prototype is shown in Figure 3.17.

Before any meshing operations are performed, it is ensured that the imported surface mesh geometries are watertight. Then, using polyhedral elements, volume meshes are created for the fluid and solid sides. As described in the best practices of Star CCM+ [53], this type of mesh element is suitable for heat transfer analysis. The boundary layers are not modeled since the flow inside the complex TPMS geometry is complex, although the flow regime is laminar [21].

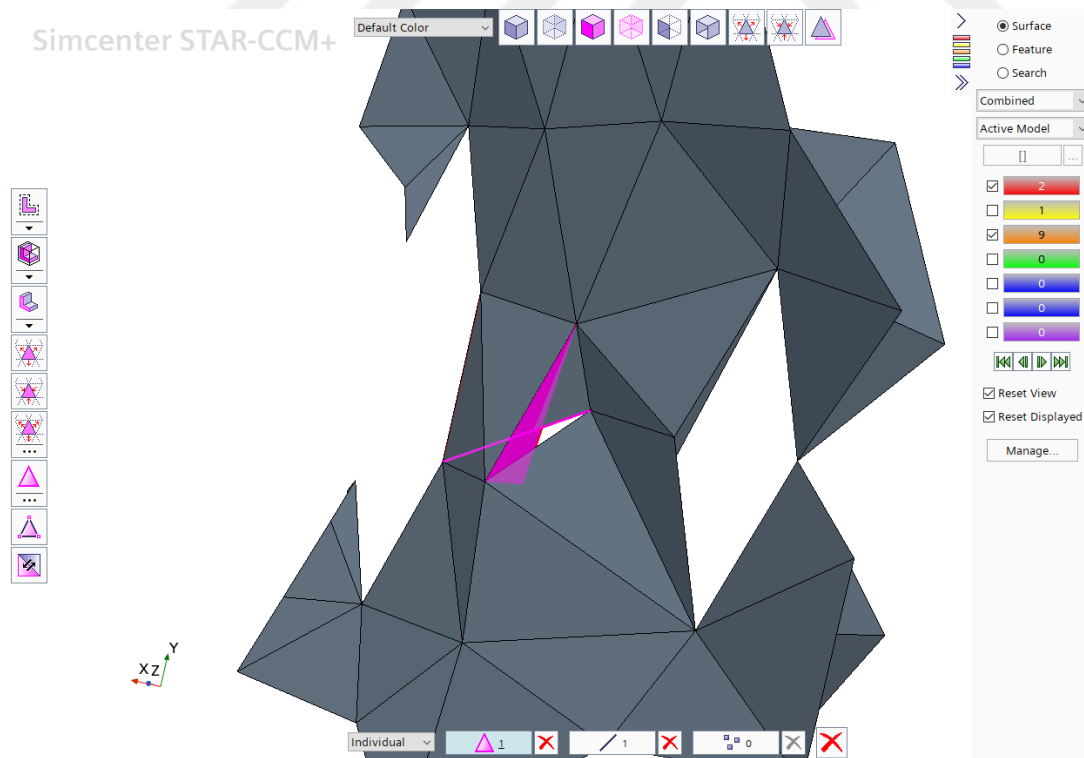


Figure 3.16: Pierced Faces Observed in Surface Repair Tool of Star CCM+

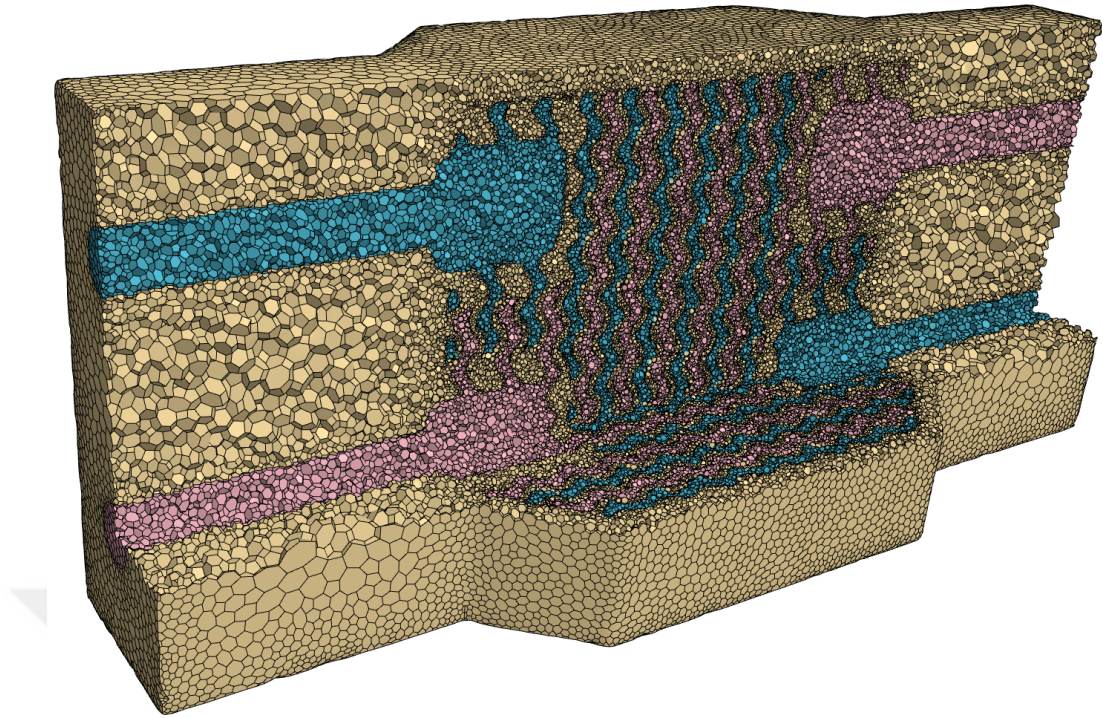


Figure 3.17: CFD Meshing on the Manufactured Prototype

3.4.3 Boundary Conditions and Assumptions for CFD Model

In the simulations, the choice and implementation of suitable boundary conditions were carried out with particular attention. These conditions specify the fluid's behavior inside the heat exchanger core, including the interactions at the inlet and outlet sections. The parameters defined include fluid mass flow rate [kg/s], temperature [°C], and pressure [bar] to ensure that the simulations represent the experimental conditions. Three physics continuums were created, which are cold side, hot side, and solid body.

Furthermore, the physics models used within Star CCM+ and the assumptions made for the solid and fluid volumes are as follows:

- Three Dimensional
- Steady State
- Incompressible Flow

- Single Phase
- Laminar Flow
- Segregated Flow
- Segregated Fluid Temperature
- Constant Density & Properties
- Coupled Solid Energy
- No Slip Wall Condition
- Pressure Outlet
- No Axial Conduction
- Free Convection Boundary Condition at the Outer Surface

In the simulations, thermal radiation, gravity, and thermal energy generation due to fluid viscosity effects were neglected. Although the assumption of constant fluid properties was made in the simulations, the hydraulic oil used in the experiments has properties that depend on the temperature. Therefore, the properties of the hot and cold side hydraulic oils are taken differently but are constant. Since the inlet and outlet temperatures for hot and cold sides are known, the average temperatures for hot and cold sides are calculated. The fluid properties used in the simulations and experimental calculations are taken at these average temperatures.

The model of segregated fluid temperature addresses the total energy equation to solve for temperature as the primary variable. This modeling approach is recommended by Star CCM+ for simulations of internal flow without combustion. The segregated flow model iteratively solves the conservation of momentum and mass and is used in nonlinear constant density flows, whereas the coupled solid energy model uses a coupled approach for the solution of the energy conservation equation [54].

Since the core structures in this study have complex geometries, a three-dimensional model was selected to provide a more robust simulation. The segregated flow solver was chosen for the fluid and solid domains. Since the equations are solved uncoupled, the segregated solver requires fewer resources than the coupled flow model [54].

Alternative heat exchanger design simulations were carried out at the same Reynolds number, temperature, and pressures as the Test # 2 in Table 3.2. Therefore, the fluid properties in the simulations for alternative heat exchanger designs were the same.

Mass flow inlet and pressure outlet boundary conditions were applied to the circular surfaces of the inlet and outlet pipe segments of the fluid domains. The choice has been made to use the mass flow inlet as the inlet boundary condition instead of the velocity because it offers improved convergence and a more realistic approach [54], which aligns with the actual experiments where flow measurements were made using mass flowmeters. The pressure outlet boundary condition was used because it imposes the working pressure of the fluid. It also allows for backflow, if it ever occurs, Star CCM+ is capable of determining if there is a backflow through an outlet boundary condition and informs the user [54]. Throughout the simulations, no warnings were encountered as such.

For surfaces where hot and cold fluids interact with the solid body of the heat exchanger, mapped contact interfaces were created in the solid to fluid interaction areas to ensure that the energy equations in these regions were coupled. Also, no slip boundary condition was applied in the solid to fluid interaction areas. Furthermore, heat transfer occurring at these interfaces could be monitored to validate the experimental results.

Furthermore, axial conduction along the fluid flow was neglected as the Peclet numbers (Pe) for flows in the experimental configurations were higher than 50 [21].

Finally, a boundary condition of free convection was applied on the outer surface of the HEX body. The ambient temperature is assumed to be 25°C. The heat transfer coefficient for free convection was calculated using empirical vertical plane correlations [47] since the normals of 12 out of 14 faces of the HEX body are perpendicular to the gravity direction during the experiments. Using Equations 3.15 and 3.16 the free convection heat transfer coefficient, $\bar{h}$ is calculated as 1.05 W/m²K. Free convection boundary condition also requires thermal resistance value. It is calculated using Equation 3.17 as 0.015 m²K/W [47]. The air properties at 25°C that are used to calculate Rayleigh number are taken from Table A4 of Incropera et al. [47]. The inlet temperatures are set as initial conditions for the fluid regions whereas the initial

condition of the HEX solid region is set to ambient temperature (25°C).

$$\overline{Nu}_L = \frac{\bar{h}L}{k} = 0.68 + \frac{0.670 Ra_L^{1/4}}{[1 + (0.492/Pr)^{9/16}]^{4/9}} \quad \text{for } Ra_L \leq 10^9 \quad (3.15)$$

$$Ra_L = \frac{g\beta(T_s - T_\infty)L^3}{\nu\alpha} \quad (3.16)$$

$$R_{t,\text{conv}} = \frac{T_s - T_\infty}{q} = \frac{1}{\bar{h}A} \quad (3.17)$$

3.5 Geometric Properties of Generated TPMS Based Heat Exchangers

In this section, the geometric properties of the alternative heat exchangers generated such as porosity, ϕ , hydraulic diameter, d_h , and the specific interface area from solid to fluid, a_{sf} will be given and compared. The formulas used in the calculation of these geometric parameters are based on different works in the literature, as mentioned in Section 3.1.

The surface area of the TPMS structures, A_{sf} , can be calculated with Equation 3.18 as described by Peng et al. [55], however, the equation is for a surface with zero thickness. In this work, since the TPMS cores inside the heat exchanger are thick and contain cutout volumes and baffle walls, the specific solid to fluid interaction area, a_{sf} , was approximated by Equation 3.19 [21]. The solid to fluid interaction area term, A_{sf} in Equation 3.19 was calculated from the mesh surface area of the TPMS core structures using nTopology.

$$A_{sf} = \iint \sqrt{1 + \left(\frac{\partial z}{\partial x}\right)^2 + \left(\frac{\partial z}{\partial y}\right)^2} dx dy \quad (3.18)$$

$$a_{sf} = \frac{\text{Total Surface Area of the Core}(A_{sf})}{\text{Total Core Volume}(V_{\text{total}})}. \quad (3.19)$$

Generated designs differ from the manufactured prototype by their TPMS core type and cell sizes. Using Gyroid, Primitive and Diamond structures, 9 different heat

exchanger geometries other than the manufactured prototype were generated in total. Generated designs have unit cell sizes of 10x10x10 mm, 15x15x15 mm, and 20x15x15 mm for each type of TPMS. The core structures of all alternative geometries have a thickness of 1 mm. The geometric parameters for all generated TPMS core structures are compared in Section 3.5.

The prototype was designed with a Gyroid core that has a unit cell size of 5x5x5 mm and a wall thickness of 1 mm, as mentioned in Section 3.1.3. In Figure 3.18, the section view of the prototype and the core structure is shown. The geometric properties of the prototype are given in Table 3.3. The geometric properties of the alternative designs will be compared with these parameters.

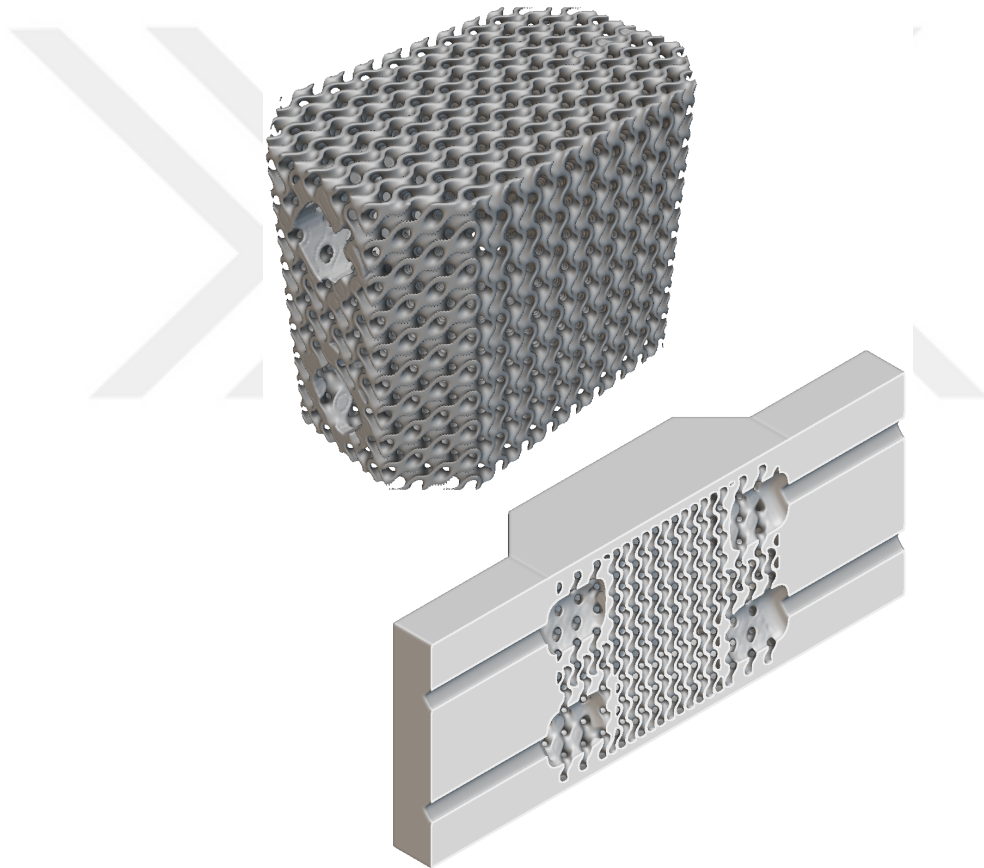


Figure 3.18: Gyroid Core, Unit Cell Size 5x5x5 mm with 1 mm of Wall Thickness (Manufactured Prototype)

In this section, the geometric properties of alternative TPMS core structures are compared. The geometric properties for all alternative HEX designs are presented in Table

Table 3.3: TPMS Geometric Properties for Gyroid Core with Unit Cell Size of 5x5x5 mm with a Wall Thickness of 1 mm

Property	Symbol	Formula	Value
Design space volume [mm ³]	V_{total}	-	83215
Wall volume [mm ³]	V_{wall}	-	33215
Fluids volume [mm ³]	V_{fluid}	-	50000
Total solid - fluid interaction area [mm ²]	A_{sf}	-	91094
Specific solid - fluid interaction area [mm ⁻¹]	a_{sf}	$a_{\text{sf}} = A_{\text{sf}}/V_{\text{total}}$	1.0947
Porosity	ϕ	$\phi = V_{\text{fluid}}/V_{\text{total}}$	0.6009
Hydraulic diameter [mm]	d_h	$d_h = 4\phi/a_{\text{sf}}$	2.1955

3.4.

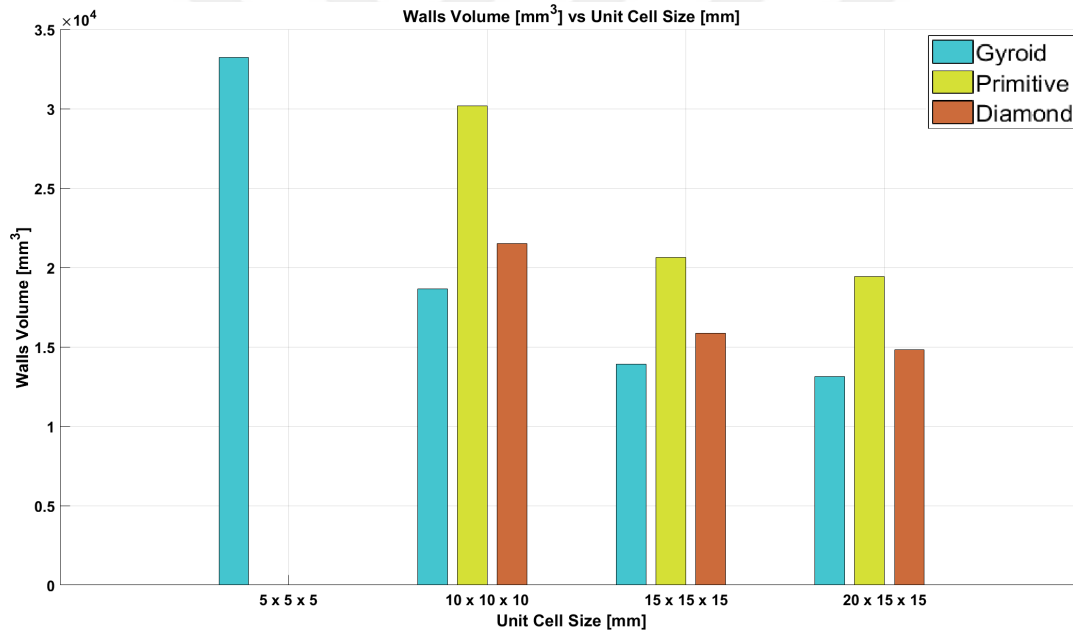


Figure 3.19: Volume of Solid Walls with respect to Unit Cell Size

When the volumes of solid walls of alternative designs are compared, it can be observed from Figure 3.19 that for a fixed unit cell size, Primitive type TPMS structures have the highest, Gyroid type TPMS structures have the lowest volume of walls within a specific design space. This can also be verified from Figure 3.20, showing the volume of fluid channels inside the core structures. As expected, Primitive type

structures have the least fluid channel volume while Gyroid type structures have the highest fluid channel volume.

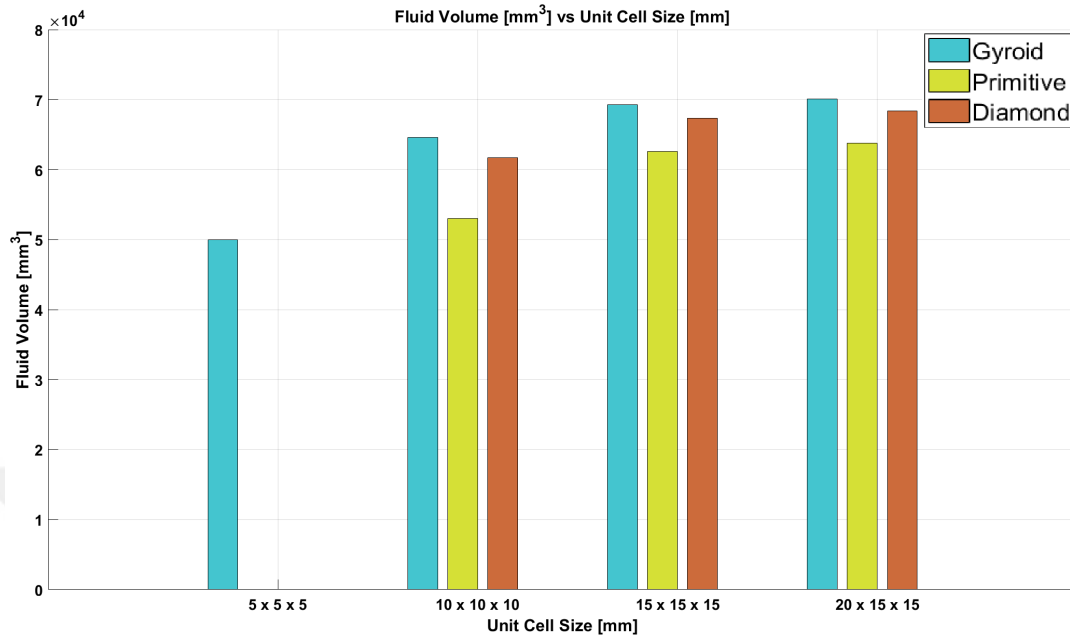


Figure 3.20: Volume of Fluid Channels with respect to Unit Cell Size

When the specific solid to fluid interaction area investigated, which is the ratio of total surface area to total volume, unexpectedly, Diamond type TPMS structures show the best performance followed by Gyroid type structures. Figure 3.21 shows that although Gyroid type structures have more fluid channel volume, Diamond type structures have more heat transfer area. The ratio of fluid volume to total volume, porosity ϕ , show exactly the same behaviour as the fluid channel volume since the total core volume is constant for all alternative TPMS structures. Lastly, the hydraulic diameters calculated by using Equation 3.8 for each alternative TPMS cores is investigated from Figure 3.22. For all unit cell sizes, Primitive structures have the highest hydraulic diameter and Diamond structures have the least. When Figure 1.2 is examined, it can be clearly seen that Primitive structures have the smoothest and continuous channels for fluid flow, while Diamond structures have the least smooth channels. Based on the hydraulic diameters, it is expected that the Diamond based HEX designs will have the most pressure drop while having excellent heat transfer rates due to having the largest heat transfer area.

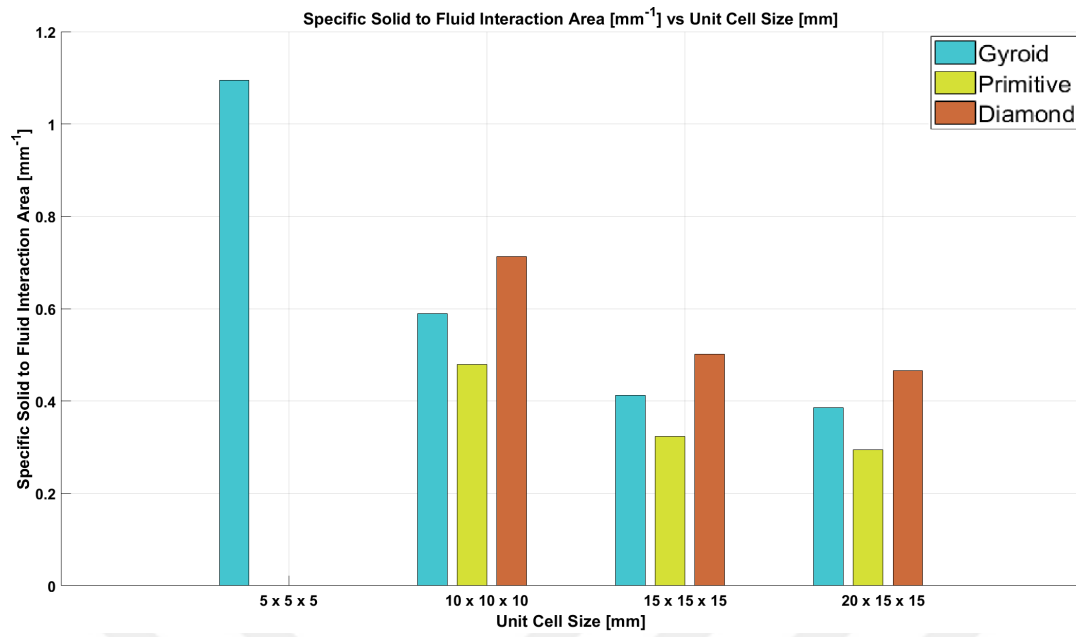


Figure 3.21: Specific Solid to Fluid Interaction Area with respect to Unit Cell Size

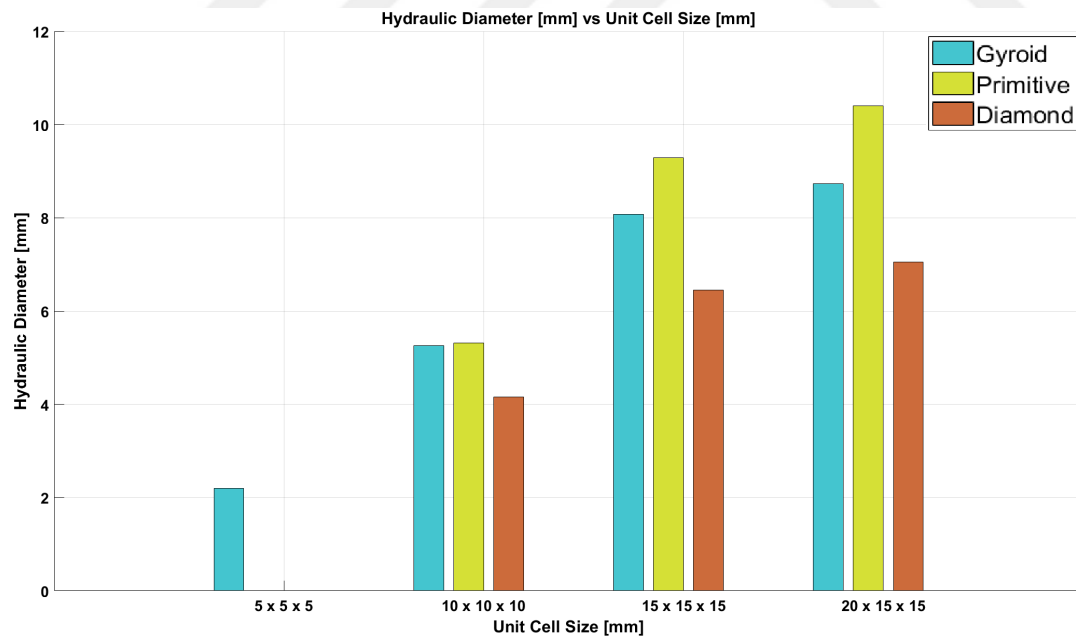


Figure 3.22: Hydraulic Diameter, d_h , with respect to Unit Cell Size

Table 3.4: Geometric Properties of the Alternative TPMS Core Structures

Property	Unit Cell Size - Wall Thickness [mm]	TPMS Type		
		Gyroid	Primitive	Diamond
Walls Volume [mm ³]	5x5x5 - 1	33215	-	-
	10x10x10 - 1	18637	30185	21494
	15x15x15 - 1	13926	20649	15845
	20x15x15 - 1	13128	19427	14817
Fluids Volume [mm ³]	5x5x5 - 1	50000	-	-
	10x10x10 - 1	64578	53030	61721
	15x15x15 - 1	69289	62566	67370
	20x15x15 - 1	70087	63788	68398
Specific Solid to Fluid Interaction Area [mm⁻¹]	5x5x5 - 1	1.0947	-	-
	10x10x10 - 1	0.5898	0.4792	0.7133
	15x15x15 - 1	0.4124	0.3236	0.5016
	20x15x15 - 1	0.3859	0.2947	0.4661
Porosity	5x5x5 - 1	0.6009	-	-
	10x10x10 - 1	0.7760	0.6373	0.7417
	15x15x15 - 1	0.8327	0.7519	0.8096
	20x15x15 - 1	0.8422	0.7665	0.8219
Hydraulic Diameter [mm]	5x5x5 - 1	2.1955	-	-
	10x10x10 - 1	5.2627	5.3198	4.1594
	15x15x15 - 1	8.0760	9.2949	6.4558
	20x15x15 - 1	8.7297	10.403	7.0541

3.6 Theoretical Formulations for Heat Exchanger Evaluation

In this section, theoretical formulations for pressure drop approximations, heat exchanger performance evaluation, and volume requirement calculation for single pass shell and tube heat exchangers are developed. The results obtained using the formulations in this section are presented in Chapter 4.

3.6.1 Pressure Drop Approximations Using Analytical and Empirical Equations

For the pressure drop values, the experimental results are compared with the calculated pressure drop values. Using Equation 3.21, the friction factor correlation developed by Iyer et al. [56] for laminar flow through Gyroid type TPMS structures, a friction factor approximation can be made. Furthermore, using the Darcy - Weisbach friction factor formula (Equation 3.20 [57]), approximate pressure drop values are calculated analytically. Figure 4.1 shows the comparison between the experimental and approximated pressure drop values.

$$f = \frac{\Delta P}{\frac{1}{2}\rho V_p^2 \frac{L}{d_h}} \quad (3.20)$$

$$f = 15.5Re^{-0.58} \quad (3.21)$$

3.6.2 Analytical Heat Exchanger Performance Evaluations

In this section, performance evaluation for the HEX designs generated is carried out using the simulation data together with the $\epsilon - NTU$ method [1].

The first step of the $\epsilon - NTU$ method is to calculate the heat capacity rates of cold and hot fluids using Equations 3.22 and 3.23. Then, by using the minimum of either heat capacity rates, the maximum theoretical heat transfer rate is calculated using Equation 3.24.

$$\dot{m}_{cold} c_{p,cold} = C_{cold} \quad (3.22)$$

$$\dot{m}_{hot} c_{p,hot} = C_{hot} \quad (3.23)$$

$$q_{max} = C_{min} \Delta T_{max} = C_{min} (T_{hot,in} - T_{cold,in}) \quad (3.24)$$

After finding the maximum theoretical heat transfer, the effectiveness ϵ , can be calculated by dividing the actual, in this case simulated, heat transfer value by the maximum theoretical heat transfer, as shown in Equation 3.25.

$$\epsilon = \frac{q_{sim}}{q_{max}} \quad (3.25)$$

The heat transfer within the heat exchanger is given by Equation 3.26. Using the heat transfer data from the simulations, the overall heat transfer coefficient U can be calculated for hot and cold sides separately. With the Equation 3.27, the logarithmic mean temperature difference ($LMTD$), ΔT_{lm} , is calculated. The area term, A is replaced by the solid to fluid interaction areas for each TPMS core.

$$q_{sim} = U A \Delta T_{lm} \quad (3.26)$$

$$\Delta T_{lm} = \frac{(T_{hot,in} - T_{cold,out}) - (T_{hot,out} - T_{cold,in})}{\ln \left(\frac{T_{hot,in} - T_{cold,out}}{T_{hot,out} - T_{cold,in}} \right)} \quad (3.27)$$

The number of transfer units (NTU) for the generated HEX designs is calculated by Equation 3.28. Using the $\epsilon - NTU$ relationships for counter flow and cross flow heat exchangers, Equation 3.29 and Equation 3.30 respectively, the effectiveness values ϵ are calculated again for each HEX design. C^* is the heat capacity rate ratio of the hot and cold fluids within the heat exchanger. It is defined as the ratio of the minimum heat capacity rate to the maximum heat capacity rate.

Equations 3.25 and 3.29 yield very close results for the heat exchangers considered in this study. However, a correction factor between 0.1 – 0.5 needs to be applied to the effectiveness values calculated by Equation 3.30 to match the results of Equation 3.25. Therefore, the flow arrangement in the TPMS structure can be classified as counter flow.

$$NTU = \frac{UA}{C_{min}} \quad (3.28)$$

$$\epsilon = \frac{1 - \exp(-NTU(1 - C^*))}{1 - C^* \exp(-NTU(1 - C^*))} \quad (3.29)$$

$$\epsilon = 1 - \exp \left[- \left(\frac{1}{C^*} (NTU)^{0.22} \exp [-C^* (NTU)^{0.78}] \right) \right] \quad (3.30)$$

Furthermore, Nusselt numbers for the flows through HEX design alternatives were also calculated. Nusselt number is the ratio of convective heat transfer to the conductive heat transfer within the fluid [1] and is calculated using Equation 3.31. The characteristic length used in Equation 3.31 is the hydraulic diameter of each TPMS core structure and k is the thermal conductivity of cold or hot fluid.

$$Nu_{d_h} = \frac{U d_h}{k} \quad (3.31)$$

Another important parameter for heat exchangers is the pumping power P . It is calculated using equation 3.32 where η_p is the pump efficiency, Δp is the pressure drop and $\dot{V}$ is the volumetric flow rate [1]. Assuming an 80% pump efficiency and a flow rate of 3 L / min, the pumping power for the generated HEX designs are calculated.

$$P = \frac{\dot{V} \Delta p}{\eta_p} \quad \text{for fully developed laminar flow} \quad (3.32)$$

3.6.3 Volume Comparison of Shell & Tube Heat Exchangers with Generated TPMS Heat Exchangers

For the comparison of a single pass shell and tube heat exchanger with the generated TPMS based heat exchangers, circular pipes that have the same hydraulic diameters as the TPMS cores are considered. The number of pipes required to achieve the same heat transfer area on the tube side as the TPMS cores is calculated using Equation 3.33. The lengths of the pipes were assumed to be the same as the length of the TPMS core, which is 60 mm.

Assuming that the thickness of the pipe is equal to the thickness of the TPMS core wall, the total volumes are calculated for the calculated number of pipes using Equation 3.34. The calculated shell and tube volume includes only the volume covered by the pipes, therefore, the volume calculated for a shell and tube heat exchanger can be larger than the volume calculated. The results of the volume requirement comparison are given in Section 4.6.

$$\frac{A_{sf}}{\pi d_h L_{core}} = N_{pipe} \quad (3.33)$$

$$\frac{\pi(d_h + 2t_{wall})^2}{4} L_{core} N_{pipe} = V_{total,pipe} \quad (3.34)$$



CHAPTER 4

RESULTS AND DISCUSSION

In this chapter, heat exchanger performance is investigated through a combination of experimental and CFD simulated data. When experimental conditions are simulated with CFD analysis using different mesh sizes, a mesh size resulting in a manageable computational time is selected. Following that, a comparison is made between the CFD simulations and the actual experimental data to assess their reliability. Different heat exchanger designs are examined, and their CFD simulation results are compared. In addition, heat exchanger performance is characterized using analytical methods. Finally, the chapter is concluded with a comparison of the volume requirements for traditional shell and tube heat exchangers with the generated TPMS based heat exchangers.

4.1 Flow Test Results

In this section, the results of the flow tests are given along with the fluid properties used throughout the study. As described in Section 3.2.1, the properties of hydraulic oil are obtained by interpolating or extrapolating the data in the work of Conte et al. [50]. They supplied the properties of MIL-H-5606 hydraulic oil for 38 °C, 93 °C, 149 °C and 204 °C at atmospheric pressure. Furthermore, properties are obtained not only for atmospheric pressure, but also for 207 bar and 552 bar. Since the pressure values for the experiments are below 5 bar, it is assumed that the properties are equal to the atmospheric pressure properties. Additionally, the hydraulic oil temperature range used in the experiments is 27 °C to 45 °C; therefore, for temperatures lower than 38 °C the properties are extrapolated.

In Table 4.1, the properties used in the CFD analysis and experimental calculations are given with the corresponding temperatures. Using these properties, CFD models are constructed as described in Section 3.4.1 for each HEX generated with different TPMS cores. The CFD model is verified by simulating the experimental conditions and mesh size selection is performed only for the flow conditions of Test #2 in Table 3.2.

Table 4.1: Properties of MIL-H-5606 Hydraulic Oil

Tempera- ture [°C]	Density [kg/m³]	Dynamic Viscosity [Pa s]	Specific Heat [J/kg/°C]	Thermal Conduc- tivity [W/m/°C]	Prandtl Value
27.05	851.2	0.01358	1894	0.1358	189.4
27.23	851.0	0.01355	1895	0.1358	189.1
27.30	851.0	0.01354	1895	0.1358	189.0
27.40	850.9	0.01353	1896	0.1358	188.8
27.88	850.5	0.01346	1898	0.1357	188.1
38.00	843.0	0.01197	1944	0.1350	172.4
38.70	842.5	0.01187	1947	0.1350	171.3
41.50	840.4	0.01146	1960	0.1350	166.7
42.98	839.3	0.01125	1967	0.1350	164.3
43.20	839.1	0.01122	1968	0.1350	164.0
44.71	838.0	0.01100	1975	0.1350	161.5
93.00	802.0	0.004331	2195	0.1310	72.60

The results of the experiments are given in Table 4.2. It can be seen that exactly equal flow rates could not be achieved for the hot and cold sides, since the flow rate adjustments are made by hand using a relief valve. In addition, the inlet temperatures and pressures are also different for each experiment. The flow control device adjusted the inlet pressures to achieve constant flow rates. Moreover, the inlet temperatures are also adjusted by the user because the experiments are started only after the oil inside the flow control device tank is heated to the desired temperature. Since the aim of the experiments is to measure the outlet temperatures and pressures, these minor

differences are not considered problematic. Heat transfer rates and pressure drops are calculated using the flow rate, inlet - outlet temperature difference, inlet - outlet pressure difference, and the specific heat property for the fluid side. Furthermore, the error margins obtained in Section 3.2.2 are also given in Table 4.2.

Although the temperature differences at the inlet and outlet sections are relatively close for each experiment, as the flow rate decreases, the heat transfer rate decreases, as expected. Heat transfer rates are calculated using Equation 3.9. Furthermore, the pressure drop values also decreased as expected. This decrease can also be verified by the friction factor formulas in Section 3.6.1. The calculated pressure drops using Equations 3.20 and 3.21 are compared with the experimental ones in Figure 4.1.

The pressure drop results are mostly in agreement with the experimental data, however, as shown in Figure 4.1, the experimental data and the calculated pressure drop data display a unique behavior. For all other tests, the hot and cold side pressure drop values for both calculated and experimental are close to each other. However, in Test #1, cold side data is close to the calculated data, whereas the hot side experimental data is not. If the sensor reading is assumed to be correct, this discrepancy can be attributed to the fact that the Reynolds number on the hot side in Test #1 is 167.1, over the transition value of 150.

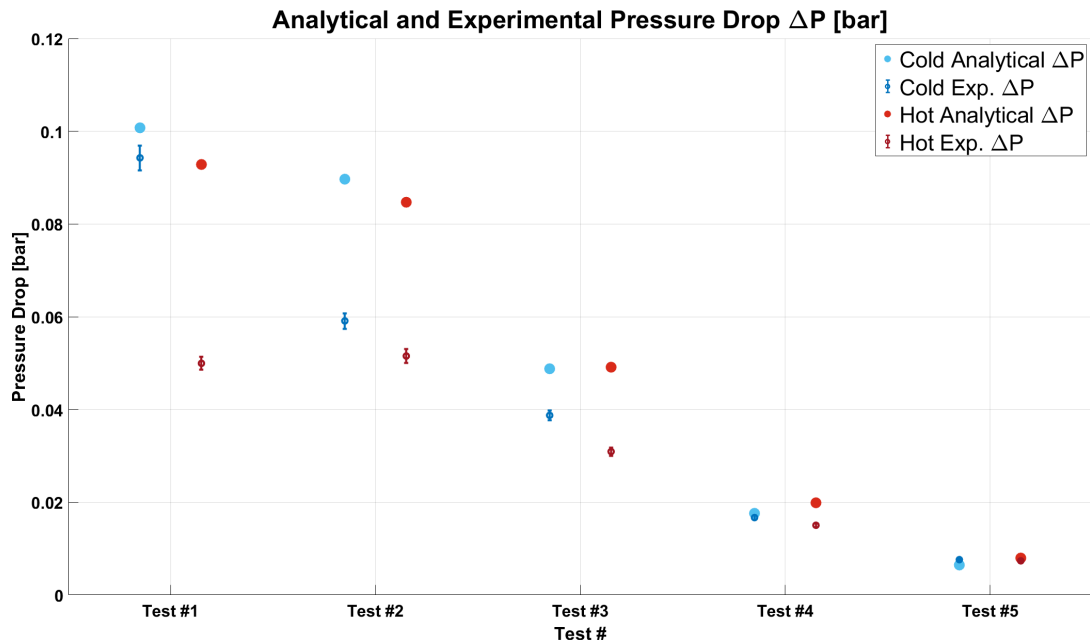


Figure 4.1: Analytical and Experimental Pressure Drop Comparison

Table 4.2: Data Obtained from the Flow Tests with the Manufactured Gyroid Based Heat Exchanger Prototype

Gyroid 5x5x5 mm with 1 mm Wall Thickness (Manufactured Prototype)							
Tests	Side	Flow Rate [L/min]	Inlet Temp. [°C]	Outlet Temp. [°C]	HEX Outer Surface Temp. [°C]	Pressure Drop [bar]	Heat Transfer Rate [W]
#1	Cold	3.19 ± 0.01	25.89 ± 0.52	29.82 ± 0.60	35.73 ± 0.71	0.0943 ± 0.0007	337.5 ± 9.6
	Hot	3.25 ± 0.01	47.02 ± 0.94	42.31 ± 0.85		0.0499 ± 0.0004	-421.3 ± 12
#2	Cold	2.93 ± 0.01	25.35 ± 0.51	29.16 ± 0.58	34.66 ± 0.69	0.0591 ± 0.0004	299.7 ± 8.5
	Hot	3.02 ± 0.01	45.26 ± 0.91	40.69 ± 0.81		0.0515 ± 0.0004	-380.1 ± 10.8
#3	Cold	1.91 ± 0.01	24.99 ± 0.50	29.85 ± 0.60	36.30 ± 0.73	0.0387 ± 0.0003	249.5 ± 7.1
	Hot	2.06 ± 0.01	45.98 ± 0.92	40.34 ± 0.81		0.0309 ± 0.0002	-320 ± 9.1
#4	Cold	0.92 ± 0.01	24.04 ± 0.48	30.05 ± 0.60	34.92 ± 0.70	0.0167 ± 0.0001	149.4 ± 4.2
	Hot	1.08 ± 0.01	45.21 ± 0.90	39.59 ± 0.79		0.0150 ± 0.0001	-166.7 ± 4.7
#5	Cold	0.46 ± 0.01	23.92 ± 0.48	30.90 ± 0.60	35.18 ± 0.70	0.0076 ± 0.0001	86.1 ± 2.4
	Hot	0.56 ± 0.01	42.62 ± 0.85	35.25 ± 0.71		0.0074 ± 0.0001	-113.8 ± 3.2

4.2 Mesh Size Selection for CFD Analysis

To observe the effects of base mesh size on the results of CFD simulations, Test #2 is chosen as a reference since the Reynolds numbers for Test #2 is close to the laminar to turbulent transition value, as shown in Table 3.2. Then, nine different CFD simulations are performed with nine different base mesh sizes and the results are compared with the experimental data. In Table 4.4, the simulation results are given, while in Table 4.5 the percent errors between the CFD simulation and the nominal experimental results are illustrated. In addition, in Table 4.3 the number of cells used

in these simulations is provided. When the base mesh size is selected to be 10 mm, the CFD simulation results do not converge, therefore, there is no simulation data in Tables 4.4 and 4.5 for the base mesh size 10 mm.

Table 4.3: Base Mesh Size versus Cell Count Distribution

Base Mesh Size [mm]	Cell Count			
	Cold Side	HEX Body	Hot Side	Overall
0.5	374706	1967600	393181	2735487
1	313257	1770918	333664	2417839
1.5	310872	1785237	330880	2426989
2	304558	1734537	323731	2362826
2.5	296525	1684364	309811	2290700
3	281457	1683045	289574	2254076
5	203379	666804	203494	1073677
7.5	141105	418861	140645	700611
10	91763	281647	91455	464865

Examining Table 4.5, it can be seen that almost all of the simulated outlet temperature results for both fluids are higher than the experimental values. In the simulations only the heat exchanger body is simulated, however, outlet temperature measurements are performed about 30 cm away from the heat exchanger due to the steel pipes. Therefore, the simulated outlet temperatures are higher than the experimental outlet temperatures.

Also, in Table 4.2, the experimental hot side heat transfer rates are higher than the experimental cold side heat transfer rates. This can also be attributed to the steel pipes, as the temperature difference between the hot outlet section and the ambient temperature is higher than the temperature difference between the cold outlet section and the ambient temperature. Therefore in Table 4.5, CFD simulation results for hot heat transfer rates are lower than the experimental results while cold heat transfer rate results are higher.

Observing the effects of base mesh size reduction in Table 4.5, it is concluded that the

errors in the CFD simulations with a base mesh size of 5 mm are reasonable. Also, in Figure 4.2, the pressure drop results in CFD simulations are compared with different base mesh sizes.

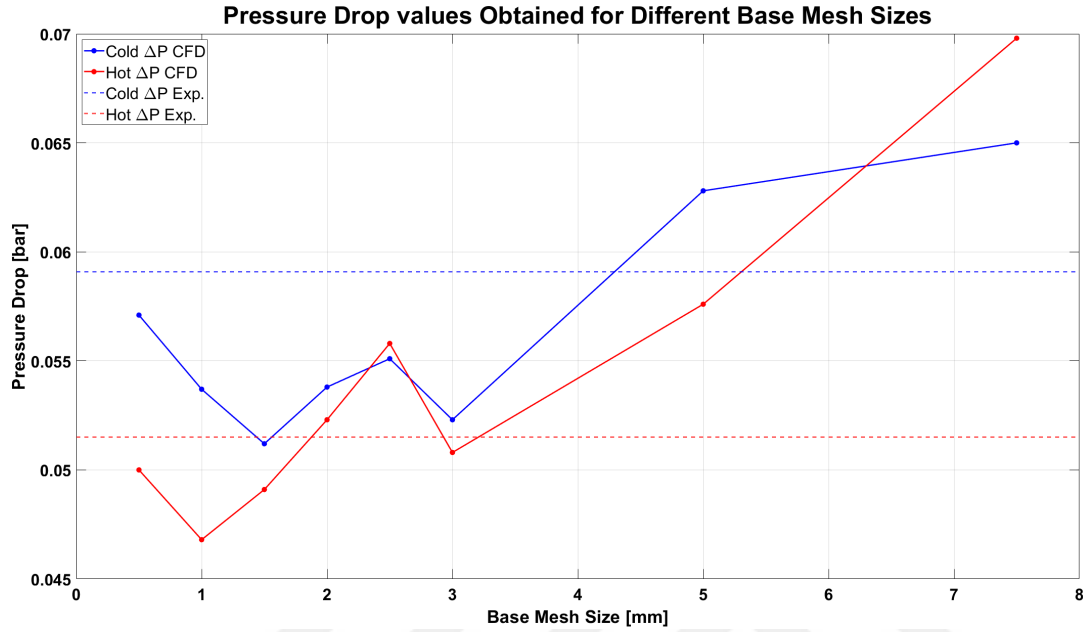


Figure 4.2: Pressure Drop Results in CFD Simulations vs. Base Mesh Size

As shown in Table 4.5, 5 mm base mesh size has around 10% accuracy with the experimental results while providing 60% reduction in total cell count compared to 0.5 mm. Therefore, the base mesh size for further simulations is selected as 5 mm. In Table 4.4 and 4.5, 5 mm base mesh size is highlighted in purple to emphasize that it is the selected base mesh size.

Considering a 10% accuracy with the experimental results, the CFD model is verified. Further CFD simulations are performed for alternative heat exchanger geometries and other flow tests conducted by changing only the geometries or the fluid properties depending on the simulation. In Section 4.3, all experimental conditions are simulated using the verified CFD model and the results are compared. In addition, in Section 4.4, the results of the CFD simulations performed for alternative heat exchanger designs are compared.

Table 4.4: CFD Simulation Results for Test #2 with Different Base Mesh Sizes

Gyroid 5x5x5 mm with 1 mm Wall Thickness (Manufactured Prototype)						
Total Cell Count	Base Mesh Size [mm]	Avg. Body Temp. CFD [°C]	HEX Temp. CFD [°C]	Outlet Temp. CFD [°C]	Heat Transfer Rate CFD [W]	Pressure Drop CFD [bar]
2735487	0.5	35.34		29.6	378.3	0.0571
				40.99	-378.4	0.0500
2417839	1	35.34		29.37	359	0.0537
				41.2	-358.6	0.0468
2426989	1.5	35.33		29.4	361.3	0.0512
				41.18	-361.1	0.0491
2362826	2	35.35		29.37	358.5	0.0538
				41.22	-358.8	0.0523
2290700	2.5	35.35		29.44	360.9	0.0551
				41.17	-361	0.0558
2254076	3	35.32		29.36	358.1	0.0523
				41.2	-358.3	0.0508
1073677	5	35.29		29.07	331.9	0.0628
				41.49	-332.1	0.0576
700611	7.5	35.29		28.7	299	0.0650
				41.9	-299.4	0.0698
464865	10	-		-	-	-
				-	-	-

Table 4.5: Percent Error in Test #2 CFD Simulation Results with Different Base Mesh Sizes Compared to Nominal Experimental Results

Gyroid 5x5x5 mm with 1 mm Wall Thickness (Manufactured Prototype)						
Total Cell Count	Base Mesh Size [mm]	Percent HEX Temp. Error	Avg. Body Temp. Error	Percent Outlet Temp. Error	Percent Transfer Rate Error	Percent Heat Rate Pressure Drop Error
2735487	0.5	1.96%		1.51%	26.23%	-3.32%
				0.74%	-0.45%	-3.01%
2417839	1	1.96%		0.72%	19.79%	-9.21%
				1.25%	-5.66%	-9.24%
2426989	1.5	1.93%		0.82%	20.55%	-13.35%
				1.20%	-5.00%	-4.69%
2362826	2	1.99%		0.72%	19.62%	-8.92%
				1.30%	-5.60%	1.42%
2290700	2.5	1.99%		0.96%	20.42%	-6.78%
				1.18%	-5.02%	8.24%
2254076	3	1.90%		0.69%	19.49%	-11.51%
				1.25%	-5.74%	-1.35%
1073677	5	1.82%		-0.31%	10.74%	6.27%
				1.97%	-12.63%	11.67%
700611	7.5	1.82%		-1.58%	-0.23%	9.92%
				2.97%	-21.23%	35.35%
464865	10	-	-	-	-	-
				-	-	-

4.3 Comparison of Experimental and Simulation Data

In Table 4.6, the percent errors between the CFD simulation and nominal experimental results for all experiments are illustrated. Furthermore, in Figure 4.3, heat transfer and pressure drop values were compared. Since the sensors used in the experiments are have high accuracy values (around $\pm 0.5\%$), it is hard to visualize the uncertainty in the experimental data in Figure 4.3.

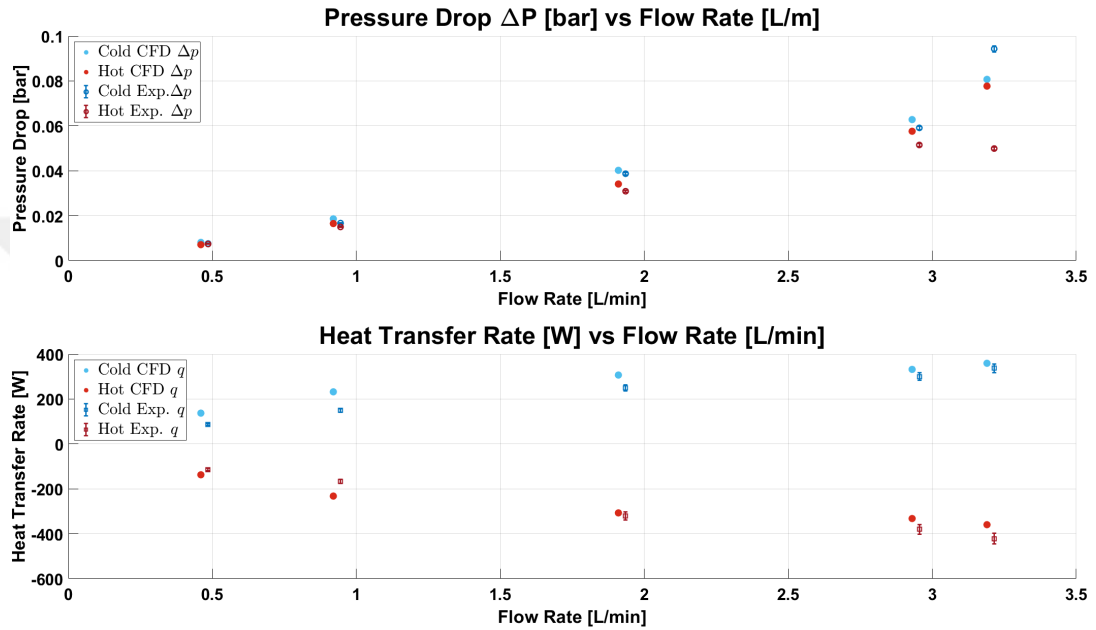


Figure 4.3: Comparison of Experimental and Simulated Heat Transfer and Pressure Drop Data

In the CFD simulations, it is assumed that the fluid properties are constant at the average fluid temperatures for the cold and hot sides at atmospheric pressure. In addition, steady state assumption is made, meaning that the inlet fluid temperatures are assumed to be constant in the CFD simulations. Although the assumptions in the CFD simulations reduce the accuracy of the model, it is shown in Figure 4.3 that the pressure drop results in the CFD simulations are consistent with the experimental data except for Test #1. Furthermore, in Figure 4.1, the experimental pressure drop values are shown to be in agreement with the analytical values calculated from Equations 3.20 and 3.21. The separation of experimental pressure drop data in Figures 4.1 and 4.3 for Test #1 from the analytical data can be attributed to the fact that the Reynolds

Table 4.6: Percent Error in CFD Simulations for Experiments Conducted using the Manufactured HEX Prototype

Tests	Gyroid 5x5x5 with 1 mm Wall Thickness (Manufactured Prototype) (Mesh Size 5mm)			
	Percent Avg. HEX Body Temp. Error	Percent Outlet Temp. Error	Percent Heat Transfer Rate	Percent Pressure Drop Error
Test #1	2.24 %	-0.35 %	6.43 %	-14.42 %
		2.39 %	-14.67 %	55.60 %
Test #2	1.82 %	-0.31 %	10.74 %	6.27 %
		1.97 %	-12.63 %	11.67 %
Test #3	-1.88 %	1.82 %	22.95 %	3.71 %
		1.34 %	-4.06 %	10.32 %
Test #4	0.29 %	7.96 %	55.28 %	11.53 %
		-4.35 %	39.27 %	9.61 %
Test #5	-4.19 %	9.76 %	59.35 %	6.18 %
		-3.61 %	20.69 %	-3.56 %

number for Test #1 is around the limits of the unstable laminar flow regime as shown in Table 3.2 and Section 3.2.1.

The outlet temperatures and the average heat exchanger body temperature obtained in the CFD simulations are within 10% of the nominal experimental values as shown in Table 4.6. However, as the flow rate decreases from Test #2 to Test #5, the error in the simulated heat transfer values increase while the error in the outlet fluid temperatures also increase. This suggests that as the flow rate decreases, the effect of unsimulated steel pipes at the inlet and outlet sections in the experimental setup increases. Since the inlet temperature readings are taken before the steel pipes and the outlet temperature readings are taken after the steel pipes, the calculated experimental heat transfer values include the heat transfer in these steel pipes. Further CFD simulations are per-

formed with the flow conditions of Test #2, therefore, the selection of 5 mm for the base mesh size is kept.

4.4 Comparison of CFD Simulation Results for Alternative Heat Exchanger Designs

A total of 9 different heat exchanger geometries are simulated for the experimental conditions of Test #2. Heat exchanger design alternatives consist of three types of TPMS structure, namely, Gyroid, Primitive, and Diamond structures. The simulation results for heat transfer, fluid outlet temperatures, and pressure drops are compared for all the generated TPMS cores.

In this section, performance of generated HEX designs are compared in Figures 4.4, 4.5, and 4.6. Additionally, the temperature distributions of the alternative heat exchanger designs at the end of the simulations are presented in Figures 4.7, 4.8 and 4.9. Having the highest solid to fluid interaction area among other TPMS core alternatives examined, HEX designs consisting of Diamond TPMS core structures showed the highest heat transfer rates in the simulations. However, because they do not have smooth channels for fluid flow, they have the smallest hydraulic diameters and hence the highest pressure drops. Figure 3.21 shows the specific solid to fluid interaction area and Figure 3.22 shows hydraulic diameters for all generated TPMS structures.

The lowest heat transfer rates are obtained for the Primitive TPMS core as they have the smallest solid to fluid interaction area, however, because they have large smooth channels, they showed the least pressure drops in the simulations. Gyroid type TPMS core alternatives performed better than the Primitive type cores in terms of heat transfer rates, resulting in lower pressure drops than Diamond cores. Therefore, Gyroid type TPMS cores show balanced HEX performance when compared to Primitive and Diamond structures.

These results showed that the heat transfer and pressure drop characteristics of TPMS structures are highly dependent on the specific area of the solid to fluid interaction and the hydraulic diameter, respectively. Although Gyroid structures have the highest porosity values, meaning the highest ratio of fluid volume to solid volume, the best

heat transfer rates are obtained with Diamond structures since they have higher specific solid to fluid interaction areas. As the cell size increases for all generated TPMS core alternatives, the heat transfer rates and pressure drop values decrease as a result of the increasing porosity. Furthermore, increasing the cell size in the fluid direction from 15 mm to 20 mm decreases the pressure drop, while not affecting the other parameters as much. Similar characteristics can also be observed in the literature [40, 21, 55].

The fluid channels inside a unit TPMS structure are symmetric, however, the design space dimensions are not exactly the multiples of the unit cell sizes. Therefore, the fluid channels inside the generated heat exchanger designs are not symmetric. Different cold and hot side pressure drops in the plots can be explained in this way.

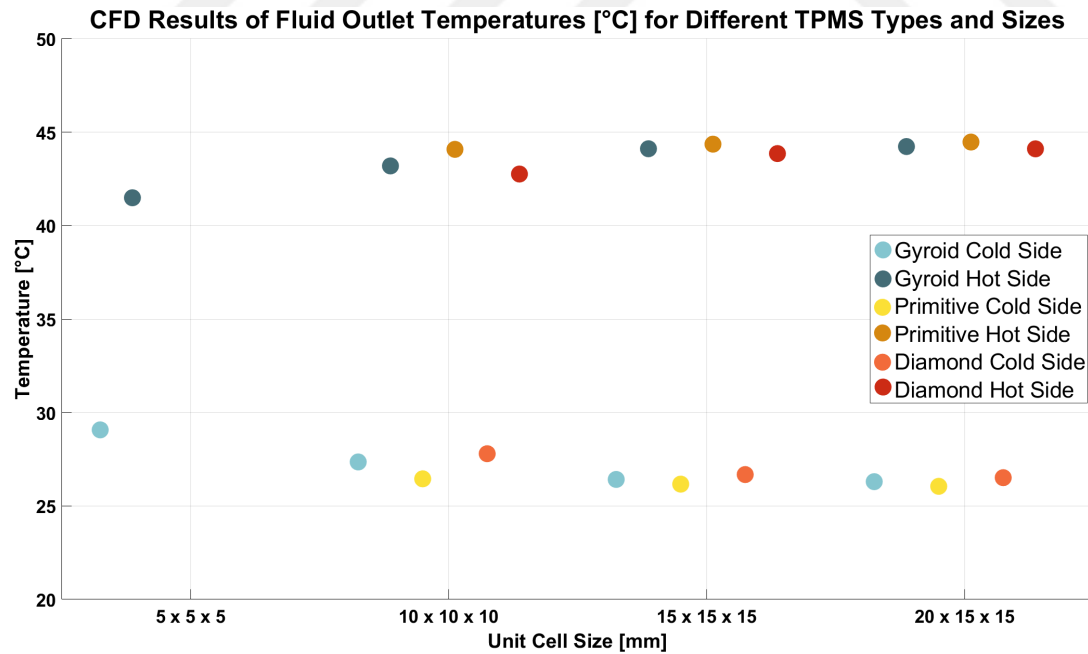


Figure 4.4: Fluid Outlet Temperature Data Obtained from Simulations for All Generated HEX Designs

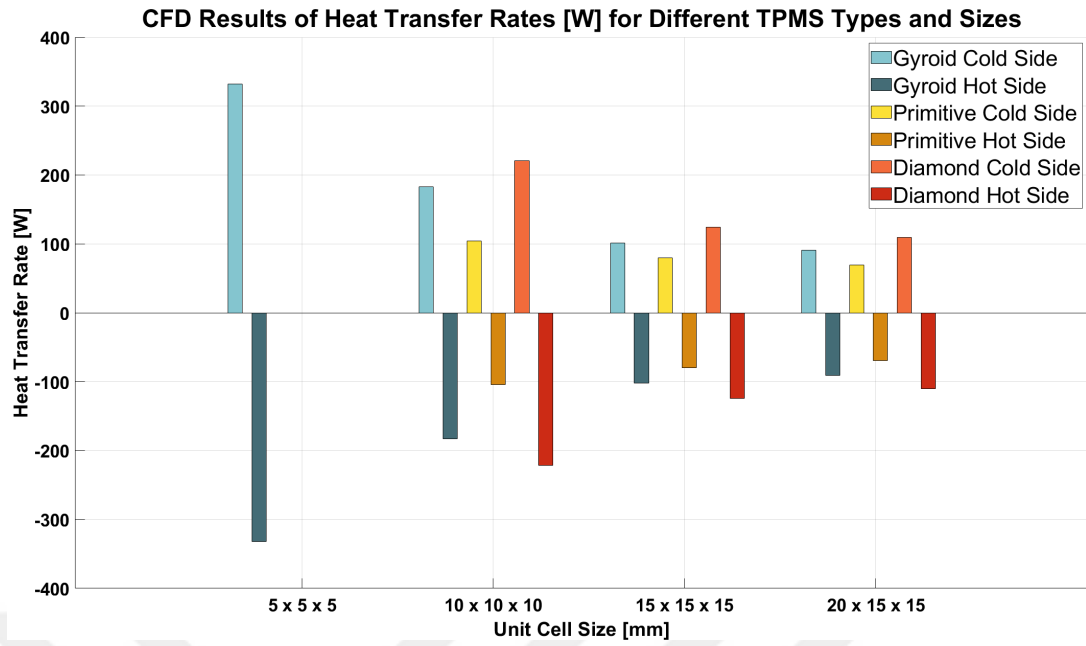


Figure 4.5: Heat Transfer Rates Obtained from Simulations for All Generated HEX Designs

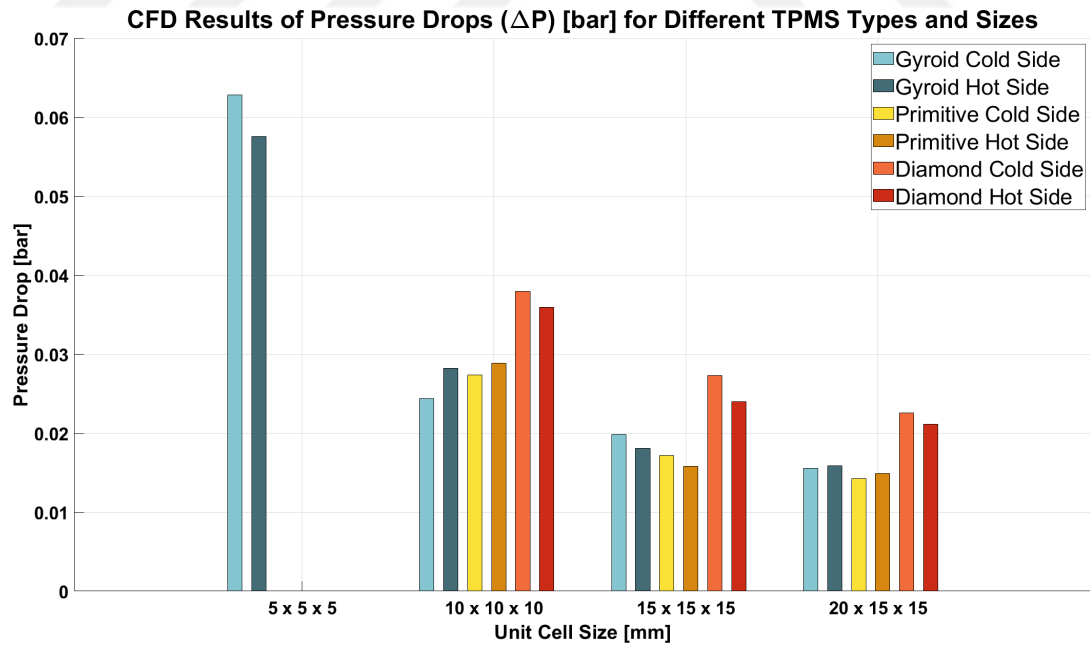


Figure 4.6: Pressure Drop Data Obtained from Analytical Calculations and Simulations for All Generated HEX Designs

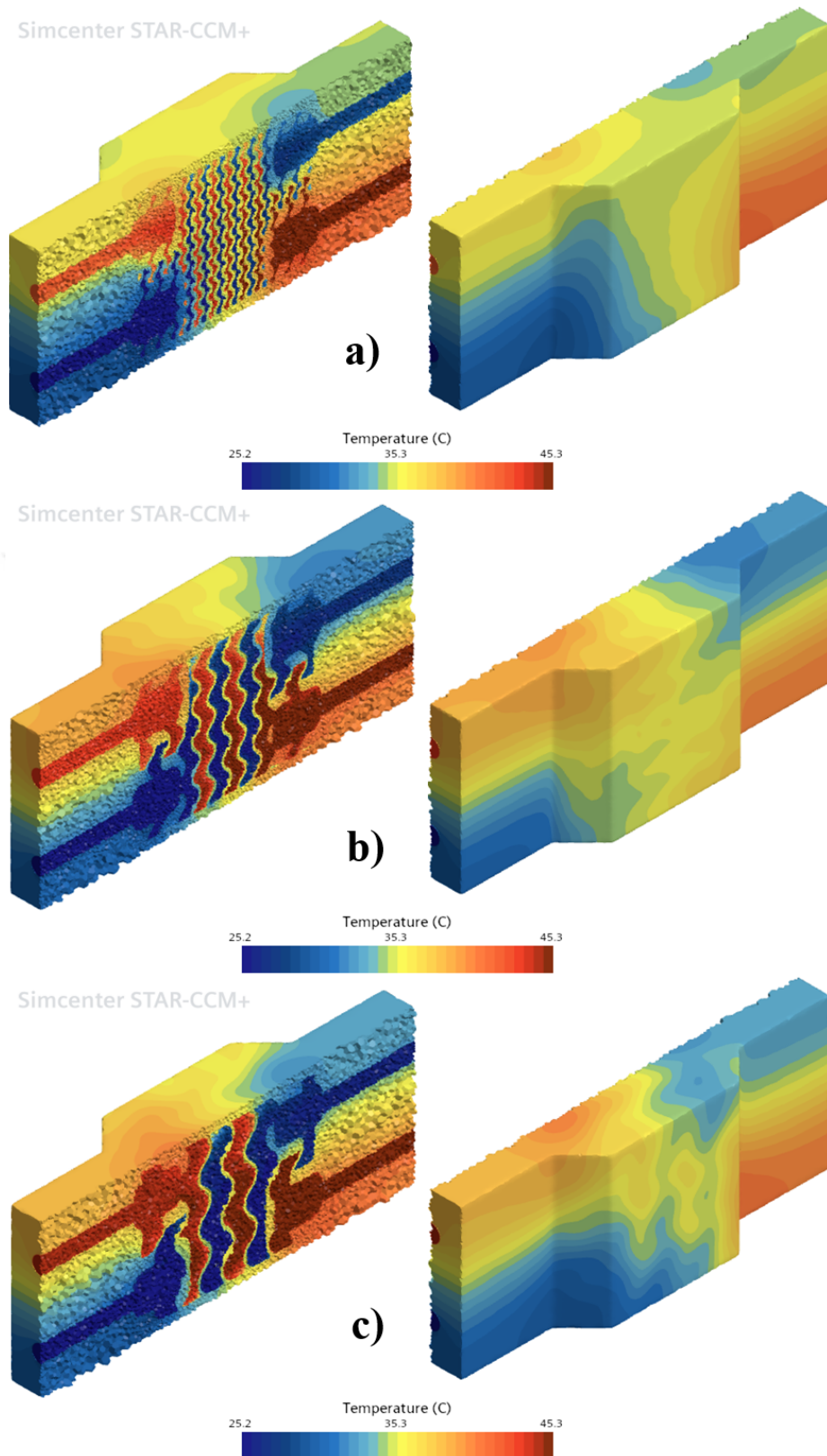


Figure 4.7: Temperature Distributions of Gyroid type Heat Exchangers Obtained in the Simulations. Unit Cell Sizes are a) 5x5x5 mm, b) 10x10x10 mm, c) 15x15x15 mm. (20x15x15 mm not shown)

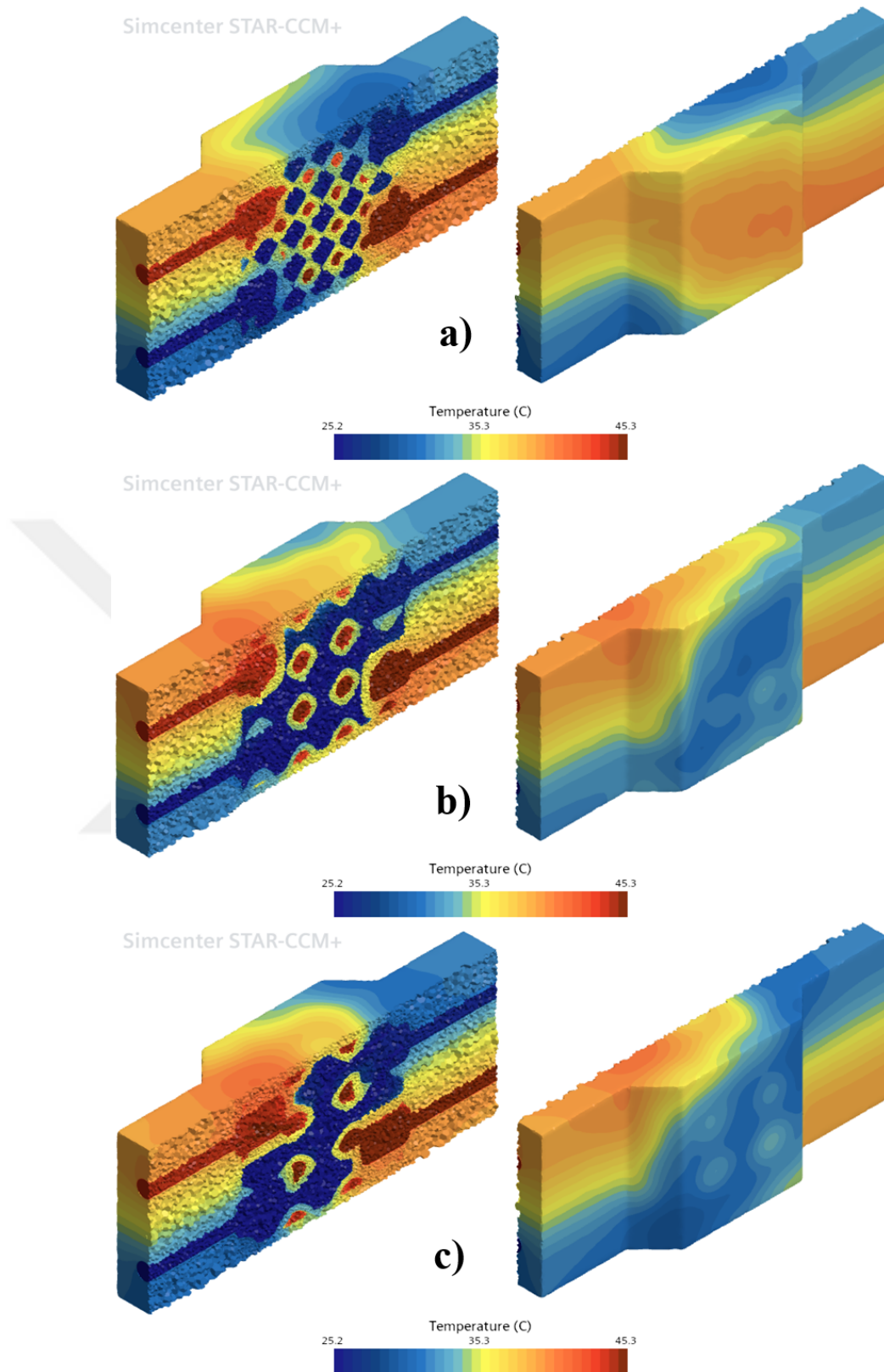


Figure 4.8: Temperature Distributions of Primitive type Heat Exchangers Obtained in the Simulations. Unit Cell Sizes are a) 10x10x10 mm, b) 15x15x15 mm, c) 20x15x15 mm.

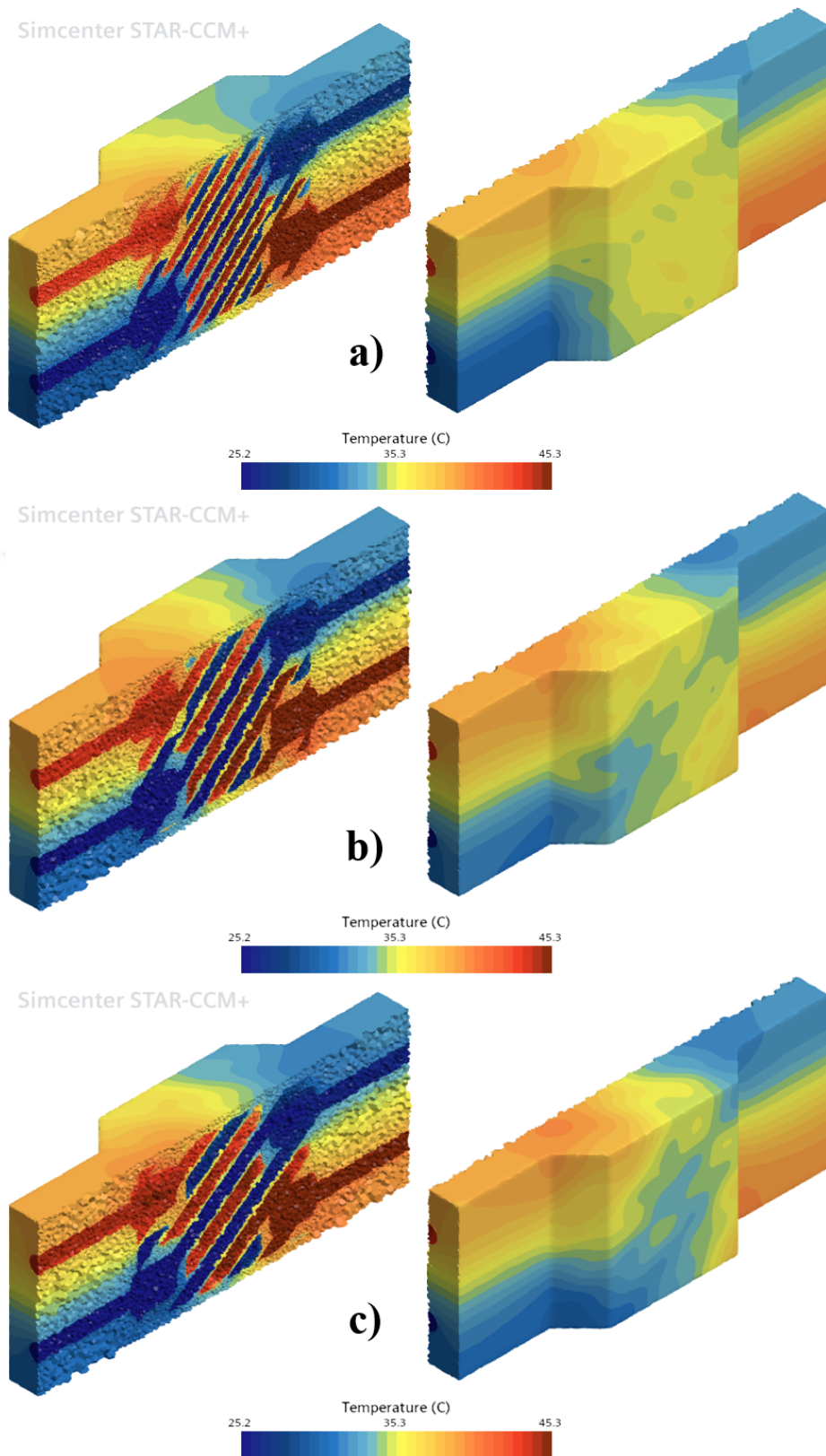


Figure 4.9: Temperature Distributions of Diamond type Heat Exchangers Obtained in the Simulations. Unit Cell Sizes are a) 10x10x10 mm, b) 15x15x15 mm, c) 20x15x15 mm.

4.5 Analytical Heat Exchanger Performance Evaluation Results

Figures 4.10, 4.11, 4.12, respectively illustrate the overall heat transfer coefficient U , number of transfer units NTU and effectiveness ϵ generated HEX designs featuring different TPMS core structures. Effectiveness values in Figure 4.12 are calculated using counter flow formulation in Equation 3.29. The effectiveness values calculated for the cross flow configuration by Equation 3.30 have errors up to 50% as described in Section 3.6.2. Therefore, the heat exchangers covered in this study can be characterized by using counter flow formulations.

The data presented in these figures are consistent with the data obtained by Passos (2019) [21] for the same Reynolds numbers. As expected, having the highest specific solid to fluid interaction area, Diamond type TPMS structures showed the best heat transfer performance followed by Gyroid type TPMS cores. Since the characteristic length in the Nusselt number calculation is the hydraulic diameter, Diamond structures have the lowest Nusselt numbers. However, although having the largest hydraulic diameters, Primitive structures could not outperform Gyroid structures in the Nusselt number because the heat transfer coefficients of Gyroid structures are higher.

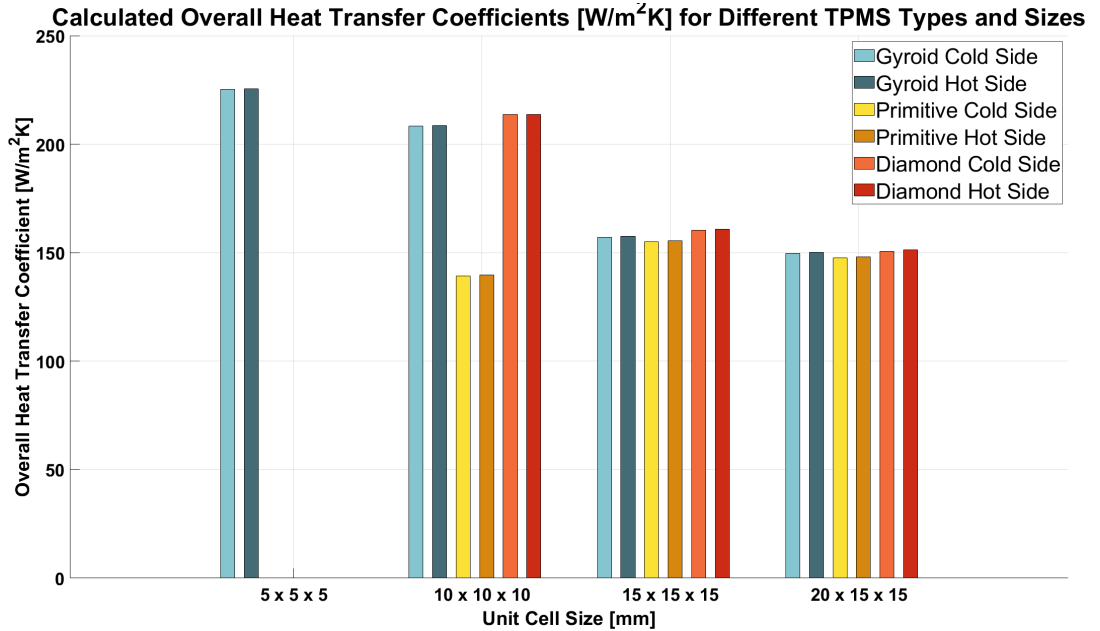


Figure 4.10: Overall Heat Transfer Coefficients within TPMS Core Alternatives

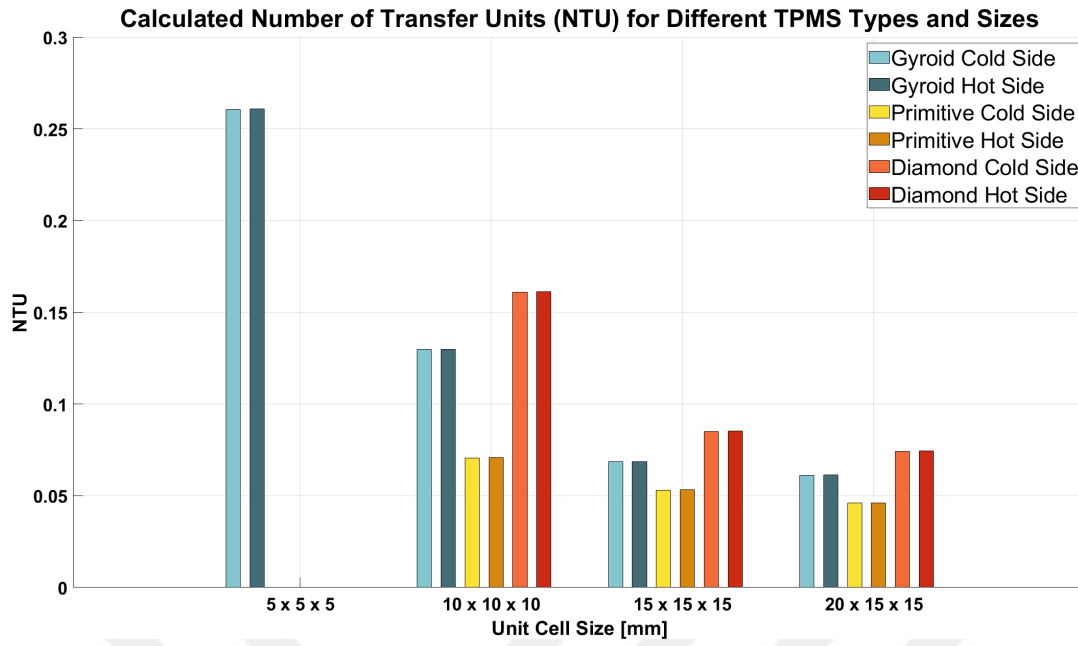


Figure 4.11: Number of Transfer Units (NTU) for TPMS Core Alternatives

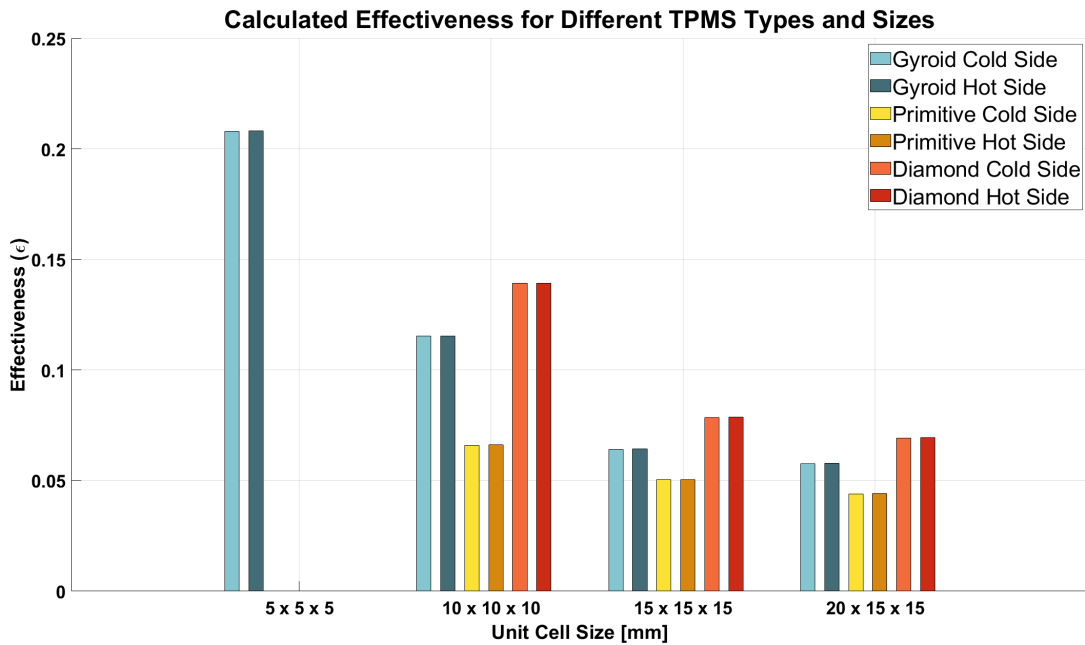


Figure 4.12: Effectiveness of TPMS Core Alternatives

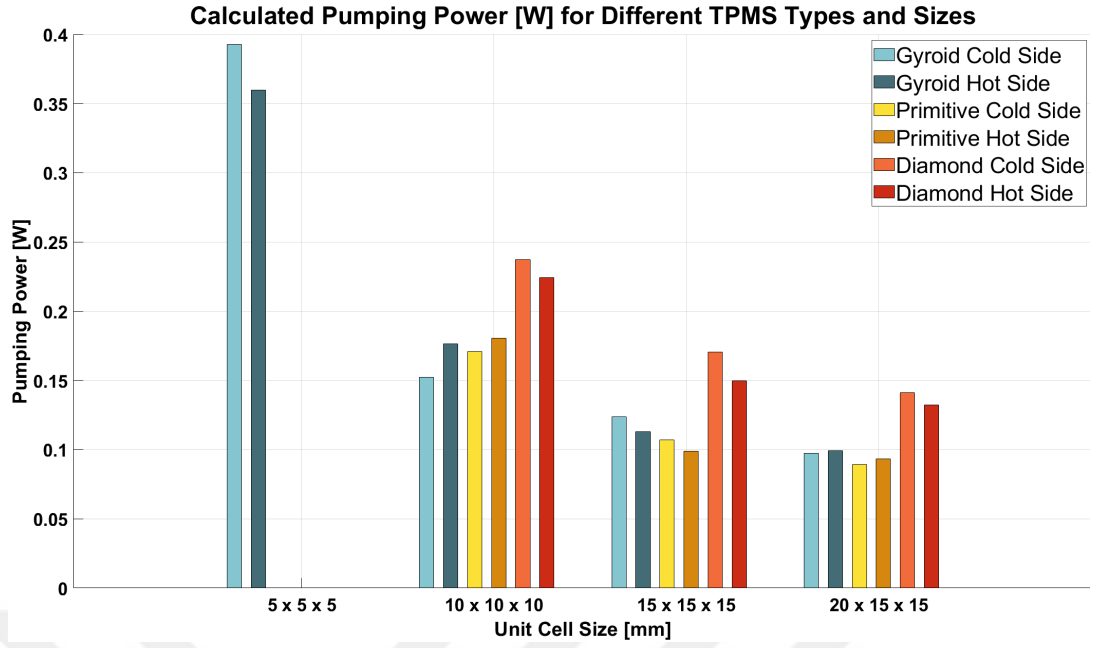


Figure 4.13: Pumping Power Required to Operate Generated Heat Exchanger Designs

The pumping power results for each heat exchanger design using Equation 3.32 are given in Figure 4.13. Since the pumping power is directly proportional to pressure drop, Figures 4.13 and 4.6 show the same behaviour among HEX design alternatives.

4.6 Volume Comparison Results of Shell & Tube Heat Exchangers with Generated TPMS Heat Exchangers

In Section 3.6.3, formulas for the volume requirement for single pass shell and tube heat exchangers are developed. Figure 4.14 shows the ratio of the TPMS core volume, which is 83215 mm^3 , to the total calculated volume for the pipes for each generated TPMS core structure using Equation 3.34. The pipe thickness increases the pipe volume quadratically whereas the design space volume was taken constant for TPMS based heat exchangers and it has little affect on the specific solid to fluid interaction area.

The most volume saving was achieved by the manufactured prototype since it has the smallest unit cell size and hence largest heat transfer (solid to fluid interaction) area. For the same unit cell sizes, the most volume saving can be obtained by utilizing the

highest specific solid to fluid interaction area property of Diamond type structures. Diamond type TPMS cores achieved around 30% to 50% volume savings according to the formulation described. A similar calculation was also performed by Chandrasekaran (2018) [58] and results showed that 32% volume reduction was achieved with Diamond type TPMS core having 15.72 mm of hydraulic diameter.

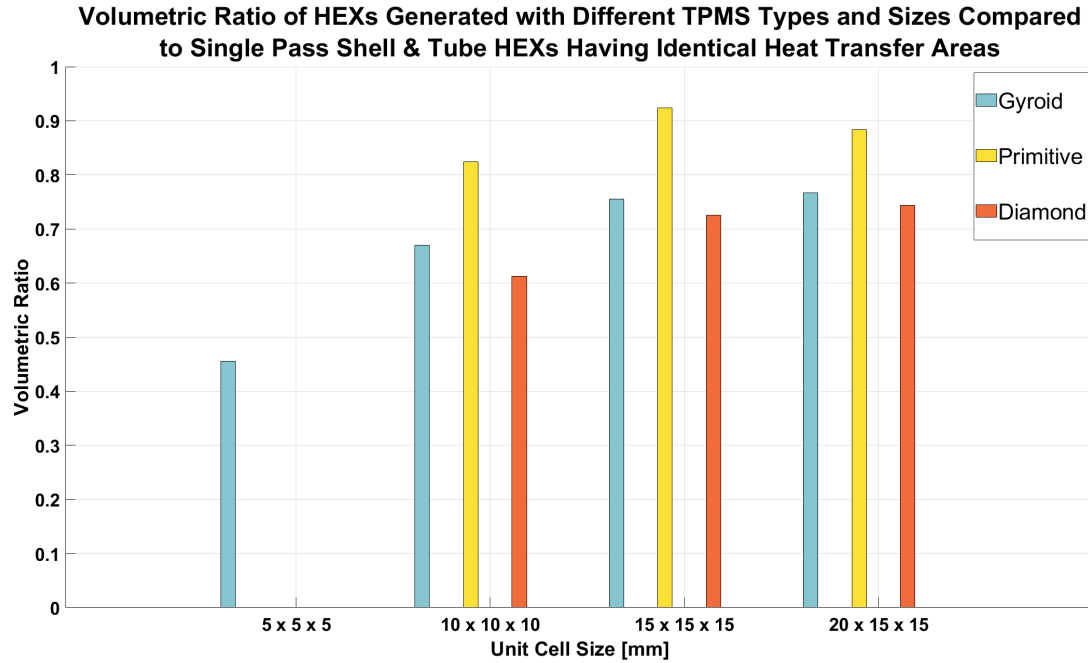


Figure 4.14: Volume Ratios of TPMS Core Alternatives to Total Required Pipe Volume

4.7 Summary of Discussions

The results of this study indicate promising heat exchanger performances by employing TPMS structures. The heat transfer properties can be geometrically tailored according to the designer's needs. For example, among the three different TPMS structures investigated in this study, the Primitive type structure shows the least pressure drop for each flow condition. However, the Primitive type cannot match the high heat transfer rates achieved by the Diamond type. In contrast, the Gyroid-type structure showed a balanced performance.

Using generative design tools such as nTopology [45] and considering manufacturing

limitations, complex parts such as TPMS type heat exchangers can be designed and manufactured. In this work, five different flow experiments are performed on the manufactured prototype, and the results are validated using analytical formulations and correlations in the literature. After the experiments, a CFD model is constructed and verified by the experimental results.

There are four key points that can cause the difference between the experimental and simulated data. The first is the difference between physical environment and the simulation environment. In the simulations, all of the physical elements in the test setup cannot be included. Therefore, the boundary conditions, inlet and outlet parameters used in the simulations do not exactly reflect the real test case. Another main point is the simulation assumptions such as steady state and constant fluid properties assumptions which are considered as simplifications for the simulation model. The third point is the location of the sensors. The temperature and pressure data for the inlet and outlet flows are sourced from distinct locations compared to the points used within the simulations since. The last main point is the geometry assumptions. In the simulations, the designed geometries are taken as perfect geometries, however in reality, the manufactured prototype can have defects such as plugged channels or high surface roughness.

By considering all of these effects, the experimental and simulation results are in accordance with the analytical calculations and the results in the literature.



CHAPTER 5

CONCLUSION

5.1 Summary of the Findings

The primary objective of this study was to develop a parametric design workflow for heat exchangers utilizing TPMS structures. Through practical tests and CFD analysis, this research investigates the characteristics of laminar flow and the properties of heat transfer in the TPMS based heat exchangers. The workflow starts by generating the heat exchanger geometry that incorporates mathematically defined core structures, namely, TPMS structures, as the heat transfer medium. After ensuring that the generated heat exchanger design was manufactured without failure, experiments for predetermined flow conditions were carried out using the constructed experimental setup. The experimental data obtained were used for the verification of the CFD simulation model so that various other heat exchanger designs generated could be simulated and evaluated. The results of this workflow showed that flow through TPMS structures can be characterized by using appropriate characteristic lengths, and analytical formulas such as the Darcy-Forchheimer pressure drop formula can be used for performance prediction.

Furthermore, it is shown in this study that the phases of designing, manufacturing, testing, and simulating a TPMS based heat exchanger each have their own unique challenges. For example, complex geometries are manipulated during the design stage while finding tools to manufacture the designed part can be difficult. Due to complex geometry and manufacturing techniques, cracks may develop on the manufactured prototype. During the testing stage, inducing pressure to the cracks can lead to unwanted consequences. On the other hand, simulation phase requires high

computational power due to complex geometry.

This study shows that the heat transfer performance of TPMS structures is directly related to their geometric properties such as porosity, hydraulic diameter, and solid to fluid interaction area. Although determining the solid to fluid interaction area of complex shapes like TPMS structures can be challenging, with the use of meshing tools, approximate values can be calculated. Pressure drop estimations can be performed using analytical friction factor formulas together with friction factor correlations for TPMS structures in the literature. Furthermore, by using generative design and simulation tools, various alternative heat exchanger designs can be characterized without prototype testing.

In this work, the flows through ten different heat exchanger designs that employ TPMS structures as heat transfer surfaces are considered. Three different types of TPMS structures are used as heat exchanger cores, namely Gyroid, Primitive, and Diamond structures, with three different unit cell sizes, are examined. An additional unit cell size is also used as a result of the manufacturing limitations encountered during the prototype production phase. The simulations verified by the experimental data showed that the Schwarz Diamond structure had the highest heat transfer rate for all unit cell sizes as a result of its highest heat transfer area. However, since there are no continuous channels inside the structure for fluid flow, the pressure drop values for the Schwarz Diamond structure are also the highest. On the other hand, the Schwarz Primitive structure showed the best performance for pressure drop while performing the worst in heat transfer results. This phenomenon can also be expected since the lowest heat transfer areas for each unit cell size alternative are calculated for the Schwarz Primitive structure. The lowest pressure drop values arise from the axial flow channels within the structure. The other alternative TPMS type, Schoen Gyroid structure, showed a balance between heat transfer and pressure drop values. A further performance evaluation of the generated heat exchangers is performed using the $\epsilon - NTU$ method. The overall heat transfer coefficient, NTU and effectiveness values obtained from this evaluation showed compatible results with the simulated heat transfer data and the literature. Additionally, effectiveness values are calculated by both counter flow and cross flow formulations and it is shown that the flow arrangement in this study is characterized better by the counter flow formulations.

Furthermore, the volume requirements of generated heat exchanger designs are compared with shell and tube heat exchangers. Assuming the same hydraulic diameter tubes and the same pipe thickness values as the TPMS wall thicknesses for the tube side of a shell and tube heat exchanger, around 30% to 50% volume savings are obtained by using TPMS structures at the core of heat exchangers.

5.2 Suggestions for Future Research

Continuing from the concepts explored in this thesis, here are some proposed directions for future research:

- Inclusion of funnels inside the heat exchanger to prevent fluids from getting trapped inside the core away from the outlet sections.
- Investigation of the effects of pool volumes at the inlet and outlet sections.
- Simulation and comparison of other types of TPMS structures such as Fischer-Koch S type.
- Data collection for different Reynolds numbers within the laminar region and turbulent region.
- Collecting a great number of data through flow tests and developing correlations for flow parameters
- Investigation of surface roughness values on the heat transfer performance of additively manufactured TPMS cores.



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