

Reduction of Flow Resistance with Hybrid TPMS Heat Exchangers

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Abstract— The design freedom enabled by additive manufacturing, combined with the unique properties of generative lattice structures, has gained increasing attention in thermal system technologies, especially for the development and fabrication of heat exchangers. In this study, a hybrid heat exchanger was developed by integrating a Diamond unit cell, known for its high thermal performance, with an IWP unit cell, which offers low flow resistance. The goal is to reduce flow resistance while maintaining heat transfer efficiency. A physical model was simulated using ANSYS software to evaluate the heat exchanger's performance. The results indicate that integrating IWP cells into a Diamond unit cell structure effectively reduces flow resistance, though with a slight compromise in heat transfer performance. This balance provides a viable design strategy for heat exchangers in applications requiring precise flow resistance control.

Keywords— Additive manufacturing, Lattice structure, TPMS, Heat exchanger.

I. INTRODUCTION

As industrial sectors like automotive, aerospace, and electronics continue to evolve rapidly, the demand for advanced thermal management technologies in electronic devices is increasing. Heat exchangers systems are essential for controlling heat transfer, cooling, and heating in these applications. [1]. The key challenge is developing innovative heat exchanger designs that meet the rigorous demands of industrial applications while maintaining manufacturability. Traditional heat exchangers, typically consisting of fins, columns, or similar components, have long been used due to the limitations of conventional manufacturing processes, which often rely on subtractive or condensation techniques [2]. However, the advent of additive manufacturing has revolutionized design possibilities. By enabling the layer-by-layer fabrication of complex geometries, it gives designers greater freedom to create more efficient and compact heat exchangers [3].

Among emerging solutions, lattice-based heat exchangers, particularly those incorporating triply periodic minimal surface (TPMS) structures have gained considerable interest [4]. TPMS geometries are characterized by continuous three-dimensional surfaces with smooth transitions and minimal volume usage. These designs provide an increased surface area

relative to their volume, making them highly suitable for efficient heat exchange [5]. Additionally, they are well-compatible with additive manufacturing, enabling the fabrication of components in various sizes, including features as small as 1 mm, with adjustable relative densities [6].

Recent studies have highlighted the superior thermal performance of TPMS structures compared to conventional designs. For example, Wei Tang et al. [7] examined the convective heat transfer performance of TPMS Gyroid, Diamond, and IWP geometries across different Reynolds numbers and compared them with conventional finned designs. Their findings showed that TPMS structures significantly outperformed their conventional counterparts, primarily due to the complex geometries inducing greater turbulence at the fluid/wall interface..

Based on these insights, researchers have continued to explore the thermal and hydraulic behavior of different TPMS unit cells to develop a comprehensive performance database. Jinghan Wang et al. [8], for example, provided simulation data for Primitive, IWP, FKS, and Neovius TPMS structures. Their analysis identified the Fischer-Koch S design as providing the best overall thermal performance and the lowest flow resistance across varying volume fractions and mass flow rates.

Despite these advancements, the practical application of TPMS heat exchangers is still limited by the reliance on standard unit cells, which can hinder customization for specific industrial needs, particularly in terms of optimizing flow resistance. To address this, Xu et al. [9] presented a hybrid approach, developing ten new TPMS unit cells derived from five standard cells. Their work aimed to expand the design space and enable new configurations for TPMS unit cells, expanding the boundaries of what is possible in thermal management design.

Numerous studies have highlighted that when evaluating the hydraulic and thermal behavior of various TPMS structures, the IWP unit cell offers the most extensive surface area for fluid flow [10]. This advantage stems from its distinctive geometry, which includes through holes that enable smooth fluid passage with minimal resistance. However, this enhanced flow capability leads to a reduced thermal exchange

effectiveness at the solid-fluid interface. Despite this limitation, the IWP structure remains a preferred choice in engineering applications, particularly for heat exchanger designs that require optimized flow resistance while maintaining specific design constraints.

In this research paper, a hybrid heat exchanger was designed by integrating a Diamond unit cell, known for its high thermal performance, with an IWP unit cell, which offers low flow resistance. The goal is to minimize flow resistance in heat exchangers while maintaining efficient heat transfer. The hybrid design features standard Diamond unit cells combined with two IWP cells positioned at the inlet and outlet, forming a 2(IWP) - 6(Diamond) - 2(IWP) configuration. This hybrid heat exchanger was compared to a standard Diamond unit cell heat exchanger. The results indicate that combining IWP unit cells significantly reduces flow resistance, albeit with a slight reduction in heat transfer performance. These findings provide a basis for designing heat exchangers that achieve an optimal balance between flow resistance and thermal efficiency.

II. PROBLEM STATEMENT

A. Generation of TPMS

TPMS surfaces are gaining increasing attention in various engineering domains, especially in thermal applications. These 3D structures display periodicity along all spatial axes (x , y , and z), enabling a continuous and uniform occupation of space. Their distinctive ability to maximize surface area within a confined volume makes them highly advantageous for thermal optimization. Furthermore, their smoothly curved geometry, free from sharp intersections, enhances heat dissipation efficiency [11].

Fourier transform equations, which rely on combinations of trigonometric functions, are widely recognized as reliable tools for generating TPMS geometries. The specific form and structure of the unit cell for the two configurations analyzed in this study are shown below (Figure 1), in alignment with the existing literature [12].

$$f_{IWP} = \cos(2\pi x) \cos(2\pi y) + \cos(2\pi y) \cos(2\pi z) + \cos(2\pi z) \cos(2\pi x) = 0.25 \quad (1)$$

$$f_{Diamond} = \cos(2\pi x) \cos(2\pi y) \cos(2\pi z) - \sin(2\pi x) \sin(2\pi y) \sin(2\pi z) = 0.155 \quad (2)$$

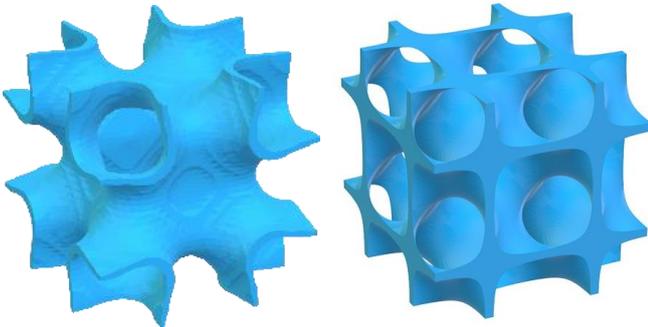


Fig. 1. Unit cell of the two structures studied: IWP and Diamond

B. The hybridization method

Improper transitions between different TPMS unit cell topologies can negatively impact fluid flow performance within the structure, potentially increasing flow resistance. To optimize the performance of hybrid heat exchangers, it is

essential to ensure a smooth transition between TPMS unit cells.

In this study, the sigmoid function method was applied to facilitate a seamless connection between two TPMS unit cells, as it is recognized as one of the most effective hybridization techniques [13].

From a mathematical perspective, the implicit function of a hybrid TPMS model incorporating two types of unit cells is expressed as follows:

$$\varphi_{Hybrid}(x, y, z) = \gamma \varphi_1(x, y, z) + (1 - \gamma) \varphi_2(x, y, z) \quad (3)$$

In this context, $\varphi_1(x, y, z)$ and $\varphi_2(x, y, z)$ correspond to the implicit functions of the two distinct components, while $\varphi_{Hybrid}(x, y, z)$ represents the resulting composite structure model. The parameter γ serves as a spatial weighting factor, ranging between 0 and 1.

This parameter γ is determined by the transition sigmoid function, expressed as follows:

$$\gamma = \frac{1}{1 + e^{kG(x,y,z)}} \quad (4)$$

In this case, $G(x,y,z)$ is a three-dimensional function that represents the spatial coordinates (x , y , z) at which the transition between TPMS unit cells takes place.

C. Physical Model

Standard Diamond and IWP TPMS unit cell heat exchangers were designed. Each exchanger consisted of ten standard unit cells, each measuring $10 \times 10 \times 10 \text{ mm}^3$, arranged in the flow direction. A 15% volume fraction was selected based on a study by El Khadiri et al. [14], which analyzed the thermal and flow characteristics of various TPMS heat exchangers. Their findings indicated that a 15% volume fraction provided optimal performance.

In contrast, a hybrid heat exchanger was designed using a combination of Diamond and IWP unit cells. Specifically, the inlet and outlet unit cells of the Diamond-based heat exchanger were replaced with IWP unit cells, resulting in a 2(IWP)-6(Diamond)-2(IWP) configuration.

The TPMS unit cells were created using MSLattice software and subsequently imported into ANSYS SpaceClaim for integration into the heat exchanger models. To ensure unobstructed flow at the inlets and outlets, extension sections of 100 mm were added.

Figure 2 illustrates the geometric model of the hybrid TPMS structures, where a boundary condition defines a fluid inlet velocity of 8 m/s at a constant temperature of 293.15 K. The outlet is modeled as a free exit. Additionally, the heat exchanger is heated from the lower surface with a uniform heat flux of 20 kW/m^2 . Symmetry boundary conditions are imposed on the remaining extension surfaces to prevent heat transfer beyond the study area.

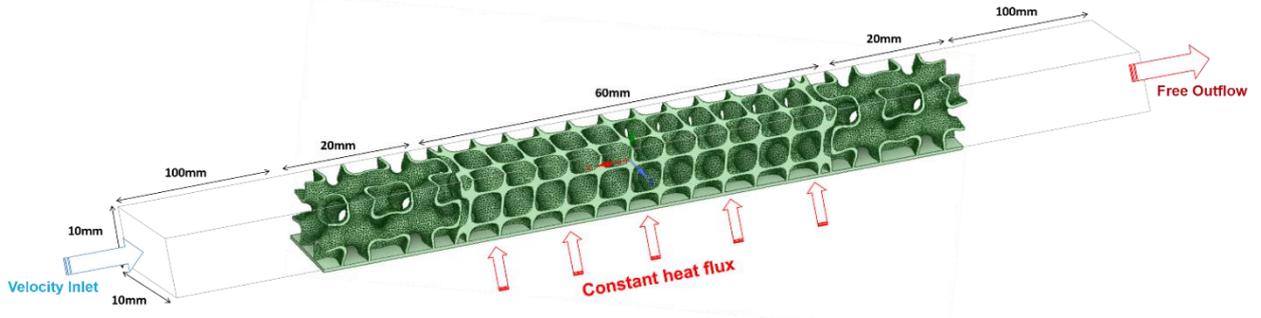


Fig. 2. Physical model.

D. CFD Method

ANSYS Fluent was used to apply the transient turbulent flow model and investigate the hydraulic and thermal behavior of the TPMS models. In the numerical simulations, aluminum was assigned as the solid material, while air served as the working fluid [7].

The fundamental governing equations, include the continuity, momentum, and energy are presented below (5-7):

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \cdot \bar{u}) = 0 \quad (5)$$

$$\frac{\partial(\rho * \bar{u})}{\partial t} + \nabla \cdot (\rho \cdot \bar{u} \cdot \bar{u}) = -\nabla_p + \nabla \cdot (\mu \left[(\nabla \cdot \bar{u} + \nabla \cdot \bar{u}^T) - \frac{2}{3} \nabla \cdot \bar{u} \right]) \quad (6)$$

$$\frac{\partial(\rho * E)}{\partial t} + \nabla \cdot (\bar{u}(\rho \cdot E + p)) = \nabla \cdot (\lambda \nabla \cdot T + \mu \left[(\nabla \cdot \bar{u} + \nabla \cdot \bar{u}^T) - \frac{2}{3} \nabla \cdot \bar{u} \right] \cdot \bar{u}) \quad (7)$$

Here: \bar{u} represents the fluid velocity, ρ is the fluid density, μ corresponds to the dynamic viscosity, p denotes the pressure, λ refers to the thermal conductivity, T indicates the fluid temperature, and E signifies the total energy.

To achieve accurate simulations of TPMS structures, the $k-\omega$ SST turbulence model was selected. This approach offers an effective compromise between computational efficiency and high precision [8]. Its strength lies in effectively capturing complex internal flow behavior caused by the intricate topology of TPMS structures.

E. Parameters

The hydraulic characteristics are assessed based on the pressure gradient (ΔP) and the Reynolds number (Re), whereas the thermal performance is evaluated using the heat transfer coefficient (h) and the Nusselt number (Nu). These parameters were calculated using the following equations:

$$Re_e = \frac{\rho d_h u_{in}}{\mu} \quad (9)$$

In this context, ρ refers to the fluid density, u_{in} indicates the fluid velocity at the entrance, and μ stands for the dynamic viscosity. The hydraulic diameter, d_h , is given by the following formula:

$$d_h = \frac{4 * V}{S} \quad (10)$$

Where, V is enclosed volume, while S represents the surface area.

The heat transfer coefficient h :

$$h = \frac{q}{T_w - T_f} \quad (11)$$

$$Nu = \frac{h d_h}{\lambda} \quad (12)$$

Where, T_w is temperature of the heating surface, while T_f represents the fluid temperature.

III. RESULTS AND DISCUSSION

Figures 3 and 4 depict the axial evolution of fluid pressure at an inlet velocity of 8 m/s, alongside the corresponding temperature distribution within the solid structure. The analysis includes both the standard IWP heat exchanger and a comparison between the standard Diamond heat exchanger and the hybrid design incorporating the IWP unit cell. The reported values represent cross-sectional averages, with red lines marking the hybridization points.

The results indicate that both standard models exhibit consistent trends in fluid pressure and structural temperature, influenced by the specific topology of their TPMS unit cells. In general, the IWP-based model experiences significantly lower pressure levels than the Diamond configuration.

However, this pressure reduction is accompanied by a decline in thermal performance for the IWP model, as the structure retains higher temperatures compared to the Diamond model.

Hybridization influences the pressure and temperature profiles, as evidenced by the transition to the IWP unit cell behavior at the hybridization points. This adaptation leads to reduced inlet and outlet pressures in the hybrid structure compared to the standard Diamond model, resulting in a notable decrease in overall pressure.

On the other hand, the structural temperature in the hybrid model is higher than that of the standard Diamond model due to the IWP unit cell's lower thermal transfer efficiency.

Table 1 compares the performance metrics of the IWP-Diamond hybrid heat exchanger, emphasizing the balance between pressure loss and thermal efficiency. The pressure drop decreases from 53.05 kPa/m to 49.34 kPa/m, indicating improved flow characteristics. However, this improvement comes at the cost of a reduced heat transfer coefficient, which declines from 2233.61 W/m²·K to 2031.93 W/m²·K. The Reynolds number increases from 1854.84 to 2024.19 in the hybrid model, while the Nusselt number remains nearly unchanged, varying only slightly from 294.55 to 294.06. This behavior is attributed to a change in hydraulic diameter caused by the integration of the IWP module cell, which enlarges the

fluid flow area. Overall, the results demonstrate that incorporating the IWP module reduces pressure losses with only a minor effect on thermal performance.

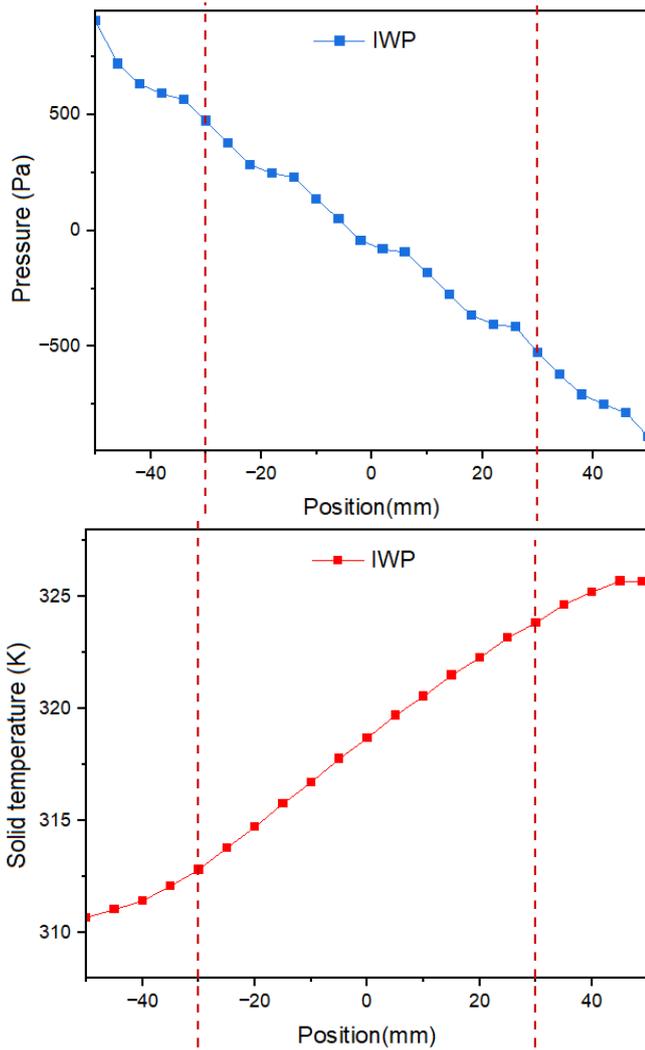


Fig. 3. Pressure and solid temperature distribution at 8 m/s fluid flow in the IWP Structure

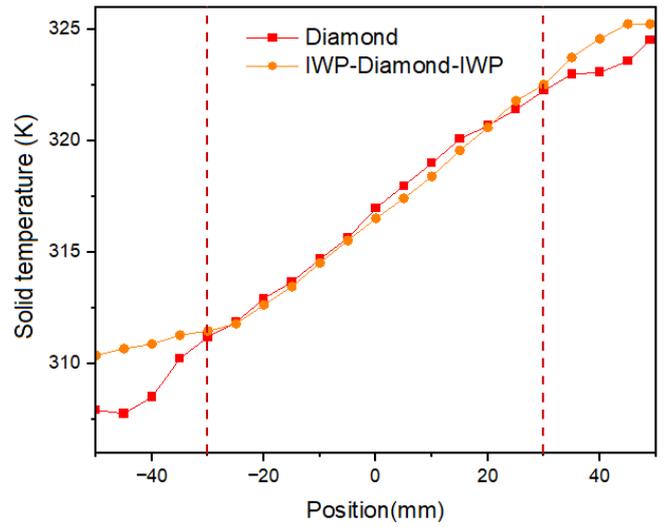
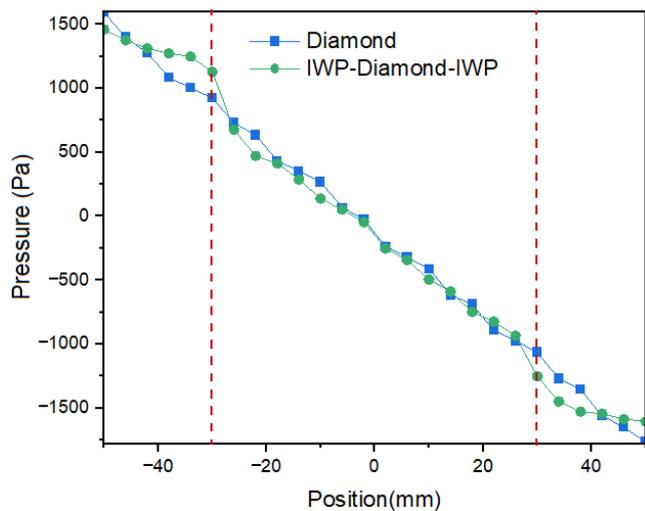


Fig. 4. Pressure and solid temperature distribution at 8 m/s fluid flow Diamond and IWP-Diamond Structures

TABLE I. FLOW AND HEAT TRANSFER PARAMETERS FOR DIAMOND AND IWP-DIAMOND STRUCTURES

Parameters	Diamond	IWP-Diamond
ΔP (kPa/m)	53.05	49.34
h (W/m ² *K)	2233.61	2031.93
Reynolds Number	1854.84	2024.19
Nusselt Number	294.55	294.06

CONCLUSION

In this study, hybridization was employed to better control fluid resistance within a TPMS-based heat exchanger. The IWP unit cell, known for its low fluid resistance among TPMS structures, was integrated into a Diamond heat exchanger, which offers superior thermal performance. This combination led to a reduction in pressure drop within the heat exchanger, albeit with a slight compromise in thermal efficiency. The results open new avenues for designing optimized TPMS-based heat exchangers tailored to specific engineering needs. Future work will extend this approach by integrating the IWP module with other TPMS structures and evaluating their thermo-hydraulic behavior under varying flow velocities. This data will inform the development of a topology optimization methodology that incorporates both thermal and hydraulic performance as objective functions, along with the integration of TPMS unit cells, enabling the identification of optimal designs tailored to specific industrial applications.

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