

Limitations of Equivalent Sand-Grain Roughness Modeling for Additively Manufactured Gyroid TPMS Heat Exchangers

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1. Introduction

Small modular reactors (SMRs) and other compact nuclear systems require high-density heat exchangers capable of meeting the required thermal load within a limited volume. Triply periodic minimal surfaces (TPMS) have attracted attention as promising geometries because of their large heat-transfer area per unit volume and enhanced mixing [1]. Among these, the gyroid structure is particularly notable because a single surface partitions the three-dimensional space into two interpenetrating yet independent regions, enabling a cold-wall-hot dual-channel heat exchanger to be constructed without separate flow-channel components.

Complex topological structures such as the gyroid can be fabricated using metal additive manufacturing (AM) processes, such as selective laser melting (SLM), which inherently result in rough internal surfaces [2]. The surface protrusions formed by partially melted powder particles adhering to the wall are on the order of tens to hundreds of micrometers, and even in a study where polishing was applied, the residual areal roughness was reduced to only about half of its original value [2]. Therefore, a substantial portion of the as-built surface roughness remains under actual operating conditions.

Yan et al. [3] reported that, in a single-phase experiment using an SLM-fabricated 316L gyroid heat exchanger, the measured pressure drop substantially exceeded the numerical prediction obtained without accounting for surface roughness. The same group of authors observed a deviation of similar magnitude in a follow-up study [2] using a separate prototype with a different geometry. Another study subsequently reported the same trend for annular TPMS heat exchangers (Diamond and Gyroid) using air as the working fluid. In that study, the measured pressure loss of the Gyroid prototype exceeded the roughness-free numerical prediction by 50–60% at low Reynolds numbers and by 67–70% at high Reynolds numbers [4].

Meanwhile, prior numerical studies applying equivalent sand-grain roughness (ESGR)-based correction models to TPMS have been limited to a Schwarz-P heat sink using a single working fluid [5]. However, that study validated the model only under smooth-wall conditions, and the roughness-corrected results were not compared with experimental data. To the

authors' knowledge, no previous study has applied a roughness correction to a gyroid heat exchanger in which two fluids flow through separate channels divided by a wall and directly compared the results with experimental data.

The objectives of this study are threefold: (1) to directly compare the roughness-free (baseline) CFD prediction for the gyroid heat exchanger with experimental data; (2) to convert the surface roughness reported in the literature into ESGR and incorporate it into the CFD model to quantify the resulting changes in performance; and (3) to determine whether the roughness correction resolves the discrepancy between the CFD predictions and experimental results and, if not, to identify the underlying cause.

2. Analysis Method

2.1 Target Geometry and Analysis Conditions

The gyroid surface is defined by the implicit function of Eq. (1).

$$\cos(2\pi x/L)\sin(2\pi y/L) + \cos(2\pi y/L)\sin(2\pi z/L) + \cos(2\pi z/L)\sin(2\pi x/L) = c \quad (1)$$

Here, L denotes the unit-cell size, and c is a constant that offsets the level surface and thereby determines the wall thickness. This study adopts the same design specifications as those of the experimental specimen used for comparison [3]: $L = 4$ mm, a $5 \times 5 \times 5$ array with a total dimension of 20 mm per side, and a wall thickness of 0.5 mm. Extension ducts, each 20 mm in length, were added to the inlet and outlet to stabilize the flow, resulting in the computational domain shown in Fig. 1. Figure 2 illustrates how the single gyroid surface partitions the unit cell into cold-side and hot-side fluid regions separated by a solid wall.

Water was used as the working fluid, with inlet temperatures of 283.15 K on the cold side and 323.15 K on the hot side. The outlets were specified as pressure outlets with a gauge pressure of 0 Pa, as shown in Fig. 1. The mass flow rate was varied from 50 to 500 kg/h, and the roughness analysis was performed at 500 kg/h, the highest flow rate examined, where the deviation in the overall heat transfer coefficient was the largest. The solid region was modeled as 316L stainless steel, and conjugate heat transfer was applied at the fluid–solid interfaces, while the outer walls were treated as adiabatic.

The SST $k-\omega$ turbulence model was employed, and the simulations were performed using the pressure-based steady-state solver in ANSYS Fluent.

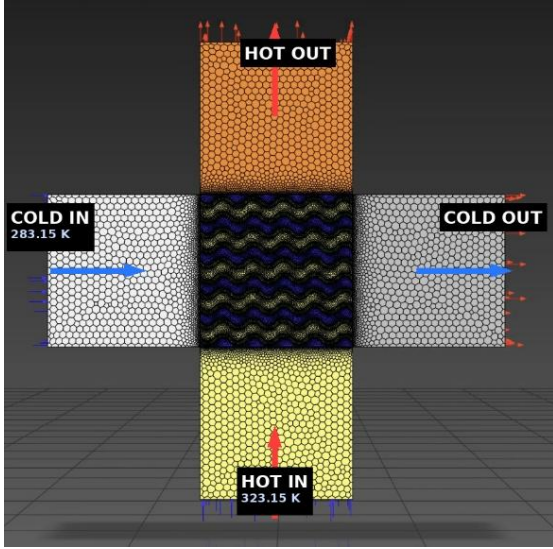


Fig. 1. Computational domain and boundary conditions.

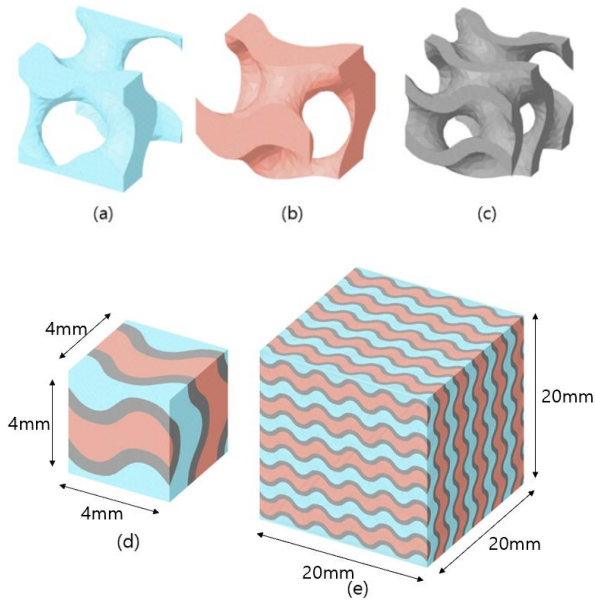


Fig. 2. Region partitioning of the gyroid unit cell and the $5 \times 5 \times 5$ core. (a) cold-side fluid region, (b) hot-side fluid region, (c) solid partition wall, (d) assembled unit cell, (e) $5 \times 5 \times 5$ core.

2.2 Mesh and Grid Convergence

Figure 3 shows the polyhedral mesh with inflation layers applied near the walls. A grid convergence analysis based on Roache's grid convergence index (GCI) [6] was performed at a mass flow rate of 500 kg/h using three mesh levels (Coarse, Medium, and Fine). The GCI for the overall heat transfer coefficient, U , was 2.65% for the Medium mesh, satisfying the 3% acceptance criterion adopted in [3]. Although the GCI for the Fanning friction factor, f , was 9.54%, exceeding this criterion, the relative

differences between the Medium and Fine meshes were only 0.12% for U and 0.43% for f , and all variables exhibited monotonic convergence. Therefore, the Medium mesh was selected for subsequent analyses, consisting of 13.47 million cells with a first wall-adjacent cell height of 0.02 mm. Convergence was assessed based on the energy and mass imbalances of the two fluid domains, together with the stability of the outlet temperatures and wall heat flux.

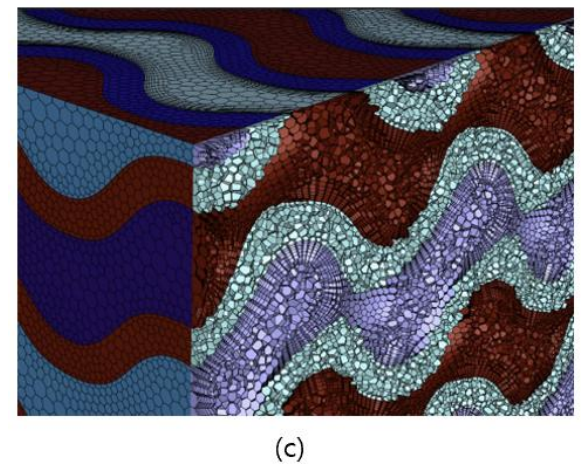
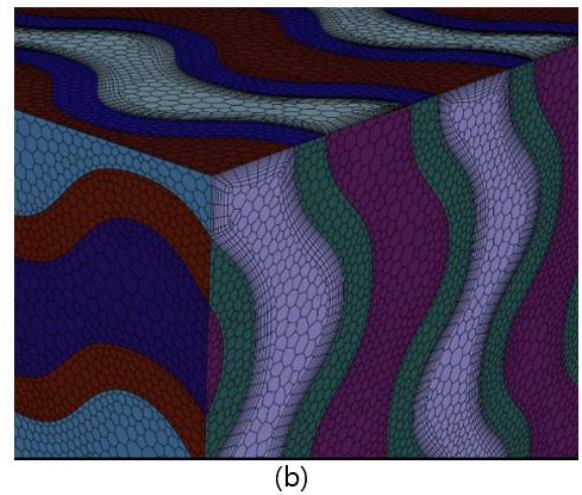
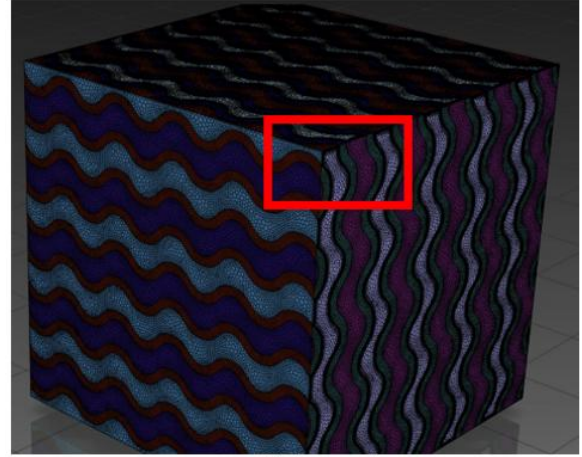


Fig. 3. Computational mesh. First wall-adjacent cell height 0.02 mm, 7 inflation layers, growth rate 1.2. (a) overall mesh of the core, (b) magnified view of the mesh, (c) wall inflation

layer viewed at a clipping cross-section.

2.3 Surface Roughness Modeling

2.3.1 Equivalent Sand-Grain Roughness and Velocity Shift

Directly resolving the irregular surface protrusions of the actual specimen in the computational mesh would require three-dimensional geometric data of the specimen surface. However, this study has access only to the roughness parameters reported in the literature. Therefore, the equivalent sand-grain roughness (ESGR) approach [7] was adopted. Based on Nikuradse's classical sand-grain pipe-flow experiments, this approach represents an arbitrary rough surface by an equivalent sand-grain height that produces the same increase in friction. The dimensionless roughness height is defined by Eq. (2)

$$k_s^+ = \frac{\rho u_\tau k_s}{\mu} \quad (2)$$

where k_s is the equivalent sand-grain roughness height [m], $u_\tau = \sqrt{\tau_w/\rho}$ is the friction velocity [m/s], ρ is the density [kg/m³], and μ is the viscosity [Pa·s]. Because u_τ is calculated from the wall shear stress τ_w , k_s^+ is updated cell-by-cell at every iteration.

For a rough wall, the logarithmic-law region is preserved but shifted downward, and the magnitude of this shift is represented by the velocity shift Δu^+ . The High Roughness (Icing) model implemented in ANSYS Fluent incorporates the approach proposed by Aupoix [7] for the SST turbulence model and takes the roughness height K_s as an input parameter. The underlying Colebrook-type relation is given by Eq. (3)

$$\Delta u^+ = \frac{1}{\kappa} \ln \left[1 + \frac{k_s^+}{\exp(\kappa(8.5 - C))} \right] \quad (3)$$

where $\kappa = 0.41$ is the von Kármán constant and $C = 5.25$ is the intercept of the smooth-wall logarithmic law. The dimensionless velocity shift, Δu^+ , represents the increase in friction caused by surface roughness.

2.3.2 Wall Boundary Conditions and Treatment in Fluent

Because k - ω -family models do not employ a wall-damping function, the effects of surface roughness can be accounted for only by directly modifying the wall values of the turbulent kinetic energy, k , and the specific dissipation rate, ω . Equations (4) and (5) represent the wall boundary conditions derived by Aupoix [7] from Colebrook data.

$$k_w^+ = \max(0, k_0^+) \quad (4a)$$

$$k_0^+ = \frac{1}{\sqrt{\beta^*}} \tanh \left\{ \left[\frac{\ln \left(\frac{k_s^+}{30} \right)}{\ln 10} + 1 - \tanh \left(\frac{k_s^+}{125} \right) \right] \cdot \tanh \left(\frac{k_s^+}{125} \right) \right\} \quad (4b)$$

$$\omega_w^+ = \frac{300}{k_s^{+2}} \left[\tanh \left(\frac{15}{4k_s^+} \right) \right]^{-1} + \frac{191}{k_s^+} \left[1 - \exp \left(-\frac{k_s^+}{250} \right) \right] \quad (5)$$

where $\beta^* = 0.09$ is an SST model constant, and $k_w^+ = k_w/u_\tau^2$, $\omega_w^+ = \omega_w \nu / u_\tau^2$ are the dimensionless wall values. For a smooth wall, $k_w = 0$ and ω_w approaches $6\nu/(\beta y^2)$ near the wall. For a rough wall, a finite value of k_w is imposed and ω_w is reduced. These modifications enhance the near-wall turbulence and thereby reproduce the increase in friction caused by surface roughness.

The default roughness treatment in ANSYS Fluent modifies the logarithmic law of the wall function and therefore requires the first wall-adjacent cell height to be greater than the roughness height. In the present analysis, the first-cell height is 20 μm , whereas K_s is on the order of 300 μm ; thus, this requirement is not satisfied. In contrast, the High Roughness (Icing) treatment does not employ wall functions but directly prescribes the wall values of k and ω according to Eqs. (4) and (5). It is applicable in the low-Reynolds-number regime and therefore not subject to this constraint. Instead, it requires sufficient near-wall resolution to maintain $y^+ < 1$. Accordingly, the High Roughness (Icing) treatment was adopted in this study, and the roughness condition was applied only to the gyroid fluid–solid interfaces, i.e., the partition walls separating the cold- and hot-side fluid regions.

The extension ducts are machined surfaces in the actual specimen and were therefore not assigned a roughness condition. The core-side walls, although also produced by additive manufacturing, were excluded because they constitute only a small fraction of the total heat-transfer area. A correction to the turbulent Prandtl number for rough-wall conditions has also been proposed within the same framework [8]; however, it was not implemented in the present analysis. Therefore, the heat-transfer results presented below should be interpreted as uncorrected values with respect to the rough-wall turbulent Prandtl number.

2.4 Roughness Parameter Estimation

Surface roughness was not measured directly in the present study. Instead, the pre-polishing roughness values reported by the same group of authors [2] for a 316L gyroid specimen fabricated using the same equipment (LiM-X260A), process parameters, and post-processing procedure (sandblasting) were adopted. The reference study reported that the surface morphologies of the two specimens were essentially similar. The arithmetic mean areal roughness, S_a , was measured at two different locations, as summarized in Table 1.

Because the measured roughness parameters cannot be directly input into the CFD model, they were converted into equivalent sand-grain roughness using the correlation proposed by Adams et al. [9], given in Eq. (6).

$$k_s = 5.863 R_a \quad (6)$$

Accordingly, two roughness cases, $K_s = 346 \mu\text{m}$ and 307 μm , were considered, as shown in Table 1. Since Eq. (6) was originally developed as a correlation using the

profile roughness R_a as its input, the present study used the areal roughness S_a as its areal counterpart of R_a .

Table 1. Roughness model input parameters

Parameter	Case A	Case B
S_a (μm)	59.07	52.45
K_s (μm)	346	307
k_s^+ (cold / hot)	102 / 192	88 / 168

3. Results and Discussion

All results below are for 500 kg/h on the Medium mesh.

The roughness-free CFD simulation predicted a cold-side pressure drop of 20,058 Pa and an overall heat transfer coefficient of 10,500 W/m²·K at a mass flow rate of 500 kg/h. Consistent with [3], it was evaluated using the cold-side fluid–solid interface area of the core ($20 \times 20 \times 20$ mm) as the reference area and calculated from the cross-flow effectiveness–NTU relation. Here, the number of transfer units (NTU) is defined as the product of the overall heat transfer coefficient, U , and the heat transfer area, A , divided by the minimum fluid heat-capacity rate, $C_{\min}$. Digitization of the curves reported in [3] yielded experimental values of 36,220 Pa and 12,065 W/m²·K, respectively, under the same operating conditions. Thus, relative to the roughness-free CFD predictions, the experimental values were approximately 80.6% higher for pressure drop and 14.9% higher for the overall heat transfer coefficient.

As shown in Fig. 4, these deviations persist across the entire mass flow rate range of 50–500 kg/h. The pressure-drop deviation ranges from +65.7% to +84.1%, while the deviation in the overall heat transfer coefficient ranges from +9.2% to +14.9%. The deviations reported in [3] relative to its own numerical predictions, namely +69.82% for pressure drop and +9.70% for the overall heat transfer coefficient, are somewhat smaller than those obtained in the present study. This difference is likely attributable to differences in the numerical solver and mesh resolution.

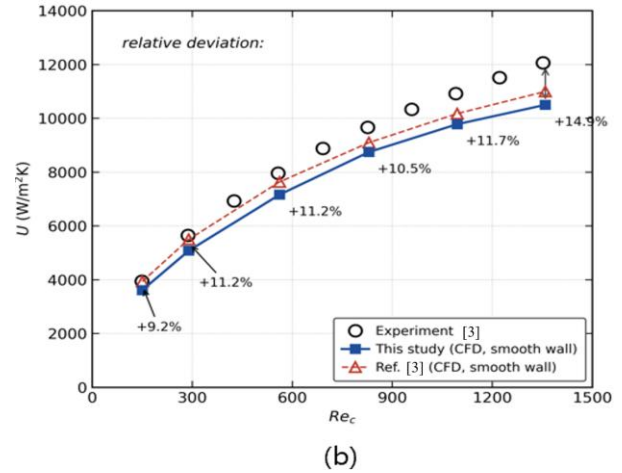
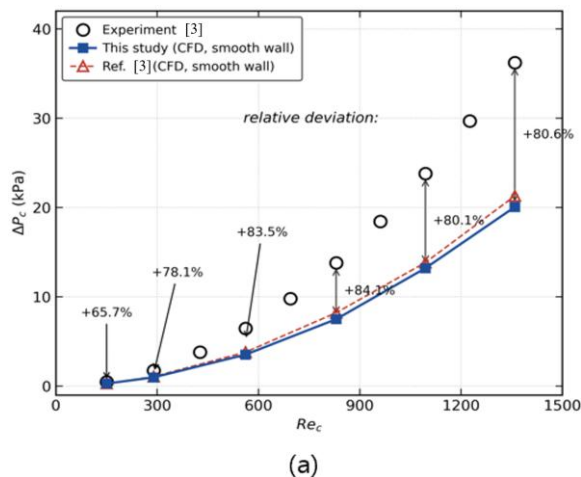


Fig. 4. Comparison of roughness-free CFD with the literature experimental values [3] over the flow-rate range 50–500 kg/h. (a) Pressure drop, (b) overall heat transfer coefficient.

3.2 Pressure Drop and Wall Shear Stress

The computed wall shear stress and pressure drop are summarized in Table 2. Under the literature-based roughness conditions, the wall shear stress increased by 61.0–69.6% on the cold side and by 81.8–86.1% on the hot side. In contrast, the pressure drop increased by only 8–10% on both sides, corresponding to approximately one-eighth of the 80.6% deviation observed experimentally.

Table 2. Performance change with roughness applied (500 kg/h). Values in parentheses are the change relative to the case without roughness. R = relative increase in pressure drop / relative increase in wall shear stress.

Variable	K_s				
	0 μm	307 μm	346 μm	700 μm	1400 μm
τ_w cold (Pa)	76.65	123.42 (+61.0%)	130.01 (+69.6%)	169.81 (+121.5%)	213.74 (+178.8%)
τ_w hot (Pa)	54.70	99.45 (+81.8%)	101.80 (+86.1%)	121.64 (+122.4%)	142.27 (+160.1%)
Δp cold (Pa)	20,058	21,668 (+8.0%)	22,092 (+10.1%)	24,423 (+21.8%)	27,599 (+37.6%)
Δp hot (Pa)	16,728	18,067 (+8.0%)	18,220 (+8.9%)	18,889 (+12.9%)	19,967 (+19.4%)
U (W/m ² ·K)	10,500	13,501 (+28.6%)	13,722 (+30.7%)	14,516 (+38.2%)	14,761 (+40.6%)
R, cold (–)	–	0.132	0.146	0.179	0.210
R, hot (–)	–	0.098	0.104	0.106	0.121

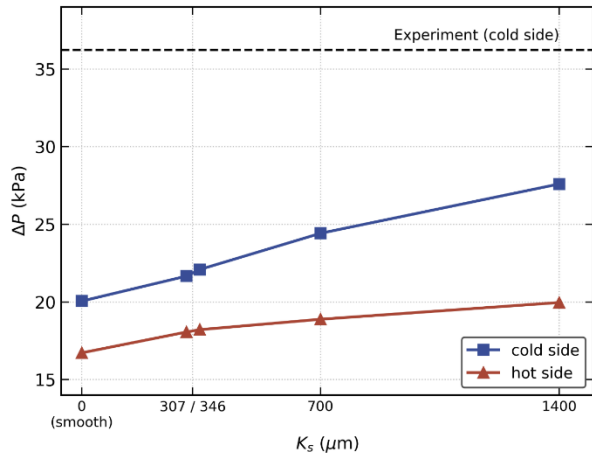


Fig. 5. Change in the cold- and hot-side pressure drop with the equivalent sand-grain roughness K_s at 500 kg/h. The horizontal line denotes the experimental value [3].

3.3 Heat Transfer

Heat transfer also increased under the roughness conditions. At $K_s = 346 \mu\text{m}$, the cold-side Nusselt number increased by 78.2%, comparable to the 69.6% increase in wall shear stress. In contrast, the increase in the Nusselt number on the hot side was smaller, at 42.7%, despite an 86.1% increase in wall shear stress.

The overall heat transfer coefficient increased from 10,500 to 13,722 $\text{W/m}^2\text{K}$, corresponding to a 30.7% increase. By comparison, the experimental result requires an increase of only 14.9%, from 10,500 to 12,065 $\text{W/m}^2\text{K}$. Thus, the roughness-corrected CFD overpredicted the overall heat transfer coefficient by 13.7%.

3.4 Relative Increase Ratio and Drag Decomposition

The ratio of the relative increase in pressure drop to the relative increase in wall shear stress, hereafter denoted as R , is also presented in Table 2. Only approximately one-fifth to one-tenth of the increase in wall shear stress is reflected in the pressure drop, and R remains within the range of 0.10–0.21 under all conditions.

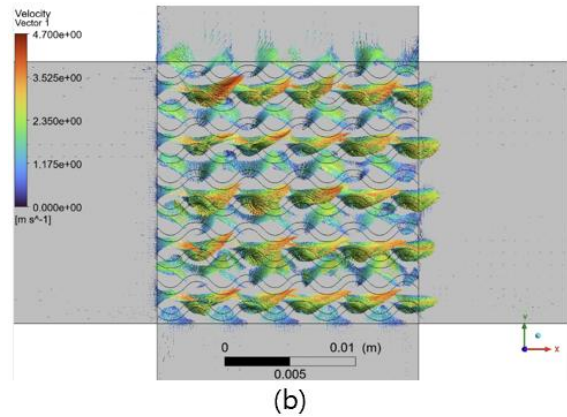
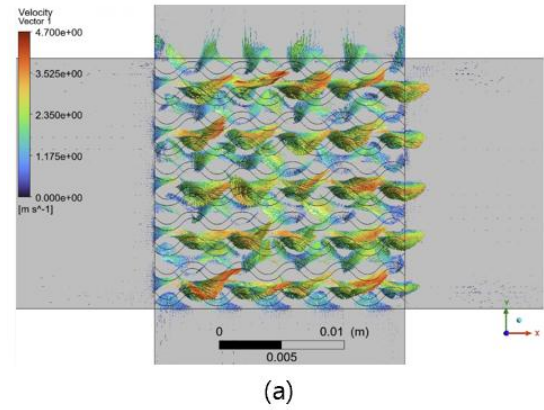
The flow-direction force acting on the gyroid partition wall was decomposed into viscous and pressure components, F_f and F_p , respectively. Both components were directly integrated over the wall by the solver and therefore do not require geometric assumptions regarding the wetted area or channel cross-sectional area, as summarized in Table 3. Without roughness, the friction component accounted for 6.6% and 6.0% of the total force on the cold and hot sides, respectively. At $K_s = 346 \mu\text{m}$, these fractions increased only to 11.8% and 11.1%, respectively. Thus, approximately 90% of the flow-direction force acting on the gyroid partition wall in this structure is attributable to form drag arising from the curvature of the flow path. As shown in Figs. 6(a) and 6(b), the roughness condition has little effect on the overall flow structure. Consequently, the increase in wall

shear stress caused by roughness is only weakly reflected in the pressure drop.

Figures 6(c) and 6(d) show that the turbulent kinetic energy increases near the wall, consistent with the increase in wall shear stress. Thus, the roughness condition primarily affects the near-wall flow region.

Table 3. Decomposition of the flow-direction force acting on the gyroid partition wall (500 kg/h). F_f is the viscous component, F_p the pressure component, and F_{total} the sum of the two.

Quantity	Cold side		Hot side	
	$K_s = 0 \mu\text{m}$	$K_s = 346 \mu\text{m}$	$K_s = 0 \mu\text{m}$	$K_s = 346 \mu\text{m}$
F_f (N)	0.177	0.350	0.134	0.271
F_p (N)	2.505	2.620	2.100	2.172
F_{total} (N)	2.681	2.970	2.233	2.443
F_f / F_{total} (%)	6.6	11.8	6.0	11.1



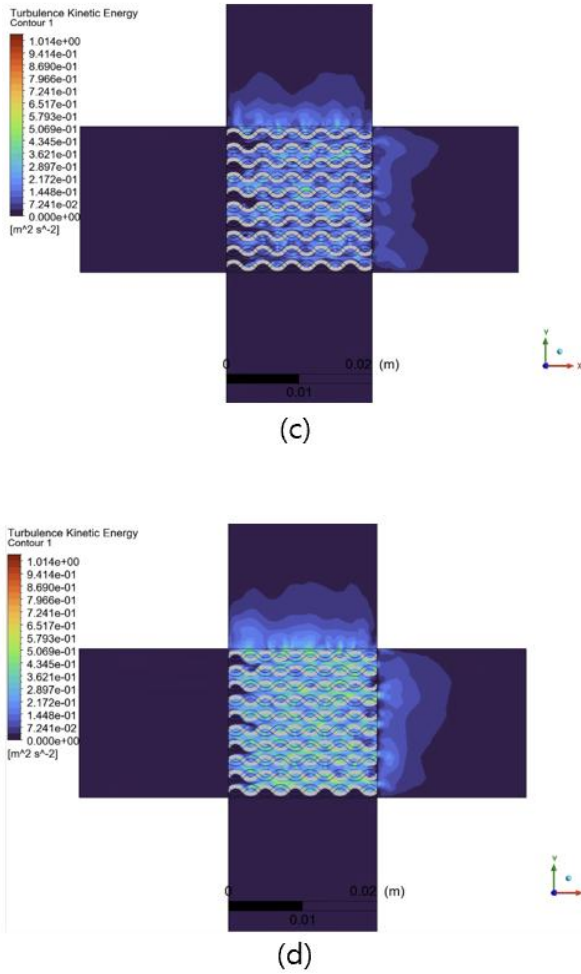


Fig. 6. Comparison at the $z = 10 \text{ mm}$ cross-section, 500 kg/h . (a) velocity vectors (no roughness), (b) velocity vectors ($K_s = 346 \mu\text{m}$), (c) turbulence kinetic energy (no roughness), (d) turbulence kinetic energy ($K_s = 346 \mu\text{m}$). Both conditions use the same color range (velocity 0–4.7 m/s, TKE 0–1.014 m^2/s^2) and the same vector scale.

3.5 Roughness Magnification Sensitivity Analysis

Because the Adams correlation may not adequately represent additively manufactured surfaces and may therefore underestimate K_s , a sensitivity analysis was performed using magnified roughness heights of $K_s = 700 \mu\text{m}$ and $1400 \mu\text{m}$, corresponding to approximately two and four times the estimated value, respectively. The purpose was to determine whether the discrepancy could be attributed to an underestimation of K_s or instead to a fundamental limitation of the wall-friction correction approach, as summarized in Table 2. Even when the wall shear stress increased to approximately three times its value in the roughness-free case, the increase in cold-side pressure drop remained less than half of the experimental deviation. The discrepancy therefore cannot be explained by an underestimation of the equivalent roughness, but arises from a structural limitation of the correction approach.

Matching the experimental pressure drop would require a K_s value exceeding the hydraulic diameter (1.56

mm), which is physically unrealistic. Moreover, the overprediction of heat transfer becomes more pronounced as K_s increases. As shown in Fig. 5, no single K_s value simultaneously improves the predictions of both pressure drop and heat transfer.

3.6 Interpretation and Design Implications

The results presented above are more appropriately interpreted as a mismatch between the modeling approach and its application target rather than as a flaw in the model itself. The Nikuradse–Colebrook family of equivalent sand-grain roughness concepts was developed primarily for pipe flow, where wall friction accounts for a substantial portion of the total pressure loss. In a form-drag-dominated structure, where this premise does not hold, the same wall-shear correction has only a limited effect on the total pressure drop. In particular, the channel-blockage effect, whereby surface protrusions reduce the effective flow cross-sectional area and thereby amplify the form drag of the flow path itself, is not captured by the ESGR approach. The ESGR approach instead increases the wall shear stress while leaving the underlying smooth geometry unchanged. This interpretation is also consistent with the observations reported in [2]. In that study, the measured pressure drop of the polished specimen agreed with the roughness-free CFD prediction within 2–4%, despite the absence of a roughness condition in the CFD analysis both before and after polishing. During the polishing process, the residual areal roughness, S_a , decreased by at most 47%, while the density of the surface protrusion microstructure decreased substantially. This suggests that the dominant source of the pressure-drop deviation is channel blockage caused by surface protrusions rather than the increase in wall friction associated with surface roughness.

In addition, because the ESGR approach represents the effect of roughness primarily through an increase in wall shear stress, the predicted heat transfer tends to increase correspondingly. Consequently, the predictions of wall shear stress and heat transfer are closely coupled. The heat-transfer overprediction observed in Section 3.3 (+13.7%) appears to be partly attributable to the turbulent Prandtl number correction described in Section 2.3.2 not being applied. The 11–14% correction reported in [5] was applied to the convective heat transfer coefficient. Because the wall conduction resistance contributes to the overall heat transfer coefficient in series and is unaffected by this correction, its effect on the overall heat transfer coefficient, U , is expected to be smaller than 11–14%.

If an AM-fabricated gyroid heat exchanger is designed using roughness-free CFD, the pressure drop may be substantially underpredicted. Moreover, even when the surface roughness reported in the literature is incorporated through the ESGR approach, most of this underprediction remains unresolved. For compact heat exchangers, an error of this magnitude may be difficult to accommodate within conventional design margins for pump power and flow distribution.

Three limitations should be considered when interpreting these results. First, the present operating conditions fall outside the range over which the ESGR framework has been extensively validated. The relative roughness, k_s/D_h , is 0.22, which exceeds the value of approximately 0.067 in the Nikuradse experiments and approximately 0.05 associated with the Moody chart. In addition, the bulk Reynolds numbers, approximately 1,400 on the cold side and 2,900 on the hot side, are substantially lower than the high-Reynolds-number, fully turbulent conditions on which the framework was originally developed. Second, although the roughness data adopted in this study were measured on the downward-facing side of the gyroid surface, they were applied uniformly over the entire wall. This assumption likely overestimates the mean roughness and therefore makes the inability of the ESGR approach to reproduce the experimental pressure drop a conservative result. Third, the near-wall resolutions of $y^+ \approx 2.8$ on the cold side and 4.7 on the hot side do not satisfy the $y^+ < 1$ requirement associated with the wall boundary condition used in the present analysis.

4. Conclusions

Converting the surface roughness reported in the literature into equivalent sand-grain roughness, incorporating it into the CFD analysis of a dual-channel gyroid heat exchanger, and directly comparing the results with experimental data led to the following conclusions.

(1) Relative to the roughness-free CFD predictions, the experimental values were approximately 80.6% higher for pressure drop and 14.9% higher for the overall heat transfer coefficient.

(2) Under the literature-based roughness conditions, the wall shear stress increased by 61–86%, whereas the pressure drop increased by only 8–10%, accounting for only approximately one-eighth of the experimental deviation. Even when K_s was increased fourfold and the wall shear stress increased by up to 179%, the cold-side pressure-drop increase reached only +37.6%, remaining less than half of the experimental deviation. Meanwhile, the predicted increase in heat transfer was approximately twice that observed experimentally.

(3) The ratio of the relative increase in pressure drop to the relative increase in wall shear stress remained within 0.10–0.21 even when K_s was increased fourfold. This behavior is attributed to the structural characteristics of the flow path, in which approximately 90% of the flow-direction force acting on the gyroid partition wall is attributable to form drag.

(4) Therefore, for form-drag-dominated TPMS structures, the application of equivalent sand-grain roughness alone is insufficient to reproduce the effects of AM-induced surface roughness. Because no single K_s value simultaneously reproduces both the pressure drop and overall heat transfer coefficient, additional modeling of mechanisms that increase form drag, such as channel

blockage, is required. A separate correction for rough-wall heat transfer is also necessary to accurately predict the overall heat transfer coefficient.

(5) Future work will implement the turbulent Prandtl number correction as a user-defined function and refine the near-wall mesh to achieve $y^+ < 1$. Further work will also focus on separately modeling the channel-blockage effect, assessing the applicability of the roughness heat-transfer correction under high-Prandtl-number conditions using water as the working fluid, and developing a correlation specifically for gyroid geometries.

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