

Article

Geometric Optimization of Corrugated Channels for Heat Transfer Enhancement Using Field Synergy and Response Surface Methodology

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Abstract

This study presents a numerical investigation of turbulent flow and heat transfer in corrugated channels, focusing on the effects of key geometric parameters on thermal–hydraulic performance. The corrugation height-to-channel height ratio (C/H), the length ratio (L_1/L_2), and the expansion angle (θ) were systematically varied, and simulations were performed for Reynolds numbers between 4×10^3 and 1×10^4 using water as the working fluid and SST $k-\omega$ turbulence model. Response Surface Methodology (RSM) was applied to develop predictive models for the Nusselt number (Nu), friction factor (f), and thermal performance index (η). The results indicate that C/H is the dominant geometric parameter controlling both heat transfer and flow resistance. Increasing C/H from 0.10 to 1.00 results in a reduction in Nu of approximately 20–22%, while the friction factor decreases by about 40–45% over the investigated Reynolds number range, revealing a clear thermal–hydraulic trade-off. In contrast, variations in L_1/L_2 (0.5–6.0) and θ (5–30°) have a relatively weak influence, typically causing changes in Nu and f below 5–7%. The thermal performance index remains consistently above unity for all configurations and varies within a narrow range ($\eta \approx 1.00$ –1.16). The maximum thermal enhancement of approximately 10–15% is achieved at lower C/H values, particularly at low Reynolds numbers, whereas higher C/H values favor reduced pressure losses. Overall, the findings quantitatively demonstrate that corrugation height governs the thermal–hydraulic behavior of corrugated channels, while L_1/L_2 and θ provide design flexibility with minimal performance penalty.

Keywords: corrugated channel; thermal performance index; response surface methodology (RSM); field synergy analysis; heat transfer enhancement



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1. Introduction

Due to the climate change caused by fossil fuels, environmental damage is continuously increasing [1]. In parallel with the increasing use of renewable energy sources, enhancing the efficiency of energy systems has become one of the primary focuses for engineers and researchers [2,3]. Due to the increasing environmental impact associated with fossil fuel consumption, climate change has become a critical global concern [1]. In parallel with the growing integration of renewable energy sources, improving the efficiency of energy systems has emerged as a primary focus for engineers and researchers [2,3]. Against this background, enhancing heat transfer performance, while minimizing energy

losses in thermal systems, is of great importance for sustainable engineering applications. Heat exchangers play a key role in energy conversion and thermal management processes, and their performance directly affects overall system efficiency. In order to enhance heat transfer between fluids and solid surfaces while limiting additional pressure losses, passive control methods based on geometric modifications have been widely investigated [4]. In particular, corrugated channel designs aim to increase convective heat transfer by modifying flow structures, promoting fluid mixing, and disturbing thermal boundary layers, thereby improving the thermal performance index (η). Among various passive enhancement techniques, corrugated channel geometries have been extensively studied and demonstrated to substantially alter flow behavior and enhance thermal performance. Previous studies have shown that, cylindrical corrugated channels and dimpled surfaces can generate vortex structures, intensify mixing, and improve heat transfer by thinning the thermal boundary layer [5]. These geometric strategies contribute to enhanced thermal performance and to more efficient energy utilization, supporting the development of sustainable heat exchanger designs. Consistent with this approach, Sahin [6] experimentally examined flow behavior in a dimpled channel using PIV, demonstrating how geometric features affect local dynamics. In addition to surface topology, geometric phase shift (φ) has been shown to influence flow patterns and heat-transfer behavior. Selecting wall geometries that boost mixing while minimizing pressure loss is therefore essential for improving thermal efficiency. Similarly, wavy channels are widely used as passive enhancements and have shown significant thermal benefits over suitable Reynolds number (Re) ranges. According to the evaluations of Zontul et al. [7] the selection of optimal waviness geometry in heat exchangers is vital for maximizing performance. This is because the waviness dimensions strongly affect flow characteristics and static pressure variations, thereby influencing the heat transfer rate. Their experimental and numerical study defined the aspect ratio (n) as the ratio of groove depth (L) to groove length (L_g), with the highest thermal performance index obtained at an aspect ratio of $n = 0.2$. Furthermore, nanofluid flows in corrugated ducts have also been shown to benefit from geometric enhancement strategies [8]. These corrugated channel configurations are frequently preferred in compact heat-transfer devices, due to their enhanced thermal performance, compactness, and ease of manufacture. As a result, they are widely employed in plate-type heat exchangers, radiators, air- and water-cooled towers, finned-tube systems, and other compact thermal systems in energy-intensive applications [9]. In addition to geometric passive enhancement techniques, another effective method for heat transfer augmentation involves introducing pulsation into the flow. A comprehensive review by Qianhao et al. [10] underscored its potential in diverse systems. For example, Kurtulmuş and Sahin [11], reported a peak thermal performance index of $\eta = 1.5$ at $Re = 4 \times 10^3$ and $St = 1.03$ in sinusoidal channels. Similarly, Zontul and Sahin [12] found that sinusoidal pulsation at $f = 1.5$ Hz enhanced η by ~40% for laminar flow ($Re = 2000$). Hamzah and Sahin [13] using nanofluids, observed up to 31% heat transfer improvement under pulsating flow at $\varphi = 180^\circ$, $f = 4$ Hz, and $Re = 7 \times 10^3$. Additional studies have also explored passive enhancement using triangular rods [14], wavy geometries [15–17] and magnetic field-assisted nanofluid flows [18–23].

For complementing aforementioned passive control techniques, the field synergy principle has been employed as a theoretical and diagnostic framework to better understand convective mechanisms. Field synergy emphasizes the alignment between the velocity vector and the temperature gradient, which significantly improves the convective heat transfer coefficient by reducing thermal resistance [24,25]. Fundamentally, field synergy reflects the degree of coordination between the fluid's velocity field and its temperature gradient. A reduced angle between these two vectors enables a more efficient and direct transfer of thermal energy from hotter to cooler regions, thereby strengthening convective

heat transfer performance [25]. As such, the field synergy principle seamlessly integrates theoretical foundations with practical applications in fluid mechanics and thermal sciences, rendering it particularly significant in the design and optimization of thermal systems [26]. Its role becomes even more critical when strategic modifications in channel geometry or flow path are applied to enhance system performance under laminar and turbulent regimes [27,28]. A prominent application of field synergy lies in the optimization of thermal system performance. Engineers employ synergy analysis to assess existing thermal conditions and determine design modifications such as altering channel geometry or redirecting flow that can enhance heat transfer efficiency without incurring excessive pressure losses [29]. Several numerical [30–33] and experimental studies [34,35] have confirmed that minimizing the angle between the velocity vector and the temperature gradient enhances heat transfer rates, providing a robust framework for designing more effective thermal management systems. In this context, combining physical interpretation tools such as field synergy analysis with statistical optimization techniques offers a comprehensive framework for both understanding and optimizing thermal–hydraulic performance. Complementing the insights derived from field synergy, recent studies have emphasized the role of Response Surface Methodology (RSM) in thermal optimization, particularly in electronics cooling, renewable energy, and industrial heat exchangers. For instance, in intelligent power modules (IPM), optimizing pipe layout reduced thermal resistance, albeit at a pressure loss cost, and RSM enabled multi-objective optimization [36]. Moreover, RSM has gained widespread acceptance for its effectiveness in analyzing multidimensional parametric influences. It employs statistical and mathematical techniques to model the relationship between input variables and desired responses, offering accurate predictions, while minimizing computational effort. This methodology has been successfully applied in various applications including fin design [37–39], elliptical tube geometry [40], plate-fin arrangements [41], and vortex generator integration [42]. Khalajzadeh et al. [43] and Han et al. [44] applied RSM to assess the combined heat transfer and resistance behavior in corrugated tubes, offering insights into overall thermal performance. These studies clearly demonstrate the effectiveness of RSM as a predictive and optimization tool for improving thermal performance and reducing pressure losses in heat transfer systems. However, most existing RSM-based investigations primarily focus on global performance indicators, such as the Nusselt number (Nu) and friction factor, without providing sufficient physical explanation of the underlying flow and heat-transfer mechanisms.

Despite the extensive literature on corrugated channels and RSM-based optimization, most previous studies have primarily focused on global performance indicators such as the Nu and friction factor, without explicitly linking geometry-induced local flow mechanisms to overall thermal–hydraulic performance. In parallel, while the field synergy principle has been widely used to interpret convective heat transfer enhancement, its application to corrugated channel geometries has remained largely descriptive, and its direct integration with geometry-driven optimization frameworks is still limited. Accordingly, the primary objective of the present study is to systematically investigate the thermal–hydraulic behavior of corrugated channels by quantitatively evaluating the effects of the corrugation height-to-channel height ratio (C/H), the length ratio (L_1/L_2), and the expansion angle (θ) over a Reynolds number range of 4×10^3 – 1×10^4 . To achieve this, turbulent flow and heat transfer are simulated using the SST k - ω turbulence model, while RSM is employed to construct predictive models for the Nu, friction factor, and thermal performance index. Furthermore, field synergy analysis is integrated to elucidate the underlying physical mechanisms by correlating local velocity–temperature gradient alignment with global performance metrics. The key contribution of this work lies in bridging geometry-induced local flow structures with global thermal–hydraulic performance through a combined RSM

and field synergy framework, thereby providing physically grounded and quantitatively supported design guidelines for optimizing corrugated channel configurations in compact heat exchanger applications.

2. Material and Method

The flow was assumed to be fully developed, and turbulent. Turbulence was modeled using the SST k – ω formulation due to its proven accuracy in predicting adverse pressure gradients, separation/reattachment zones, and strong near-wall effects typical of corrugated channels. Numerical simulations were performed in ANSYS Fluent 2025 R2 for Reynolds numbers between 4×10^3 and 1×10^4 , and the governing equations were solved using the finite volume method (FVM). The corrugated channel geometry was defined by three primary parameters: the corrugation height-to-channel height ratio (C/H), the length ratio (L_1/L_2), and the expansion angle (θ). For each parameter, five different values were analyzed, and the geometry was systematically varied to evaluate their influence on hydrodynamic and thermal behavior. These parameters particularly C/H , L_1/L_2 , and θ were selected because their effects on flow development, boundary-layer behavior, and heat-transfer enhancement are well documented in the literature. The expansion angle θ was varied between 5° and 30° to capture transitions from attached flow ($\theta \approx 10^\circ$) to regimes where separation and recirculation become more pronounced ($\theta \geq 20^\circ$), consistent with diffuser studies by Reneau et al. [45]. The narrow-section length L_1 governs fluid acceleration and boundary-layer thinning, whereas L_2 affects vortex residence and secondary-flow development, both of which play key roles in the overall thermal performance index (η). To ensure a fully developed inlet condition, the entry height was set to H with an upstream length of $10H$. The outlet region was similarly defined as $10H$ in length with a height of H , where H represents the minimum channel height. The corrugated configuration consists of a gradual expansion defined by θ , a straight extension of length L_2 , and a downstream contraction returning to the minimum channel height. Figure 1 illustrates the channel configuration, parameter definitions, and mesh structure (not to scale). A two-dimensional (2D) modeling approach was adopted in this study to maintain computational efficiency and to enable an extensive parametric investigation within the Response Surface Methodology framework. This approach allows the isolation of the dominant thermal–hydraulic mechanisms such as boundary layer disruption, flow separation–reattachment, and synergy-angle variation without introducing unnecessary three-dimensional end wall effects. Although three-dimensional corrugated channels may exhibit transverse and end-wall-induced secondary flows, previous studies have demonstrated that 2D simulations can reliably capture the essential heat-transfer behavior of corrugated and grooved channels with sufficient accuracy when spanwise variations are negligible [46–56]. Validation against Gnielinski and Blasius correlations confirmed that, the 2D model provides reliable predictions of Nu and f . Therefore, the 2D configuration was considered the most efficient and appropriate choice for isolating geometric effects and constructing accurate predictive models.

Table 1 summarizes the geometric specifications of the computational model, including the channel height (H), lengths of the straight and corrugated sections (L_1 and L_2), corrugation height (C), corrugation radius (R), and expansion angle (θ). These values were selected to represent a realistic corrugated channel configuration commonly used in engineering applications. The dimensions in Table 1 correspond to the baseline geometry; in subsequent analyses, these parameters were systematically varied within predefined ranges to evaluate their effects on thermal and hydraulic performance.

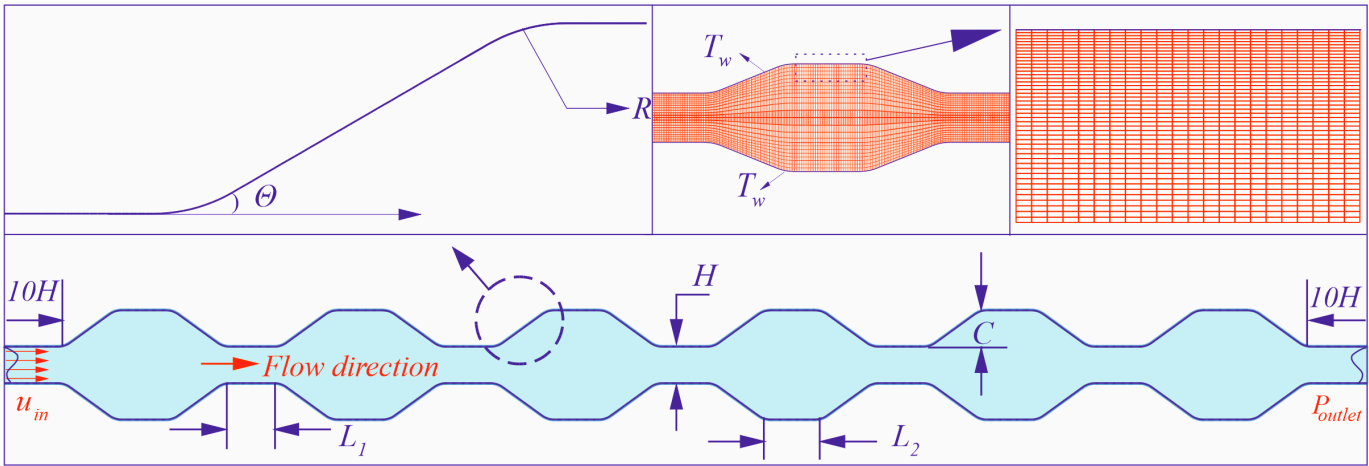


Figure 1. Schematic diagram of the corrugated channel.

Table 1. The dimensional specifications of the model.

Parameter	Value
H (channel height)	0.6 (mm)
L1 (length of corrugated)	3 (mm)
L2 (length of channel)	1.2 (mm)
C (corrugation height)	0.06 (mm)
R (radius of the corrugation)	0.14 (mm)
θ (expansion angle)	5°

Table 2 presents the thermophysical properties of water at 300 K, which were assumed constant to isolate the effects of geometric variations on flow and heat transfer. The governing equations were discretized using a second-order upwind scheme, and velocity–pressure coupling was handled using the SIMPLE algorithm [57], with a convergence criterion set at 10^{-8} .

Table 2. The thermophysical characteristics of the water at 300K.

Property	Symbol	Value
Density	ρ (kg/m ³)	996.5
Dynamic viscosity	μ (kg / m · s)	8.3×10^{-4}
Specific heat capacity	C_p (J/kg · K)	4071.6
Thermal conductivity	k (W/m · K)	0.61519

The computational study was conducted by systematically varying the key design parameters as summarized in Table 3. Reynolds number (Re) varied from 4×10^3 to 1×10^5 in increments of 1000. The corrugation height-to-channel height ratio (C/H), the length ratio (L_1/L_2), and the expansion angle (θ) were modified within specified ranges, each with constant incremental steps. This parametric setup ensured a comprehensive evaluation of the thermal and hydraulic responses across a wide spectrum of flow and geometrical conditions.

Table 3. Summary of the studied parameters, ranges, and increment intervals used in the simulations.

Variable	Range	Increment
Reynolds Number, Re	$4 \times 10^3 \leq \text{Re} \leq 10 \times 10^4$	$\Delta = 1000$
Corrugation height-to-channel height ratio, C/H	$0.1 \leq C/H \leq 1$	$\Delta = 0.1$
Length of corrugated to length of channel, (L_1/L_2)	$0.5 \leq L_1/L_2 \leq 6$	$\Delta = 0.5$
Expansion angle, θ	$3^\circ \leq \theta \leq 30^\circ$	$\Delta = 3^\circ$

2.1. Governing Equations

The governing equations for mass, energy, and momentum were applied assuming a two-dimensional (2D), incompressible, turbulent, and steady-state condition to calculate and analyze the relevant parameters. The related equations are provided below. Firstly, the continuity (1) and momentum (2) equations are given as follows:

$$\frac{\partial(\rho u_i)}{\partial x_i} = 0 \quad (1)$$

$$\frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_k}{\partial x_k} \right) - \rho \overline{u'_i u'_j} \right] \quad (2)$$

The following symbols, for example, P , μ , δ_{ij} , u' and $\overline{u'_i u'_j}$ presented in Equation (2) are the pressure, the dynamic viscosity, the Kronecker delta function, the velocity fluctuation, and the Reynolds stress. The required energy equation is also given below:

$$\frac{\partial}{\partial x_i} [u_i (\rho E + P)] = \frac{\partial}{\partial x_j} \left[\lambda + \left(\frac{C_p \mu_t}{Pr_t} \right) \frac{\partial T}{\partial x_j} + u_i (\tau_{ij})_{eff} \right] \quad (3)$$

Here, in Equation (3) symbols C_p , λ , T , Pr , t and E represent the specific heat (C_p), the thermal conductivity (λ), the temperature (T), the Prandtl number (Pr), the turbulence (t) and the total energy (E). Here, the equation of total energy is presented by Equation (4):

$$E = C_p T + \left(\frac{P}{\rho} \right) + \left(\frac{u^2}{2} \right) \quad (4)$$

The determination of the deviatoric tensor is presented by Equation (5):

$$(\tau_{ij})_{eff} = \mu_{eff} \left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j} \right) - \frac{2}{3} \mu_{eff} \frac{\partial u_k}{\partial x_k} \delta_{ij} \quad (5)$$

In this study, the SST k - ω turbulence model was selected because it reliably predicts flows with adverse pressure gradients, separation, and strong near-wall effects conditions naturally present in corrugated channel geometries. As shown by Menter [58] and Walker et al. [59], the model combines the free-stream robustness of k - ϵ with the near-wall accuracy of k - ω , enabling precise prediction of recirculation zones, vortex formation, and wall-bounded turbulence. Given that geometric contractions and expansions in corrugated channels intensify local separation and velocity gradients, the SST k - ω model has been widely and successfully used in similar ribbed and corrugated duct studies [60–66], making it the most suitable closure model for the present numerical analysis. The governing transport equations for this model are as follows [58]:

$$\frac{\partial(\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left(\Gamma_k \frac{\partial k}{\partial x_j} \right) + \tilde{G}_k - Y_k + S_k \quad (6)$$

$$\frac{\partial(\rho\omega u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left(\Gamma_\omega \frac{\partial \omega}{\partial x_j} \right) + G_\omega - Y_\omega + D_\omega + S_\omega \quad (7)$$

Here, in Equations (6) and (7) the following symbols such as k presents turbulent kinetic energy, Γ_k presents the effective diffusivity of k , Γ_ω symbolize the effective diffusivity of ω , and $Y_\omega = \rho\beta\omega^2$ and $Y_k = \rho\beta^*k\omega$ present the dissipation of k and ω due to turbulence with β and β^* model constants. S_k and S_ω present possible source terms.

$G_k = -\rho \overline{u'_i u'_j} \left(\frac{\partial u_i}{\partial x_j} \right)$: Production of turbulence kinetic energy, k due to the mean velocity gradients, G_ω is the generation of ω . On the other hand, $\tilde{G}_k$ and G_ω parameters are determined by

$$\tilde{G}_k = \min(G_k, 10\beta^*k\omega) \quad (8)$$

$$G_\omega = \frac{\alpha}{v_t} G_k \quad (9)$$

v_t : The kinematic turbulent viscosity is presented by v_t and β^* is the model constant. Thermal diffusivity (α) is stated as:

$$\alpha = \alpha_\infty \frac{\left(\alpha_0^* + \frac{Re_t}{R_\omega} \right)}{\left(1 + \frac{Re_t}{R_\omega} \right)} \quad (10)$$

$$Re_t = \frac{\rho k}{\mu \omega}, \quad R_\omega = 2.95 \text{ and } \alpha_0^* = \frac{\beta_i}{3}$$

The term α_∞ is represented by:

$$\alpha_\infty = F_1 \alpha_{\infty,1} + (1 - F) \alpha_{\infty,2} \quad (11)$$

The terms $\alpha_{\infty,1}$ and $\alpha_{\infty,2}$ are represented by:

$$\alpha_{\infty,1} = \frac{\beta_{i,1}}{\beta_\infty^*} - \frac{\kappa^2}{\sigma_{\omega,1} \sqrt{\beta_\infty^*}} \quad (12)$$

$$\alpha_{\infty,2} = \frac{\beta_{i,2}}{\beta_\infty^*} - \frac{\kappa^2}{\sigma_{\omega,1} \sqrt{\beta_\infty^*}} \quad (13)$$

$\kappa = 0.41$ and $\beta_i = 0.072$

σ : Turbulent Prandtl number is presented by σ and the terms Γ_k and Γ_ω are mentioned as

$$\Gamma_k = \mu + \frac{\mu_t}{\sigma_k} \quad (14)$$

$$\Gamma_\omega = \mu + \frac{\mu_t}{\sigma_\omega} \quad (15)$$

$$\sigma_k = \frac{1}{\frac{F_1}{\sigma_{k,1}} + \frac{(1-F_1)}{\sigma_{k,2}}} \Gamma_\omega = \mu + \frac{\mu_t}{\sigma_\omega} \quad (16)$$

$$\sigma_\omega = \frac{1}{\frac{F_1}{\sigma_{\omega,1}} + \frac{(1-F_1)}{\sigma_{\omega,2}}} \quad (17)$$

μ_t denoted the turbulent viscosity which is presented by

$$\mu_t = \alpha^* \frac{\rho k}{\omega} \quad (18)$$

The term α^* stands for the turbulent viscosity which is defined by

$$\alpha^* = \alpha_\infty^* \frac{\left(\alpha_0^* + \frac{Re_t}{R_k}\right)}{\left(1 + \frac{Re_t}{R_k}\right)} \quad (19)$$

$$Re_t = \frac{\rho k}{\mu \omega}, R_k = 6, \alpha_0^* = \frac{\beta_i}{3} \text{ and } \beta_i = 0.072$$

Here, $\alpha^* = \alpha_\infty^* = 1$ for high Re numbers. The blending equation, F_1 , is stated as:

$$F_1 = \tan\left(\Phi_1^4\right) \quad (20)$$

Φ_1 was specified by:

$$\Phi_1 = \min \left[\max \left(\frac{\sqrt{k}}{0.09\omega y}, \frac{500\mu}{\rho y^2 \omega} \right), \frac{4\rho k}{\sigma_{\omega,2} D_\omega^+ y^2} \right] \quad (21)$$

D_ω^+ : positive portion of the cross-diffusion term and is described by:

$$D_\omega^+ = \max \left[2\rho \frac{1}{\sigma_{\omega,2}} \frac{1}{\omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j}, 10^{-10} \right] \quad (22)$$

Instead of having a constant value, β_i is computed with the aid of the equation given below:

$$\beta_i = F_1 \beta_{i,1} + (1 - F_1) \beta_{i,2} \quad (23)$$

D_ω can be calculated by Equation (24).

$$D_\omega = 2(1 - F_1) \rho \sigma_{\omega,2} \frac{1}{\omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} \quad (24)$$

The model constants of the SST k- ω turbulence model can be found in detail in the original formulation by Menter [58].

2.2. Boundary Conditions

For the inlet section of the channel constant temperature and uniform velocity are applied $u = u_{in}$, $v = 0$, $T = T_{in}$ and for the channel walls No-slip condition and constant wall temperature are applied. So, $u = v = 0$, $T = T_w$ (for considered geometric section). For other walls, adiabatic boundary condition is used: $\frac{\partial T}{\partial y} = 0$ (for smooth walls). For channel outlet: Zero gradient boundary conditions were taken for $\frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial T}{\partial x} = 0$ and the pressure at the outlet was set as zero.

2.3. Definitions of Key Parameters

The simulation considered several key dimensionless parameters, including the Reynolds number (Re), friction factor (f), Nusselt number (Nu), and thermal performance index (η). These quantities were evaluated to clarify the fundamental flow and heat-transfer mechanisms. Re was used to capture the effect of flow regime on heat transfer and frictional losses, while the friction factor quantified viscous energy dissipation. The Nusselt number provided a measure of convective heat-transfer efficiency, and the thermal performance index assessed the balance between heat-transfer enhancement and the associated pressure penalty. Together, these analyses enabled a clearer understanding of the interaction between

flow behavior and thermal performance, supporting the optimization of channel designs for improved efficiency. The Reynolds number can be defined as in Equation (25):

$$Re = \frac{\rho \bar{u} D_h}{\mu} \text{ where } D_h = \frac{4A}{P} \quad (25)$$

The rate of heat propagation can be determined concerning the Nusselt number as:

$$\overline{Nu} = \frac{\bar{h} D_h}{k} \quad (26)$$

The average convective heat transfer coefficient, $\bar{h}$ is:

$$\bar{h} = \frac{q_{wall}}{(T_{wall} - T_m)} \quad (27)$$

Here, q_{wall} is the heat flux, T_{wall} is the wall temperature and T_m is the time-mean temperature. Simply, the time-mean temperature (T_m) is determined with Equation (28):

$$T_m = \frac{1}{\bar{u} A_c} \int u T dA_c \quad (28)$$

Here, the cross-sectional area of the channel is presented with A_c and the mean velocity ($\bar{u}$) calculated using Equation (29):

$$\bar{u} = \frac{1}{A_c} \int u dA_c \quad (29)$$

The thermal performance index presented with Equation (31) is a key parameter employed to evaluate the heat transfer efficiency of a system, representing a balance between the energy consumption and heat transfer capability of the system. Frictional losses are defined as energy losses that occur during the movement of the fluid in a pipe or channel, and these losses vary depending on parameters such as the fluid's viscosity, flow rate, and surface properties of the pipe. The friction factor is a coefficient employed to quantify these losses and is typically dependent on the average speed of the fluid. This coefficient plays a critical role in engineering calculations to estimate energy losses due to friction. The evaluation of frictional losses and the thermal performance index in tandem is imperative to enhance the energy efficiency of the system and optimize heat transfer performance.

$$f = \frac{\Delta P D_h}{2 \rho \bar{u}^2 L} \quad (30)$$

Furthermore, the thermal performance index, denoted by η , is indicative of the overall performance of the developed heat propagation elements. It is defined as the enhanced convective heat propagation rate (Nu/Nu_s) in relation to the ratio of friction factors (f/f_s) under conditions of identical pumping power.

$$\eta = \frac{\frac{Nu}{Nu_s}}{\left(\frac{f}{f_s}\right)^{\frac{1}{3}}} \quad (31)$$

where Nu_s is the smooth channel's Nusselt number, Nu is the grooved channel Nusselt number, f_s is the smooth channel friction factor, and f is the grooved channel friction factor. The angle between the velocity vector and gradients are shown in Figure 2 and these synergy angles can be calculated using Equations (32) and (33), which are given below.

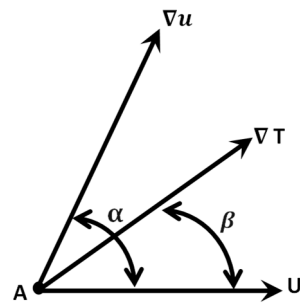


Figure 2. The angle between the velocity vector and gradients.

The angle between the velocity vector and velocity gradient (α) and the angle between the velocity vector and temperature gradient (β) [67].

$$\vec{U} \cdot \nabla T = |\vec{U}| |\nabla T| \cos \beta \quad (32)$$

Here at $\beta = 0^\circ$ the heat transfer becomes maximum.

$$\vec{U} \cdot \nabla u = |\vec{U}| |\nabla u| \cos \alpha \quad (33)$$

The synergy angles α and β can be calculated from numerical results using the following relations [68]:

$$\alpha = \arccos \left(\frac{\left| u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right|}{\sqrt{u^2 + v^2} \sqrt{\left(\frac{\partial u}{\partial x} \right)^2 + \left(\frac{\partial u}{\partial y} \right)^2}} \right) \quad (34)$$

$$\beta = \arccos \left(\frac{\left| u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right|}{\sqrt{u^2 + v^2} \sqrt{\left(\frac{\partial T}{\partial x} \right)^2 + \left(\frac{\partial T}{\partial y} \right)^2}} \right) \quad (35)$$

2.4. Response Surface Methodology (RSM)

Response Surface Methodology is a structured and efficient optimization technique widely used to model and improve processes where several design parameters affect one or more performance metrics. It employs a combination of statistical and mathematical methods to construct predictive models and explore complex parameter interactions in engineering applications. RSM is particularly valuable in numerical approaches such as computational fluid dynamics (CFD), where it enables modeling with reduced computational cost by minimizing the number of required simulations [69,70]. A common representation of the response surface is a second-order polynomial regression model, expressed as:

$$Y = b_0 + \sum_{i=1}^n b_i x_i + \sum_{i=1}^n b_{ii} x_i^2 + \sum_{i=1}^{n-1} \sum_{j=i+1}^n b_{ij} x_i x_j + \varepsilon \quad (36)$$

Here, Y is the predicted system response, x_i and x_j are the independent input variables, b_0, b_i, b_{ii}, b_{ij} are the regression coefficients, and ε represents the random error [8,27]. The second-order polynomial regression model is widely used in RSM applications because it provides a good balance between model accuracy and computational efficiency. Unlike linear models, it is capable of capturing curvature and interaction effects among design variables, which are common in nonlinear engineering systems. At the same time, its relatively simple mathematical structure allows efficient optimization and straightforward

physical interpretation of parameter effects, making it particularly suitable for CFD-based parametric studies. RSM is not only effective for identifying the influence of individual variables but also for revealing interaction effects and optimizing system behavior within a defined design space. In the present study, the ranges of the three independent geometric parameters were first defined, CFD simulations were then conducted within these ranges, and the resulting data were subsequently analyzed using second-order polynomial regression models within the RSM framework.

3. Validation

3.1. Validation of CFD Results for Parallel Plate Configuration

To validate the numerical simulation results, the Nusselt number (Nu) and friction factor (f) values obtained from CFD analyses for a smooth parallel-plate channel were compared with well-established empirical correlations from the literature specifically, Gnielinski [71] for Nu, and Moody [72] and Blasius [73] for f across a Reynolds number range of 4×10^3 to 10×10^3 , as illustrated in Figure 3. To validate the numerical model, the Nusselt numbers obtained from CFD were compared with the Gnielinski correlation. The deviations decrease with increasing Reynolds number, ranging from 28.15% at $Re = 4 \times 10^3$ to 4.94% at $Re = 10 \times 10^3$. The friction factor predictions also showed good agreement with reference correlations. When compared with the Moody diagram, the deviations remained within 3–7%, and when compared with the Blasius correlation, they were within 1.5–7%. These results confirm that the numerical model provides sufficiently accurate thermal and hydrodynamic predictions for the subsequent parametric and RSM analyses.

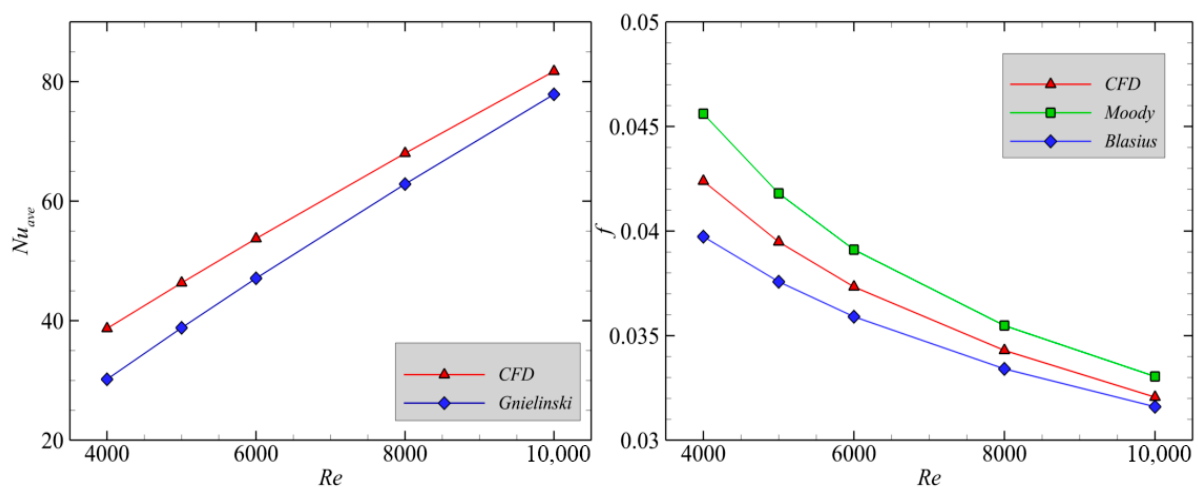


Figure 3. Comparison of Nusselt number (Nu) and friction factor (f) with correlations from literature.

3.2. Grid Size Independence Study

The mesh size and structure play a crucial role in accurately predicting flow and heat transfer characteristics. In the present study, a structured non-uniform mesh was generated throughout the computational domain. The grid was systematically refined near the corrugated walls to accurately resolve steep velocity and temperature gradients, while coarser elements were employed in the core flow region to reduce computational cost. To properly capture near-wall effects, boundary layer resolved prism type cells were incorporated along the channel walls in a structured manner. Rather than using a separately defined inflation block, the prism-layer resolution was achieved by gradually clustering quadrilateral cells normal to the wall, ensuring smooth cell growth and adequate near-wall resolution. This approach allows accurate representation of the viscous sublayer and thermal boundary layer, which is essential for SST k- ω -based simulations. Mesh quality

along the corrugated surfaces was evaluated using the dimensionless wall distance y^+ , defined as

$$y^+ = \frac{\rho u_\tau y}{\mu}$$

where y is the thickness of the first layer of the lattice, u_τ is the friction velocity, and μ is the dynamic viscosity. When $y^+ < 1$, the combination of Wilcox k - ε and SST k - ω turbulence models results in sensitively precise results. The results revealed that starting with the third grid, the values of y^+ were less than 1. The results showed $y^+ < 1$ starting from the third grid. Several grid resolutions were tested to ensure mesh independence. For $Re = 8 \times 10^3$ (Table 4), both the mean Nu and friction factor showed negligible changes beyond the third grid. Therefore, the third mesh configuration was adopted for all simulations.

Table 4. Grid size independence analyze for all cases.

Grid	$C/H = 0.5$			$L_1/L_2 = 3$			$\theta = 15^\circ$		
	Nodes	Nu	f	Nodes	Nu	f	Nodes	Nu	f
1	26,031	28.10	0.0255	19,581	32.2	0.0345	15,988	33.2	0.036
2	52,062	27.30	0.024	39,162	31.05	0.033	31,975	32.1	0.035
3	104,125	26.78	0.0239	78,325	30.31	0.0325	63,950	31.16	0.03423
4	208,250	26.75	0.0237	156,650	30.30	0.0324	127,900	31.10	0.0341
5	416,500	26.70	0.0235	313,300	30.29	0.0323	255,800	30.93	0.0340

4. Results and Discussions

4.1. Effect of Geometric Parameters on Temperature and Velocity Fields

This chapter presents a comprehensive investigation into the impact of the expansion angle, the length ratio, and the corrugation height-to-channel height ratio on the temperature and velocity distributions within a corrugated channel. Each geometric parameter was systematically varied while maintaining the remaining parameters constant in order to isolate its individual influence on the thermal and hydrodynamic behavior of the flow. Detailed contour plots are provided to visualize the spatial evolution of the temperature and velocity fields across the diverse configurations examined. The findings pertaining to each geometric variation are rigorously interpreted and discussed to facilitate an in-depth understanding of the flow dynamics and thermal transport mechanisms engendered by the channel modifications.

Figure 4 investigates the effect of the expansion angle, θ on the temperature and velocity distributions, considering $L_1/L_2 = 2.5$ and $C/H = 0.1$. As illustrated in Figure 4, the notation defines two-dimensional diffusers that expand and contract progressively, connected to channels of two different widths. Based on this notation, the axial lengths of the diffusers that connect two channels of widths C and H are represented as L . The expansion and contraction angles, commonly called included angles, are expressed as θ with the inlet height as C and the outlet height as H . The wall temperature was maintained at a constant 339 K, and temperatures were varied between 300 K and 339 K. At the same time, the time-averaged velocity range was adjusted between 0.5 m/s and 5.7 m/s using a color scale. Generally, a diffuser is characterized by three main parameters: the expansion angle θ , and the dimensionless numbers L/C and L/H . According to Reneau et al. [45], in a two-dimensional diffuser where $\theta = 10^\circ$, flow separation does not occur until $L/C = 6$. Under these conditions, the flow regime within the diffuser can be described as attached flow, where the velocity and pressure distributions are symmetric about the central axis. On the other hand, when $\theta = 20^\circ$ and 30° , the flow regime within the diffuser transitions to a large transitory stall. In this regime, the flow becomes erratic, characterized by significant

fluctuations in the entire flow pattern. Reverse flow structures are continually formed and dissipated within the diffuser, causing substantial pressure fluctuations throughout the system [74]. The turbulent nature of the flow naturally accelerates the mixing of the fluid's vortical flow structures, thereby enhancing the mass and momentum mixing process and expediting heat transfer. Also, pressure recovery is relatively high in flow regimes characterized by attached flow and transitory stall. When considering the overall geometry, it is well-established that pressure loss is lower compared to separated flow regimes.

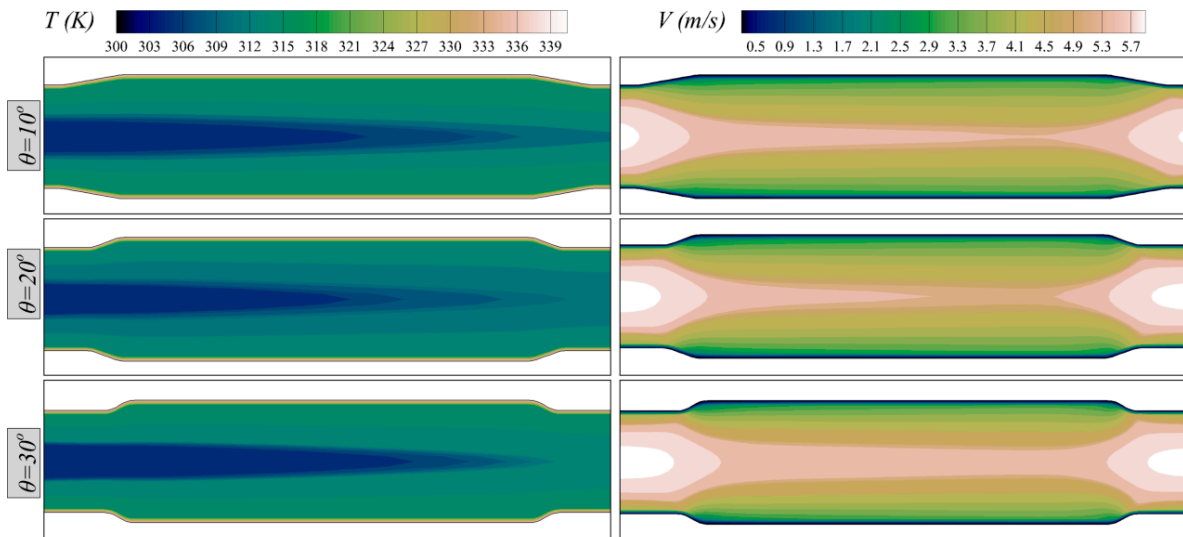


Figure 4. Temperature and velocity contours for different θ values at $Re = 8000$, $L_1/L_2 = 2.5$, and $C/H = 0.1$.

Figure 5 examines the effect of the length ratio (L_1/L_2) on the temperature and velocity distributions, considering $C/H = 0.1$ and $\theta = 5^\circ$. As depicted in Figure 5, the influence of the length ratio (L_1/L_2) on the thermal and velocity fields is investigated for a fixed corrugation height ratio ($C/H = 0.1$) and a fixed expansion angle ($\theta = 5^\circ$) under $Re = 8000$. When L_1/L_2 is small (e.g., $L_1/L_2 = 0.5$), the temperature distribution exhibits restricted thermal development along the channel centerline, and the velocity field indicates more frequent expansions and contractions. These abrupt geometric variations induce localized flow acceleration and deceleration, thereby enhancing turbulence and mixing in the proximity of the walls. As L_1/L_2 increases to intermediate values ($L_1/L_2 = 3$), a more structured thermal field develops with extended low-velocity core regions, indicating enhanced channeling of the main flow and a more uniform convective heat transfer behavior. For higher L_1/L_2 ratios ($L_1/L_2 = 6$), the channel geometry becomes more elongated, resulting in a further stabilization of the flow. In this case, velocity gradients are reduced, flow reattachment is enhanced, and thermal diffusion from the heated walls towards the centerline becomes more significant. Overall, an increase in L_1/L_2 improves flow uniformity but reduces secondary flow intensity, leading to a trade-off between the augmentation of thermal mixing and the magnitude of flow resistance.

Figure 6 illustrates the effect of varying the corrugation height-to-channel height ratio (C/H) on the temperature and velocity distributions at $Re = 8000$, with fixed $L_1/L_2 = 2.5$ and $\theta = 5^\circ$. As presented in Figure 6, elevating the corrugation height-to-channel height ratio (C/H) exerts a significant influence on both the thermal and hydraulic performance of the channel. For smaller C/H values, the temperature field remains relatively uniform with streamlined velocity distributions. However, as C/H increases, more pronounced expansions and contractions lead to augmented flow disturbances, enhanced convective mixing, and improved thermal boundary layer disruption. Although these effects enhance

heat transfer by promoting secondary flows and vortex generation, they also increase hydraulic losses through higher pressure drops and flow instability. Sudden expansions and contractions in corrugated channels are known to cause significant pressure penalties compared to smoother transitions. To alleviate these abrupt changes, diffusers and nozzles are used between narrow and wide sections, providing a more gradual flow transition, reducing turbulence generation, and improving flow uniformity. Incorporating such gradual transition elements helps enhance convective performance while balancing the trade-off between heat transfer enhancement and hydraulic resistance, leading to a more optimized channel design.

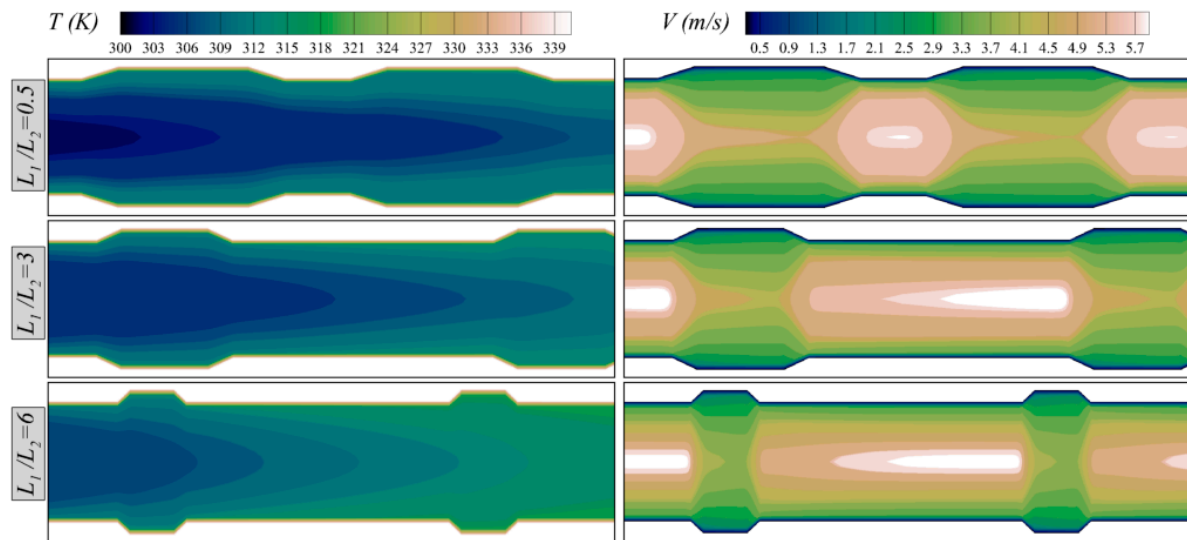


Figure 5. Temperature and velocity contours for different L_1/L_2 values at $Re = 8000$, $C/H = 0.1$, and $\theta = 5^\circ$.

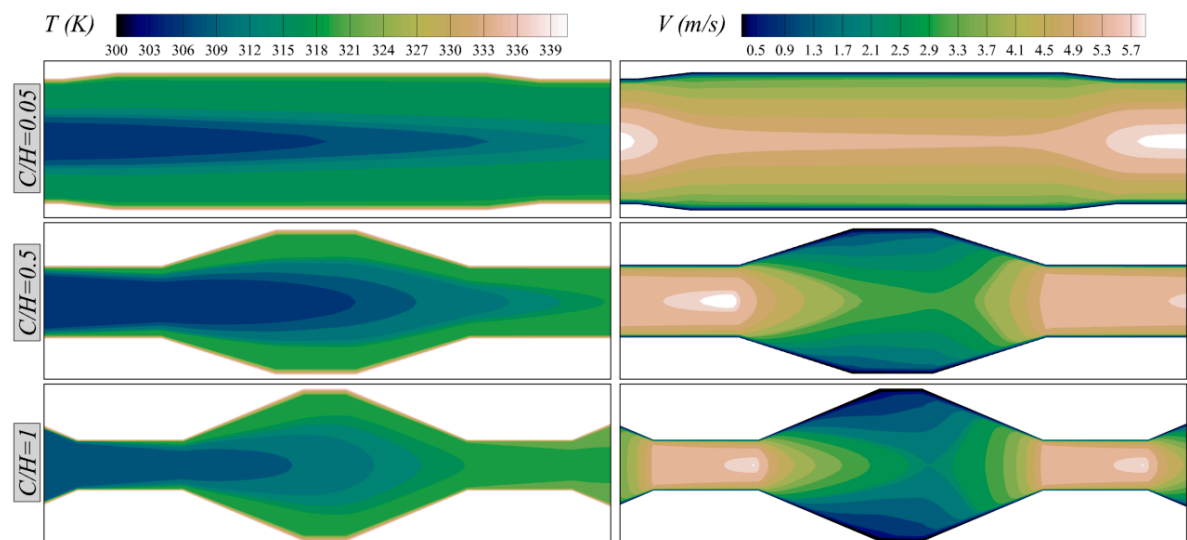


Figure 6. Temperature and velocity contours for different C/H values at $Re = 8000$, $L_1/L_2 = 2.5$, and $\theta = 5^\circ$.

4.2. Field Synergy Analysis of Thermal and Flow Structures Under Geometric Variations

In this section, the impact of key geometric parameters on local field synergy characteristics is systematically investigated. Specifically, the distributions of the field synergy angle α and β are analyzed to assess the alignment between velocity vectors and temperature gradients under varying corrugation height-to-channel height ratios (C/H), length

ratios (L_1/L_2), and expansion angles (θ). The aim is to reveal how geometric modifications influence convective heat transfer performance through the enhancement or deterioration of field synergy within the flow domain. Figures 7 and 8 presents the distribution of local field synergy angle (α) and (β) for different geometric configurations under a Reynolds number of $Re = 8000$. These figures systematically evaluate the impact of three key geometric parameters corrugation height-to-channel height ratio (C/H), length ratio (L_1/L_2), and expansion angle (θ) on the α distribution. The α angle characterizes the alignment between velocity and temperature gradient vectors, serving as a fundamental indicator of heat transfer efficiency. Lower α values (depicted in blue and green) indicate a stronger velocity-temperature gradient alignment, which enhances convective heat transfer by minimizing thermal resistance. In contrast, higher α values (shown in yellow and red) suggest weaker synergy, leading to reduced heat transfer efficiency. The progression of α values across different cases provides valuable insights into how geometric modifications influence thermal performance. Figure 7 illustrates the effect of geometric parameters on α distribution. In the first column, increasing the corrugation height-to-channel height ratio (C/H) improves field synergy. At $C/H = 0.05$, the α field is mostly uniform and orange-dominated, with minor low- α zones along the centerline, indicating suboptimal velocity-temperature alignment. As C/H increases to 0.4 and 0.7, green and blue regions emerge in the core, reflecting enhanced alignment and heat transfer. However, red zones near the inlet and outlet also intensify, signaling secondary flows that may reduce local efficiency. At $C/H = 1.0$, low α values persist in the core, but strong red bands near the walls suggest excessive turbulence and flow separation, potentially offsetting the gains. The second column examines the effect of length ratio (L_1/L_2). At $L_1/L_2 = 0.5$, the α field remains largely unchanged, dominated by orange with minor centerline green bands. As the ratio increases to 2.0 and 3.5, slight structuring occurs in the low- α core, but the improvements remain minimal. The third column examines the effect of expansion angle (θ). At $\theta = 7^\circ$, the α field is mostly uniform and orange-dominated. As θ increases to 10° , 20° , and 30° , green and blue regions emerge in the core, reflecting enhanced alignment and heat transfer. However, red zones near the inlet and outlet also intensify, signaling secondary flows that may reduce local efficiency.

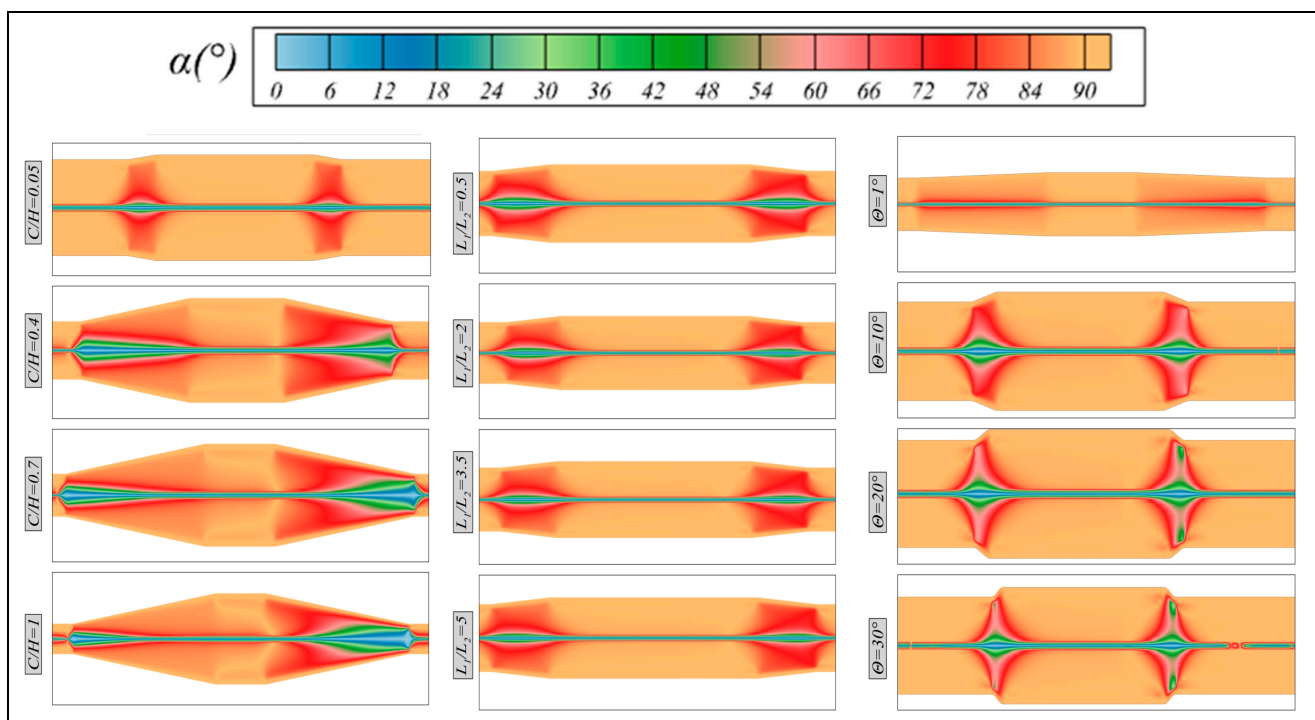


Figure 7. Distribution of field synergy angle α ($^\circ$) for various values of C/H , L_1/L_2 , and θ at $Re = 8000$.

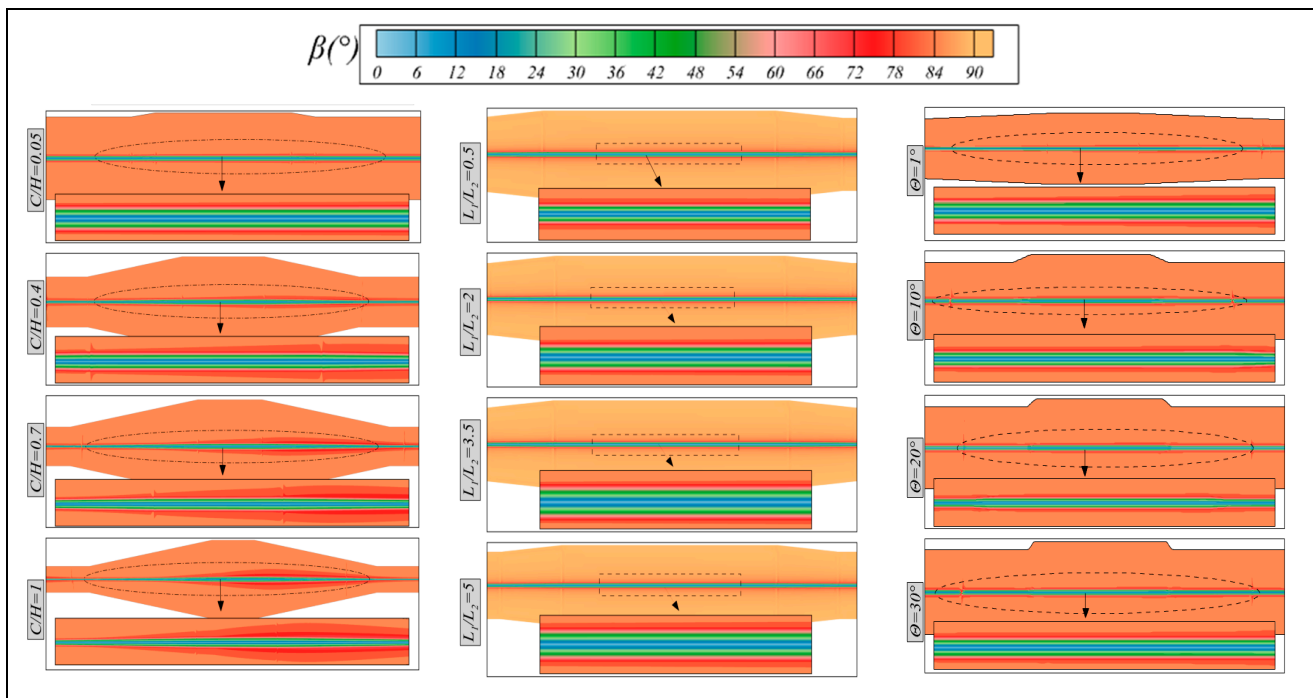


Figure 8. Distribution of field synergy angle β (°) for various values of C/H , L_1/L_2 , and θ at $Re = 8000$.

Even at $L_1/L_2 = 5.0$, only minor local changes are observed, indicating that length ratio adjustments alone do not significantly enhance synergy. The third column evaluates the influence of expansion angle (θ). At $\theta = 1^\circ$, the α field is highly uniform with limited synergy. Moderate angles (10 – 20°) introduce localized green bands along the centerline, suggesting slight improvements. However, high α values persist near the walls, limiting overall impact. At $\theta = 30^\circ$, red zones near expansion areas become more prominent, indicating greater flow separation and reduced alignment. Overall, α distributions reveal that C/H has the most significant impact on synergy, enhancing core alignment but potentially introducing turbulence near walls at high values. In contrast, L_1/L_2 and θ show only marginal, localized effects, making them secondary factors in heat transfer optimization.

The β angle, reflecting the alignment between velocity vectors and temperature gradients, is a key indicator of convective heat transfer performance. Lower β values (blue/green) indicate stronger synergy and improved heat transfer, while higher values (yellow/red) suggest weaker alignment and increased thermal resistance. Figure subfigures and magnified insets provide detailed views of these variations. In the first column of Figure 8, the effect of corrugation height-to-channel height ratio (C/H) on β distribution is examined.

At $C/H = 0.05$, the β field appears mostly uniform with orange-dominated zones, indicating moderate synergy. A narrow centerline band of lower β values suggests slight improvement, though near-wall regions remain unaffected. As C/H increases to 0.4 and 0.7, distinct green and blue zones emerge in the core, especially near recirculation areas, showing improved alignment. Insets confirm significant local β reductions. However, at $C/H = 1.0$, red zones intensify near the walls, indicating turbulence and flow separation that counteract the synergy benefits. The second column focuses on the length ratio (L_1/L_2). At $L_1/L_2 = 0.5$, the β distribution remains unchanged, with dominant orange areas and minimal variations. Increasing the ratio to 2.0 and 3.5 introduces weak green bands along the centerline, implying marginal improvement. Yet near-wall regions continue to exhibit high β values. Even at $L_1/L_2 = 5.0$, the overall distribution remains mostly unaffected, and insets confirm that length ratio alone has a limited impact on β alignment and heat

transfer. In the third column, the effect of expansion angle (θ) is assessed. At $\theta = 1^\circ$, the field remains stable and uniform, offering no significant enhancement. At 10° and 20° , localized green bands appear, mainly along the centreline, showing slight synergy gains. However, these effects do not extend to near-wall regions. At $\theta = 30^\circ$, red zones emerge, reflecting flow separation and decreased performance. Insets reveal that despite minor local benefits, θ has limited influence compared to C/H . Overall, among the three parameters, C/H exerts the most substantial effect on β distribution by enhancing core flow synergy, though excessive values introduce turbulence. Conversely, L_1/L_2 and θ produce only minor, localized changes. Thus, optimizing C/H is key, while L_1/L_2 and θ serve as secondary design parameters.

4.3. Thermal-Hydraulic Response Analysis Based on RSM Prediction

This section presents the predicted responses of the corrugated channel using Response Surface Methodology (RSM), a robust statistical technique for modelling complex interactions between design variables and performance metrics. The analysis focuses on the influence of three geometric parameters: (i) corrugation height-to-channel height ratio (C/H), (ii) length ratio (L_1/L_2), and (iii) expansion angle (θ). To isolate each parameter's individual effect on the Nusselt number (Nu), friction factor (f), and efficiency index (η), a parametric approach was applied, holding two variables constant while varying the third. For clarity, the results are presented row-wise across all figures: the first row (a) shows Nu variations, the second row (b) presents f , and the third row (c) displays η . This layout facilitates direct comparison of how each design parameter independently affects the key thermal-hydraulic responses.

For the length ratio (L_1/L_2), RSM-based simulations were conducted over a range from 0.5 to 6.0 in increments of 0.5. As illustrated in Figure 9, the resulting 3D response surfaces demonstrate how variations in Reynolds number (Re) and L_1/L_2 affect the system's thermal and hydraulic performance. The Nusselt number increases monotonically with both Re and L_1/L_2 , indicating that higher Reynolds numbers, reflecting stronger inertial forces, enhance convective heat transfer. Likewise, L_1/L_2 leads to more extended flow path interactions and heightened secondary motion. These phenomena promote thermal boundary layer disturbance and mixing, thereby yielding higher Nu values across the board. In contrast, the second surface reveals a nonlinear behaviour in the friction factor (f) with respect to L_1/L_2 and Re . While there is a general trend of increasing f with larger L_1/L_2 ratios, especially noticeable at lower Re , the irregular surface topology points to the presence of local flow separation and complex vortex dynamics introduced by the varying channel geometry. Although the influence of Re on f is somewhat subdued, it still contributes incrementally to frictional losses due to enhanced turbulence. The third plot provides a visual representation of the efficiency index which is a comprehensive metric that quantifies the trade-off between two conflicting processes such as the enhancement of heat transfer and the reduction in pressure. In conclusion, the results indicate that η increases with L_1/L_2 , particularly in lower Reynolds number conditions. This signifies that the geometric extension can substantially improve overall performance. However, as the length ratio continues to grow, the rate of η 's improvement slows down and the trend starts to level off, indicating diminishing marginal returns beyond a certain threshold, where the hydraulic penalties begin to offset thermal gains. Nevertheless, the consistent upward trend in η with Re supports the robustness of the geometric configuration across various flow regimes.

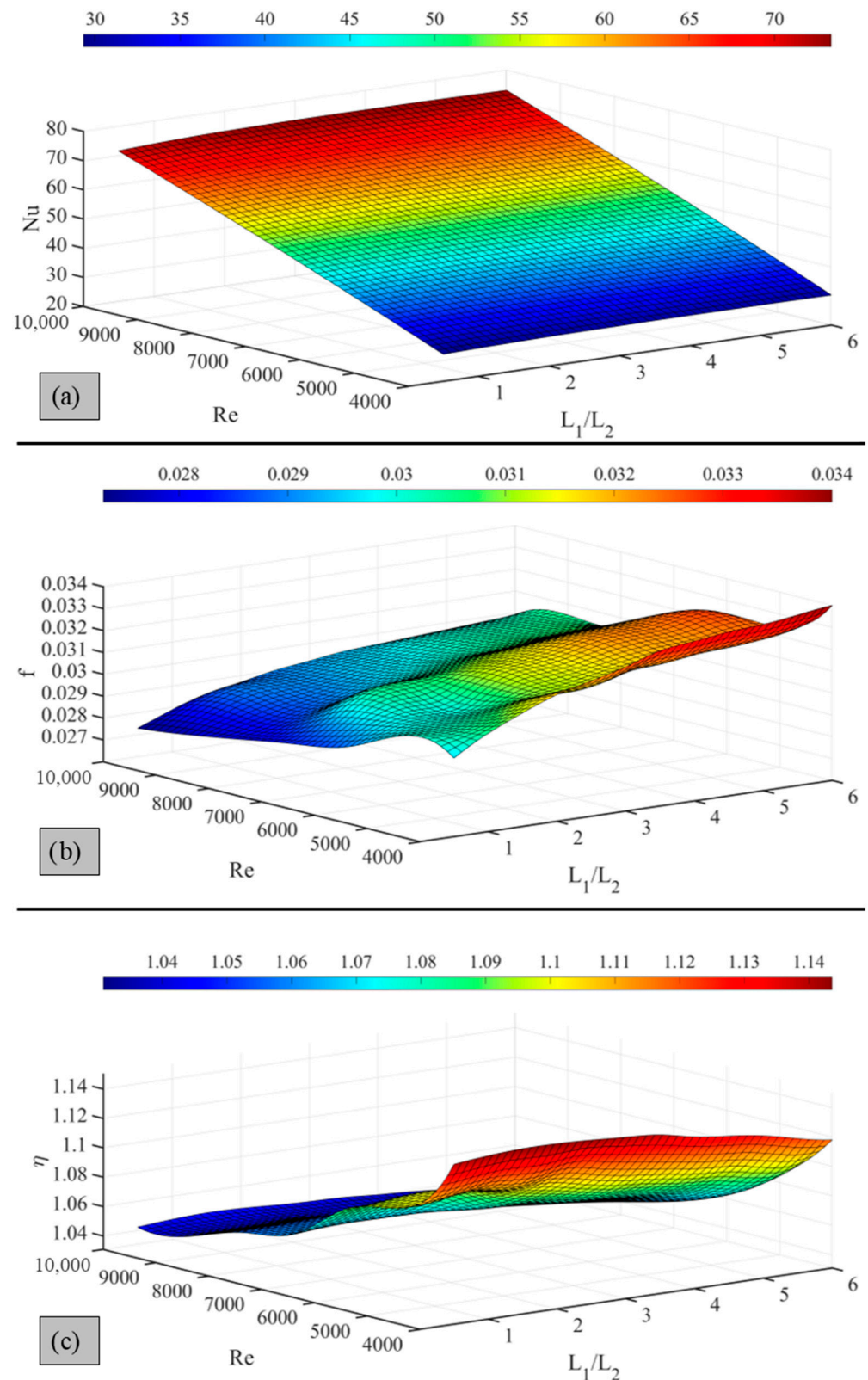


Figure 9. Variation in (a) Nu , (b) f , and (c) η with L_1/L_2 .

The influence of the corrugation height-to-channel height ratio (C/H) on the thermal and hydraulic performance of the corrugated channel was systematically analysed using RSM, as presented in Figure 10. To construct the response surfaces, the C/H ratio was varied from 0.03 to 0.3 in increments of 0.03, while other geometric parameters were held constant. This parametric variation enabled the generation of predictive models for key

thermal-fluidic performance indicators such as (i) the Nusselt number, (ii) the friction factor, and (iii) the efficiency index. The 3D surface plots clearly illustrate the effects of Reynolds number (Re) and C/H ratio on these performance measures. The Nu rises consistently with both Re and C/H, due to intensified boundary layer thinning and convective mixing. Additionally, a higher C/H ratio leads to enhanced turbulence and disruption near the channel walls, resulting in further improvement in heat transfer. Conversely, the friction factor shows a mild but consistent increase with increasing C/H, suggesting a balance, while deeper corrugations promote thermal enhancement, they also introduce elevated hydraulic resistance. The influence of Re on f is present but less pronounced in comparison. The efficiency index, combining both thermal and hydraulic effects, demonstrates an initial improvement with increasing C/H, especially under lower Reynolds number conditions. However, the response surface reveals that η eventually reaches a performance plateau beyond a certain C/H threshold, where further increases no longer yield proportionally significant gains. This underscores the importance of optimizing C/H to balance heat transfer benefits with pressure drop penalties.

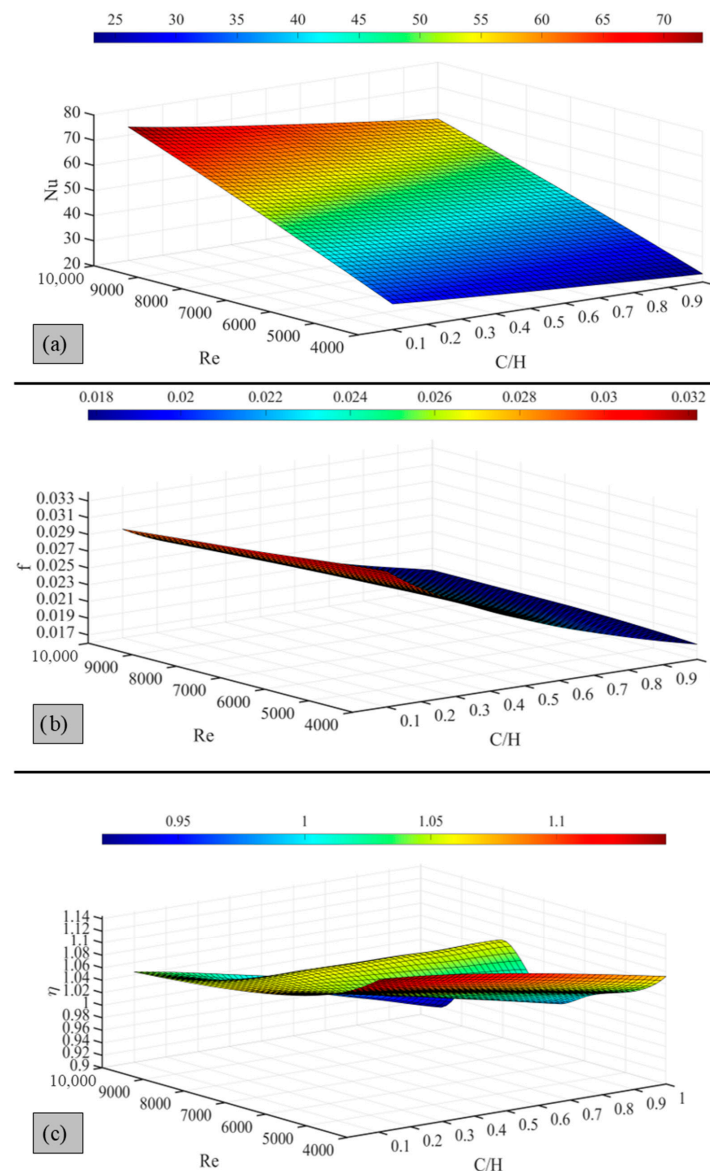


Figure 10. Variation in (a) Nu, (b) f , and (c) η with C/H.

In continuation of the parametric analysis conducted through Response Surface Methodology (RSM), the influence of the expansion angle (θ) on thermal and hydraulic performance was examined, as illustrated in Figure 11. To construct the response surfaces, θ was systematically varied from 0° to 30° in 1° increments while other geometric parameters were held constant. This fine-grained resolution enabled a detailed investigation of how subtle geometric changes impact the Nusselt number, the friction factor, and the efficiency index. The surface plot for Nu indicates a steady enhancement in heat transfer with increasing θ across all Reynolds numbers (Re). This trend suggests that larger expansion angles promote more vigorous flow separation and secondary motion, thereby intensifying convective heat transfer within the channel. A similar, though less dramatic, trend is observed for the friction factor, which also increases with higher θ . This behaviour is consistent with the greater flow disturbance and wall shear stress introduced by larger expansion angles, which lead to elevated pressure losses. The efficiency index, which reflects the combined effect of thermal enhancement and hydraulic penalty, demonstrates a clear increasing trend with θ . This result implies that the thermal gains achieved by increasing the expansion angle outweigh the corresponding rise in pressure drop, especially under low to moderate Reynolds number conditions. Unlike the C/H analysis, no evident asymptotic behaviour is observed in this case, suggesting a more linear benefit from increasing θ within the tested range. These findings underscore the capability of RSM to pinpoint design regions where performance parameters can be simultaneously optimized.

Overall, the RSM framework proved to be a highly effective tool in visualizing and optimizing the thermal and hydraulic performance of the corrugated channel under varying geometric configurations. The methodology enabled a comprehensive evaluation of the individual and combined effects of key design variables, namely the length ratio, the corrugation height-to-channel height ratio, and the expansion angle, on critical performance indicators including the Nusselt number, the friction factor, and the efficiency index. As demonstrated in Figures 8–10, the use of RSM not only facilitated the identification of performance-enhancing trends and geometrical thresholds but also revealed nonlinear interactions and asymptotic behaviours that would be challenging to capture through conventional trial-and-error methods. This structured approach supported a deeper understanding of trade-offs between heat transfer improvement and pressure loss penalty and ultimately contributed to informed design decisions for maximizing system efficiency in a broad range of thermo-fluid applications.

4.4. Effect of Geometric Parameters on Thermal-Hydraulic Performance

In this section, the influence of key geometric configurations on flow and heat transfer characteristics is systematically examined. The variations in the Nusselt number, friction factor, and thermal performance index are analysed with respect to the Reynolds number by independently modifying three geometric parameters: the length ratio, the corrugation height-to-channel height ratio, and the expansion angle.

Figure 12 presents the variation in the Nusselt number as a function of Reynolds number, the length ratio (L_1/L_2), the corrugation height-to-channel height ratio (C/H), and the expansion angle (θ). The results show a consistent increase in Nu with rising Reynolds number, confirming the expected behaviour for forced convection in corrugated channels. Variations in L_1/L_2 and θ within the studied ranges have only a modest influence on Nu: at a given Reynolds number, changing L_1/L_2 from 0.5 to 6 alters Nu by less than about 3–4%, while varying θ between 3° and 30° leads to changes typically below 7%. This indicates that boundary-layer development and velocity distribution are governed more strongly by Reynolds number and C/H than by L_1/L_2 or θ . In contrast, C/H exhibits a pronounced effect on heat transfer. A smaller C/H value (i.e., deeper corrugation and a narrower flow

passage) produces higher velocity gradients and enhances convective transport, whereas increasing C/H reduces the intensity of mixing and lowers Nu . The influence of θ is consistent with diffuser-flow literature (e.g., Reneau et al. [50]) where attached flow persists for moderate expansion angles when the expansion ratio is sufficiently large, preventing major separation in the present geometries. In summary, Figure 12 demonstrates that while C/H strongly modulates Nu , variations in L_1/L_2 and θ provide greater design flexibility without severely compromising heat transfer efficiency. For instance, at $Re = 8000$, Nu decreases from 59.53 to 46.49 as C/H increases from 0.10 to 1.00, corresponding to an approximately 22% reduction in heat-transfer performance. This clearly confirms the high sensitivity of Nu to the C/H ratio.

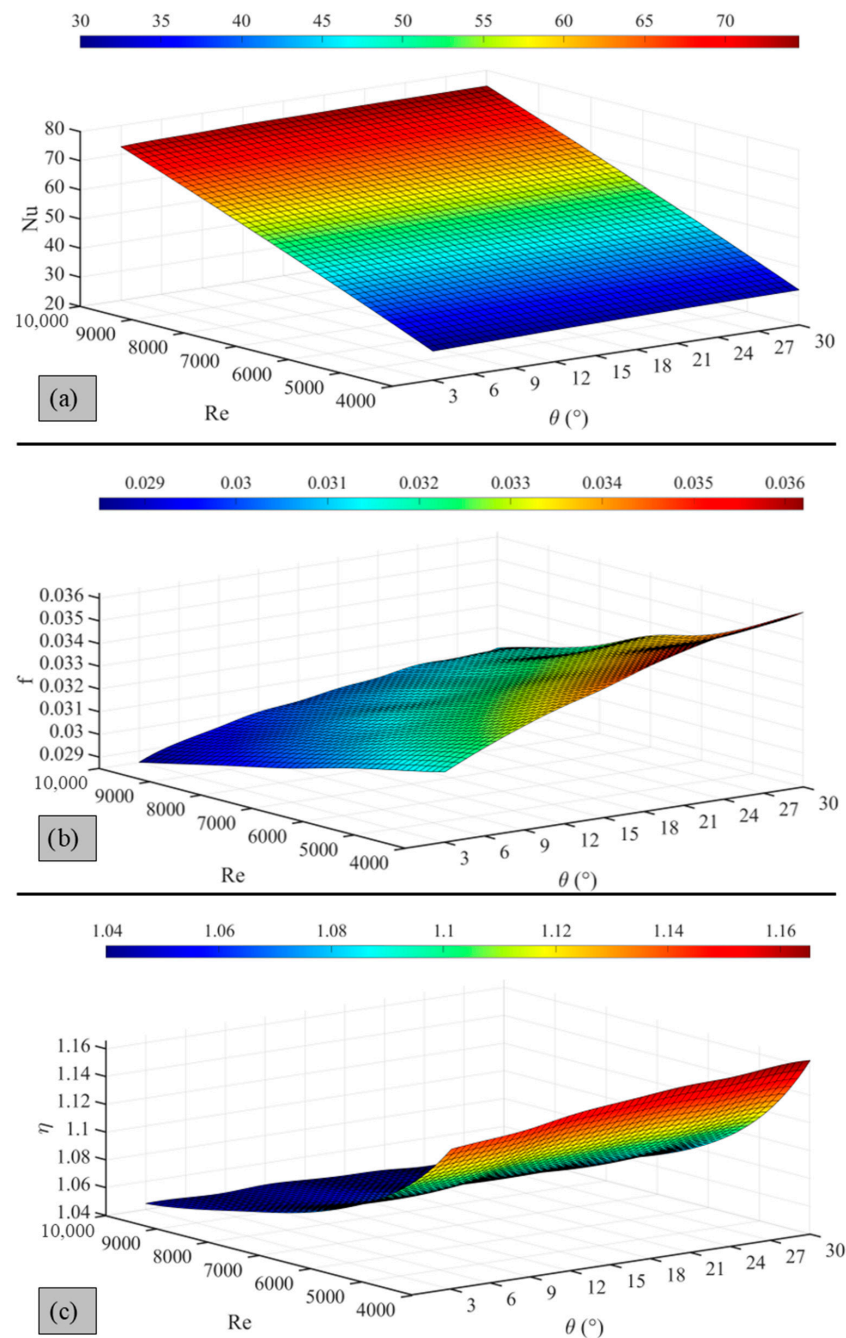


Figure 11. Variation in (a) Nu , (b) f , and (c) η with θ .

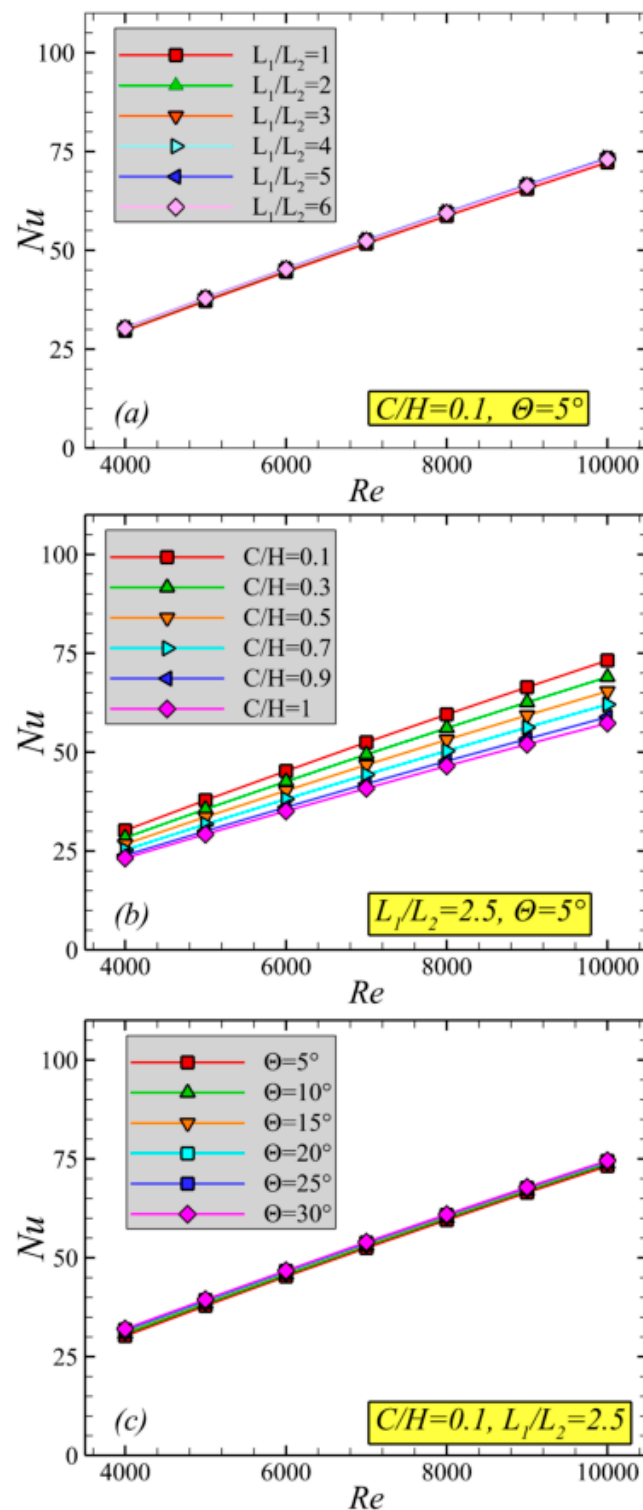


Figure 12. Variation in Nusselt number with Reynolds number: (a) For the ratio of L_1/L_2 within the range $1 \leq L_1/L_2 \leq 6$, (b) For the ratio of the, C/H $0.1 \leq C/H \leq 1$, (c) For the expansion angle, θ within the range $5^\circ \leq \theta \leq 30^\circ$.

Figure 13 demonstrates that the friction factor (f) decreases gradually as the Reynolds number increases from 4×10^3 to 1×10^4 , consistent with classical turbulent channel-flow behaviour. This occurs because increasing Reynolds number enhances momentum transport within the boundary layer, enabling the flow to overcome wall-induced resistance more effectively. Within the investigated geometric ranges, the length ratio (L_1/L_2) shows only a modest influence on f : at a given Reynolds number, changing L_1/L_2 from 0.5

to 6.0 increases the friction factor by approximately 10–14%, indicating a comparatively minor effect relative to other parameters. In contrast, (C/H) exerts a much stronger impact. A smaller C/H value (deeper corrugation) leads to higher flow blockage and increased wall shear, producing substantially larger friction factors. As C/H increases from 0.10 to 1.00, the friction factor decreases by approximately 40–45% across all Reynolds numbers—for example, at $Re = 4000$, f drops from 0.0322 to 0.0178. This pronounced reduction highlights the dominant role of C/H in controlling flow resistance. The expansion angle also affects friction, but to a lesser degree. Increasing θ from 3° to 30° results in a rise of approximately 10–15% in f , depending on Reynolds number. This moderate sensitivity aligns with diffuser-flow literature, where larger expansion angles increase pressure losses but do not necessarily induce severe flow separation within the studied geometric limits. Overall, these findings show that while L_1/L_2 and θ produce relatively small variations in friction factor, C/H remains the primary geometric parameter governing hydrodynamic resistance in corrugated channels.

Figure 14 examines the influence of the corrugated channel geometry—specifically the length ratio (L_1/L_2), the corrugation height-to-channel height ratio (C/H), and the expansion angle (θ) on the thermal performance index (η) across the Reynolds number range $4 \times 10^3 \leq Re \leq 1 \times 10^4$. Regardless of geometric configuration, η decreases slightly with increasing Reynolds number but remains consistently above unity, indicating that corrugation enhances thermal performance compared to a straight channel. The straight channel is implicitly represented by the reference line $\eta = 1$, which corresponds to the baseline straight channel performance. In the first row of Figure 14, the effect of L_1/L_2 is relatively small. Changing L_1/L_2 from 0.5 to 6 results in variations of only about 1–2% at low Reynolds numbers and less than 3% at higher Reynolds numbers. For example, at $Re = 4000$, η changes from 1.1398 ($L_1/L_2 = 0.5$) to 1.1279 ($L_1/L_2 = 6$), corresponding to a variation of roughly 1%. This indicates that the length ratio plays a secondary role in modifying thermal performance. In the second row of Figure 14, C/H shows the strongest influence on η . At $Re = 4000$, η decreases from 1.1434 at $C/H = 0.10$ to 1.0720 at $C/H = 1.00$, corresponding to a reduction of approximately 6.2%. At higher Reynolds numbers, however, the influence of θ becomes less pronounced, indicating that the positive effect of larger expansion angles diminishes as inertial effects dominate the flow. This behaviour suggests that θ has a measurable but limited impact on overall thermal performance, which is strongly dependent on the flow regime. In the third row of Figure 14, increasing the expansion angle θ from 3° to 30° produces a moderate enhancement in η particularly at low to moderate Reynolds numbers. For instance, at $Re = 4000$, η increases from 1.1408 ($\theta = 3^\circ$) to 1.1650 ($\theta = 30^\circ$), corresponding to a 2.1% improvement. Even at $Re = 10000$, the increase remains around 1%, indicating that θ has a measurable but limited effect on overall thermal performance.

Overall, η enhancements in this study fall in the range of 5–15%, primarily driven by C/H , while L_1/L_2 and θ yield smaller but consistent improvements. These behaviours align well with previous research on grooved and corrugated channels, including the synergy-angle analyses reported by Cai et al. [67] and Zhai et al. [68], which confirm that improved alignment between velocity and temperature gradients corresponds to enhanced thermal performance.

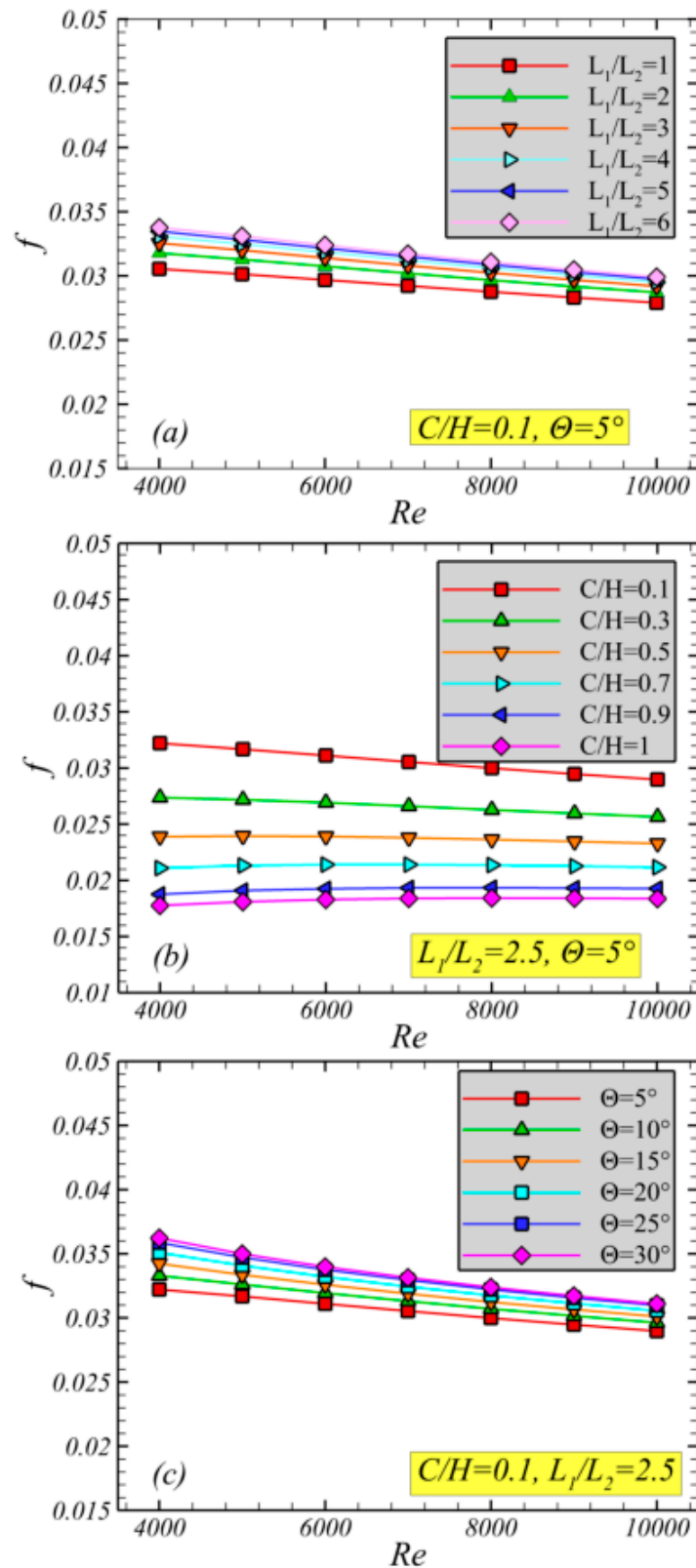


Figure 13. Variation in the friction factor (f) with Reynolds numbers: (a) For the ratio of L_1/L_2 within the range $1 \leq L_1/L_2 \leq 6$; (b) For the ratio of, C/H $0.1 \leq C/H \leq 1$; (c) For the Expansion angle, θ within the range $5^\circ \leq \theta \leq 30^\circ$.

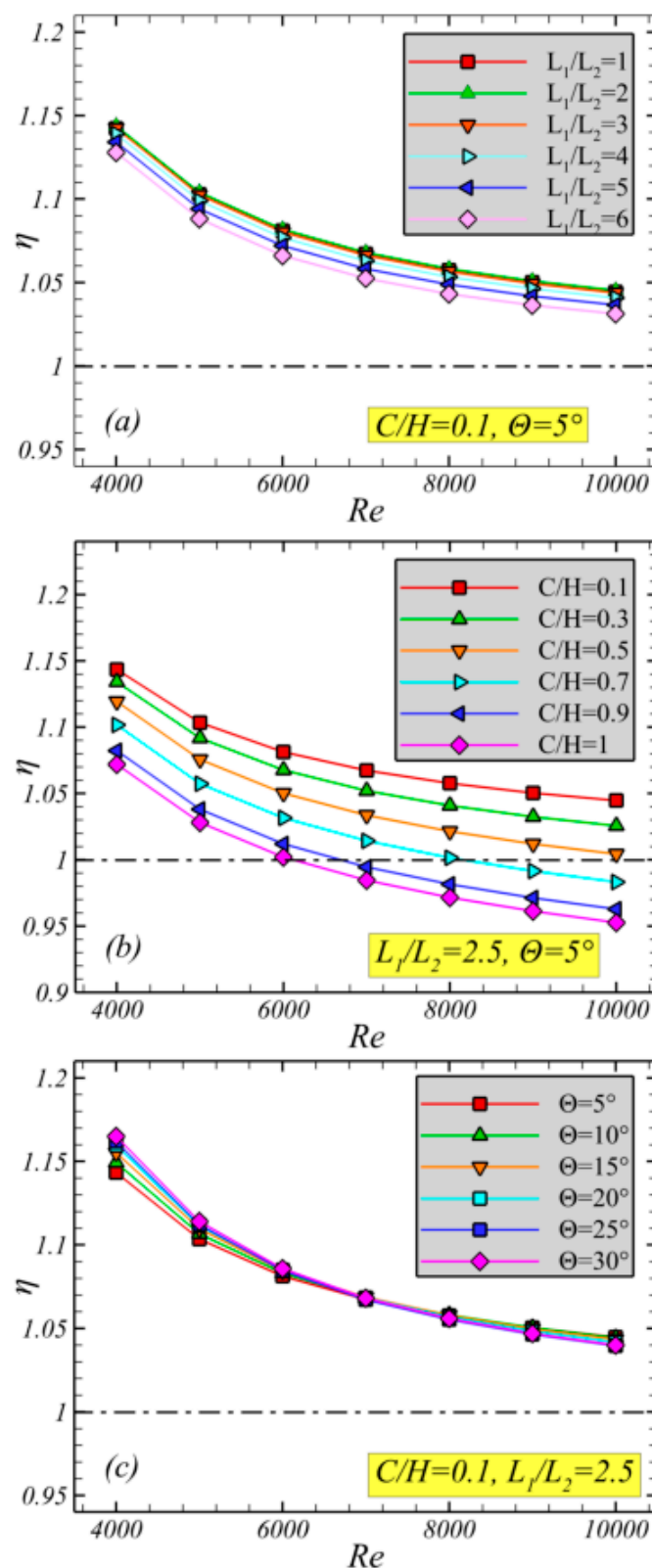


Figure 14. Variation in the thermal performance index with Reynolds numbers: (a) For the ratio of L_1/L_2 within the range $1 \leq L_1/L_2 \leq 6$, (b) For the ratio of the corrugation height-to-channel height, C/H $0.1 \leq C/H \leq 1$, (c) For the expansion angle within the range $5^\circ \leq \theta \leq 30^\circ$.

5. Conclusions

This study numerically investigated the thermal–hydraulic behaviour of corrugated channels by systematically analysing the effects of three geometric parameters—the corru-

gation height-to-channel height ratio (C/H), the length ratio (L_1/L_2), and the expansion angle (θ)—over a Reynolds number range of 4×10^3 – 1×10^4 . Response Surface Methodology (RSM) was used to construct predictive models for the Nu, friction factor (f), and thermal performance index (η), while field synergy analysis provided additional mechanistic insight into how geometric variations influence velocity–temperature gradient alignment. The results confirmed that C/H is the dominant geometric parameter affecting both heat transfer and flow resistance. Increasing C/H from 0.10 to 1.00 reduced Nu by approximately 22% at $Re = 8000$ and decreased the friction factor by 40–45%, indicating a clear trade-off between thermal enhancement and hydraulic loss. Lower C/H values generated stronger recirculation, improved mixing, and better synergy-angle alignment, leading to higher η values. Conversely, higher C/H resulted in a more streamlined flow with reduced turbulence intensity.

The length ratio (L_1/L_2) exhibited only minor influence. Across the full range (0.5–6), its effect on Nu, f , and η remained within 1–3%, demonstrating that this parameter primarily offers geometric flexibility without significantly altering overall performance. The expansion angle θ produced moderate but consistent improvements. Increasing θ from 3° to 30° yielded a 1–2% rise in η , with modest increases in Nu and f . These trends align with diffuser-flow behaviour, where larger angles introduce mild separation yet do not significantly deteriorate flow stability within the tested range. Field synergy analysis supported these findings; lower α and β angles were primarily achieved at small C/H values, confirming stronger alignment between velocity and temperature gradients and explaining the enhanced convective transport observed in these configurations.

Based on these results, practical design recommendations can be proposed for corrugated channels operating within the investigated Reynolds number range. Lower corrugation height ratios ($C/H = 0.10$ – 0.30) are recommended when thermal performance enhancement is prioritized, as these configurations yield stronger recirculation, improved field synergy, and higher η values, particularly at moderate Reynolds numbers. Higher C/H values ($C/H \geq 0.70$) may be preferable in applications where pressure-loss reduction is the primary concern, despite the associated decrease in heat-transfer enhancement. Given its negligible influence on performance, the length ratio (L_1/L_2) may be selected mainly based on geometric or manufacturing constraints. Moderate expansion angles ($\theta = 15^\circ$ – 30°) provide a balanced improvement in thermal performance with acceptable increases in friction losses and can be applied consistently across the examined Reynolds number range. The combined use of RSM and field synergy analysis therefore offers a robust predictive–diagnostic framework for the design of compact heat-exchanger systems requiring enhanced heat transfer, controlled turbulence, and acceptable pressure losses.

Future studies may complement the present numerical findings through experimental investigations, such as Particle Image Velocimetry (PIV) measurements and detailed heat-transfer and pressure-drop experiments, to further validate the predicted flow structures and thermal–hydraulic performance. Such experimental validation would enhance the robustness and practical applicability of the proposed design guidelines.

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