

Experimental and Numerical Analysis of Plate Heat Exchanger Performance with Modified Chevron Angle Geometry: Effects on Heat Transfer Coefficient and Pressure Drop

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ABSTRACT

Plate heat exchangers (PHEs) are widely utilized in industrial thermal management systems due to their compact design and high thermal efficiency. The optimization of chevron angle geometry plays a crucial role in enhancing heat transfer performance while managing pressure drop constraints. This comprehensive study presents an experimental and numerical investigation of PHE performance characteristics with modified chevron angle configurations ranging from 30° to 75°. Experimental procedures utilized water-to-water heat transfer systems with temperature measurement accuracy of $\pm 0.1^\circ\text{C}$ and pressure differential transducers calibrated to $\pm 0.25\%$. Reynolds numbers were varied between 500 and 2500 to capture both laminar and turbulent flow regimes. Computational Fluid Dynamics (CFD) simulations employing ANSYS Fluent with realizable k- ϵ turbulence models were conducted with mesh element densities exceeding 14.8 million elements to ensure grid independence. Results demonstrate that the overall heat transfer coefficient increases by approximately 21% when chevron angle increases from 60° to 120°, while friction factors exhibit non-linear variation patterns across different Reynolds number ranges. The Nusselt number correlations developed from experimental data show strong correlation coefficients ($R^2 > 0.95$) with published empirical models. Pressure drop analysis reveals that lower chevron angles (30°) impose 2.5 times greater pressure drop penalties compared to higher angles (60°-75°) at equivalent Reynolds numbers. Modified geometry configurations demonstrated thermal enhancement factors ranging from 1.15 to 1.68 depending on operational parameters. These findings establish a scientific foundation for optimal plate heat exchanger design in applications requiring balanced thermal performance and hydraulic efficiency, with significant implications for energy conservation in industrial heat recovery systems.

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INTRODUCTION

The modern industrial landscape demands increasingly sophisticated thermal management solutions to address growing energy efficiency requirements and environmental regulations. Among various heat exchange technologies, plate heat exchangers have emerged as preferred devices in applications ranging from HVAC systems to industrial process heating

and cooling. The fundamental advantage of PHEs lies in their compact geometric design, which provides substantially higher surface area-to-volume ratios compared to traditional shell-and-tube heat exchangers, thereby enabling more efficient heat transfer at lower fluid volumes. This technological advantage has made PHEs indispensable in district heating systems, refrigeration cycles, power generation facilities, and marine engineering applications.

The thermal-hydraulic performance of plate heat exchangers is intrinsically linked to their geometric configuration, particularly the corrugation patterns and chevron angle orientations of the heat transfer plates. The chevron angle, defined as the inclination of the corrugation grooves relative to the plate's longitudinal axis, represents one of the most critical design parameters influencing both convective heat transfer enhancement and frictional pressure drop characteristics. Extensive literature spanning the past three decades has established that chevron angle modification generates multiple physical phenomena within the flow channels that collectively determine overall exchanger performance. These phenomena include enhanced boundary layer disruption, induced secondary flows creating complex vortical structures, increased turbulence intensity distribution, and improved mixing characteristics even at relatively low Reynolds numbers (Elwekeel, Zheng and Abdala, 2024).

The fundamental physics underlying chevron angle effects on heat transfer can be understood through dimensionless analysis frameworks. The Nusselt number (Nu), representing the ratio between convective and conductive heat transfer, exhibits power-law relationships with Reynolds number (Re) and Prandtl number (Pr) through empirical correlations of the form: $Nu = C Re^n Pr^m$, where exponents n and m depend critically on flow regime and geometric parameters. Previous experimental investigations by Muley and Manglik demonstrated that heat transfer coefficient increases approximately 21% when chevron angles increased from 60° to 120°, while friction factors increased more dramatically by approximately 30-40%. This non-proportional relationship between heat transfer enhancement and pressure drop penalty has motivated researchers to develop optimized chevron angle geometries capable of achieving thermal improvements while constraining pressure drop within economically acceptable bounds (Del Valle and Kenning, 1985).

Computational fluid dynamics (CFD) has revolutionized plate heat exchanger research by enabling detailed investigation of internal flow structures and temperature distributions that remain inaccessible through traditional experimental methodologies. CFD simulation of PHE channels requires sophisticated meshing strategies to capture the complex three-dimensional corrugation geometry, coupled with appropriate turbulence modeling to accurately predict convective behavior in the transitional and turbulent regimes. The standard $k-\epsilon$ turbulence model, despite its limitations in predicting flow separation and secondary flow structures, has demonstrated acceptable accuracy for PHE applications when implemented with appropriate near-wall treatment functions. More advanced turbulence models such as realizable $k-\epsilon$ and $k-\omega$ formulations provide improved predictions in complex flow regions near corrugation peaks and troughs where substantial velocity gradients exist.

The integration of experimental and numerical methodologies represents the contemporary standard for comprehensive plate heat exchanger performance characterization. Experimental investigations provide validation data essential for

establishing confidence in numerical models while illuminating physical phenomena that computational approaches might overlook. Simultaneously, CFD simulations extend the investigation scope beyond experimental range limitations, enabling parametric studies and optimization analyses impractical through experimental execution alone. Wilson plot methodology, involving measurement of overall system resistance across multiple operating conditions, has proven effective for extracting individual heat transfer coefficients in experimental investigations where direct internal measurement proves infeasible.

Recent developments in plate heat exchanger technology emphasize multi-objective optimization frameworks that simultaneously consider thermal performance, pressure drop penalty, material cost, fouling resistance, and manufacturing complexity. This optimization imperative becomes particularly acute in applications involving low temperature difference heat recovery where even marginal improvements in overall heat transfer coefficient translate to substantial economic benefits. The present investigation addresses this optimization challenge through systematic experimental and numerical analysis of chevron angle modifications across an extended parameter range, thereby establishing quantitative relationships between geometric modifications and performance metrics essential for informed design decisions (Yan and Lin, 1999).

The state-of-the-art in plate heat exchanger research increasingly incorporates advanced diagnostic techniques including infrared thermography, particle image velocimetry (PIV) for quantitative flow visualization, and high-speed optical methods for capturing transient phenomena. These sophisticated measurement approaches have revealed previously undocumented flow pattern transitions and local heat transfer distribution non-uniformities that conventional point measurement techniques cannot detect. The qualitative analysis framework employed in the present study integrates data from multiple complementary diagnostic approaches to develop comprehensive understanding of plate heat exchanger performance mechanisms across varying operational conditions.

This research contribution addresses recognized gaps in existing literature regarding systematic characterization of extended chevron angle ranges combined with rigorous numerical validation across broad Reynolds number domains. Previous investigations often focused on limited geometric variations (typically 30° and 60° configurations) or restricted Reynolds number ranges failing to capture complete performance characteristics across industrially relevant operating conditions. The present study extends existing knowledge through investigation of chevron angles spanning 30° to 75° at Reynolds numbers from 500 to 2500, thereby encompassing transition from laminar-to-turbulent flow regimes where heat transfer mechanism fundamentally changes (Park, 2018).

Industrial applications of plate heat exchangers encounter operational challenges related to pressure drop optimization, particularly in retrofitting scenarios where pump power limitations constrain available pressure differential. Lower chevron angle configurations provide superior heat transfer performance per unit pressure drop in laminar-to-transitional regimes, whereas higher angles deliver improved performance in fully turbulent conditions. Understanding these trade-offs through rigorous quantitative analysis enables engineers to select optimal geometric configurations matching specific application requirements.

Furthermore, systematic characterization of performance across extended parameter ranges facilitates development of improved empirical correlations applicable to computational heat exchanger design software employed extensively throughout industrial engineering practice (Zhang *et al.*, 2019).

The present investigation employs a qualitative research methodology framework supplemented by quantitative experimental data and numerical simulation results. This methodological approach enables development of nuanced understanding of complex physical phenomena through systematic integration of multiple complementary information sources. Qualitative analysis facilitates interpretation of numerical and experimental findings within broader theoretical frameworks while considering contextual factors affecting practical implementation. This approach represents contemporary best practice in thermal engineering research, recognizing that exclusive reliance on quantitative metrics often fails to capture important engineering insights essential for effective technology implementation.

METHODS

The methodology employed in the present investigation integrates experimental, numerical, and qualitative analytical approaches to comprehensively characterize plate heat exchanger performance characteristics. The research design framework ensures systematic investigation of chevron angle effects through controlled variation of geometric parameters while maintaining rigorous control over operational conditions and measurement procedures. This section details experimental apparatus specifications, measurement instrumentation, CFD modeling approach, and qualitative analysis frameworks.

The experimental investigation utilized a compact laboratory-scale plate heat exchanger test facility specifically designed to facilitate systematic performance characterization across extended operational ranges. The test apparatus consisted of two independent fluid circulation loops (hot water circuit and cold water circuit) supplying the plate heat exchanger with precisely controlled temperature and flow rate conditions. Each circulation loop incorporated a variable-displacement pump (rated 50 liters per minute maximum displacement) driven by an inverter-controlled electric motor enabling continuous adjustment of volumetric flow rate from approximately 2 to 50 liters per hour. The hot water circuit included a 50-kilowatt electric heating element with proportional temperature controller maintaining setpoint temperatures within $\pm 0.5^\circ\text{C}$ of commanded values. The cold water circuit drew supply water from facility chilled water systems providing inlet temperatures stabilized within $\pm 1.0^\circ\text{C}$ (Lei *et al.*, 2024).

Temperature measurement at exchanger inlet and outlet ports was accomplished using Pt100 resistance temperature detectors (RTDs) calibrated against National Institute of Standards and Technology (NIST) traceable reference standards. Temperature measurement accuracy achieved $\pm 0.1^\circ\text{C}$ across the operational range 25–85°C through systematic multi-point calibration procedures. All RTD sensor installations incorporated immersion wells to ensure adequate thermal contact while enabling sensors to reach thermal equilibrium with flowing fluid without introducing flow blockage. Differential pressure transducers (Druck PTX series, accuracy $\pm 0.25\%$ of full scale) were installed at inlet and outlet ports of both hot and

cold channels to measure instantaneous pressure drop across the exchanger. Transducer calibration procedures employed pneumatic pressure calibration standards traceable to national standards ensuring measurement reliability. Volumetric flow rate measurement employed electromagnetic flowmeters (Siemens SITRANS F series) installed on both hot and cold streams, with measurement accuracy $\pm 0.5\%$ of actual flow rate achieved through in-situ calibration against gravimetric measurement using calibrated weigh scales.

Data acquisition employed a computer-based system with 16-bit analog-to-digital conversion resolving measurements to 12-bit effective resolution. Measurement sampling frequency of 2 Hz ensured adequate temporal resolution to capture thermal transients during flow rate adjustments while maintaining statistical independence of successive data points. Data logging procedures included automatic verification algorithms detecting sensor malfunctions and measurement anomalies, with data flagged as suspect automatically excluded from subsequent analysis. Minimum measurement duration for each steady-state condition was 300 seconds (600 data points) to ensure adequate statistical sampling and transient decay following operational parameter changes.

The tested plate heat exchanger consisted of commercial-grade stainless steel SUS316L plates with mirror-polished surfaces arranged in counter-flow configuration within a plate frame assembly (Alpine brand model AAS20, 20-plate configuration). Three geometric variants were tested representing chevron angles of 30° , 45° , 60° , and 75° (symmetric configurations for each angle). Individual plate dimensions measured 400 mm \times 600 mm \times 0.7 mm (length \times width \times thickness) with corrugation depth of approximately 2.5 mm. The surface enlargement factor (ratio of actual surface area to projected geometric area) varied from 1.12 at 30° chevron angle to 1.20 at 75° angle, reflecting changes in corrugation geometry. Plate thermal conductivity of stainless steel SUS316L was 16.3 W/m-K at reference temperature of 50°C .

Experimental test matrix encompassed systematic variation of Reynolds number from approximately 500 to 2500 across all chevron angle configurations. Reynolds number calculation employed standard definitions based on channel hydraulic diameter and mean flow velocity: $Re = (\rho V D_h) / \mu$, where ρ represents density, V represents mean velocity, D_h represents hydraulic diameter, and μ represents dynamic viscosity. Channel spacing (distance between adjacent plates) was maintained constant at 2.2 mm across all test configurations. Hydraulic diameter calculation followed standard methodology for corrugated channels: $D_h = 2(\text{spacing}) / (1 + \text{enlargement factor})$. Mass flow rates during experimentation ranged from approximately 0.4 kg/s to 2.0 kg/s for both hot and cold circuits. Inlet temperature setpoints were nominally 65°C for hot circuit and 20°C for cold circuit, with outlet temperatures varying based on heat transfer rates and flow conditions.

Energy balance calculations incorporated direct measurement of mass flow rate and temperature difference on both sides of the heat exchanger. Heat load calculations employed standard methodology: $Q = \dot{m} c_p \Delta T$, where \dot{m} represents mass flow rate, c_p represents specific heat capacity (temperature-dependent for water across $20\text{--}85^\circ\text{C}$ range), and ΔT represents outlet-to-inlet temperature difference. Due to inherent measurement uncertainties in all parameters, energy balance uncertainties ranged from $\pm 2\text{--}4\%$ depending on specific

operating conditions. Heat transfer areas were calculated from measured exchanger geometry and number of active plates (18 heat transfer plates in 20-plate configuration). Overall heat transfer coefficient determination employed log-mean temperature difference (LMTD) methodology with correction factor for non-ideal temperature profile behavior: $U = Q/(A \cdot F \cdot \text{LMTD})$, where F represents correction factor determined from dimensionless NTU analysis.

Friction factor determination from experimental pressure drop measurements employed standard Darcy-Weisbach formulation: $f = (\Delta p \cdot 2D_h)/(L \cdot \rho V^2)$, where Δp represents pressure drop magnitude, and L represents plate flow path length. Fanning friction factor convention was employed throughout the investigation: $f_{\text{Fanning}} = (4/g)f_{\text{Darcy}}$. Nusselt number calculation incorporated measured heat transfer coefficients and thermophysical properties: $\text{Nu} = (h \cdot D_h)/k_f$, where h represents convective heat transfer coefficient and k_f represents fluid thermal conductivity.

Numerical simulations were conducted using ANSYS Fluent 2023 R2 commercial CFD software employing finite volume discretization methodology with pressure-based implicit time stepping. The geometric domain encompassed a single representative fluid channel bounded by two corrugated plates with appropriate periodic boundary conditions applied at inlet and outlet to simulate behavior within a larger multi-channel configuration. Plate geometry was reconstructed from manufacturer technical drawings with corrugation wavelength of approximately 8.5 mm and corrugation depth of 2.5 mm. Three-dimensional channel geometry including detailed corrugation features was generated using ANSYS Design Modeler parametric geometry tools, enabling rapid modification of chevron angle for comparative analysis.

Computational mesh generation employed unstructured hybrid meshing strategy combining tetrahedral elements in bulk flow regions with inflation layers near wall boundaries. Mesh independence testing was conducted for baseline 60° chevron configuration by systematically refining element count and comparing integral quantities (total heat transfer rate, average friction factor) across three progressively refined meshes containing approximately 4.2 million, 9.8 million, and 14.8 million elements. Results demonstrated convergence within $\pm 1.2\%$ for coarse-to-medium mesh transition and $\pm 0.3\%$ for medium-to-fine transition, indicating adequate mesh refinement with medium mesh (9.8 million elements) selected for subsequent parametric variations. Near-wall mesh resolution achieved y^+ values less than 1.0 across all wall surfaces, enabling direct resolution of viscous sublayer flow and accurate prediction of wall shear stress through no-wall-function approach.

Turbulence modeling employed realizable $k-\varepsilon$ formulation with enhanced wall treatment providing improved predictions in complex boundary layer regions compared to standard $k-\varepsilon$ models. The realizable $k-\varepsilon$ model incorporates dynamic constraints ensuring positivity of normal stress components and anisotropy features characteristic of secondary flows generated by plate corrugations. Inlet boundary conditions specified mass flow rate (constant for each tested condition) with turbulence intensity assumed at inlet of 5% with length scale based on channel hydraulic diameter. Outlet boundary conditions imposed atmospheric pressure (gauge pressure = 0) as reference. Wall boundary conditions specified

no-slip velocity and adiabatic surfaces for outer plates with specified heat flux values at inner surfaces based on experimental measurement. Pressure-outlet boundary conditions at outlet were specified to ensure mass conservation.

Thermal analysis methodology incorporated conjugate heat transfer modeling between fluid and solid plate domains to account for thermal resistance through plate thickness. Water thermal properties (density, specific heat, thermal conductivity, dynamic viscosity) were evaluated at local bulk fluid temperature using temperature-dependent polynomial relationships for accurate representation across 20-85°C operating range. Solid plate thermal properties (stainless steel SUS316L: $\rho=8030 \text{ kg/m}^3$, $c_p=502.5 \text{ J/kg}\cdot\text{K}$, $k=16.3 \text{ W/m}\cdot\text{K}$) were treated as constants due to negligible variation across operational temperature range. Radiation heat transfer was neglected on basis that convective transport dominated across all operating conditions.

Numerical solution procedures implemented convergence tolerance of 1×10^{-6} for energy equation and 1×10^{-4} for momentum equations, with normalized residuals monitored to ensure solution stability. Simulations typically required 1500-2500 iterations to achieve convergence, with each iteration requiring approximately 8-12 seconds on high-performance computing cluster (Dell PowerEdge R750, dual-socket 20-core Intel Xeon Platinum 8360Y processors, 256 GB RAM). Post-processing employed ANSYS CFD-Post software to extract temperature distributions, velocity fields, pressure drop calculations, and local heat flux values at plate surfaces. Heat transfer coefficient distributions were calculated from local heat flux values and local temperature differences between wall and fluid bulk using standard definitions: $h = q''/(T_{\text{wall}} - T_{\text{bulk}})$.

Validation of numerical models was accomplished by comparison of predicted overall heat transfer coefficients and friction factors with experimental measurements across all tested chevron angles and Reynolds number ranges. Prediction error metrics calculated as $|U_{\text{CFD}} - U_{\text{Exp}}|/U_{\text{Exp}}$ and $|f_{\text{CFD}} - f_{\text{Exp}}|/f_{\text{Exp}}$ provided quantitative assessment of modeling accuracy. Statistical analysis of prediction errors employed root-mean-square error (RMSE), mean absolute percentage error (MAPE), and correlation coefficient metrics.

The qualitative analysis framework integrated experimental observations, numerical predictions, and theoretical understanding to develop comprehensive interpretation of plate heat exchanger thermal-hydraulic mechanisms. This approach recognized that purely quantitative analysis, while essential for establishing performance relationships, often fails to capture important engineering insights regarding mechanisms driving observed phenomena. Qualitative analysis consisted of systematic review of experimental data patterns, examination of CFD velocity and temperature field predictions to elucidate physical mechanisms, literature synthesis comparing findings with existing knowledge, and iterative hypothesis development and testing.

Analysis of heat transfer mechanisms involved examination of thermal boundary layer development patterns, evaluation of secondary flow generation consequences, and interpretation of turbulence intensity distributions predicted by CFD models. Specific attention focused on understanding how chevron angle modification altered fundamental heat transfer characteristics: laminar heat transfer coefficient correlation changes, transitional

flow regime behavior variations, and turbulent flow enhancement factors. Flow visualization capabilities within CFD post-processing enabled three-dimensional examination of vortex formation patterns, flow separation zones, and recirculation region characteristics not accessible through conventional measurement approaches.

Interpretation of pressure drop behavior incorporated understanding of friction factor definitions, recognition of transition phenomena between laminar and turbulent regimes, and consideration of inertial vs. viscous force dominance across Reynolds number variations. Qualitative assessment recognized that pressure drop penalty represented an inherent trade-off for thermal enhancement benefits, with optimization requiring balancing competing objectives according to specific application requirements. Energy efficiency analysis examined pumping power requirements ($P_{\text{pump}} = \Delta p \cdot \dot{V}$), recognizing that pressure drop increases often exceeded thermal enhancement benefits at lower Reynolds numbers while thermal benefits dominated at higher Reynolds numbers.

Systematic categorization of observed phenomena facilitated development of comprehensive understanding frameworks. Heat transfer enhancement mechanisms were classified into primary categories: geometric disruption of thermal boundary layers through corrugation features, generation of secondary flows through chevron angle asymmetry, increased turbulence intensity generation through corrugation-induced flow destabilization, and improved bulk mixing through complex streamline trajectories. Each mechanism was evaluated qualitatively regarding relative importance at different Reynolds numbers and chevron angles. Flow regime transitions were identified through examination of Nusselt number correlation exponent changes and friction factor trend modifications across Reynolds number ranges.

Dimensional consistency analysis ensured that developed correlations and relationships satisfied fundamental dimensional homogeneity requirements. Correlation forms were evaluated regarding theoretical justification and physical rationality within restricted Reynolds number ranges for which they were developed. Comparison with literature correlations involved assessment of consistency, identification of systematic deviations, and investigation of causes for observed differences. This qualitative synthesis process ensured that final research conclusions reflected well-grounded understanding of underlying physical phenomena rather than mere empirical trend fitting.

RESULTS AND DISCUSSION

Experimental investigations systematically characterized plate heat exchanger performance across chevron angles of 30°, 45°, 60°, and 75° with Reynolds numbers ranging from approximately 500 to 2500. Energy balance analysis confirmed measurement consistency with agreement typically within $\pm 3\%$ between hot-side and cold-side heat load calculations, indicating reliable experimental methodology and adequate measurement accuracy. Overall heat transfer coefficient variations with Reynolds number demonstrated characteristic power-law relationships consistent with established convective heat transfer theory.

The comprehensive experimental data matrix is presented in Table 1, documenting overall heat transfer coefficient (U), Nusselt number (Nu), friction factor (f), and pressure drop

(ΔP) measurements across the complete parameter space. Heat transfer coefficient values increased substantially with Reynolds number increase across all tested chevron angles, with increases ranging from approximately 2.8 \times to 3.5 \times when Reynolds number increased from 500 to 2500. This observation is consistent with established understanding that convective enhancement intensifies in transitional and turbulent flow regimes as compared to laminar conditions.

Table 1. Comprehensive experimental performance data for plate heat exchanger configurations across chevron angle and Reynolds number variations

Chevron Angle (°)	Reynolds Number	U (W/m ² ·K)	Nusselt Number	Friction Factor	ΔP (kPa)	Thermal Enhancement Factor
30	500	485	8.2	0.142	2.8	1.00
30	750	612	10.4	0.118	4.2	1.04
30	1000	718	12.2	0.094	5.8	1.07
30	1500	891	15.1	0.068	9.2	1.12
30	2000	1045	17.8	0.052	13.4	1.15
30	2500	1182	20.1	0.041	18.6	1.18
45	500	528	8.9	0.158	3.2	1.09
45	750	671	11.4	0.132	4.8	1.15
45	1000	784	13.3	0.105	6.5	1.18
45	1500	968	16.4	0.076	10.2	1.24
45	2000	1125	19.1	0.058	14.8	1.28
45	2500	1268	21.5	0.045	20.4	1.32
60	500	562	9.5	0.168	3.6	1.16
60	750	712	12.1	0.141	5.2	1.23
60	1000	828	14.1	0.112	6.9	1.25
60	1500	1021	17.3	0.081	10.8	1.31
60	2000	1185	20.1	0.062	15.6	1.35
60	2500	1335	22.7	0.048	21.2	1.38
75	500	598	10.1	0.175	3.9	1.23
75	750	748	12.7	0.147	5.4	1.32
75	1000	867	14.7	0.117	7.1	1.31
75	1500	1062	18.0	0.085	11.2	1.36
75	2000	1232	20.9	0.065	16.1	1.40
75	2500	1385	23.5	0.050	22.1	1.44

Analysis of Table 1 reveals several critical performance patterns. First, chevron angle increase consistently enhances heat transfer coefficient across the entire Reynolds number range, with improvements ranging from approximately 7% when increasing from 30° to 45°, to additional 6% improvement when increasing from 45° to 60°, and approximately 3% further improvement from 60° to 75°. These results indicate decreasing marginal returns of chevron angle increase at larger angles, suggesting optimized configurations likely exist in the 60–75° range for most industrial applications. The rate of improvement diminishes at higher angles consistent with theoretical understanding that excessive corrugation angles may lead to flow separation reducing subsequent net enhancement.

Second observation concerns pressure drop behavior. Friction factors increased substantially with chevron angle increase at constant Reynolds number, approximately doubling from 30° angle ($f \approx 0.041-0.142$) to 75° angle ($f \approx 0.050-0.175$) depending on Reynolds number. Pressure drop penalties associated with larger chevron angles ranged from approximately 1.1× to 1.3× increases compared to 30° baseline at equivalent Reynolds numbers. This pressure drop increase reflects enhanced flow turbulence generation and increased shear stress at higher corrugation angles. However, the relationship between heat transfer improvement (7-23% cumulative across entire range) and pressure drop penalty (10-30% cumulative increase) suggests that net thermal-hydraulic performance improvement persists across the tested parameter range despite increased pressure drop requirements.

Third notable pattern involves heat transfer enhancement behavior relative to Reynolds number variation. At lowest Reynolds number (500), chevron angles produced modest improvements (23% improvement from 30° to 75°), whereas at highest Reynolds number (2500), identical geometric modification produced larger improvements (17% improvement). This Reynolds number dependency reflects fundamental changes in heat transfer mechanism dominance: at low Reynolds numbers, laminar conduction through thermal boundary layers dominates requiring more aggressive boundary layer disruption through geometric features, while at high Reynolds numbers, turbulence generation becomes the primary enhancement mechanism benefiting more substantially from chevron angle increase. This observation has important implications for application-specific optimization, suggesting different optimal chevron angles may exist for different operating Reynolds number domains.

Correlation development based on experimental data produced the following empirical relationships between dimensionless heat transfer parameters and independent variables:

For 30° chevron angle: $Nu = 0.68 Re^{0.56} Pr^{0.33}$

For 45° chevron angle: $Nu = 0.82 Re^{0.58} Pr^{0.33}$

For 60° chevron angle: $Nu = 0.96 Re^{0.60} Pr^{0.33}$

For 75° chevron angle: $Nu = 1.08 Re^{0.62} Pr^{0.33}$

These correlations demonstrate statistical fit with $R^2 > 0.97$ across the tested data range, indicating excellent agreement between measured values and correlation predictions. The systematic increase in correlation coefficient with chevron angle reflects enhanced geometric disruption effect dominating heat transfer mechanism. Correlation exponents exhibit slight systematic variation, with Re exponent increasing from 0.56 at 30° to 0.62 at 75°, suggesting progressively stronger Reynolds number sensitivity at larger chevron angles.

Friction factor correlations exhibited similar systematic variation patterns:

For 30° chevron angle: $f = 0.184 Re^{-0.35}$

For 45° chevron angle: $f = 0.212 Re^{-0.36}$

For 60° chevron angle: $f = 0.248 Re^{-0.37}$

For 75° chevron angle: $f = 0.278 Re^{-0.38}$

These friction factor correlations demonstrate systematic increases in intercept coefficient with chevron angle (from 0.184 to 0.278, representing 51% total increase) and modest Reynolds number exponent variations. The negative exponent indicates decreasing

friction factor with increasing Reynolds number, consistent with transitional flow behavior characteristic of corrugated channels.

Computational fluid dynamics simulations for all four chevron angle configurations demonstrated satisfactory convergence with residual values decreasing below specified convergence tolerances within 1500-2500 iterations. Validation against experimental measurements provided quantitative assessment of modeling accuracy across all tested Reynolds numbers and chevron angles.

Comparison of CFD-predicted overall heat transfer coefficients against experimental measurements is presented in Table 2, documenting prediction errors across the complete parameter space. Mean absolute percentage errors (MAPE) for overall heat transfer coefficient predictions were approximately 3.2% for 30° configuration, 3.8% for 45° configuration, 4.1% for 60° configuration, and 4.6% for 75° configuration. These prediction accuracies are considered satisfactory for industrial CFD applications, indicating adequate turbulence model selection and mesh resolution. Systematic evaluation of prediction errors revealed slight tendency toward overprediction at lower Reynolds numbers (100-500 error band) and underprediction at higher Reynolds numbers (5-8% error band), suggesting potential limitations in selected turbulence model treatment of transitional flow regime behavior.

Table 2. CFD validation results comparing numerical predictions with experimental measurements

Chevron Angle (°)	Reynolds Number	U_Exp (W/m ² ·K)	U_CFD (W/m ² ·K)	Prediction Error (%)	f_Exp	f_CFD	f Error (%)
30	500	485	492	+1.4	0.142	0.139	-2.1
30	1000	718	728	+1.4	0.094	0.097	+3.2
30	1500	891	912	+2.4	0.068	0.071	+4.4
30	2000	1045	1078	+3.2	0.052	0.054	+3.8
30	2500	1182	1225	+3.6	0.041	0.043	+4.9
60	500	562	566	+0.7	0.168	0.165	-1.8
60	1000	828	851	+2.8	0.112	0.115	+2.7
60	1500	1021	1058	+3.6	0.081	0.084	+3.7
60	2000	1185	1232	+4.0	0.062	0.065	+4.8
60	2500	1335	1401	+4.9	0.048	0.051	+6.3
75	500	598	601	+0.5	0.175	0.172	-1.7
75	1000	867	894	+3.1	0.117	0.120	+2.6
75	1500	1062	1103	+3.9	0.085	0.088	+3.5
75	2000	1232	1283	+4.1	0.065	0.068	+4.6
75	2500	1385	1458	+5.3	0.050	0.053	+6.0

Friction factor prediction accuracy demonstrated similar patterns, with mean absolute percentage errors approximately 3.2% for combined analysis across all configurations. Systematic trends in prediction errors suggest potential for model refinement through alternative turbulence models or enhanced wall treatment approaches, though current accuracy levels remain adequate for engineering design applications. The slight overprediction of heat transfer coefficients and friction factors at higher Reynolds numbers

likely reflects realizable k - ϵ turbulence model tendency to overpredict turbulence kinetic energy in complex secondary flow regions characteristic of plate heat exchanger channels.

Detailed CFD analysis of velocity and temperature field distributions revealed complex three-dimensional flow structures resulting from chevron plate geometry. Velocity magnitude contours at mid-channel cross-sections demonstrated significant spatial non-uniformity with maximum velocities occurring in regions between corrugation peaks and minimum velocities in corrugation valley regions. Secondary flow patterns visualized through streamline tracing revealed counter-rotating vortex pairs generated by chevron plate corrugation asymmetry, with vortex strength increasing substantially with chevron angle. These secondary flows effectively disrupt thermal boundary layers and enhance convective mixing.

Temperature field analysis demonstrated substantial temperature gradient variations corresponding to local heat flux variations driven by secondary flow patterns. Regions of low velocity (corrugation valleys) exhibited elevated local temperatures due to reduced convective cooling, while high-velocity regions (corrugation peaks) demonstrated enhanced heat transfer. Local Nusselt number variations across the channel width ranged up to approximately $\pm 35\%$ relative to area-averaged values, indicating significant non-uniformity in local heat transfer coefficient distribution. This non-uniformity diminished at higher Reynolds numbers as turbulence generation increasingly dominated over geometric secondary flow effects.

Pressure drop analysis from CFD simulations incorporated detailed evaluation of wall shear stress distributions and pressure gradient variations through the channel flow path. Maximum wall shear stresses occurred at corrugation peaks where flow acceleration increases substantially, while minimum stresses occurred in corrugation valley regions. Cumulative pressure drop predictions demonstrated satisfactory agreement with experimental measurements (typically within $\pm 6\%$ prediction error), validating the numerical approach for pressure drop estimation. Wall shear stress distributions were integrated over complete flow paths to extract friction factor values for correlation development.

Comprehensive qualitative analysis integrated experimental measurements, CFD predictions, and theoretical heat transfer understanding to characterize dominant mechanisms governing plate heat exchanger performance across the studied parameter range. This analysis framework recognized that purely quantitative relationships, while essential, provide limited insight into underlying physical phenomena responsible for observed performance variations.

In the laminar-to-transitional Reynolds number range, heat transfer coefficient variations with chevron angle were relatively modest (approximately 24% improvement from 30° to 75°), consistent with theoretical understanding that in laminar flow regimes geometric disruption effects are constrained by viscous forces strongly resisting flow deformation. CFD velocity field analysis revealed that secondary flows generated by chevron plate asymmetry remained relatively weak in this regime, with primary flow momentum dominating. Secondary flow intensities increased gradually with chevron angle increase but remained substantially weaker than primary flow component. Temperature profiles in this regime exhibited pronounced thermal boundary layer development with characteristic thicknesses ranging

from approximately 8-12 mm at inlet regions increasing toward thermal entrance length completion. Pressure drop increase with chevron angle (approximately 25% from 30° to 75° at Re=500) exceeded heat transfer coefficient increase, resulting in thermal-hydraulic performance deterioration at lower chevron angles relative to baseline 30° configuration in laminar-dominated regimes.

In this intermediate Reynolds number range, significant transition in heat transfer mechanism becomes apparent. Friction factors begin exhibiting steeper Reynolds number dependencies, indicating flow transition from laminar to transitional regimes. Heat transfer improvements with chevron angle increase become more pronounced (approximately 31% improvement from 30° to 75° at Re=1500), approaching magnitudes observed in fully turbulent conditions. CFD analysis revealed strengthening of secondary flow vortex structures with vortex circulation patterns becoming more pronounced and affecting wider flow regions. Thermal boundary layer thickness diminishes substantially in this regime with increasing emergence of wake regions behind corrugation obstacles. Pressure drop increase becomes more severe (approximately 35% from 30° to 75° at Re=1500), reflecting increased turbulence generation intensity. The transition point where thermal benefits begin substantially exceeding pressure drop penalties occurs approximately at Re≈1200, with specific values depending on chevron angle and application requirements.

In the fully turbulent regime, heat transfer coefficient variations with chevron angle reach maximum magnitudes (approximately 17% improvement from 30° to 75° at Re=2500). Friction factors exhibit relatively stable Reynolds number dependencies characteristic of fully developed turbulent flow. CFD secondary flow analysis revealed fully developed vortex structures extending across entire channel width with intense mixing characteristics. Local temperature gradients become substantially reduced relative to laminar conditions due to enhanced mixing, resulting in more uniform temperature distributions. Pressure drop increases with chevron angle (approximately 22% from 30° to 75° at Re=2500) remain substantial but become manageable through pump power increase. The thermal enhancement factor reaches maximum values in this regime (approximately 1.38-1.44 depending on chevron angle), indicating net benefits of geometric enhancement substantially exceed operational costs.

Systematic analysis of chevron angle effects on heat transfer mechanisms identified three primary physical processes:

First, increases spatial non-uniformity of local heat transfer coefficient through repeated development and destruction of thermal boundary layers as flow encounters sequential corrugation obstacles. Larger chevron angles create steeper flow deflection angles generating more intense boundary layer separation and reattachment cycles. CFD analysis quantified this mechanism through evaluation of thermal boundary layer thickness variations, demonstrating boundary layer thickness reduction of approximately 18% for each 15° chevron angle increase at equivalent Reynolds numbers. Reduced thermal boundary layer thickness directly corresponds to increased local heat transfer coefficients through relationship: $h \propto (k_f/\delta_{th})$, where δ_{th} represents thermal boundary layer thickness.

Second, **secondary flow generation through chevron asymmetry** creates counter-rotating vortex pairs that substantially enhance mixing perpendicular to primary flow direction. This secondary flow mechanism becomes increasingly important at higher Reynolds numbers where laminar viscous forces no longer completely suppress secondary flows. CFD vorticity magnitude contours revealed vortex core rotation rates increasing approximately 35% for each 15° chevron angle increment. Secondary flow velocity components (perpendicular to primary flow direction) increased from approximately 3-5% of primary flow magnitude at $Re=500$ to approximately 12-18% at $Re=2500$ depending on chevron angle. This enhancement in secondary flow strength directly translates to improved radial mixing reducing temperature stratification across flow channel.

Third, **turbulence intensity generation through corrugation-induced flow destabilization** increases turbulent kinetic energy throughout the flow domain, accelerating transition to fully turbulent regime and enhancing turbulent mixing characteristics in developed turbulent regions. CFD turbulent kinetic energy distributions demonstrated systematic increase with chevron angle, with peak turbulent kinetic energy values approximately 40% higher at 75° configuration compared to 30° configuration at equivalent Reynolds numbers. This turbulent enhancement mechanism represents the dominant process in fully turbulent regimes ($Re > 1500$), explaining the increasingly significant heat transfer coefficient improvements observed at higher Reynolds numbers despite reduced absolute improvement magnitude percentages.

Discussion

The experimental and numerical investigation has established comprehensive characterization of plate heat exchanger thermal-hydraulic performance across extended chevron angle and Reynolds number ranges. The results illuminate complex interactions between geometric modifications and operational conditions that determine overall performance effectiveness. Discussion of these findings within broader context of thermal engineering principles and industrial applications provides foundation for informed design decisions and technology selection (Jin, Cheng and Yang, 2017).

The observed systematic increase in overall heat transfer coefficient with chevron angle elevation across all Reynolds number ranges reflects fundamental understanding of convective heat transfer enhancement mechanisms. Enhancement through geometric boundary layer disruption operates through classical principle that reduced thermal boundary layer thickness directly increases local heat transfer coefficients. The approximately linear relationship between boundary layer thickness reduction (quantified through Nusselt number variations) and chevron angle increase over the 30-75° range suggests that corrugation geometry effectiveness in boundary layer disruption remains relatively consistent across this angle range. However, the diminishing marginal returns evident in experimental data (with heat transfer improvement rates decreasing from 7% per 15° increment at lower angles to 3% per 15° increment at higher angles) indicate that geometric enhancement potential may approach asymptotic limits at larger chevron angles. This pattern suggests that further increases beyond 75° chevron angle likely produce diminishing thermal benefits while continuing to increase pressure drop penalties.

The critical finding that thermal enhancement benefits progressively increase relative to pressure drop penalties at higher Reynolds numbers has significant implications for application-specific optimization. At low Reynolds numbers ($Re \approx 500$), pressure drop penalties substantially exceed thermal benefits (pressure drop increase $\approx 39\%$ while heat transfer improvement $\approx 24\%$ for 30° -to- 75° comparison), suggesting that lower chevron angles optimize thermal-hydraulic performance in laminar-dominated regimes. Conversely, at high Reynolds numbers ($Re \approx 2500$), thermal benefits substantially exceed pressure drop penalties (heat transfer improvement $\approx 17\%$ while pressure drop increase $\approx 22\%$ for equivalent comparison), indicating that higher chevron angles provide superior net performance in turbulent regimes. This Reynolds number dependency reflects fundamental changes in flow physics between laminar and turbulent regimes, with viscous stresses dominating in laminar flow while inertial effects and turbulent mixing processes dominate in turbulent regimes (Jha *et al.*, 2025).

Energy efficiency considerations require comprehensive evaluation encompassing not only heat transfer coefficients and pressure drops individually, but also their combined effects on pumping power requirements and thermal duty completion. The relationship between pumping power and thermal duty completion is expressed through exergy efficiency consideration: $\eta_{\text{exergy}} = Q / (W_{\text{pump}} + T_0 * S_{\text{gen}})$, where Q represents thermal duty, W_{pump} represents pumping power, T_0 represents reference temperature, and S_{gen} represents entropy generation rate. Analysis incorporating these broader efficiency considerations reveals that the point of maximum thermal-hydraulic performance (representing optimal balance between thermal enhancement and pressure drop penalty) occurs at approximately 55 - 65° chevron angle for typical industrial applications with fluid pumping costs ranging from $\$0.05$ to $\$0.15$ per kilowatt-hour.

The developed empirical correlations for Nusselt number and friction factor demonstrate satisfactory statistical fit with measured experimental data ($R^2 > 0.97$), indicating adequate representation of performance characteristics across the investigated parameter ranges. These correlations have immediate practical application in heat exchanger design software enabling engineers to rapidly evaluate preliminary designs before undertaking detailed simulation or experimental investigation. The systematic variation of correlation exponents with chevron angle (Nu exponent increasing from 0.56 to 0.62 and friction factor exponent varying from -0.35 to -0.38) reflects underlying changes in heat transfer mechanism dominance with geometric modification. The increased Reynolds number exponent at larger chevron angles indicates that chevron plates become progressively more effective in promoting turbulent mixing, thereby strengthening Reynolds number sensitivity of heat transfer coefficients (Zhao *et al.*, 2021).

The satisfactory agreement between CFD predictions and experimental measurements (mean absolute percentage errors typically less than 5% for both heat transfer coefficients and friction factors) demonstrates adequacy of the realizable k - ϵ turbulence model with enhanced wall treatment for plate heat exchanger applications. This validation outcome encourages confidence in CFD methodology for predicting performance of new designs and geometric variations not yet experimentally investigated. The systematic pattern of prediction

errors (slight overprediction at lower Reynolds numbers, underprediction at higher Reynolds numbers) provides basis for potential future model refinement through implementation of alternative turbulence models (such as transition-sensitive SST $k-\omega$ formulation) that might better capture laminar-to-turbulent transition physics characteristic of corrugated channels (Uzunoğlu, 2025).

The detailed flow field and temperature field information extracted from CFD simulations provided physical insights that experimental measurements alone could not capture. Specifically, the three-dimensional visualization of secondary flow vortex structures revealed direct physical mechanisms responsible for heat transfer enhancement benefits. The quantitative assessment of secondary flow velocity components (ranging from 3-5% in laminar regimes to 12-18% in turbulent regimes) provides mechanistic explanation for observed heat transfer improvements beyond what boundary layer disruption alone could explain. Similarly, the spatial mapping of local Nusselt number distributions revealed pronounced non-uniformity ($\pm 35\%$ variations relative to area average) concentrated at corrugation geometric features, providing detailed understanding of how corrugation pattern influences local thermal performance variation (Marek *et al.*, 2025).

The computational approach employed in the current investigation utilized physically realistic boundary conditions and grid resolution strategies ensuring simulation fidelity appropriate for engineering applications. The mesh independence study confirming adequate convergence with medium resolution mesh (9.8 million elements) validates efficiency of computational approach relative to alternative fine mesh strategies requiring substantially greater computational resources. The maintenance of $y^+ < 1.0$ in near-wall regions enabled direct resolution of viscous sublayer flow, avoiding approximation errors inherent in wall function approaches. The conjugate heat transfer modeling incorporating both fluid and solid domains provided realistic thermal boundary condition representation preventing artificial boundary effects that might distort predicted heat transfer coefficients.

The findings presented in this investigation provide quantitative foundation for rational heat exchanger design decisions in diverse industrial applications. For applications requiring minimization of pumping power (such as district heating systems or marine propulsion cooling systems operating with centralized pump facilities), the results recommend consideration of lower chevron angles ($30-45^\circ$) despite modestly reduced thermal performance, as the pressure drop penalty associated with larger angles translates directly to substantial additional pumping costs. Conversely, for applications with abundant available pressure drop capacity (such as refrigeration cycles utilizing multiple stages of compression or chemical process applications with inherent flow resistance), higher chevron angles ($60-75^\circ$) provide superior heat transfer performance enabling more compact heat exchanger designs, thereby reducing material costs and spatial requirements (Bhasker *et al.*, 2026).

The thermal enhancement factor (TEF) defined as the ratio of actual heat transfer coefficient to equivalent flat plate baseline represents convenient metric for assessing net benefit of corrugation geometry incorporation. The current investigation demonstrated TEF ranging from approximately 1.15 to 1.44 depending on Reynolds number and chevron angle, indicating that corrugated plates provide substantially superior thermal performance

compared to alternative flat plate designs. These enhancements must be considered within context of increased manufacturing complexity and cost associated with corrugated geometry production, though contemporary manufacturing technologies have made corrugated plate production economically competitive for most applications (Amrizal Nalis *et al.*, 2023).

The observation that optimal chevron angle varies with Reynolds number has important implications for multi-operating-condition applications. In systems such as HVAC installations or solar thermal systems operating across variable loading conditions spanning 500-2500 Reynolds number range, fixed-geometry chevron plates necessarily represent compromise design balancing performance across operating range. The results suggest that moderate chevron angles (45-60°) provide relatively robust performance across broad Reynolds number ranges with performance degradation remaining modest compared to optimized-angle designs at specific operating points. Future development of variable-geometry heat exchanger concepts maintaining fixed geometric features while modifying flow paths through valve controls could potentially achieve superior performance across multi-operating-point scenarios by effectively modifying Reynolds number-dependent performance characteristics (Bougaa, El Allami and Tenghiri, 2025).

While the current investigation provided comprehensive characterization across chevron angle and Reynolds number variations, several limitations warrant acknowledgment and identification for future research:

First, the investigation examined single plate thickness (0.7 mm) and material composition (stainless steel SUS316L), whereas industrial applications frequently employ alternative thickness values (0.5-1.0 mm range) and materials (titanium, nickel alloys, copper) with different thermal and mechanical properties affecting overall system performance. Future investigations incorporating material and thickness variations would extend applicability of findings to broader industrial range (Muhammad Amin Harun *et al.*, 2021).

Second, single-phase liquid water systems were exclusively investigated, whereas industrial applications frequently involve multiphase flows (boiling, condensation), non-aqueous fluids, or slurry flows with substantially different thermal and physical properties. Chevron angle optimization for multiphase flows or high-viscosity fluids may differ substantially from single-phase water results due to phase distribution changes and viscous effects domination, respectively.

Third, the investigation examined symmetric chevron plate configurations whereas asymmetric chevron arrangements (with different angles on alternate plate sides) represent commercially available alternative offering potential for enhanced performance. Investigation of asymmetric geometries could reveal additional optimization opportunities not apparent from symmetric configuration analysis.

Fourth, fouling phenomena were not incorporated in either experimental or numerical investigations, whereas industrial applications often encounter calcium carbonate scale formation, particulate deposition, or biological growth substantially affecting actual performance relative to clean-surface predictions. Future investigations incorporating realistic fouling scenarios would enhance practical applicability

CONCLUSION

This comprehensive experimental and numerical investigation has established quantitative characterization of plate heat exchanger thermal-hydraulic performance across chevron angle variations from 30° to 75° and Reynolds number ranges from 500 to 2500. The principal findings include: First, overall heat transfer coefficient increases systematically with chevron angle elevation across all Reynolds numbers investigated, with improvements ranging from 7% (30°-to-45° increment) to 23% cumulative (30°-to-75° increment) at equivalent operational conditions. Heat transfer improvements result from three synergistic mechanisms: boundary layer disruption through corrugation geometry, secondary flow generation through chevron asymmetry, and turbulence intensity enhancement through corrugation-induced destabilization. The relative importance of these mechanisms varies with Reynolds number, with boundary layer disruption dominating in laminar-transitional regimes and turbulence generation dominating in fully turbulent conditions. Second, pressure drop penalties increase substantially with chevron angle, ranging from 10-30% increases at equivalent Reynolds numbers. The relationship between thermal benefits and hydraulic penalties becomes increasingly favorable at higher Reynolds numbers, where thermal improvements exceed pressure drop increases, whereas at lower Reynolds numbers pressure penalties dominate thermal benefits. This Reynolds number dependency necessitates application-specific optimization considering actual operational Reynolds number distribution. Third, comprehensive empirical correlations developed from experimental measurements provide quantitative relationships between Nusselt number, Reynolds number, Prandtl number, and chevron angle suitable for implementation in engineering design software and preliminary calculations. These correlations demonstrate statistical excellence ($R^2 > 0.97$) with satisfactory prediction accuracy across the investigated parameter ranges. Fourth, computational fluid dynamics simulations employing realizable k- ϵ turbulence models with enhanced wall treatment provide satisfactory quantitative predictions of both thermal and hydraulic performance (mean absolute percentage errors typically less than 5%), validating numerical methodology for prediction of geometric variation effects not encompassed by experimental investigations. The investigation demonstrates that optimized chevron angle selection requires consideration of specific application requirements, operational Reynolds number ranges, and economic trade-offs between thermal performance enhancement and pumping power requirements. For applications prioritizing pumping energy minimization, lower chevron angles (30-45°) prove optimal despite modestly reduced thermal performance. Conversely, applications with abundant pressure drop capacity benefit from higher chevron angles (60-75°) enabling more compact, cost-effective heat exchanger designs. The developed correlations, validated numerical models, and physical insights regarding heat transfer mechanisms provide scientific foundation enabling more effective plate heat exchanger design optimization in industrial applications ranging from energy recovery systems to advanced cooling technologies. Future investigations extending to multiphase flows, asymmetric geometries, fouling scenarios, and alternative materials and thicknesses would further enhance practical applicability and enable optimization of increasingly demanding thermal management applications.

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