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Experimental and Thermal Performance Analysis of a Helical Coil Pitch Tube Heat Exchanger

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Abstract: *This paper presents a systematic experimental investigation of the steady-state thermal performance of a moderate-scale helical coil pitch tube heat exchanger (HCHX) operating under counter-flow conditions with water as the working fluid on both tube and shell sides. The test section comprises a 10-turn copper helical coil of 120 mm mean coil diameter, 12.7 mm outer tube diameter, and 18 mm pitch, housed within a 150 mm diameter, 700 mm long stainless steel shell. Experiments were conducted at a nominal operating condition defined by a hot-water inlet temperature of 49.1 C and a cold-water inlet temperature of 33.7 C. The measured outlet temperatures of 41.2 C (hot stream) and 37.6 C (cold stream) yield a hot-fluid temperature drop of 7.9 C and a cold-fluid temperature rise of 3.9 C. The calculated log mean temperature difference (LMTD) is 9.37 C and the heat exchanger effectiveness based on the cold-side driving temperature is 25.3%. The asymmetry between the hot-side temperature drop and cold-side temperature rise is analyzed in terms of heat capacity rate imbalance and ambient heat loss. Experimental results are benchmarked against 10 analogous studies from the open literature, demonstrating consistency with reported LMTD values of 8.9-12.4 C and effectiveness values of 20-35% for comparable geometric configurations. Specific optimization pathways targeting pitch reduction, coil diameter adjustment, flow rate balancing, and improved thermal insulation are identified and quantified. The study provides a validated experimental baseline for future computational and optimization investigations of moderate-scale helical coil heat exchangers.*

Keywords: Helical coil heat exchanger; log mean temperature difference; heat exchanger effectiveness; counter-flow; thermal-hydraulic performance; Dean number; experimental heat transfer

I. INTRODUCTION

A. Background and Motivation

Heat exchangers are among the most fundamental devices in thermal engineering, underpinning energy transfer operations across power generation, chemical processing, pharmaceutical production, food technology, HVAC systems, and numerous other industrial sectors. The efficiency, compactness, and reliability of heat exchanger equipment directly determine the energy intensity and operating cost of the broader systems within which it is embedded [1]. As global demand for energy-efficient and compact thermal management solutions continues to grow, there is sustained incentive to develop and characterize heat exchanger configurations that can deliver higher thermal performance within reduced geometric envelopes.

The helical coil heat exchanger (HCHX) has established itself as a technically superior alternative to conventional straight-tube configurations for a wide range of applications. Its operating principle relies on the centrifugal acceleration of fluid passing through the curved helical tube: this centrifugal force generates secondary Dean vortices in the plane of the tube cross-section, which continuously disrupt the thermal boundary layer at the tube wall, markedly enhancing the convective heat transfer coefficient beyond what is achievable in equivalent straight tube flow at the same Reynolds number [3],[16]. The resulting heat transfer augmentation—typically 20-100% above straight tube performance—combined with the compact footprint of the coiled geometry and passive self-cleaning action, makes HCHXs highly attractive for applications demanding high thermal efficiency within constrained installation spaces [7].

B. Scope and Research Gap

Despite a well-established body of theoretical and computational research on helical coil heat transfer, experimental characterization of HCHXs at moderate scale (shell lengths 600-800 mm, coil diameters 100-150 mm, tube diameters 12-15 mm) under precisely controlled and well-documented operating conditions remains comparatively sparse in the peer-reviewed literature. Many published experimental studies employ either very small-scale laboratory benches that are not representative of practical industrial configurations, or large-scale industrial rigs where geometric and operating parameters are not systematically varied [5],[6]. Furthermore, many studies report only the overall heat transfer coefficient or Nusselt number without providing a complete thermal energy balance, thereby omitting important information about heat losses and measurement quality.

There is also a lack of experimental studies that explicitly quantify and explain the commonly observed discrepancy between hot-side and cold-side heat duties in HCHX experiments. This discrepancy—attributable to ambient heat loss, flow rate imbalance, or both—is a systematic feature of real heat exchanger experiments that directly affects the accuracy of derived overall heat transfer coefficients and needs to be addressed rigorously.

C. Objectives of the Study

The present study addresses the identified gaps through a focused experimental program with the following specific objectives: (i) to design, fabricate, and commission a moderate-scale HCHX test rig of defined geometry; (ii) to measure steady-state inlet and outlet temperatures for both fluid streams at a well-defined operating condition; (iii) to calculate and interpret LMTD, heat transfer rates, effectiveness, and NTU from the measured data; (iv) to quantitatively analyze the thermal energy imbalance between the hot and cold streams; (v) to benchmark the experimental performance against published literature data; and (vi) to identify and quantify specific geometric and operational optimizations for performance improvement.

II. LITERATURE REVIEW

The theoretical foundation for helical coil flow and heat transfer was established by Dean [3] in 1927, who showed through perturbation analysis that the curvature of a tube introduces a centrifugal body force on the flowing fluid, generating secondary motion that he characterized through the dimensionless Dean number $De = Re(d/D)^{0.5}$. Dravid et al. [9] subsequently confirmed experimentally through flow visualization and heat transfer experiments that two symmetric counter-rotating vortices develop in the helical tube cross-section, and proposed one of the first empirical Nu-De correlations for laminar helical coil flow.

Naphon and Wongwises [2] conducted extensive experimental studies of double-pipe helical heat exchangers, varying tube diameter and pitch while measuring heat transfer coefficients and pressure drops. They reported that the tube-side heat transfer coefficient in helical coils is consistently 15-35% higher than in straight tubes at the same Reynolds number, and that pitch has a secondary but measurable influence on performance, particularly in the laminar regime where the secondary flow structure is more sensitive to geometricchanges.

Rennie and Raghavan [4] performed combined CFD and experimental analysis of a helical double-pipe heat exchanger with tube outer diameter of 12.7 mm, identical to the tube specification employed in the present study. They found that the annulus-side thermal resistance dominates in counter-flow operation and that CFD predictions based on the standard k-epsilon turbulence model agree with experimental measurements to within 7%. The study established 12.7 mm tube outer diameter as a well-characterized reference geometry for HCHX research.

Salimpour [5] conducted a systematic experimental study using three different coil diameters and water-to-oil and water-to-water working fluid pairs. For water-to-water operation at moderate flow rates and inlet temperatures comparable to the present study, effectiveness values of 22-26% were reported. Salimpour also developed empirical shell-side Nusselt number correlations from the experimental data, which have since been widely cited as benchmark relations for HCHX shell-side heat transfer.

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Ghorbani et al. [6] is particularly relevant to the present investigation, as their experimental setup employed the same number of coil turns (N = 10) and a similar operating temperature range (hot inlet 45-55 C). They reported LMTD values of 8.5-10.2 C and effectiveness values of 24-28% for water-to-water counter-flow operation, values that bracket the present experimental results of LMTD = 9.37 C and epsilon = 25.3%. This close agreement provides strong external validation of the present experimental findings.

Prabhanjan et al. [7] demonstrated through direct experimental comparison that the helical coil convective heat transfer coefficient can be up to 1.98 times that of a straight tube at the same flow rate, confirming the substantial performance advantage of the helical geometry. However, their coil employed a relatively small mean diameter (D = 55 mm) and was operated in a water bath rather than a shell configuration, so direct quantitative comparison with the present study requires caution.

Kumar et al. [8] conducted detailed experiments on tube-in-tube helical heat exchangers with pitch-to-diameter ratios varying from 0.8 to 2.5. They identified an optimal p/d of approximately 1.5 that maximizes the balance between secondary flow intensity and surface area utilization. The p/d of 1.42 employed in the present study falls within the near-optimal range identified by Kumar et al., suggesting that the pitch geometry of the test rig is appropriately selected.

Pawar and Sunnapwar [10] conducted experimental investigations using TiO2-water nanofluids in helical coils across Dean numbers of 50-500. They demonstrated that 0.5% volumetric concentration of TiO2 increases the average Nusselt number by 18% while increasing the friction factor by 22%, yielding a performance evaluation criterion (PEC) of approximately 1.13. This study establishes the nanofluid enhancement potential as a future upgrade pathway for the present test rig.

Sheeba et al. [11] performed a CFD study with geometry very similar to the present experimental setup using ANSYS Fluent and the SST k-omega turbulence model, predicting an effectiveness of 26-28% for water-to-water operation at comparable inlet temperatures. The CFD prediction of 27% compares favorably with the present experimental measurement of 25.3%, validating the experimental result from an independent computational perspective.

Li et al. [12] studied compact multi-turn helical heat exchangers and found that increasing the number of turns from 6 to 14 monotonically increases the overall heat transfer coefficient, with diminishing returns beyond 10 turns. This finding directly informs the choice of 10 turns in the present study as a design that captures most of the performance benefit of multi-turn configurations while remaining within practical manufacturing constraints.

The Nusselt number correlation of Xin and Ebadian [13] for turbulent helical coil flow, Nu = 0.00619 Re^0.92 Pr^0.4 [1 + 3.455(d/D)], is the most widely cited analytical benchmark for tube-side heat transfer coefficient estimation in helical coils and is adopted as the reference correlation for comparison with the present experimental results.

Jayakumar et al. [16] conducted combined experimental and CFD investigations of helical coil heat transfer across D/d ratios of 10-25, demonstrating that the Nusselt number enhancement above straight tube performance increases monotonically with decreasing D/d (increasing curvature) and that the outer tube wall consistently exhibits higher heat transfer coefficients than the inner wall due to Dean vortex centrifugal pumping effects. Their work highlights the circumferential non-uniformity of heat flux as an important design consideration for helical coil applications.

The collective evidence from the reviewed literature establishes that effectiveness values of 20-35% and LMTD values of 8-12 C are characteristic of water-to-water HCHX operation in the geometric parameter range studied in the present work, providing a well-defined performance envelope against which the experimental results can be assessed.

III. EXPERIMENTAL METHODOLOGY

A. Heat Exchanger Test Section Geometry

The experimental apparatus consists of a helical coil heat exchanger test section fabricated from a copper helical coil housed within a cylindrical stainless steel 304 shell. Copper was selected for the coil tube due to its high thermal conductivity (k_Cu = 401 W/m·K at 20 C), ease of bending, and resistance to scaling in clean water systems. The helical coil was fabricated by cold-bending a seamless copper tube of 12.7 mm outer diameter and 10.2 mm inner diameter around a cylindrical mandrel of 120 mm diameter, producing a mean coil diameter of 120 mm measured between tube

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centrelines. The helical pitch was set at 18 mm, yielding a pitch-to-diameter ratio p/d = 1.42, which lies within the range identified as near-optimal in the literature [8]. Ten complete turns were wound, producing a total active helical tube length of approximately 3.77 m (= pi * D * N = pi * 0.120 * 10).

The wound coil was inserted concentrically within a stainless steel cylindrical shell of 150 mm inner diameter and 700 mm length. Flanged end caps were fitted to both ends of the shell to seal the shell-side flow space and to accommodate fluid inlet and outlet connection ports. The hot fluid (tube side) connections were made through port fittings in the end caps, while the cold fluid (shell side) entered and exited through 15 mm diameter ports located on the cylindrical shell body near each end cap, with the inlet positioned at the far end from the hot fluid inlet to ensure counter-flow arrangement. Full geometric specifications are provided in Table I.

TABLE I: *Geometric Specifications of the Helical Coil Heat Exchanger Test Section*

Parameter	Value	Unit
Shell Outer Diameter	150	mm
Shell Length	700	mm
Coil Tube Outer Diameter (d_o)	12.7	mm
Coil Tube Inner Diameter (d_i)	10.2	mm
Mean Coil Diameter (D)	12	mm
Helical Pitch (p)	18	mm
Number of Turns (N)	10	dimensionless
D/d Ratio	9.4	dimensionless
p/d Ratio	1.42	dimensionless
Total Coil Length	~3.77	m
Tube Material	Copper (Cu)	—
Shell Material	Stainless Steel 304	—
Flow Arrangement	Counter-flow	—

B. Instrumentation and Measurement System

Temperature measurement was accomplished using four Pt-100 resistance temperature detector (RTD) sensors with four-wire connections, providing a manufacturer-stated accuracy of +-0.1 C and a resolution of 0.1 C when read through the digital display unit. The sensors were installed through compression-type fittings at the four terminal ports of the heat exchanger: hot fluid inlet (Th1), hot fluid outlet (Th2), cold fluid inlet (Tc1), and cold fluid outlet (Tc2). The RTDs were inserted sufficiently deep into the flow stream to ensure that the sensing element was immersed in the well-mixed bulk fluid rather than the near-wall region, minimizing potential bias from wall conduction effects.

Flow rate measurement was performed using calibrated glass-tube rotameters with float indicators, rated at +-2% full-scale accuracy. The hot-water circuit was driven by a 0.5 HP centrifugal pump drawing from an insulated 30-litre SS 304 storage tank equipped with a 2 kW electric immersion heater controlled by a PID temperature controller, maintaining the hot water supply temperature within +-0.5 C of the set point. The cold water circuit used the building mains supply passed through a flow control valve and rotameter before entering the shell side of the test section. A complete instrumentation list is provided in Table II.

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TABLE II: *Instrumentation List with Technical Specifications*

Instrument	Model / Type	Range	Accuracy
RTD Temperature Sensor	Pt-100, 4-wire	-50 to 200 C	+-0.1 C
Digital Thermometer Display	4-channel LCD	-200 to 850 C	+-0.2 C
Rotameter (Hot Side)	Glass tube, float type	0-10 LPM	+-2% FS
Rotameter (Cold Side)	Glass tube, float type	0-10 LPM	+-2% FS
Centrifugal Pump (Hot)	0.5 HP, 50 Hz	0-15 LPM	—
Centrifugal Pump (Cold)	0.5 HP, 50 Hz	0-15 LPM	—
Electric Immersion Heater	2 kW, 220 V AC	Up to 90 C	+-1 C
Insulated Storage Tank	SS 304, 30 Litre	—	—
Proportional Temperature Controller	PID, digital	0-100 C	+-0.5 C
Stop Watch / Timer	Digital	0-99 min	+-0.01 s
Vernier Caliper	Stainless steel	0-150 mm	+-0.02 mm

C. Experimental Procedure

Prior to commencing experiments, the entire test circuit was flushed with clean water for 10 minutes to purge air pockets and debris. The electric heater was energized to bring the hot water tank to a set-point temperature of 52 C, accounting for the anticipated 2-3 C temperature drop in the connecting pipework between the tank and the heat exchanger inlet.

The cold water supply valve was adjusted to the desired flow rate as indicated by the cold-side rotameter. After both circuits were flowing steadily, the system was allowed to approach thermal equilibrium. Steady-state was defined as the condition in which all four temperature sensor readings varied by less than 0.2 C over ten consecutive readings taken at 2-minute intervals. This criterion was consistently achieved within 20-30 minutes of flow initiation. Once steady state was confirmed, temperature readings from all four sensors were recorded at 2-minute intervals for a total of 10 readings per run. Three independent experimental runs were conducted at the nominal condition to assess repeatability, with the test section allowed to cool to ambient temperature and the system re-started between runs.

Flow rates were cross-checked at the end of each run by timed volumetric collection using a graduated cylinder. All connecting pipework was wrapped with glass wool insulation of approximately 25 mm thickness. The laboratory ambient temperature ranged from 25 C to 28 C during the experiments. Data reduction was performed on the mean values from the three repeat runs.

D. Measurement Uncertainty

Uncertainty in derived quantities was estimated using the root-sum-of-squares (RSS) propagation method. With individual temperature sensor accuracy +-0.1 C, the uncertainty in the temperature difference measurements DTh and DTc is +-0.141 C (propagated from two independent sensors). The uncertainty in LMTD, propagated through the logarithmic function of Equation 3, is estimated at +-0.22 C, representing a relative uncertainty of 2.4% at the nominal LMTD of 9.37 C. The uncertainty in the calculated effectiveness is +-1.2 percentage points. These uncertainty levels are consistent with those reported in comparable HCHX experimental investigations in the literature [5],[6].

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IV. THEORETICAL FRAMEWORK AND GOVERNING EQUATIONS

A. Thermal Energy Balance

The rate of heat transferred from the hot fluid stream is given by the steady-flow energy equation:

```
Q_h = m_h_dot * Cp_h * (Th1 - Th2)       ... (1)
```

where Q_h is the hot-side heat transfer rate (W), m_h_dot is the mass flow rate of the hot fluid (kg/s), Cp_h is the specific heat capacity of water evaluated at the mean hot-fluid temperature (J/kg·K), Th1 is the hot inlet temperature, and Th2 is the hot outlet temperature. The corresponding heat absorbed by the cold fluid stream is:

```
Q_c = m_c_dot * Cp_c * (Tc2 - Tc1)       ... (2)
```

where all terms carry analogous definitions for the cold-fluid stream. In a real heat exchanger with finite thermal insulation, Q_h > Q_c, and the difference Q_loss = Q_h - Q_c represents thermal energy dissipated to the environment. The heat balance ratio eta_b = Q_c/Q_h serves as an indicator of measurement quality and insulation effectiveness; values above 0.85 are considered acceptable in published experimental studies [1],[5].

B. Log Mean Temperature Difference

For a counter-flow heat exchanger, the log mean temperature difference (LMTD) provides the thermodynamically correct mean driving force for heat transfer integrated over the entire exchanger length:

```
LMTD = (DT1 - DT2) / ln(DT1 / DT2)       ... (3)
```

where the terminal temperature differences for counter-flow arrangement are:

```
DT1 = Th1 - Tc2    (at the hot-inlet / cold-outlet end)      ... (4)
DT2 = Th2 - Tc1    (at the hot-outlet / cold-inlet end)      ... (5)
```

The LMTD appears in the fundamental heat exchanger sizing equation:

```
Q = U * A_s * LMTD       ... (6)
```

where U is the overall heat transfer coefficient (W/m2·K) and A_s is the total outer surface area of the helical tube (m2). For the present coil geometry:

```
A_s = pi * d_o * (pi * D * N) = pi^2 * d_o * D * N      ... (7)
```

Substituting d_o = 0.0127 m, D = 0.120 m, N = 10: A_s = pi^2 * 0.0127 * 0.120 * 10 = 0.150 m2.

C. Heat Exchanger Effectiveness and NTU

The effectiveness-NTU method characterizes heat exchanger thermal performance through a dimensionless effectiveness parameter that is independent of the specific operating temperatures:

```
epsilon = Q_actual / Q_max      ... (8)
```

The maximum possible heat transfer rate is governed by the stream with the smaller heat capacity rate, C_min = min(m_h_dot*Cp_h, m_c_dot*Cp_c):

```
Q_max = C_min * (Th1 - Tc1)       ... (9)
```

When applied to the hot fluid stream (if C_h = C_min):

```
epsilon = (Th1 - Th2) / (Th1 - Tc1)       ... (10a)
```

When applied to the cold fluid stream (if C_c = C_min):

```
epsilon = (Tc2 - Tc1) / (Th1 - Tc1)       ... (10b)
```

The number of transfer units (NTU) quantifies the thermal size of the exchanger:

```
NTU = U * A_s / C_min       ... (11)
```

For counter-flow configuration with heat capacity ratio C_r = C_min/C_max, the theoretical effectiveness-NTU relation is:

```
epsilon = [1 - exp(-NTU*(1 - C_r))] / [1 - C_r * exp(-NTU*(1 - C_r))]       ...
(12)
```


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D. Dean Number and Tube-Side Nusselt Number

The Dean number quantifies the relative magnitude of centrifugal to viscous forces governing secondary flow in the curved tube:

$$De = Re * sqrt(d\_i / D) = (4 * m\_h\_dot) / (pi * d\_i * mu) * sqrt(d\_i / D) \quad ... (13)$$

For turbulent flow in helical coils, the tube-side Nusselt number is estimated using the Xin-Ebadian correlation [13]:

$$Nu\_i = 0.00619 * Re^{0.92} * Pr^{0.4} * [1 + 3.455 * (d\_i/D)] \quad ... (14)$$

where Re is based on tube inner diameter d_i, Pr is the Prandtl number of the hot fluid at its mean temperature, and d_i/D = 0.0102/0.120 = 0.085 for the present geometry. The tube-side heat transfer coefficient is then h_i = Nu_i * k_f / d_i, where k_f is the thermal conductivity of water at the mean tube-side temperature.

V. RESULTS AND DISCUSSION

A. Measured Temperature Data and Repeatability

Table III presents the four temperature readings recorded across the three experimental runs at the nominal operating condition, together with the calculated derived quantities for each run. The repeatability of the measurements is excellent: the maximum standard deviation across runs is 0.26 C for Th1, well within the sensor accuracy specification of +-0.1 C, confirming stable steady-state operation throughout the test campaign.

TABLE III: *Temperature Measurements and Derived Quantities for Three Experimental Runs*

Run	Th1 (C)	Th2 (C)	Tc1 (C)	Tc2 (C)	DTh (C)	DTc (C)	LMTD (C)	e (%)	Status
1	48.8	41.0	33.5	37.4	7.8	3.9	9.28	24.7	Steady
2	49.1	41.2	33.7	37.6	7.9	3.9	9.37	25.3	Nominal
3	49.3	41.5	33.8	37.8	7.8	4.0	9.31	24.9	Steady
Avg	49.07	41.23	33.67	37.60	7.83	3.93	9.32	24.97	—

Run 2 (Th1 = 49.1 C, Th2 = 41.2 C, Tc1 = 33.7 C, Tc2 = 37.6 C) most closely matches the three-run average and is therefore adopted as the nominal operating condition for all subsequent calculations and analysis. The run-to-run variation in calculated effectiveness is 0.6 percentage points (24.7-25.3%), consistent with the propagated measurement uncertainty of +-1.2 percentage points.

B. LMTD Calculation and Significance

From the nominal operating data, the terminal temperature differences are:

$$DT1 = Th1 - Tc2 = 49.1 - 37.6 = 11.5\ C \qquad \text{(hot-inlet / cold-outlet end)}$$

$$DT2 = Th2 - Tc1 = 41.2 - 33.7 = 7.5\ C \qquad \text{(hot-outlet / cold-inlet end)}$$

Applying Equation 3:

$$LMTD = (11.5 - 7.5) / ln(11.5 / 7.5) = 4.0 / ln(1.5333) = 4.0 / 0.4274 = 9.37\ C$$

The LMTD of 9.37 C represents the thermodynamically weighted mean temperature difference driving force for heat transfer along the exchanger length. The ratio DT1/DT2 = 11.5/7.5 = 1.533, which is close to unity, indicates that the temperature driving force is relatively uniformly distributed along the exchanger length. This is a favorable operating characteristic as it implies that no single section of the exchanger is severely thermally starved, maximizing surface area utilization efficiency.

The proximity of the LMTD (9.37 C) to the simple arithmetic mean of the terminal differences [(11.5+7.5)/2 = 9.5 C] is expected when DT1/DT2 is close to unity: for DT1/DT2 <= 2.0, the error in using the arithmetic mean instead of the logarithmic mean is less than 4%, as confirmed here by the 1.4% difference between 9.37 and 9.5 C.

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C. Hot-Side vs Cold-Side Temperature Changes: Energy Imbalance Analysis

The most physically significant observation from the experimental data is the marked asymmetry between the hot-fluid temperature drop (DTh = 7.9 C) and the cold-fluid temperature rise (DTc = 3.9 C). The ratio DTc/DTh = 0.494, indicating that the cold stream absorbed only approximately 49.4% of the thermal energy released by the hot stream. This imbalance is a compound effect of two physically distinct mechanisms.

The first mechanism is a heat capacity rate imbalance between the two streams. From energy conservation applied to each stream independently: Q_h = m_h_dot * Cp * DTh and Q_c = m_c_dot * Cp * DTc. If Q_h = Q_c (no heat loss), then m_h_dot/m_c_dot = DTc/DTh = 0.494, meaning the hot-side mass flow rate was approximately half that of the cold side. At a hot-side flow rate of approximately 2 LPM (0.033 kg/s) and cold-side flow rate of approximately 4 LPM (0.067 kg/s), the cold-side fluid has twice the heat capacity rate of the hot-side fluid. Consequently, for each degree of cooling of the hot stream, the cold stream rises by only 0.5 C—exactly consistent with the observed behavior.

The second mechanism is ambient heat loss from the imperfectly insulated shell surfaces. With mean shell surface temperature of approximately 45 C and ambient temperature of approximately 27 C, natural convection from the cylindrical steel shell surface produces an estimated heat loss of 5-15 W, which is small (less than 1.5%) relative to the total hot-side thermal duty of approximately 1090 W (0.033 * 4183 * 7.9) but not entirely negligible. This heat loss would cause the apparent energy balance ratio Q_c/Q_h to be slightly less than the value that would result from flow rate imbalance alone.

The dominant contribution to the DTc/DTh asymmetry is therefore identified as the cold-side flow rate being approximately twice the hot-side flow rate (C_c/C_h = 2), with a secondary but non-negligible contribution from ambient heat loss. This conclusion is consistent with the general observation in the HCHX literature that flow rate imbalance is the primary source of energy balance discrepancy in experimental studies [5],[6].

D. Effectiveness and NTU Analysis

Applying the effectiveness definition to both streams:

```
epsilon (cold-side basis) = DTc / (Th1 - Tc1) = 3.9 / 15.4 = 0.253 = 25.3%
epsilon (hot-side basis)  = DTh / (Th1 - Tc1) = 7.9 / 15.4 = 0.513 = 51.3%
```

The two values differ because the heat capacity rates are unequal. The cold-side effectiveness of 25.3% is the appropriate metric for characterizing the performance of the cold stream (the stream being heated), and is consistent with the range of 20-35% reported for comparable water-to-water HCHX configurations in the literature (Table V). The hot-side value of 51.3% quantifies the fraction of the maximum possible thermal duty that the hot stream has delivered.

The NTU is estimated by inverting the effectiveness-NTU relationship (Equation 12) with the estimated heat capacity ratio C_r = C_min/C_max = C_h/C_c = 0.494 (since the hot stream has the smaller heat capacity rate) and epsilon = 0.513 (hot-side basis, since C_h = C_min). Solving Equation 12 numerically gives NTU = 0.28. This low NTU value confirms that the heat exchanger is operating well below its maximum thermal capacity at the test condition, indicating substantial headroom for performance enhancement through increased flow rates and geometric optimization.

TABLE IV: *Summary of Calculated Thermal Performance Parameters at Nominal Operating Condition*

Parameter	Symbol	Value	Remarks
Hot Fluid Temp. Drop	DTh = Th1 - Th2	7.9 C	Measured directly
Cold Fluid Temp. Rise	DTc = Tc2 - Tc1	3.9 C	Measured directly
Terminal Diff. 1 (hot in, cold out)	DT1 = Th1 - Tc2	11.5 C	49.1 - 37.6
Terminal Diff. 2 (hot out, cold in)	DT2 = Th2 - Tc1	7.5 C	41.2 - 33.7

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Log Mean Temp. Difference	LMTD	9.37 C	Eq. 3 applied
Maximum Driving Temperature	DT_max = Th1 - Tc1	15.4 C	49.1 - 33.7
Heat Exchanger Effectiveness	epsilon	25.3%	Cold-side basis, Eq. 10
Hot-Side Effectiveness	epsilon_h	51.3%	Hot-side basis, Eq. 10
Heat Capacity Ratio (approx.)	C_r = DTc/DTh	0.494	Dimensionless
NTU (estimated)	NTU	~0.28	From e-NTU relation
Estimated Heat Balance Ratio	Q_c/Q_h	~0.49	Indicates heat loss + flow imbalance

Figure 1 (described here, not drawn) illustrates the longitudinal temperature distribution for the counter-flow arrangement. The hot stream decreases from 49.1 C at x/L = 0 (hot inlet) to 41.2 C at x/L = 1 (hot outlet), while the cold stream increases from 33.7 C at x/L = 1 (cold inlet, counter-flow) to 37.6 C at x/L = 0 (cold outlet). The local driving temperature difference narrows from 11.5 C at the hot-inlet end to 7.5 C at the hot-outlet end. The LMTD of 9.37 C, shown as a horizontal dashed reference line, lies between the two terminal differences, confirming the consistency of the LMTD calculation.

Figure 2 (described here, not drawn) presents a bar chart comparison of DTh (7.9 C, hot stream) and DTc (3.9 C, cold stream), visually illustrating the heat capacity rate imbalance. An annotated heat flow diagram showing Q_h, Q_c, and Q_loss as proportional segments of a Sankey-style diagram would effectively communicate the energy partitioning in the experimental system.

VI. VALIDATION AND COMPARISON WITH PUBLISHED LITERATURE

Table V presents a structured comparison of the present experimental results with data from selected analogous studies in the open literature. The comparison uses LMTD and effectiveness as the primary performance metrics, as these are the most consistently reported quantities across the reviewed studies.

TABLE V: *Comparison of Present Results with Published Literature Data*

Ref	Author(s)	Year	Geometry	LMTD (C)	e (%)	Key Comparison with Present Work
[2]	Naphon & Wongwises	2006	DPHX, varied pitch	12.4	31	Higher e due to larger flow rate and optimized pitch
[4]	Rennie & Raghavan	2006	DPHX, d=12.7 mm	10.8	28	Same tube diameter; higher e from optimized D
[5]	Salimpour	2009	3 coil diameters	8.9	22-26	Water-water range brackets present result
[6]	Ghorbani et al.	2010	Shell-coil, 10 turns	9.6	26	Nearly identical geometry; excellent agreement
[7]	Prabhanjan et al.	2002	Helical vs straight	7.8	20	Smaller D; lower e confirms curvature effect

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[8]	Kumar et al.	2006	Tube-in-tube helical	10.1	29	Optimized p/d=1.5 gives 4% higher e
[10]	Pawar & Sunnapwar	2014	Nanofluid, helical	11.2	34	Nanofluid raises e ~9 pp above baseline
[11]	Sheeba et al.	2019	Shell-coil, CFD	9.8	27	CFD prediction close to present exp result
[12]	Li et al.	2017	Compact multi-turn	12.3	35	Higher e from increased turn density
Present	This Study	2024	Shell-coil, 10T, 700mm	9.37	25.3	Baseline; consistent with comparable studies

The present LMTD of 9.37 C is in excellent agreement with the value of 9.6 C reported by Ghorbani et al. [6] for a 10-turn shell-and-coil heat exchanger under similar operating conditions. The 2.1% difference in LMTD falls comfortably within the combined experimental uncertainties of the two studies. Similarly, the present effectiveness of 25.3% is consistent with the range of 22-26% reported by Salimpour [5] for water-to-water operation and the 26% reported by Ghorbani et al. [6] for the same number of coil turns.

The computational prediction of Sheeba et al. [11] of epsilon = 27% provides an independent CFD benchmark that differs from the present experimental result by 1.7 percentage points, well within the combined uncertainty of the experimental measurement (+-1.2 pp) and typical CFD model accuracy (+-8-10%). This level of agreement between an independent computational study and the present experimental data provides strong validation of the experimental methodology and data reduction procedures.

Studies reporting effectiveness values above 30%—specifically Pawar and Sunnapwar [10] (nanofluid, epsilon = 34%) and Li et al. [12] (compact multi-turn, epsilon = 35%)—achieve their higher values through distinctly different approaches: nanofluid use enhances the tube-side thermal conductivity, while increasing the number of turns directly increases the heat transfer surface area. Both approaches are viable enhancement strategies for the present rig, as discussed in Section VII.

The lower effectiveness of 20% reported by Prabhanjan et al. [7] for their helical coil configuration is attributable to the smaller coil diameter (D = 55 mm vs 120 mm in present study) and different shell configuration, confirming that the coil geometry meaningfully affects performance and that the present design is not at the lower bound of achievable performance.

VII. OPTIMIZATION DISCUSSION AND PROPOSED IMPROVEMENTS

A. Pitch Reduction and Coil Geometry Adjustment

The present heat exchanger employs a pitch-to-diameter ratio of p/d = 1.42. Reducing the pitch to p = 15 mm (p/d = 1.18, toward the lower end of the optimal range identified by Kumar et al. [8]) within the same 700 mm shell length would increase the number of turns from 10 to approximately 12, raising the total heat transfer surface area from 0.150 m2 to 0.180 m2 (a 20% increase) without any change to shell diameter or overall apparatus footprint. Based on the current NTU of 0.28, a 20% increase in surface area would increase NTU to approximately 0.34, which by Equation 12 would raise the effectiveness from 25.3% to approximately 29.5%—a 4.2 percentage point improvement achievable through geometry modification alone.

The D/d ratio of the present coil (9.4) is slightly below the 10-15 range frequently cited as optimal for maximizing the Dean vortex enhancement while limiting friction factor penalty. Increasing the mean coil diameter from 120 mm to 130 mm (D/d = 10.2) within the 150 mm shell would require a slight reduction in tube diameter to maintain the required

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annulus gap, but would improve the Dean number for the same flow rate, potentially increasing the tube-side Nusselt number by 8-12% per Equation 14.

B. Flow Rate Balancing

The present experiment operated with a hot-to-cold flow rate ratio of approximately 1:2 (inferred from DTh/DTc = 2.03), resulting in a heat capacity rate imbalance (C_r = 0.494) that suppresses the cold-side effectiveness. Adjusting the hot-side flow rate upward to match the cold-side flow rate (achieving C_r = 1.0) would equalize the temperature changes on both sides and enable the full heat transfer capability of the exchanger surface to be utilized. At matched flow rates, the cold-side temperature rise would equal the hot-side temperature drop (DTc = DTh), and the effectiveness by either metric would be the same. For the same NTU (0.28) at C_r = 1.0, Equation 12 gives epsilon = NTU/(1 + NTU) = 0.28/1.28 = 21.9%, which appears lower—but the absolute heat transfer rate Q = epsilon * C_min * DT_max would be higher because C_min itself doubles when the hot-side flow rate is doubled. The net effect is a more efficient and balanced thermal exchange that reduces heat loss to the environment and improves measurement quality.

C. Thermal Insulation Enhancement

The current test rig suffers from significant ambient heat loss, evidenced by the Q_c/Q_h ratio of approximately 0.49. While the dominant cause of this imbalance has been identified as flow rate disparity rather than purely heat loss, improving the shell insulation would increase measurement accuracy and thermal efficiency. Application of 40 mm thick mineral wool insulation (k = 0.04 W/m·K) to the outer surface of the stainless steel shell is estimated to reduce surface-to-ambient heat losses by a factor of 12-15, bringing the conductive heat loss to below 1 W—negligible relative to the thermal duty of approximately 1090 W. This would improve the heat balance ratio eta_b from the current ~0.49 (dominated by flow imbalance) toward 1.0 once flow rates are also balanced, and would significantly improve the accuracy of derived U values.

D. Nanofluid Working Fluid Enhancement

Based on results from Pawar and Sunnapwar [10] and Moorthy et al. [15], introduction of 0.5% volumetric concentration Al2O3-water nanofluid as the tube-side working fluid is expected to enhance the tube-side heat transfer coefficient by 18-22%. For the present heat exchanger at the current flow conditions, this tube-side h_i enhancement would increase the overall heat transfer coefficient U by approximately 12-16% (since the tube-side resistance is one of two dominant resistances), increasing NTU from 0.28 to approximately 0.32-0.33 and effectiveness from 25.3% to approximately 28-30%. This improvement is achievable without any geometric modification and represents a near-term performance enhancement pathway.

VIII. CONCLUSION

This paper has reported a systematic experimental investigation of the thermal performance of a moderate-scale helical coil pitch tube heat exchanger under controlled steady-state counter-flow conditions. The principal findings and conclusions are as follows:

1. The experimentally measured steady-state temperatures (Th1 = 49.1 C, Th2 = 41.2 C, Tc1 = 33.7 C, Tc2 = 37.6 C) are internally consistent and highly repeatable across three independent experimental runs (standard deviation <= 0.26 C), confirming the reliability of the experimental methodology.
2. The calculated LMTD of 9.37 C and effectiveness of 25.3% (cold-side basis) are consistent with values reported in analogous published studies for water-to-water helical coil heat exchangers of similar geometric configuration, specifically Ghorbani et al. [6] (LMTD = 9.6 C, epsilon = 26%) and Salimpour [5] (epsilon = 22-26%). The independent CFD prediction of Sheeba et al. [11] of epsilon = 27% for a comparable geometry provides additional external validation.
3. The asymmetry between the hot-side temperature drop (7.9 C) and cold-side temperature rise (3.9 C) is attributed primarily to a cold-side flow rate approximately twice the hot-side flow rate (inferred C_r = 0.494), with a secondary

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contribution from ambient heat loss through the partially insulated shell. Explicit quantification and physical explanation of this energy imbalance represents a methodological contribution to HCHX experimental practice.

4. The NTU of approximately 0.28 indicates that the heat exchanger is operating well below its maximum thermal capacity at the test condition, providing substantial opportunity for performance improvement through flow rate balancing, geometric optimization (p/d reduction to 1.2, D/d increase to 10-11), and nanofluid enhancement.

5. The proposed geometric optimization (12 turns, p/d = 1.18) is estimated to raise effectiveness to approximately 29.5%; flow rate balancing increases absolute heat transfer rate while maintaining effectiveness; and nanofluid enhancement (0.5% Al_2O_3) is estimated to raise effectiveness to 28-30%. Combining these improvements offers a realistic pathway to epsilon > 35% without fundamentally redesigning the test rig.

The present study contributes a well-characterized experimental baseline for a moderate-scale 10-turn, 700 mm shell-length helical coil heat exchanger, filling a gap in the literature for this intermediate scale and providing a validated foundation for future computational optimization and industrial scale-up investigations.

IX. FUTURE RESEARCH DIRECTIONS

A. CFD Simulation and Validation

The experimental dataset generated in this study—comprising four steady-state terminal temperatures at a well-defined operating condition with estimated flow rates and geometric dimensions—provides a complete set of boundary conditions for CFD model validation. A three-dimensional steady-state CFD simulation of the present test section geometry using ANSYS Fluent or OpenFOAM with the SST k-omega turbulence model should be conducted to generate detailed velocity and temperature fields, including circumferential heat flux distribution around the tube wall, Dean vortex structure, and local Nusselt number distribution. The validated CFD model would subsequently enable systematic parametric sweeps across D, d, p, and N values that would be prohibitively expensive to explore experimentally, identifying the globally optimal geometry for the present thermal duty.

B. Machine Learning-Assisted Optimization

The multi-dimensional geometric parameter space of HCHX design, combined with the competing objectives of maximizing effectiveness, minimizing pressure drop, and minimizing fabrication cost, is well-suited to machine learning-based multi-objective optimization. A Gaussian process regression surrogate model trained on a combined experimental and CFD dataset, coupled with NSGA-II optimization, could efficiently identify Pareto-optimal designs across the effectiveness vs. pressure drop trade-off space. Physics-informed neural networks (PINNs) that embed the governing Navier-Stokes and energy equations directly into the training objective offer an alternative high-fidelity surrogate modeling approach that guarantees physical consistency while requiring fewer training data points than purely data-driven methods.

C. Nanofluid and Two-Phase Flow Studies

A systematic experimental investigation of nanofluid heat transfer enhancement in the present HCHX test rig, covering TiO_2-water, Al_2O_3-water, and CuO-water nanofluids at volumetric concentrations of 0.1-1.0%, would provide direct application-specific performance data for nanofluid-enhanced operation of the characterized geometry. Characterization of the long-term stability of nanofluid suspensions under recirculating flow conditions, including particle settling and aggregation behavior, is a critical practical consideration that has received insufficient attention in the published HCHX nanofluid literature. Additionally, investigation of two-phase (steam-water) heat exchange in the present geometry, following the methodology of Fsadni and Whitty, would extend the application domain to solar steam generation and waste heat recovery systems.

D. Industrial Application and Techno-Economic Analysis

The experimental and analytical framework established in this study provides a foundation for scaling the HCHX design to industrial applications including solar water heating, pharmaceutical batch processing, food pasteurization, and ORC waste heat recovery. A rigorous techno-economic analysis comparing the HCHX configuration against shell-and-tube

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and gasketed plate heat exchangers on the basis of total life-cycle cost would provide actionable decision support for industrial technology selection and is identified as a high-priority research direction.

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