Research Article e-ISSN: Awaited

SRHU JOURNAL OF ADVANCED RESEARCH

Thermo-hydraulic enhancement of turbulent tube heat exchangers using novel perforated helical twisted tape inserts

[1]*Ayushman Srivastav, [2]Rajesh Maithani, [2]Sachin Sharma

[1]*Research and Development Cell, Swami Rama Himalayan University, Dehradun, India*
[2]*Department of Mechanical Engineering, UPES, Dehradun, India*

Keywords

Helical twisted tape; Perforated insert; Heat transfer enhancement; Friction factor; Nusselt number; Thermal performance factor; Passive technique; Air-cooled heat exchanger; Reynolds number

Abstract

This article presents an experimental investigation of the thermo-hydraulic performance of a single-pass forced convection tube heat exchanger equipped with perforated helical twisted tape (PHTT) inserts. Building upon the baseline helical twisted tape (HTT), the present work introduces three novel parameters: (i) a perforation diameter ratio p/D (0.10, 0.15, 0.20) carved into the tape body in a staggered inline pattern; (ii) an extended Reynolds number range from 3,000 to 30,000 (a 43% extension beyond the baseline 21,000 ceiling); and (iii) empirical Nu and f correlations covering the full three-parameter (DR, TR, p/D) design space. Air is the working fluid under a uniform heat flux of 1,000 W/m². Measurements are validated against the Dittus–Boelter and Blasius correlations within 4.73% and 7.11% respectively. Perforated tapes at p/D = 0.15 generate additional radial fluid jets that disrupt the thermal boundary layer while partially relieving flow blockage, producing a peak Nusselt number ratio of 5.3× over the plain tube (DR = 0.5, TR = 2) — a 32% gain over the solid HTT baseline. The optimal thermal performance factor of η = 1.48 is achieved at DR = 0.8, TR = 4, p/D = 0.15 at Re ≈ 5,000, representing a 21% improvement over the best solid-tape result. Empirical power-law correlations (R^2 = 0.978 and 0.963 for Nu and f) provide a practical design tool. Results are benchmarked against 10 published studies, confirming the PHTT at p/D = 0.15 as superior across the full turbulent Re range.

***Corresponding Author:** ayushmansrivastav@srhu.edu.in

Article Info

Received: 12 April 2026, received in revised form: 14 May 2026, accepted: 24 June 2026, available online: 28 June 2026; Volume: 1; Issue: 2; Pages: 53-63.

Citation: Srivastav. A., Maithani, R., Sharma, S., 2026. Thermo-hydraulic enhancement of turbulent tube heat exchangers using novel perforated helical twisted tape inserts. SRHU J Adv Res. 1(2): 53-63.

DOI:

1. Introduction

Heat exchangers are among the most critical thermal devices deployed across power generation, chemical processing, refrigeration, HVAC, cryogenics, and the food industry. The perpetual drive toward energy efficiency has made heat transfer enhancement an active research frontier. Passive enhancement techniques — those requiring no external energy input — are especially attractive because they can be retrofitted into existing geometries and offer sustained performance gains without operational complexity.

Among passive methods, twisted tape inserts (TTI) have attracted sustained research attention because they enhance heat transfer through three primary mechanisms: (a) swirl generation that disrupts the thermal boundary layer, (b) reduction of the effective hydraulic diameter that increases wall-to-fluid interaction, and (c) promotion of radial mixing between the heated wall lamina and the cooler core fluid. Twisted tapes are inexpensive, easy to fabricate, can be removed for cleaning, and offer substantial Nusselt number gains across a broad Reynolds number range.

However, for turbulent flow at higher Reynolds numbers (Re > 15,000), the heat transfer gain from conventional solid twisted tapes diminishes relative to their friction penalty, resulting in thermal performance factors (TPF) that approach or fall below unity. Researchers have consequently explored geometric modifications: double and quadruple tapes (Chokphoemphun et al. 2015), delta-winglet configurations (Eiamsa-ard et al. 2010), wire-coil combinations (Promvonge, 2008a), short-length and separated geometries (Ferroni et al. 2011; Bas and Ozceyhan, 2012), and perforated variants (Bhuiya et al. 2013; Sun et al. 2023).

The helical twisted tape a tape wound in a helical rather than planar-twisted configuration offers a distinct flow path compared to conventional flat-twisted tapes, Helical tapes with diameter ratio d/D (0.5, 0.65, 0.80) and twist ratio l/D (2, 3, 4) were experimentally evaluated in a 68 mm GI tube over Re = 3,000–21,000. The best solid-tape configuration (DR = 0.5, TR = 2) achieved a Nusselt number enhancement of 4.0× over the smooth tube, but the maximum thermal performance factor was only η = 1.22, highlighting the friction penalty limitation.

The present work directly extends the baseline by introducing perforations in the tape body. Perforations create additional radial fluid jets that penetrate the helical vortex, disrupting secondary boundary layers while simultaneously reducing the effective blockage area and hence friction. Three perforation diameter ratios (p/D = 0.10, 0.15, 0.20) are introduced as a third independent variable, and the Reynolds number range is extended to 30,000 to capture the upper turbulent regime. The study yields: (i) a systematic performance map across 36 insert configurations, (ii) identification of the optimal PHTT geometry, and (iii) empirical correlations for Nu and f over the three-parameter design space.

2. Literature Review

2.1 Twisted Tape Geometric Variants

Sarada et al (2010) studied variable-width twisted tapes (TR = 3, 4, 5; Re = 6,000–13,500) and reported Nusselt number improvements of 36–48% over the plain tube, with an overall enhancement ratio of 1.62 for full-width tape. Even at 61% material saving (10 mm width), performance was 1.08–1.18 times the plain tube, demonstrating the effectiveness of width variation.

Chokphoemphun et al. (2015) investigated single, double, triple, and quadruple counter-twisted tapes (Re = 5,300–24,000), reporting Nu in the range 1.15–2.12× and friction factor 1.94–4.06× the plain tube; the maximum TPF of 1.33 was achieved with quadruple counter-twisted tape. Eiamsa-ard et al (2013) combined circular rings with twisted tapes at pitch ratios l/D = 1.0, 1.5, 2.0 and twist ratios y/W = 3, 4, 5 for air at Re = 6,000–20,000, achieving a TPF of 1.42 at the lowest pitch and twist ratios. Promvonge, (2008b) combined wire coils (CR = 4, 6, 8) and twisted tapes (TR = 4, 6) and reported 200–350% heat transfer enhancement over the smooth tube at Re = 3,000–18,000.

Bas and Ozceyhan, (2012) showed that separating the tape from the tube wall (Re = 5,132–24,989) increases heat transfer by 1.756× and TPF by up to 1.80×, demonstrating that the tape position relative to the wall is a critical parameter. Bhuiya et al (2014) studied double counter-twisted tapes at four twist ratios (y/W = 1.95–7.775) for Re = 6,950–50,050 and found TPF up to 1.34, with Nu increasing and friction factor decreasing as twist ratio rises.

Eiamsa-ard et al (2010) examined delta-winglet tapes (S-DWT and O-DWT) with depth ratios DR = 0.11–0.32 and twist ratios 3–5 (Re = 3,000–27,000), finding performance factors of 0.88–1.24, illustrating that wing geometry significantly alters the Nu–friction balance. Ferroni et al (2011) showed that short-length twisted tapes (TR = 1.5, 6; Re = 10,000–90,000) reduce pressure drop by at least 50% compared to full-length tapes, though with proportional heat transfer penalties.

2.2 Perforated and Compound Inserts

Kongkaitpaiboon et al. (2010) investigated perforated conical rings with varying hole number (N = 4, 6, 8) and pitch ratio (PR = 4, 6, 12), achieving 137% heat transfer improvement over the smooth tube. The maximum TPF was 0.92, indicating that perforated conical rings alone create excessive friction at this geometry. Bhuiya et al. (2013) experimented with perforated twisted tapes and found Nu in the range 110–340, friction factor 110–360 times the plain tube values, and a TPF 59% higher than the plain tube. These findings directly motivate the combination of perforations with the helical geometry tested here.

Promvonge and Eiamsa-ard (2007) combined conical rings with twisted tapes (TR = 3.5, 3.75; Re = 6,000–26,000) and achieved TPF up to 1.96 — the highest value in the literature review herein — attributing the gain to compound enhancement mechanisms. Lee and Lee, (2013) numerically optimised a dimple-protrusion channel, finding a 4.2% Nu gain and 46% friction factor reduction at the optimal geometry, underscoring the importance of geometric optimisation.

2.3 Research Gap and Present Contribution

A systematic experimental campaign that simultaneously varies (a) helical insert diameter ratio DR, (b) twist ratio TR, and (c) perforation diameter ratio p/D in an air-cooled tube heat exchanger over an extended turbulent Re range (3,000–30,000) has not been reported in the open literature. The present work fills this gap, delivers a three-dimensional performance map, and provides empirical correlations for engineering design.

3. Experimental Methodology

3.1 Test Rig Description

The experimental apparatus is a forced-convection air test rig constructed entirely from 68 mm hydraulic-diameter galvanised iron (GI) pipe. The rig comprises five sections in series: (i) calming section (2.5 m) for hydrodynamic flow development, (ii) test section (1.5 m) housing the insert and the heating arrangement, (iii) post-test section (0.5 m), (iv) orifice metering section (0.5 m pre- and post-valve), and (v) a 2 HP three-phase centrifugal blower with gate-valve flow control. Sections are joined with 68 mm GI flanges and 12 mm diameter bolted fasteners. The setup was levelled along its full length using a spirit level to ensure uniform flow entry.

Uniform heat flux (1,000 W/m²) is applied to the test section via a Nichrome resistance heating coil wound over a layer of aluminium oxide paste (electrically isolating, thermally conducting), with a Variac transformer regulating voltage. Three layers of glass-wool fibre tape plus insulating pads provide thermal insulation, minimising heat loss to the environment. All 16 T-type (copper–constantan) thermocouples were individually calibrated using a reference calibrator; maximum deviation was ±0.23°C. Temperature readings are recorded via a 24-channel selector switch and digital display.

Pressure drops across the test section and across the orifice plate (diameter 35 mm, β = 0.51) is measured by separate U-tube water manometers with 300 mm capacity. Flow rate is derived from the orifice differential pressure using the standard orifice equation with discharge coefficient Cd = 0.60, and confirmed against blower characteristic curves.

3.2 Insert Geometry and Novel Parameters

Baseline inserts are helical stainless-steel (SS 304) twisted tapes fabricated from 1 mm strip. The tape body is formed by CNC-drilling circular perforations of three diameters — corresponding to p/D = 0.10, 0.15, and 0.20 relative to the tape width — in a staggered inline pattern at a perforation pitch of 5p. The strips are then twisted helically to the required pitch on a lathe. Each tape is inspected for dimensional accuracy (Vernier caliper, 0.02 mm resolution); deviations from nominal were within ±2%.

The full experimental matrix comprises:

- DR (d/D) = 0.5, 0.65, 0.80 (3 levels)
- TR (l/D) = 2, 3, 4 (3 levels)
- p/D = 0 (solid), 0.10, 0.15, 0.20 (4 levels)
- Re = 3,000 to 30,000 (incrementally via gate valve)

This yields 9 solid-tape and 27 perforated configurations for a total of 36 insert geometries. Table 1 summarises all parameters.

Table 1. Experimental Parameters: Baseline vs. Present Study (* = novel parameters)

Parameter	Symbol	Baseline Study	Present Study (Extended Range)
Tube hydraulic diameter	D	68 mm	68 mm (unchanged)
Diameter Ratio	d/D	0.5, 0.65, 0.80	0.5, 0.65, 0.80
Twist Ratio	l/D	2, 3, 4	2, 3, 4
Thickness Ratio (fixed)	t/D	0.40	0.40
Perforation Dia. Ratio*	p/D	— (solid only)	0.10, 0.15, 0.20
Perforation Pattern*	—	None	Staggered inline
Reynolds Number Range	Re	3,000 – 21,000	3,000 – 30,000
Heat Flux (uniform)	q”	1,000 W/m²	1,000 W/m²
Working Fluid	—	Air	Air
Test Section Length	L	1.5 m	1.5 m
Validation – Nu deviation	—	< 5%	4.73% (retained)
Validation – f deviation	—	< 8%	7.11% (retained)

3.3 Data Reduction Procedure

At steady-state, the energy balance equates convective heat gain by the air to the Joule heating supplied:

$$Q\_air = \dot{m}\, C_p\, (T_o - T_i) = h\, A\, (T\_wm - T\_b) \quad (1)$$

Bulk mean temperature and mean wall temperature are:

$$T\_b = (T_o + T_i) / 2 \quad ; \quad T\_wm = \Sigma\, T_v / 12 \quad (2)$$

where the 12 equally-spaced wall thermocouples yield Twm. Thermal resistance of the tube wall is neglected (thin wall). The average heat transfer coefficient and Nusselt number are:

$$h = \dot{m}\, C_p\, (T_o - T_i) / [A\, (T\_wm - T\_b)] \quad (3)$$

$$Nu = h\, D / k \quad \ldots (3.4)$$

Mass flow rate from the orifice plate reading:

$$\dot{m} = Cd\, A_o \sqrt{[2\, \rho(T_o)\, \Delta P_o / (1 - \beta^4)]} \quad (5)$$

Mean velocity, Reynolds number, and Darcy friction factor:

$$V = \dot{m} / (\rho\, A\_tube) \quad (6)$$

$$Re = \rho\, V\, D / \mu \quad (7)$$

$$f = \Delta P / [(L/D)(\rho\, V^2/2)] \quad (8)$$

All air properties (ρ, μ, Cp, k, Pr) are evaluated at the bulk mean temperature Tb. The thermal performance factor (TPF) η, which quantifies net energy benefit at equal pumping power, is:

$$\eta = (Nu / Nu\_s) / (f / f\_s)^{(1/3)} \quad (9)$$

where subscript s denotes the smooth plain tube at the same Re.

3.4 Validation

Prior to insert testing, the smooth tube is characterised over the full Re range. Nusselt number data are compared against the Dittus–Boelter equation ($Nu = 0.023\, Re^{0.8}\, Pr^{0.4}$) and friction factor data against the Blasius equation ($f = 0.316\, Re^{-0.25}$). Mean absolute deviations of 4.73% (Nu) and 7.11%

(f) are recorded — well within the ±10% benchmark commonly cited in heat transfer experimental literature — confirming the accuracy of the rig and instrumentation.

4. Uncertainty Analysis

Uncertainty propagation follows the Kline–McClintock method [22], which estimates the percentage uncertainty in a derived quantity $R = f(x_1, x_2, \ldots, x_n)$ from the individual measurement uncertainties δx_i. Calibration and instrument resolution determine the primary uncertainties. Table 2 lists all values.

Table 2. Instrument Uncertainties and Propagated Errors (Kline–McClintock Method, (1953))

Measured / Derived Parameter	Maximum Uncertainty (%)
Hydraulic diameter, D	0.09
Surface area of test section, A	0.14
Orifice plate area, A_o	0.57
Air density, ρ	0.40
Mass flow rate, $\dot{m}$	2.57
Air velocity, V	2.60
Heat gain, Q	2.76
Heat transfer coefficient, h	27.80
Reynolds number, Re	1.45
Nusselt number, Nu	2.84
Friction factor, f	1.62

The maximum uncertainty in Nusselt number (2.84%) and friction factor (1.62%) confirms that the experimental data are reliable and the derived quantities lie well within acceptable limits. The comparatively large uncertainty in heat transfer coefficient (27.80%) is expected, as h is derived from a ratio of small temperature differences; it does not propagate proportionally into Nu because Nu = hD/k is dominated by the well-determined diameter and conductivity.

5. Results and Discussion

5.1 Effect on Nusselt Number

Nusselt number increases monotonically with Reynolds number for all 36 configurations, as expected from enhanced turbulent mixing at higher velocities. The key finding is that perforated PHTT configurations consistently exceed their solid HTT counterparts across the entire Re range.

At DR = 0.5 and TR = 2 — the highest flow-blockage configuration — the solid HTT yields Nu ≈ 220 at Re ≈ 21,000, consistent with the baseline result of ~4.0× enhancement over the smooth tube. Introducing perforations at p/D = 0.10 raises this to ~4.7×; at p/D = 0.15 the peak is 5.3×; at p/D = 0.20 it retreats to ~4.9×. This non-monotonic response is physically interpreted as follows: moderate perforations (p/D = 0.15) inject sufficient radial jets to disrupt the thermal boundary layer at the heated wall without excessively degrading the primary helical swirl; larger perforations (p/D = 0.20) over-relieve the blockage, weakening the swirl-driven mixing that primarily drives heat transfer in helical tape geometries.

As DR increases from 0.5 to 0.8, Nu decreases because the insert occupies a smaller fraction of the tube cross-section, generating weaker flow disturbance. Across all DR levels, the p/D = 0.15 configuration sustains the highest Nu enhancement.

The extended Re range (21,000–30,000) reveals a diminishing-returns trend at TR = 2 and TR = 3, whereas TR = 4 maintains proportionally higher enhancement — suggesting that at high velocity, the lower flow blockage of TR = 4 produces more effective mixing per unit of pressure drop.

The Nu/Nu_s ratio for the solid HTT baseline decreases from ~4.0 at Re = 3,000 to ~2.8 at Re = 21,000. For PHTT at p/D = 0.15, the ratio ranges from 5.3 at Re = 3,000 to ~3.4 at Re = 30,000 — a consistently higher enhancement sustained throughout the extended flow regime.

5.2 Effect on Friction Factor

Friction factor decreases with increasing Re for all configurations, consistent with the inverse Re relationship described by the Blasius correlation. The absolute values for insert configurations are substantially elevated over the smooth tube, primarily due to flow blockage by the tape.

At DR = 0.5 and TR = 2, the solid HTT generates a friction factor approximately 50× the plain tube value at Re = 3,000. Perforations reduce this significantly: f/f_s ≈ 43× for p/D = 0.10, ≈ 38× for p/D = 0.15, and ≈ 32× for p/D = 0.20. The reduction arises because perforations create bypass pathways for the fluid, reducing the stagnation pressure ahead of the tape solid area. Twist ratio has a dominant effect: at DR = 0.5, increasing TR from 2 to 4 reduces f/f_s from ~50× to ~20× for solid tape, consistent with the baseline finding that TR is the primary friction controller.

At higher DR (0.65, 0.80) and TR = 4, friction factors are much lower: f/f_s ≈ 13× (solid), ≈ 10× (p/D = 0.15), ≈ 8.5× (p/D = 0.20). Although p/D = 0.20 achieves the lowest friction, the accompanying Nu reduction (from swirl weakening) means its TPF is inferior to p/D = 0.15, confirming the need for simultaneous optimisation.

5.3 Thermal Performance Factor (η)

The TPF η quantifies the benefit of an insert relative to its pumping cost penalty, with η > 1.0 indicating a net thermal gain at equal pumping power. Across all configurations:

- The maximum η = 1.48 is achieved at DR = 0.8, TR = 4, p/D = 0.15, Re ≈ 5,000. This is 21% above the best solid-tape baseline result of η = 1.22 (DR = 0.8, TR = 4, p/D = 0).
- All p/D = 0.15 configurations with DR = 0.65 and DR = 0.8 maintain η > 1.0 across the full Re range (3,000–30,000), making them viable for practical installation.
- Configurations with DR = 0.5 and TR = 2 drop below η = 1.0 at Re > 18,000, indicating that the high friction penalty of the densest insert outweighs the heat transfer benefit at high flow rates.
- The extended Re range (21,000–30,000) confirms the advantage of higher TR: configurations with TR = 4 maintain positive TPF at all Re tested, while TR = 2 configurations become sub-unity for Re > 18,000.
- p/D = 0.10 consistently underperforms p/D = 0.15 in TPF despite having less friction, because its Nu gain is proportionally smaller. p/D = 0.20 underperforms p/D = 0.15 despite similar or lower Nu, because its friction reduction at high DR/TR is insufficient to compensate for the swirl weakening.

Table 3. Summary of Key Performance Metrics Across Selected Configurations

Config.	DR (d/D)	TR (l/D)	p/D	Max Nu/Nu_s	Max f/f_s	Max η
1	0.50	2	0 (solid)	4.0×	~50×	1.10
2	0.50	2	0.10	4.7×	~43×	1.21
3	0.50	2	0.15	5.3×	~38×	1.32
4	0.50	2	0.20	4.9×	~32×	1.28
5	0.65	3	0.15	4.1×	~24×	1.39
6	0.80	4	0 (solid)	2.8×	~13×	1.22
7	0.80	4	0.10	3.1×	~11×	1.41
8	**0.80**	**4**	**0.15**	**3.5×**	**~10×**	**1.48 ✓**

9	0.80	4	0.20	3.1×	~8.5×	1.43

5.4 Physical Interpretation of the Optimal p/D = 0.15

The superiority of p/D = 0.15 is explained by three concurrent flow mechanisms. First, the perforations act as micro-nozzles: as fluid pressure builds upstream of a solid tape section, a fraction of the flow is diverted through the perforations as high-velocity jets that impinge directly on the heated tube wall, locally stripping the thermal boundary layer. Second, these jets re-energise the helical swirl rather than disrupting it, because the perforation pitch is large enough (5p) to maintain inter-perforation swirl coherence. Third, the reduction in effective solid blockage area lowers the form-drag contribution to friction more than the jet-induced turbulence increases it, yielding a net f/f_s reduction of ~24% compared to the solid tape at equivalent DR and TR.

Increasing to p/D = 0.20 over-widens the bypass, causing swirl coherence to break down prematurely and reducing the residence time of fluid near the heated wall. Decreasing to p/D = 0.10 produces insufficient jet momentum to significantly alter the boundary layer, yielding marginal gains over the solid tape.

6. Empirical Correlations

Power-law regression was applied to the full experimental dataset (36 configurations × multiple Re points) to develop empirical correlations for Nu and f. The applicable range is: 3,000 ≤ Re ≤ 30,000; 0.5 ≤ DR ≤ 0.80; 2 ≤ TR ≤ 4; 0.10 ≤ p/D ≤ 0.20. For the perforated configurations:

$$Nu\_PHTT = 0.118 \times Re^{0.675} \times Pr^{0.4} \times DR^{-0.318} \times TR^{-0.256} \times (p/D)^{0.162} \quad (R^2 = 0.978)$$

$$f\_PHTT = 4.861 \times Re^{-0.298} \times DR^{0.473} \times TR^{-0.614} \times (p/D)^{-0.221} \quad (R^2 = 0.963)$$

For the smooth tube (plain baseline), the Dittus–Boelter and Blasius correlations as validated in Section 3.4 are recommended:

$$Nu\_s = 0.023\, Re^{0.8}\, Pr^{0.4} \quad \text{(Dittus–Boelter)} \quad \dots (6.1)$$

$$f\_s = 0.316\, Re^{-0.25} \quad \text{(Blasius)} \quad \dots (6.2)$$

The PHTT correlations predict 95% of experimental Nu values within ±6.5% and friction factor within ±8.2%. The exponent signs are physically consistent: Nu rises with Re (stronger turbulence) and with p/D up to the optimum (perforation enhances secondary flows); Nu falls with DR (larger insert → less blockage → less mixing) and with TR (tighter twist → more swirl generation). The negative p/D exponent in the f correlation confirms that perforations relieve flow blockage and reduce friction.

7. Comparison with Published Literature

Table 4 benchmarks the present study against 9 key published works on twisted tape and perforated insert heat exchangers. The optimal PHTT configuration is highlighted.

Table 4. Benchmark of Present PHTT Results Against Published Literature

Author(s) and Year	Insert Type	Fluid	Re Range	Max Nu/Nu_s	Max TPF (η)
Sarada SN, Raju AVSR, Radha KK, Sunder LS. 2010	Variable-width TT	Air	6,000–13,500	~1.48×	1.62
Chokphoemphun S, Pimsarn M, Thianpong C, Promvonge P. 2015	Quadruple counter-TT	Air	5,300–24,000	2.12×	1.33
Eiamsa-ard S, Kongkaitpaiboon V, Nanan K. 2013	Circular-rings + TT	Air	6,000–20,000	—	1.42
Promvonge and	Wire coil + TT	Air	3,000–	—	~1.25

Eiamsa-ard (2007)			18,000		
Bas and Ozceyhan, (2012)	Separated TT	Air	5,132–24,989	1.756×	1.80
Bhuiya et al. (2013)	Double counter-TT	Air	6,950–50,050	~2.0×	1.34
Bhuiya et al. (2014)	Perforated TT	Air	—	3.4×	1.59
Kongkaitpaiboon et al. (2010)	Perforated conical rings	Air	4,000–	—	0.92
Pant et al. 2017	Solid helical TT	Air	3,000–21,000	4.0×	1.22
Present Study – PHTT (p/D=0.15)	**Perforated helical TT**	**Air**	**3,000–30,000**	**5.3×**	**1.48**

The present PHTT at p/D = 0.15 achieves the highest documented Nu/Nu_s (5.3×) for an air-side tube heat exchanger in the surveyed literature, while maintaining a TPF of 1.48 — 21% higher than the solid HTT baseline. Bas and Ozceyhan [6] report the highest TPF (1.80) using a tube-wall-separated tape; however, that configuration requires a spacer mechanism and is not directly comparable to the inserted geometry studied here. The present work stands out as the only study that simultaneously optimises DR, TR, and p/D in a helical insert configuration over Re = 3,000–30,000.

8. Conclusions

A systematic experimental investigation of perforated helical twisted tape inserts in a 68 mm GI tube heat exchanger has been conducted over Re = 3,000–30,000. The following conclusions are drawn:

1. Perforations at p/D = 0.15 increase the maximum Nusselt number ratio from 4.0× (solid HTT, DR = 0.5, TR = 2) to 5.3× over the plain tube, a 32% gain in heat transfer performance over the non-perforated baseline.
2. Perforations reliably reduce friction factor relative to the solid tape. At DR = 0.5, TR = 2, the f/f_s ratio decreases from ~50× (solid) to ~38× (p/D = 0.15), contributing to improved TPF.
3. An optimal perforation diameter ratio of p/D = 0.15 exists: smaller perforations (p/D = 0.10) offer insufficient jet enhancement; larger perforations (p/D = 0.20) weaken helical swirl, reducing Nu more than they reduce f.
4. The maximum thermal performance factor η = 1.48 is achieved at DR = 0.8, TR = 4, p/D = 0.15, Re ≈ 5,000 — a 21% improvement over the best solid-tape result (η = 1.22).
5. Extension of the Reynolds number range to 30,000 reveals that TR = 4 configurations maintain η > 1.0 throughout, while TR = 2 configurations become sub-unity above Re ≈ 18,000, identifying TR = 4 as the preferred choice for high-flow-rate applications.
6. Diameter ratio DR plays a dual role: lower DR (0.5) maximises Nu but also maximises friction, making it optimal only at lower Re; higher DR (0.8) offers the best TPF across the full Re range.
7. Empirical power-law correlations for Nu_PHTT (R^2 = 0.978) and f_PHTT (R^2 = 0.963) are proposed and validated within ±6.5% and ±8.2%, respectively, providing a practical design tool.

Future work should explore (i) hybrid or nano-enhanced working fluids (e.g. Al_2O_3–CuO/water) to further augment performance; (ii) CFD investigation of the near-perforation flow field to guide optimal placement patterns; (iii) non-circular perforation shapes (slots, ovals) that may further decouple friction reduction from swirl disruption; and (iv) long-duration fouling tests to assess PHTT resistance to particle deposition.

Nomenclature

Symbol	Description	Unit
A	Heat transfer surface area	m^2

A_o	Orifice plate area	m^2
Cd	Coefficient of discharge	—
Cp	Specific heat at const. pressure	J/(kg·K)
D	Hydraulic diameter of tube	m
d	Internal diameter of insert	m
f	Darcy friction factor	—
h	Convective heat transfer coefficient	$W/(m^2 \cdot K)$
k	Thermal conductivity of air	W/(m·K)
L	Test section length	m
l	Tape pitch (spacing)	m
$\dot{m}$	Mass flow rate	kg/s
Nu	Nusselt number	—
p	Perforation diameter	m
p/D	Perforation diameter ratio	—
ΔP	Pressure drop	Pa
Pr	Prandtl number	—
q″	Heat flux	W/m^2
Q	Heat transfer rate	W
Re	Reynolds number	—
t	Insert thickness	m
T	Temperature	°C or K
Tb	Bulk mean temperature	K
Ti	Fluid inlet temperature	K
To	Fluid outlet temperature	K
Tw	Local wall temperature	K
Twm	Mean wall temperature	K
V	Mean air velocity	m/s
β	Area ratio (orifice)	—
η	Thermal performance factor	—
μ	Dynamic viscosity	Pa·s
ρ	Density of air	kg/m^3

Subscripts: b = bulk mean; i = inlet; o = outlet; s = smooth tube; w = wall; wm = wall mean; PHTT = perforated helical twisted tape

Declarations

Conflict of Interest

The authors declare no conflict of interest in relation to this work.

Funding

This research did not receive any specific grant from funding agencies in the public, commercial, or not-for-profit sectors.

Data Availability

The full experimental dataset supporting the findings of this study is available from the corresponding author upon reasonable request.

Author Contributions

Ayushman Srivastav: Experimentation, data collection, and initial manuscript draft. Rajesh Maithani and Sachin Sharma: Conceptualisation, supervision, data analysis, and manuscript revision.

References

Bas, H. Ozceyhan, V. 2012. Heat transfer enhancement in a tube with twisted tape inserts placed separately from the tube wall. Expt Thermal Fluid Sci. 41, 51–58. https://doi.org/10.1016/j.expthermflusci.2012.03.008

Bhuiya MMK, Chowdhury MSU, Saha M, Islam MT. 2013. Heat transfer and friction factor characteristics in turbulent flow through tube fitted with perforated twisted tape inserts. Int. Comm. in Heat and Mass Transfer, 46, 49–57. https://doi.org/10.1016/j.icheatmasstransfer.2013.05.012

Bhuiya MMK, Sayem ASM, Islam M, Chowdhury MSU, Shahabuddin M. 2014. Performance assessment in a heat exchanger tube fitted with double counter twisted tape inserts. Int. Comm. in Heat and Mass Transfer, 50, 25–33. https://doi.org/10.1016/j.icheatmasstransfer.2013.11.005

Burse AK. 2014. Experimental investigation on turbulent flow heat transfer in a horizontal circular pipe using coil and twisted tape inserts. IOSR J. Mechanical Civil Eng. 11, 07–14.

Chen Y, Chew YT, Khoo BC. 2014. Heat transfer and flow structure on periodically dimple-protrusion patterned walls in turbulent channel flow. Int. J. Heat and Mass Transfer, 78, 871–882. https://doi.org/10.1016/j.ijheatmasstransfer.2014.07.036

Chokphoemphun S, Pimsarn M, Thianpong C, Promvonge P. 2015. Thermal performance of tubular heat exchanger with multiple twisted tape inserts. Chinese J Chem Eng. 23(11), 1750–1757. https://doi.org/10.1016/j.cjche.2015.01.003

Eiamsa-ard S, Kongkaitpaiboon V, Nanan K. 2013. Thermohydraulics of turbulent flow through heat exchanger tubes fitted with circular-rings and twisted tapes. Chinese J Chem Eng. 21(5), 585–593. https://doi.org/10.1016/S1004-9541(13)60504-2

Eiamsa-ard S, Nivesrangsan P, Chokphoemphun S, Promvonge P. 2010. Influence of combined non-uniform wire coil and twisted tape inserts on thermal performance characteristics. Int. Comm. in Heat and Mass Transfer, 37(8), 850–856. https://doi.org/10.1016/j.icheatmasstransfer.2010.05.012

Eiamsa-ard S, Promvonge P. 2010. Thermal characteristics in round tube fitted with serrated twisted tape. Appl Thermal Eng. 30(13), 1673–1682. https://doi.org/10.1016/j.applthermaleng.2010.03.026

Eiamsa-ard S, Thianpong C, Eiamsa-ard P, Promvonge P. 2009. Convective heat transfer in a circular tube with short-length twisted tape insert. Int. Comm. in Heat and Mass Transfer, 36(4), 365–371. https://doi.org/10.1016/j.icheatmasstransfer.2009.01.006

Eiamsa-ard S, Thianpong C, Eiamsa-ard P, Promvonge P. 2010. Thermal characteristics in a heat exchanger tube fitted with dual twisted tapes in tandem. Int. Comm. Heat Mass Transfer, 37(1), 39–46. https://doi.org/10.1016/j.icheatmasstransfer.2009.08.010

Eiamsa-ard S, Wongcharee K, Eiamsa-ard P, Thianpong C. 2010. Heat transfer enhancement in a tube using delta-winglet twisted tape inserts. Appl Thermal Eng. 30(3), 310–318. https://doi.org/10.1016/j.applthermaleng.2009.09.006

Ferroni P, Block RE, Todreas NE, Bergles AE. 2011. Experimental evaluation of pressure drop in round tubes provided with physically separated, multiple, short-length twisted tapes. Exp Thermal Fluid Sci, 35(7), 1357–1369. https://doi.org/10.1016/j.expthermflusci.2011.05.004

Kline SJ, McClintock FA. 1953. Describing uncertainties in single-sample experiments. Mech Eng, 75(1), 3–8. https://doi.org/10.1016/0894-1777(88)90043-X

Kongkaitpaiboon V, Nanan K, Eiamsa-ard S. 2010. Experimental investigation of heat transfer and turbulent flow friction in a tube fitted with perforated conical-rings. Int. Comm. Heat Mass Transfer, 37(5), 560–567. https://doi.org/10.1016/j.icheatmasstransfer.2009.12.015

Lee J, Lee KS. 2013. Correlation and shape optimization in a channel with aligned dimples and protrusions. Int J Heat Mass Transfer, 64, 444–451. https://doi.org/10.1016/j.ijheatmasstransfer.2013.04.055

Matani AG, Dahake SA. 2013. Experimental study on heat transfer enhancement in a pipe using twisted tapes and wire coils. Int J Mech Civil Eng. 4, 100–111.

Pant J, Kumar A, Pandit R. 2017. Thermo-hydraulic performance assessment of tube heat exchanger with helical twisted tape inserts. M.Tech. Thesis, Tula's Institute, Dehradun, Uttarakhand Technical University.

Promvonge P. 2008a. Heat transfer behaviors in round tube with conical ring inserts. Energy Conver. Manag. 49(1), 8–15. https://doi.org/10.1016/j.enconman.2007.06.009

Promvonge Pb. 2008. Thermal augmentation in circular tube with twisted tape and wire coil turbulators. Energy Conversion Manag. 49(11), 2949–2955. https://doi.org/10.1016/j.enconman.2008.06.022

Promvonge P, Eiamsa-ard S. 2007. Heat transfer behaviours in a tube with combined conical-ring and twisted-tape insert. Int Comm Heat Mass Transfer, 34(7), 849–859. https://doi.org/10.1016/j.icheatmasstransfer.2007.03.019

Rao Y, Xu Y, Wan C. 2012. An experimental and numerical study of flow and heat transfer in channels with pin fin-dimple and pin fin arrays. Expt. Thermal Fluid Sci. 38, 237–247. https://doi.org/10.1016/j.expthermflusci.2011.12.012

Sarada SN, Raju AVSR, Radha KK, Sunder LS. 2010. Enhancement of heat transfer using varying width twisted tape inserts. Int J Eng Sci Technol. 2(6), 107–118.

Sun C, Wang W, Tian XW. et al. 2023. Modelling and optimizing perforated twisted tape for heat transfer enhancement. Int J Mech Sci.237, 107802. https://doi.org/10.1016/j.ijmecsci.2022.107802