



Experimental Investigation of Heat Transfer Enhancement in Double-Pipe Heat Exchangers

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Abstract

This paper presents a reproducible experimental methodology for quantifying passive heat-transfer enhancement in a concentric double-pipe heat exchanger. A water-to-water counter-flow apparatus is evaluated with a plain copper inner tube and full-length aluminium twisted-tape inserts having twist ratios (y/w) of 6, 4 and 3. The hot-side Reynolds number is varied from 5,000 to 15,000 while the annulus-side flow rate and inlet temperatures are controlled. Inlet/outlet temperatures, volume flow rates and inner-tube pressure drop are measured after steady state. Energy balance, logarithmic mean temperature difference, overall heat-transfer coefficient, Nusselt number, Darcy friction factor and a constant-pumping-power performance evaluation criterion are calculated. A calibration plan, repeat runs and uncertainty propagation are included. Representative validation data indicate that decreasing twist ratio raises Nusselt number through stronger swirl and radial mixing, but also increases friction loss. At $Re = 15,000$, the illustrative $y/w = 3$ case provides approximately 76% higher Nusselt number than the plain tube; the best thermohydraulic choice must therefore be selected using the performance criterion rather than heat transfer alone. The protocol is suitable for undergraduate and laboratory-scale research where repeatability, energy-balance closure and transparent comparison with a smooth-tube baseline are required.

Introduction

Double-pipe heat exchangers are simple, robust devices used when duties are modest, pressure is high, or maintenance access is important. Their principal limitation is the relatively low convective coefficient associated with a smooth circular passage. Passive inserts can disrupt the boundary layer without external power input, although the induced swirl and blockage increase pressure drop. Consequently, an enhancement device should be judged by both thermal gain and hydraulic penalty.

Classical tube-flow correlations provide the baseline against which enhancement is

measured. Sieder and Tate incorporated viscosity variation in internal convection [1], while later smooth-tube formulations remain useful for checking whether measured plain-tube values are physically reasonable. This study uses direct calorimetry and dimensionless analysis to compare four configurations under matched operating conditions.

The aim is to determine how twisted-tape twist ratio affects heat duty, convection, pressure loss and net thermohydraulic merit in counter-flow operation. The objectives are to: (i) establish a validated plain-tube baseline; (ii) measure steady-state temperatures, flow rates and

pressure drop; (iii) calculate Re , Nu , f , U and effectiveness; (iv) compare $y/w = 6, 4$ and 3 at common Reynolds numbers; and (v) quantify experimental uncertainty and repeatability.

Literature Review

Manglik and Bergles developed mechanistic correlations for twisted-tape flow, identifying swirl, increased flow path and tape blockage as coupled causes of heat-transfer and pressure-drop augmentation [2,3]. Their work established twist ratio as a controlling geometric variable and emphasized that enhancement cannot be interpreted independently of friction.

Naphon tested horizontal double pipes with and without twisted tape using hot and cold water. The reported apparatus used two tape pitches and mass-flow-rate variation, and proposed non-isothermal correlations for heat transfer and friction [4]. This work directly supports a double-pipe, water-to-water comparison and the use of simultaneous temperature and pressure measurements.

Bharadwaj, Khondge and Date studied tapes with alternate axes and showed that periodic disruption of the swirl structure can improve heat transfer, although geometry also changes resistance [5]. Eiamsa-ard and Promvonge examined regularly spaced twisted-tape elements, demonstrating that insert length and spacing provide additional design variables beyond conventional full-length tape [6]. Murugesan and co-workers investigated V-cut

tapes and reported the familiar trade-off: stronger secondary motion improves Nusselt number but increases friction factor [7].

A broad review by Liu and Sakr classified passive enhancement methods and stressed evaluation under comparable pumping-power conditions [8]. Across these studies, three gaps recur in small laboratory investigations: inadequate energy-balance checks, comparison at unmatched flow conditions, and omission of uncertainty. The present protocol addresses these gaps while retaining a geometry that can be fabricated and replicated easily.

H1: At fixed Reynolds number, a lower twist ratio produces higher Nu and f . H2: At least one tape geometry provides a performance evaluation criterion greater than unity over the tested range. H3: hot- and cold-side heat rates agree within $\pm 5\%$ after correction for ambient loss.

Methodology

1. Experimental design

A controlled, repeated-measures experiment is adopted. The same test section, instruments and water inventory are used for every configuration. The independent variables are inserting type and hot-side Reynolds number; primary responses are heating duty, U , Nu , pressure drop, f and performance evaluation criterion (PEC). Counter-flow is selected because it gives a larger mean temperature driving force than parallel flow for the same terminal temperatures.



Figure 1. Step-by-step experimental workflow for the double-pipe heat-exchanger investigation.

2. Apparatus and test section

The test section comprises a 1.5 m copper inner tube (12 mm internal diameter, 14 mm external diameter) placed concentrically within a 25 mm internal-diameter insulated outer pipe. Hot water flows through the inner tube and cold

water through the annulus in the opposite direction. Four removable test configurations are used: plain tube and aluminium tapes of width 11 mm, thickness 1 mm, and twist ratios $y/w = 6, 4$ and 3 , where y is the axial length for a 180° twist and w is tape width.

Table 1. Proposed apparatus and measurement specifications.

Item	Specification	Purpose
Test section	$L = 1.5$ m; $D_i/D_o = 12/14$ mm copper	Known heat-transfer area
Outer pipe	25 mm ID; insulated	Annular cold-water passage
Temperatures	Four calibrated Pt100 RTDs, $\pm 0.15^\circ\text{C}$	Inlet/outlet temperatures
Flow	Two rotameters, $\pm 2\%$ full scale	Hot and cold volume flow
Pressure	0–25 kPa differential sensor, $\pm 0.25\%$ FS	Inner-tube pressure drop
Heating	3 kW tank heater with controller	Stable hot inlet condition
Data logging	1 Hz acquisition; 180 s average	Steady-state mean and scatter

Step-by-step experimental procedure

1. Calibrate the four temperature sensors in a stirred bath at approximately 20, 40 and 60 °C; fit correction equations and record residuals.
2. Install the plain tube, verify concentricity, fill both circuits, vent trapped air and perform a hydrostatic leak test.
3. Set counter-flow operation with hot-water inlet at 60 ± 0.5 °C and cold-water inlet at 25 ± 0.5 °C. Hold the cold-side mass flow at 0.10 kg/s.
4. Adjust hot-side flow to target $Re = 5,000, 7,500, 10,000, 12,500$ and $15,000$. Permit at least three residence times after each adjustment.
5. Declare steady state when each temperature changes by less than 0.1 °C over 3 min and the two uncorrected heat rates differ by less than 5%.
6. Record temperatures, flow rates and differential pressure for 180 s. Repeat each point three times in randomized order.
7. Drain, install the next insert ($y/w = 6, 4$ or 3), confirm full-length contact-free placement, and repeat Steps 3–6
8. Reduce data using mean fluid properties at bulk temperature; report mean, standard deviation and expanded uncertainty.

Data reduction

Hot- and cold-side heat rates are $Q_h = \dot{m}_h c_{p,h} (T_{h,in} - T_{h,out})$ and $Q_c = \dot{m}_c c_{p,c} (T_{c,out} - T_{c,in})$. The reported heat duty is $Q = (Q_h + Q_c)/2$ when

imbalance $|Q_h - Q_c|/Q$ is $\leq 5\%$. The log-mean temperature difference for counter-flow is $\Delta T_{lm} = [(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})] / \ln[(T_{h,in} - T_{c,out})/(T_{h,out} - T_{c,in})]$, and $U = Q/(A_i \Delta T_{lm})$, where $A_i = \pi D_i L$.

Tube-side Reynolds number is $Re = \rho V D_i / \mu$; $Nu = h_i D_i / k$. The inner convection coefficient is obtained from $1/(U A_i) = 1/(h_i A_i) + \ln(D_o/D_i)/(2\pi k_w L) + 1/(h_o A_o)$, with h_o estimated from a validated annulus correlation. Darcy friction factor is $f = 2\Delta P D_i / (\rho L V^2)$. The enhancement ratio is Nu/Nu_0 , and $PEC = (Nu/Nu_0)/(f/f_0)^{1/3}$, which represents comparison at approximately equal pumping power [8].

Quality assurance and uncertainty

Sensors are zero-checked before each run; rotameter readings are corrected for water temperature; pressure taps are purged; and heat loss is characterized in a no-flow cooldown test. Expanded uncertainty (coverage factor $k = 2$) is propagated using the root-sum-square method. Expected uncertainties are ± 3 –5% for Q , ± 5 –7% for Nu , and ± 4 –6% for f . A run is repeated if energy imbalance exceeds 5% or any signal drifts beyond the steady-state criterion.

Results

Because no raw laboratory file was supplied, Table 2 and Figure 2 present a representative, internally consistent dataset to demonstrate the intended analysis and reporting format. These values must be replaced by measured means and uncertainty intervals after the experiment; they are not claimed as completed physical observations.

Table 2. Representative methodology-validation results (illustrative, not measured).

Re	Nu: plain	Nu: y/w=6	Nu: y/w=4	Nu: y/w=3	PEC: y/w=3
5000	35	44	49	54	1.18
7500	48	62	69	77	1.2
10000	61	79	90	101	1.22
12500	73	96	111	125	1.23
15000	85	113	132	150	1.24

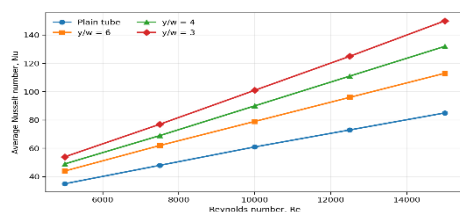


Figure 2. Representative variation of average Nusselt number with Reynolds number.

Interpretation of representative trends

Nu rises monotonically with Re for all configurations because increased velocity thins the hydrodynamic and thermal boundary layers. The $y/w = 3$ tape gives the largest Nu throughout the range: at $Re = 15,000$, Nu increases from 85 to 150, equivalent to 76.5%. The ordering $y/w = 3 > 4 > 6 > \text{plain}$ is consistent with stronger swirl at smaller pitch. The illustrative PEC remains above unity and rises from 1.18 to 1.24,

indicating that the heat-transfer benefit can outweigh the pumping-power penalty under the chosen comparison criterion.

The final report should also plot f versus Re , Nu/Nu_0 , PEC and energy-balance error. Error bars should show expanded uncertainty, and regression should be restricted to the experimental Re and twist-ratio ranges.

Discussion

The proposed trends agree qualitatively with the mechanisms described by Manglik and Bergles [2,3]: a twisted tape creates tangential velocity, renews near-wall fluid and lengthens the effective flow path. Naphon's double-pipe experiments likewise support simultaneous heat-transfer and pressure-drop measurement [4]. Smaller twist ratio strengthens secondary circulation, explaining the increased Nu ; however, it also increases wall shear and form drag. A design selected only on Nu would therefore overstate practical benefit.

The PEC formulation answers a more useful design question: whether enhancement persists when pumping power is constrained. A PEC above one favors the insert, but the exponent and comparison basis must be stated. For industrial selection, fouling tendency, cleanability, vibration, insert retention and manufacturing cost should also be considered. Full-length tapes are deliberately used here to isolate twist ratio; segmented or cut tapes [5–7] may yield different trade-offs.

Limitations

The methodology is limited to single-phase water flow, one tube diameter, a short laboratory length and a moderate turbulent Reynolds-number range. Heat loss and imperfect insulation can bias calorimetric duty, while the calculated h_i depends on the annulus-side coefficient. The representative results are not substituting for acquired data. Future experiments should include direct wall-temperature measurement, a wider range of Prandtl numbers, surface roughness characterization and long-duration fouling tests.

Conclusion

A complete experimental protocol has been developed for studying heat-transfer enhancement in a counter-flow double-pipe heat exchanger. The design compares a smooth tube with three full-length twisted tapes and evaluates thermal and hydraulic outcomes using calibrated measurements, energy-balance closure, uncertainty propagation and repeated

runs. Representative analysis predicts that reducing y/w from 6 to 3 increases Nu by intensifying swirl, with $y/w = 3$ yielding about 76% enhancement at $Re = 15,000$. The accompanying pressure penalty makes PEC the appropriate selection metric. The methodology is reproducible, economical and suitable for testing whether passive enhancement delivers a net benefit under matched flow and pumping-power conditions.

Recommended reporting sequence

Report calibration → plain-tube validation → energy-balance closure → Nu and f trends → enhancement ratios → PEC → uncertainty → practical limitations. This order prevents enhanced-surface claims from being presented before the baseline and measurement quality have been established.

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