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Increasing Heat Recovery in Heat Exchanger Design Using a Combined Twisted Tape and Twisted Tube Technology

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This work introduces a heat exchanger design approach for units using combined twisted tape and twisted tube heat transfer enhancement technology. In design and retrofit, the use of these systems is characterized by heat transfer enhancement on both sides of the unit. It is shown that the rate of increase of heat transfer is much higher than in single twisted tubes leading to improved designs. Thermal and hydraulic experimental published data on these types of surfaces are employed in developing the design approach. In the case of design, an area reduction of up to 32.2 % with a length reduction of 31.4 % for the same pressure drop levels is achieved. In the case of retrofit, the use of a combined twisted tube and twisted tape led to a heat load increase of 29.2 % with an increment of 40.4 % in pressure drop. The methodology is demonstrated in a case study from the open literature.

1. Introduction

The thermal performance of tubular heat exchangers can be enhanced using different combined technologies. For instance, inside the tubes, turbulence promoters are a prominent option; on the outside of the tube, the substitution of segmental baffles for helical baffles offers improved performance. The search for new surface designs has led to the development of new techniques to enhance heat transfer at the expense of increased pressure drop, such as the case of the combined twisted tube and twisted tape geometry. In this work, the combination of twisted tubes with twisted tape inserts is analysed to demonstrate its suitability in design and retrofit applications. The aim of this work is to show the use of published thermal-hydraulic data on these types of devices on a new design methodology based on the full use of pressure drop for the sizing of heat exchangers and the use of existing ones for improved heat recovery.

When new heat transfer surfaces appear, they are accompanied by a thermal-hydraulic study (either numerical or experimental) where the heat transfer and friction performance is obtained, which is basic for design purposes (Durán et al., 2021). The literature review in this work is focused on the studies performed on the combined twisted tube and twisted tape technology. In this regard, a very innovative tube design consisting of a six-start spirally corrugated tube was proposed by Qian et al. (2018), who studied the thermal and hydraulic performance within a Reynolds number between 10,000 and 60,000. They found that the Nusselt number and friction factor decreased with the corrugation pitch. In a later study, Qian et al. (2020) analysed multi-corrugated tubes finding that an eight-start spirally corrugated tube exhibits improved performance compared to a six-start spirally corrugated tube.

The use of combined concentric oval twisted tubes has shown performance improvements of up to 24 % to 39 % for small twist pitches and large aspect ratios. The relative twist direction of the tubes also results in higher thermal performance (Li et al., 2021). One step forward to improve the performance of twisted tubes is using twisted tapes inside the tube. An example is the combination of three-start tubes with a triple-channel twisted tape introduced by Eiamsa et al. (2016). The relative position of the tube and the insert is crucial for improved performance, finding that the belly-to-neck arrangement gives better results. Compared to single twisted tubes, these arrangements provide 17 % increased thermal performance for Reynolds around 5,000. A trapezoidal

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twisted tape was analysed in combination with twisted tubes finding that the performance is improved when smaller twist ratios are used (Samruaisin et al., 2019). Similar studies with modified twisted tape designs have demonstrated improved performance in a wide Reynolds number range (Yu et al., 2022).
The amount of thermohydraulic information on these combined technologies is large; however, the amount of work published on the actual application of these systems in design is scarce. Recently, Barraza and Picón (2022) introduced a new design approach to replace smooth tubes with twisted tubes in existing heat exchangers. Their design approach is based on developing a thermohydraulic model that relates heat transfer and pressure drop under the full use of allowable pressure drop. In the case of high-viscosity fluids, they demonstrated a 45 % surface area reduction with a lower pressure drop consumption.
The ongoing literature review reveals that more work on implementing new enhancement techniques is required in the design and retrofit of heat exchangers. This work seeks to bridge this gap and introduces a thermal-hydraulic approach to size new units and rate existing ones using combined twisted tube and twisted tape technology. Two case studies are analysed for design and retrofit using two different enhanced twisted tube geometry: single-channel twisted tube with twisted tape (TT-C), and 5-channel twisted tube with twisted tape (TT-c-5R). A case study from the open literature is used as a base case for comparison purposes.

2. Thermohydraulic design

For the tube side, the numerical data reported by Yu et al. (2022) for a Reynolds number between 5,000 and 55,000, and Prandtl number between 2 and 7 is used in this work. The data is plotted in Figure 1 and a simple regression analysis is applied to determine the expressions for the Nusselt number and friction factor (Table 1). The data in Figure 1 apply to a twisted oval tube and a simple twisted tape with a coincident twist angle (TT-C) and a 5-channel twisted tape with an opposing twist angle (TT-C-5R).

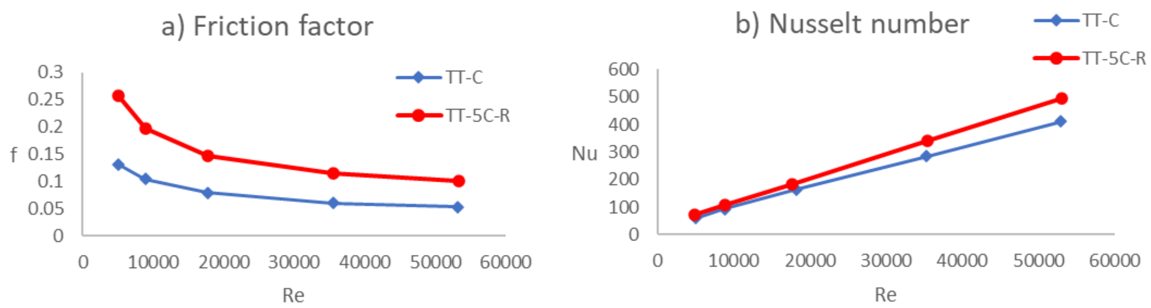

Figure 1: Thermohydraulic behaviour for selected combined twisted tube and twisted tape a) Friction factor b) Nusselt number

Table 1: Expressions for Nu and friction factor for selected combined twisted tube and twisted tape

Tube type	Expression	Reynolds Range	Prandtl range
	Nusselt number		
TT-C	$Nu = 0.062Re^{0.8062}$	$5{,}000 < Re < 53{,}000$	
TT-5C-R	$Nu = 0.0694Re^{0.8115}$	$5{,}000 < Re < 53{,}000$	2<Pr<7
	Friction factor		
TT-C	$f = 3.4595Re^{-0.385}$	$5{,}000 < Re < 53{,}000$	
TT-5C-R	$f = 7.417Re^{-0.398}$	$5{,}000 < Re < 53{,}000$	

The thermohydraulic performance of the Shell side is computed using the expression reported.

$$Nu_{sh} = 0.02379{Re_{sh}}^{0.8}Pr^{0.33}{F_{rm}}^{-0.4347}\left[1 + 3.6{F_{rm}}^{-0.357}\right] \tag{1}$$

$$f_{sh} = 9.461{Re_{sh}}^{-0.5}{F_{rm}}^{-0.078}\left[1 + 3.6{F_{rm}}^{-0.357}\right] \tag{2}$$

3. Design methodology

In this work, the stream whose pressure drop is to be maximised is called the critical stream and the opposing is the reference stream. Figure 2 shows the main geometrical parameters of twisted tube and twisted tapes. These parameters along with the tube count are taken from available commercial information.

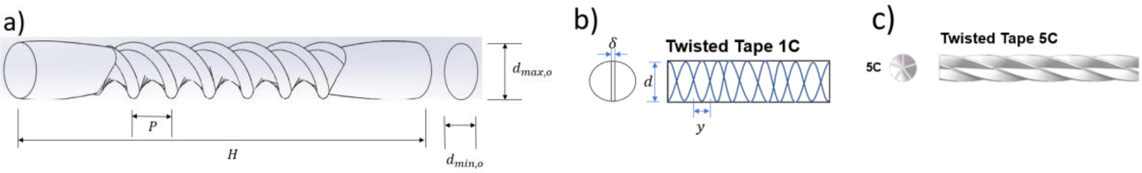

Figure 2: Geometrical parameters: a) Twisted tube, b) Single-channel twisted tape, c) Five-channel twisted tape

3.1 Tube side analysis

From the tube geometry, the hydraulic diameter is obtained from:

$$d_h = \frac{4A_{ci}}{P_w} \tag{3}$$

Where A_{ci} the tube flow area (m^2) and P_w is the wetted perimeter (m). In the case of an ellipse, these terms are computed from:

$$A_{ci} = \frac{\pi d_{max} d_{min}}{4} \tag{4}$$

$$P_w = \pi \left[3(d_{max} + d_{min}) - \sqrt{(3d_{max} + d_{min})(d_{max} + 3d_{min})}\right] \tag{5}$$

Combining Eqs (7), (8) and (9) and rearranging gives:

$$d_h = d_{max} d_{min} / \left[3(d_{max} + d_{min}) - \sqrt{(3d_{max} + d_{min})(d_{max} + 3d_{min})}\right] \tag{6}$$

The free flow area for the tube side is:

$$A_{ct} = \pi d_h^2 N_{tubes} / 4N_{pass} \tag{7}$$

Using Eq (7), the mass flow rate $(\dot{m}_t)$ and the density (ρ_t), the fluid velocity inside the tubes can be calculated as:

$$v_t = \dot{m}_t / \rho_t A_{ct} \tag{8}$$

The Reynolds number is obtained from:

$$Re_t = \rho_t v_t d_h / \mu_t \tag{9}$$

For the twisted tube, the expressions in Table 1 are used to calculate the Nusselt number (Nu_t) and the heat transfer coefficient (h_t).

$$Nu_t = 0.062 Re_t^{0.8062} \tag{10}$$

$$h_t = \frac{Nu_t k}{d_h} \tag{11}$$

The friction factor on the tube side is obtained from Eq(12) and used in Eq(13) to calculate the pressure drop, where H is the tube length.

$$f_t = 3.4595 Re_t^{-0.385} \tag{12}$$

$$\Delta P_t = \frac{f_t \rho_t v_t H}{2d_h} \tag{13}$$

3.2 Shell side analysis

The model for the Shell side also requires the calculation of the free flow area. The wetted perimeter (P_{ws}) and the equivalent diameter (d_{eq}) are:

$$P_{ws} = \frac{\pi d_h}{2} \tag{14}$$

$$d_{eq} = 1.10\left(P_t^2 - 0.917 d_h^2\right) / d_h \tag{15}$$

For a triangular tube arrangement, the free flow area in the axial direction is obtained from:

$$A_{c,sh} = (N_{tubes} - 2)\left(\frac{0.87P_t^2}{2} - \frac{\pi d_h^2}{8}\right) \quad (16)$$

For highly viscous fluids, the Colburn and friction expressions proposed are employed. These expressions contain the modified Froude number (F_{rm}), that is related to the degree of torsion expressed in terms of the diameter of the longest axis (d_{max}), the tube pitch (P) and the tube hydraulic diameter (d_h):

$$F_{rm} = P^2/d_{max}d_h \quad (17)$$

The mass flux (G_{sh}) and the Reynolds number in the shell side are computed from:

$$G_{sh} = \dot{m}_{sh}/Ac_{sh} \quad (18)$$

$$Re_{sh} = G_{sh}d_{eq}/\mu_{sh} \quad (19)$$

Eq (20) is used to determine the Nusselt number from where the heat transfer coefficient is obtained:

$$Nu_{sh} = 0.02379{Re_{sh}}^{0.8}Pr^{0.33}{F_{rm}}^{-0.4347}\left[1 + 3.6{F_{rm}}^{-0.357}\right] \quad (20)$$

$$h_{sh} = \frac{Nu_{sh}k_{sh}}{d_h} \quad (21)$$

The shell side friction factor and the pressure drop are obtained from:

$$f_{sh} = 9.461{Re_{sh}}^{-0.5}{F_{rm}}^{-0.078}\left[1 + 3.6{F_{rm}}^{-0.357}\right] \quad (22)$$

$$\Delta P_{sh} = f_{sh}G_{sh}P^2D_s/2d_{eq}\rho_{sh} \quad (23)$$

From the heat transfer coefficients of the tube and Shell side, the overall heat transfer coefficient is calculated using Eq(24)

$$U = 1/\left(\frac{1}{h_t} + \frac{1}{h_{sh}} + \frac{1}{ln(\frac{d_{max,o}}{d_{max,i}})/2\pi kH}\right) \quad (24)$$

Assuming a counter current arrangement, the heat transfer area is computed from the design equation:

$$Q_1 = UA\Delta T_{LM} \quad (25)$$

Where ΔT_{LM} is the log mean temperature difference and A is the surface area from where the tube length is computed as:

$$A = \pi d_h H \quad (26)$$

4. Case studies

The operating data and physical properties of a case study taken from Yan et al. (2017) are shown in Table 2, where an epoxy resin is heated using hot water. The thermal conductivity of the tube wall is 50 (W/m °C).

Table 2: Operating conditions, physical properties, and geometry for case study

Process data	Hot stream (Shell side)	Cold stream (Tube side)
Mass flow rate (kg/s)	1.88	1.57
Pressure drop (Pa)	31,000	31,000
Inlet temperature (°C)	89.2	30
Outlet temperature (°C)	80	54.1
Density (kg/m^3)	971.8	1,160
Heat capacity (J/kg °C)	4,196	1,920
Thermal conductivity (W/m °C)	0.67	0.21
Viscosity (kg/m s)	0.000354	0.000354
Heat load (kW)	72.58	
Number of tubes (-)	23	
Shell diameter (m)	0.12	
Tube hydraulic diameter (m)	0.018	

4.1 Design applications

New designs for the TT-C and TT-C-5R geometries are performed and shown in Table 3. The design results are compared to the single twisted tube exchanger design reported by Barraza and Picón (2022). For a TT-C tube, the heat transfer area is 7.59 m^2 with an overall heat transfer coefficient of 227.1 W/(m^2 °C). This means an area and tube length reduction of 32.2 % and 31.4 %. For a TT-C-5R tube, the required surface area is 7.54 m^2 and the overall heat transfer coefficient is 238.3 W/(m^2 °C). This represents an area reduction of 32.7 % with respect to the simple twisted tube. In both cases, the design was carried out for a similar tube side pressure drop.

Table 3: Design results for case study

	Twisted tube Barraza and Picón (2022)	Combined twisted tube and twisted tape (This work)	
		TT-C	TT-C-5R
Surface area (m^2)	11.2	7.59	7.54
Number of tubes (-)	23	22	41
Tube length (m)	8.6	5.9	3.1
Overall heat transfer coefficient (W/m^2 °C)	145.4	227.1	238.3
Tube side pressure drop (Pa)	31,000	30,931	30,471
Tube side velocity (m/s)	0.7	1.13	1.14

4.2 Retrofit applications

In the case of retrofit, the assumption is that the existing tube bundle is fully replaced by the combined twisted tube and twisted tape geometry. Due to space limitations, only the results are shown here. Results indicate that, as expected, higher heat duties can be achieved using the same shell and tube dimensions compared to the use of single twisted tube geometry. When the tube bundle is replaced by TT-C tubes, an increment of 29.2 % on the heat load, and 40.4 % (43,513 Pa) on the pressure drop are obtained. The use of the TT-C-5R results in a heat load increment of 29.7 % and a pressure drop rise in the order of 263 % (81,584 Pa).

5. Conclusions

This work has introduced a new design approach for the sizing of heat exchangers using a combination of twisted tube and twisted tape heat transfer enhancement technology. The methodology is extended to the retrofit of existing units for increased heat load. The results show that the combined use of a twisted tube and twisted tape technology is advantageous in design as it allows the transfer of the same heat duty with less surface area. The surface area is reduced by 32.7 % for the same available pressure drop with respect to the simple twisted tube design. The use of these combined systems is also an excellent option in the case of retrofit applications where for the same installed shell dimension, the replacement of the tube bundle can give heat load increments of 29.2 % for the simple twisted tube with an increment of pressure drop of 40.4 %. The TT-C gives better results than the TT-C-5R combination.

Nomenclature

A – Heat transfer area, m^2
A_c – Free flow area, m^2
d_{eq} – Equivalent diameter, m
d_h – Hydraulic diameter, m
d_{max} – Long axis internal diameter, m
d_{min} – Short axis internal diameter, m
ΔP– Pressure drop, Pa
ΔT_{LM}– Log mean temperature difference, °C
f – Friction factor, -
F_{rm}– Froude number, -
G– Mass flux, kg/ m^2 s
h –Heat transfer coefficient, W/(m^2·°C)
H– Tube length, m
$\dot{m}$ – Mass flow rate, kg/s
N_{pass} – Number of passes, -
N_{tubes} – Tube count, -
Nu – Nusselt number, -
P – Twist pitch, m
Pr – Prandtl number, -
P_t – Tube pitch, m
P_W – Wetted perimeter, m
Re – Reynolds number, -
U – Overall heat transfer coefficient, W/(m^2·°C)
v – Fluid velocity, m/s
ρ – Fluid density, kg/m^3
k – Thermal conductivity, W/(m·°C)
μ – Fluid viscosity, - kg m/s

Subscripts
i – Inner side
o – Outer side
t – Tube side
sh – Shell side

Abbreviations
TT – Twisted tube
TT-C– Twisted tube with simple twisted tape
TT-C-5R– Twisted tube with 5-channel twisted tape

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