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Retrofit of Tubular Heat Exchangers Using Twisted Tubes for a Target Thermal Effectiveness

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This study introduces a systematic approach for retrofitting tubular heat exchangers with twisted tubes, considering the tube twist pitch as a design variable. Twisted tubes enhance the thermal operation of shell and tube heat exchangers, which increases the overall heat transfer coefficient without significant pressure drop penalties. The methodology is based on expressions for the friction factor and Nusselt number, in the range of twist pitch from 0.05 m to 0.2 m. These expressions are developed using CFD and validated with experimental data, with an average absolute error of 10.40 % and 6.47 % for the Nusselt number and friction factor. Using a twist pitch P=0.05 m, an increase of 130 % in convective heat transfer coefficient is achieved; this represents an increase of 30.44 % in thermal load. The results show that under the assumption that twist pitch is a continuous variable, the specific twist pitch can be determined to achieve a fixed increment in heat load. A 15 % target in thermal load increment can be achieved with a twisted tube with P=0.11 m without any changes in total operating cost. A case study is presented to demonstrate the application of the design approach.

1. Introduction

Shell and Tube Heat Exchangers (STHE) are one of the most widely used equipment in the chemical industry due to their versatility and relative ease of construction. One of the most widely used strategies to improve the performance of these devices and increase the thermal load is to improve the overall heat transfer coefficient at the expense of pressure drop. This can be achieved by modifying the convective coefficient by installing turbulence promoters or by modifying the surface to increase local turbulence. An alternative that achieves these objectives is the use of twisted tubes, which have a corrugated surface due to the twisting of the tube and generate turbulence inside and outside the tubes. These types of tubes can also work with twisted tape type turbulence promoters, as demonstrated by Samruaisin et al. (2019), who experimentally studied a combined twisted tube and twisted tape. These works reported that with a twist ratio of 2, a greater increase in the Thermal Performance Factor is obtained, achieving an increase of 5.4 % compared to a twisted tube with a twist ratio of 5. Work in this direction has included the use of new configurations, such as the work done by Yu et al. (2022), where they found that with a twisted tube and a center-clear twisted tape, a 35.58 % increase in Nusselt can be achieved compared to a smooth tube. The combination of these technologies has extended to the implementation of retrofit projects to reduce the use of external services. Crespo et al. (2024) evaluated the use of different configurations of twisted tubes with twisted tapes with different numbers of channels, finding that the best performance for these applications is obtained by combining a twisted tube with a triple-channel twisted tape. While the combination of technologies allows for increased thermal load, its use has been observed to generate increased pressure drop levels, leading to higher pumping costs. The importance of its geometry and the impact on thermal and hydraulic performance have been highlighted in the literature. Some works have shown that the variation in the aspect ratio is an important parameter, and for a larger aspect ratio and a smaller twist pitch, a greater increase in the convective coefficient is achieved without significant increases in the pressure drop (Yang et al., 2011). For this reason, research has focused on finding the best combination of geometric variables to utilize only twisted tubes and improve their thermohydraulic performance. For example,

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Li et al. (2021) numerically studied twisted oval tubes by varying geometric aspects such as twist direction, aspect ratio, twist pitch, and inner radius of the tube; the best performance was obtained for the lowest twist pitch with the largest inner radius. Twisted tubes also offer improvements in tube exterior due to their external shape, which allows for increases in the Nusselt number without substantial changes in pressure drop. The use of a twisted tube bundle at STHE has revealed that temperatures inside the equipment are more uniform due to the turbulence and vortex generation processes. The results indicate that the use of small twist pitch and aspect ratios close to 1 generates increases of 12.8 % in the Nusselt number and 15.2 % in the friction factor (Gu et al., 2020). While the thermohydraulic study of different systems to increase heat transfer is essential, its use in design or retrofit situations is of utmost importance. When comparing the designs in terms of surface area and pressure drop with respect to conventional exchangers, the advantages of using these systems become evident (Barraza-Colón and Picón-Núñez, 2022), where the required surface area can be reduced by 45 % and even with pressure drop reductions of 15 %. The state of the art shows that the design of twisted tube geometry for specific applications in shell-and-tube exchangers has not been addressed, so this work seeks to fill this gap. An approach to the design of twisted tubes is introduced by setting a desired thermal load as a target through the determination of the degree of twist of the tube using the twist pitch P as the design variable. The methodology uses correlations for the friction factor and the Nusselt number, obtained using CFD, in a P range of 0.05 m to 0.2 m. It is applied to a practical case.

2. Thermohydraulic data of twisted tubes

The thermohydraulic data of the tube interior were generated using CFD. These were validated with experimental data provided by Yang et al. (2011), who studied twisted tubes with different twist pitches (P). The main geometric characteristics of a twisted tube are shown in Figure 1.

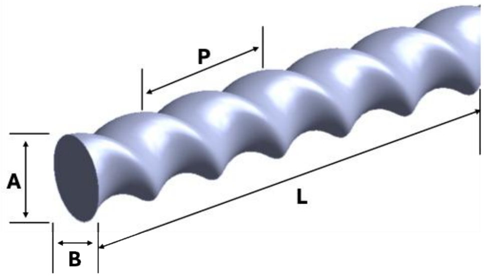

Figure 1 : Main geometrical parameters of twisted tube

For validation purposes, a twisted tube with a length L=1.75 m, aspect ratio A/B=1.6, and P=0.104 m is studied, using water as the working fluid. The model was solved using the $k-\omega$ turbulence model in Ansys Fluent, while mesh sensitivity analysis showed mesh independence with 897,373 elements. Figure 2 shows the comparison of the Nusselt number and friction factor obtained by CFD against the experimental data obtained by Yang et al. (2011).

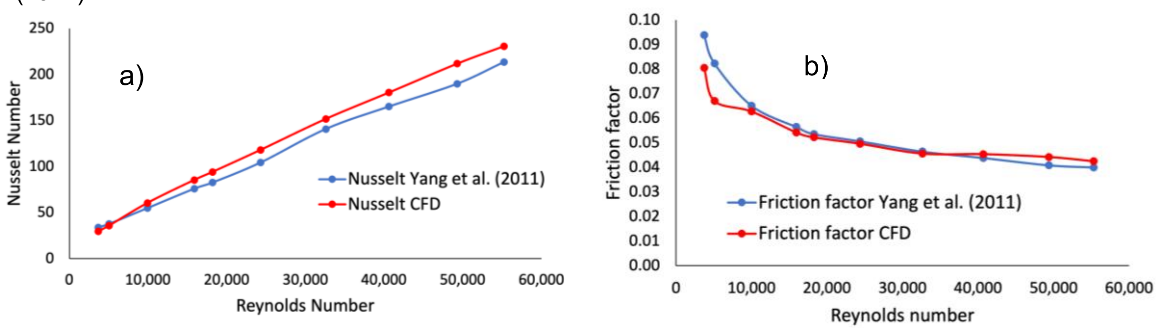

Figure 2: Validation of CFD data with experimental data (Yang et al., 2020). a) Nusselt number b) Friction factor

Nusselt number predictions show small errors at Reynolds values below 5,000; beyond this point, the values remain within the average, with the error increasing again at Reynolds values above 40,000. The minimum and maximum errors for the entire Reynolds range are 5 % and 14 %, with an average absolute error of 10.40 %. The same trend is observed in the case of the friction factor; starting at a Reynolds number of 10,000, there is an improvement, with values very similar to the experimental ones. The minimum error is 1.8 % and the maximum is 18.65 %, with an average absolute error of 6.48 % for the entire Reynolds range. A more detailed analysis of the prediction errors for Nu and f is presented in Table 1.

Table 1: Absolute error for CFD data versus experimental data

Reynolds	Nusselt (Yang et al., 2011)	Nusselt CFD	Error Nu (%)	Friction factor (Yang et al., 2011)	Friction factor CFD	Error f (%)
3,694	34.10	29.81	12.56	0.094	0.080	14.24
5,060	37.85	35.94	5.06	0.082	0.067	18.65
9,959	55.09	60.72	10.22	0.065	0.063	3.67
15,903	76.08	85.62	12.54	0.056	0.054	3.96
18,232	82.84	94.40	13.96	0.054	0.052	2.53
24,337	104.58	118.31	13.13	0.051	0.050	2.00
32,610	140.91	151.86	7.77	0.046	0.046	1.80
40,642	165.29	180.55	9.23	0.044	0.045	3.48
49,317	190.05	211.89	11.49	0.041	0.044	8.19
55,261	213.65	230.88	8.06	0.040	0.043	6.24

According to the deviations shown in Figure 2 and Table 1, the average absolute errors for Nu and f are 10.40 % and 6.50 %. Using the numerical model, data are generated for Nu and f for P values of 0.05, 0.1, 0.15, and 0.2 m. The generated data are shown in Figure 3.

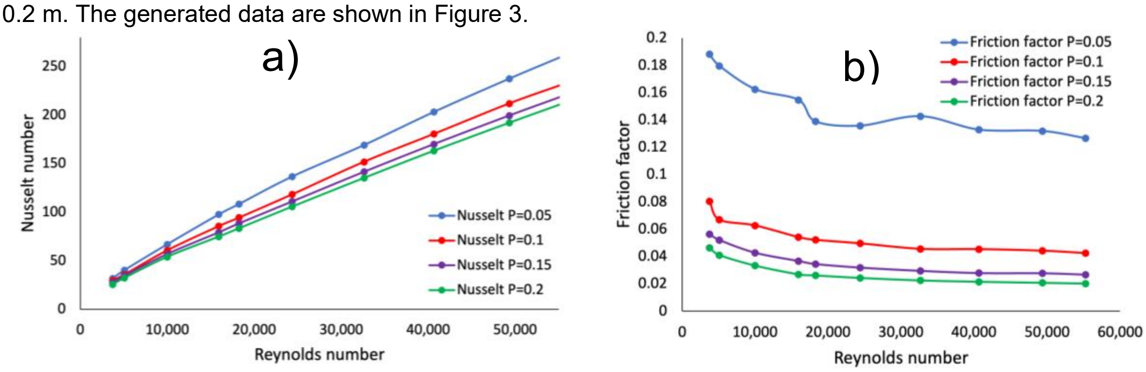

Figure 3: Data generated for twist pitch values from 0.05 to 0.2 m, a) Nusselt number and b) Friction factor

In Figure 3a, it can be observed that for values of P=0.05 m, the Nusselt number presents higher values. This is because the greater the degree of twist, the greater the local turbulence generated, which increases the Nusselt number and the convective heat transfer coefficient. The trend shows that decreasing the twist pitch results in a higher Nusselt number. The friction analysis shows a similar behaviour, for values of P=0.05 m the friction factor increases up to 66 % with respect to P=0.1 m, while the increase of the latter with respect to P=0.2 m is 53 %. From this numerical information, the correlations of Eq(1) and Eq(2) are generated using the methodology proposed by Picón et al. (2023).

$$Nu_t = (-0.0716P + 0.0587)Re^{(0.0016P+0.771)} \tag{1}$$

$$f_t = (0.0834P + 0.5431)Re^{(-1.138P-0.097)} \tag{2}$$

Eq(1) and Eq(2) were generated for discrete values; however, for the purposes of this study, they are considered continuous in the range of P between 0.05 and 0.2 m, a Reynolds range of 3,678 to 55,261, and a Prandtl number between 2 and 20. When comparing CFD data with predictions using correlations, these have an average error of 2.5 % for Nu and 21 % for f over the entire range of P. For the shell-side analysis, the correlations proposed by Gu et al. (2020) are used for a coupled vortex configuration in the Froude number range of 9.3 to 312.9.

$$Nu_{sh} = 0.155Re^{0.64}Pr^{0.4}\left(\frac{A}{B}\right)^{0.4}(P/d_h)^{-0.1} \tag{3}$$

$$f_{sh} = 3.831 Re^{-0.33} Pr^{-0.25} \left(\frac{A}{B}\right)^{0.62} (P/d_h)^{-0.4} \quad (4)$$

3. Twisted tube Design Approach

The approach to designing a twisted tube to achieve a thermal performance uses the thermal effectiveness-Number of Transfer Units model. The methodology consists of the following steps: 1. Set an initial twist pitch P, 2. Calculate the convective coefficients inside the tubes and for the shell side using Eqs(1) to (4), 3. Obtain the new overall heat transfer coefficient (U) and the number of transfer units (Ntu), 4. Determine the new thermal effectiveness for the shell and tube heat exchangers, and finally 5. Obtain the new outlet temperatures and the thermal load increase ΔQ. The procedure is iterative until the target thermal load is reached. Figure 4a graphically shows the design approach, and Figure 4b shows the block diagram of the methodology.

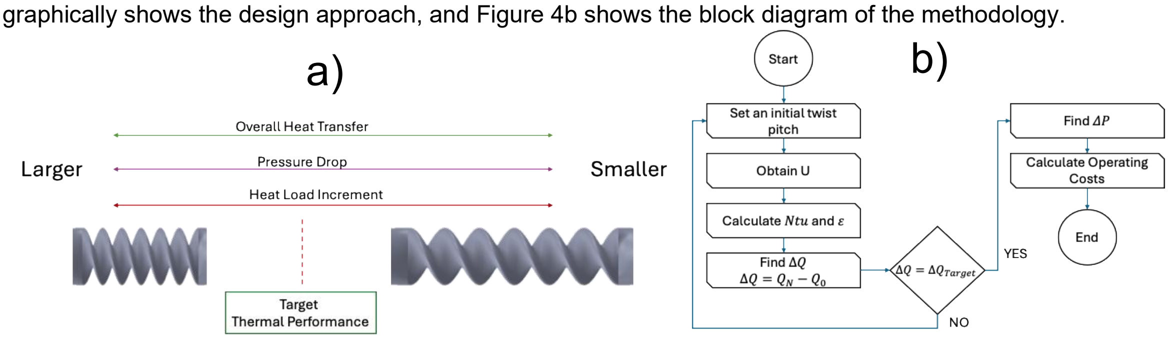

Figure 4: Design approach for designing twisted tubes a) Pictorial diagram, b) Flowchart

4. Case study

The case study consists of a tubular exchanger with a surface area of 410 m^2 and a length of 6.63 m, with a shell diameter of 0.889 m and 1,031 tubes of 0.019 m external diameter and a thickness of 0.002 m. The operating data and physical properties are shown in Table 2. The smooth tube bundle will be replaced by twisted tubes with A=0.024 m, B=0.019 m, and a thickness of 0.002 m. The cost of the shell and tube heat exchanger is determined using the expression proposed by Wang et al. (2020) and the operating cost analysis according to the methodology proposed by Picón et al. (2024). By using the same shell and tube count, the available area for heat transfer is increased due to the torsion effect of the tube.

Table 3 shows the main design results for the twist pitch range between 0.05 m and 0.2 m. From a thermal perspective, it is observed that for small values of P, there are higher convective heat transfer coefficients. This allows for increases in the overall U coefficient in the range of 14 % to 34.1 %. Compared with the original design, by using P=0.05 m, the convective coefficient on the tube side increases by 130 %, resulting in an increase in thermal load ΔQ of 30.44 %, while the thermal effectiveness increases from 0.58 to 0.84, representing an increase of 43.8 %.

Table 1: Operating data for case study

Parameter	Shell side	Tube Side
Inlet temperature [°C]	28	250
Outlet temperature [°C]	96	120
Thermal effectiveness [-]	0.58	
Mass flow rate [kg/s]	16.17	10
Pressure drop [Pa]	45,030	120
Heat capacity mass flow rate [kW/°C]	67.91	35.00
Convective heat transfer coefficient [W/m^2 °C]	234.56	193.12
Overall heat transfer coefficient [W/m^2 °C]	91.84	
Viscosity [kg/m s]	0.00080	0.00040
Heat capacity [J/kg °C]	4,200	3,500
Conductivity [W/m °C]	0.140	0.130
Density [kg/m^3]	999	560

Table 3: Design results for case study for the entire range of twist pitch

$P[m]$	h_t $[W/m^2\ °C]$	h_{sh} $[W/m^2 °C]$	U $[W/m^2 °C]$	ΔQ [%]	A_s $[m^2]$	$Total\ \Delta P$ $[Pa]$	$Pumping_{Cost}$ $[\$/y]$	$Total_{Cost}$ $[\$/y]$
0.05	444	174.4	123.2	30.44	747.6	13,114.7	482.70	14,967.73
0.06	438.3	171.3	121	26.62	638.2	11,059.6	407.06	13,131.34
0.08	426.9	166.4	117.5	20.69	529.5	8,390.4	308.82	11,245.65
0.1	415.4	162.8	114.7	16.57	479.2	6,648.3	244.70	10,339.26
0.12	404	159.8	112.3	13.61	451.8	5,377.2	197.91	9,829.06
0.14	392.6	157.4	110.2	11.35	435.4	4,396.1	161.80	9,513.90
0.16	381.1	155.3	108.2	9.551	424.7	3,616.0	133.09	9,302.42
0.18	369.7	153.5	106.4	8.036	417.3	2,986.0	109.90	9,152.50
0.2	358.2	151.9	104.7	6.71	412.1	2,470.9	90.94	9,044.31

Pressure drop analysis shows that the increase is primarily due to the effect within the tubes. However, the levels are not as high, remaining within normal operating ranges. This is mainly due to the self-supporting twisted tube technology and the absence of baffles. The overall effect is a reduction in velocity but an increase in internal turbulence due to the geometry of the tube.

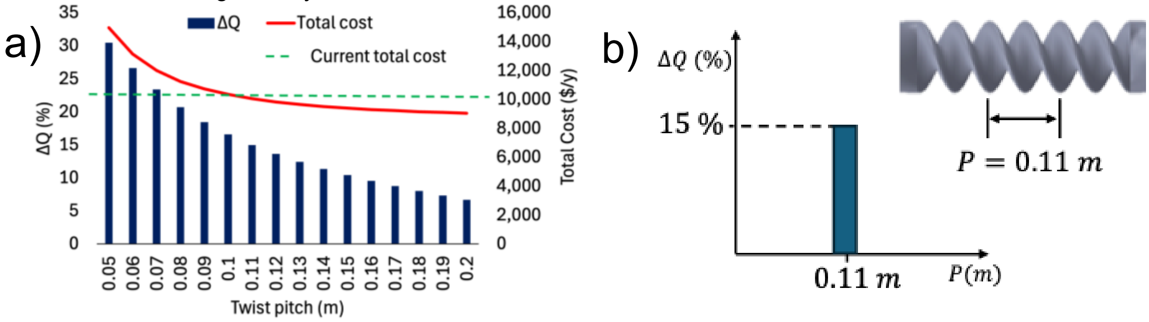

Figure 5: Design results. a) ΔQ and total cost for the entire range of P, b) Twisted tube for a target $\Delta Q = 15$ %

Figure 5a shows the increase in thermal load due to the use of twisted tubes for different values of P. The current equipment operating cost is 10,427.80 \$/y. Designs with P greater than 0.09 m are below current operating costs. For this reason, replacing smooth tubes with twisted tubes is an attractive and economically viable alternative to increasing thermal load. To illustrate this approach, if a 15 % increase in heat load is desired, it can be achieved by using a twisted tube with a twist pitch of P=0.11 m (Figure 5b). This option would have an operating cost of 10,052.50 \$/y.

5. Conclusions

The use of twisted tubes as a tool for the retrofit of tubular heat exchangers is presented. A design approach is introduced that allows a twisted tube to be designed by modifying its twist pitch to achieve an increase in thermal load. The approach is based on Nusselt and friction factor correlations generated by CFD and validated with experimental data. The main conclusions of this work are:

- The methodology allows for evaluating the increase in thermal load as a function of the twist pitch in the operating range of the correlations and for a Prandtl range between 2 and 20.
- The twist pitch has a direct effect on the thermal effectiveness, twisted tubes with a P=0.05 m increase the convective coefficient on the tube side by 130 %, achieving a ΔQ of 30.44 % and an increase in thermal effectiveness from 0.58 to 0.84.
- The increase in thermal load is accompanied by an increase in pressure drop on the tube side, but a reduction on the shell side. The combined effect translates into competitive operating costs. A 15 % increase in thermal load can be achieved with a twisted tube with a P of 0.11 m, while maintaining nearly constant operating costs.

Nomenclature

A – Twisted tube long axis, m
A_s – Heat transfer area, m^2
B – Twisted tube short axis, m
D – Twisted tape diameter, m
d_h – Hydraulic diameter, m
ΔP – Pressure drop, Pa

ΔQ – Thermal load increment, %
f – Friction factor, -
h – Heat transfer coefficient, W/m^2 °C
L – Tube Length, m
Nu – Nusselt number, -
Ntu – Number of heat transfer units, -
P – Twist pitch, m
Pr – Prandtl number, -
Re – Reynold number, -
U – Overall heat transfer coefficient, W/m^2 °C

Abbreviations

CFD – Computational Fluid Dynamics
STHE – Shell and tube heat exchanger

Subscripts

t – tube side
Sh - Shell side

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