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Counteracting Downstream Effects Using Turbulence Promoters in Retrofit

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One of the simplest strategies for retrofitting heat recovery networks with the goal of reducing energy consumption in a process is the implementation of turbulence promoters. However, an adverse effect of this measure in heat recovery networks is the generation of downstream temperature disturbances that impact the operation of equipment within the network. This study introduces the use of Perforated Twisted Tapes (PTT) in retrofit schemes to enhance thermal load while simultaneously mitigating adverse effects related to network structure. General thermohydraulic correlations have been developed to analyse the implementation of twist ratios between 2 and 7. Through a case study with multiple scenarios, it has been observed that when using higher-density promoters (with a twist ratio of 2), energy savings are reduced by 38.38 % due to adverse effects, resulting in net savings of 3.66 MW. In the case of using promoters with lower density (with a TR=7), the reduction in savings is 26.67 %, with net savings of 3.52 MW.

1. Introduction

Several approaches are employed in retrofit strategies for heat exchanger networks. These include replacing existing units with more thermally effective equipment, installing additional heat transfer area, and enhancing heat transfer through the implementation of turbulence promoters. One such promoter is the Twisted Tape (TT), which consists of twisted ribbons.

One of the modifications that has shown significant benefits is the implementation of perforations in the twisted tapes, either axially or transversely. In previous experimental studies, Thianpong and Eiamsa (2018) analysed the thermal performance when using Perforated Twisted Tapes (PTT) within heat exchanger tubes. Orifices with diameters of 2, 2.5, and 3 mm were spaced at distances of 7.5, 11, and 15 mm, respectively. It was demonstrated that the heat transfer rate increases when the distance between the orifices is smaller, and the diameter is larger. Compared to a smooth tube, an increase of up to 86.7 % is obtained and compared with simple twisted tape an increase of 27.4 % is achieved. In an experimental study conducted by Dagdevir et al. (2021), different types of Twisted Tape were explored, including simple TT and PTT. Using water as the working fluid, it was observed that the Nusselt number obtained with PTT is up to 15 % higher compared to a simple TT. On the other hand, Bhuiya et al. (2020) designed a novel type of perforated Twisted Tape called Perforated Triple Twisted Tape (PTTT), which was inserted into a smooth tube. These PTTT had perforations with diameters of 2, 4, 6, and 8 mm, arranged in 3 rows with a total of 180 holes. The most notable results were obtained with a pore diameter of 4 mm, which increased the Nusselt number by 320 % compared to a plain tube.

Other researchers have explored these geometries using Computational Fluid Dynamics. For instance, Afsharpanah et al. (2018) conducted a 3D numerical analysis of a converging-diverging tube with dual PTT for a Reynolds number range between 10,000 and 20,000, with varying hole quantities in the Twisted Tape (1, 2, and 3). The most relevant finding was that the friction factor decreased by up to 396 % compared to bare converging-diverging tubes with simple twisted tubes, while the Nusselt number increased by only 9 %. Additionally, Ruengpayungsak et al. (2019) performed numerical simulations using a centrally perforated

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Twisted Tape (CP-TT). The space cut ratio s/w (which represents the perforation diameter relative to the CP-TT diameter) varied between 0.5, 0.7, and 0.9. The results indicated that in laminar flow, the maximum thermal enhancement factor (TEF) is 8.92 at s/w=0.5 and for turbulent flows, the maximum TEF=1.33 value was obtained with s/w = 0.9. Additionally, Saylroy and Eiamsa (2017) conducted a numerical study using Twisted Tapes with square cuts (SC-TT) within a smooth tube under turbulent conditions. They varied the ratio between the perforation width and the SC-TT diameter (WR). The maximum TEF obtained was 1.37 using a WR of 0.9.
Retrofit is the strategy that involves the modification to existing heat exchanger networks to increase heat recovery with minimal possible modifications. In this regard, Xu et al. (2022) proposed the use of plate heat exchangers (PHEs) based on a Pinch Analysis to identify necessary modifications within a heat exchange network, including their location and cost. They employed a step-by-step methodology to maximize heat exchanger efficiency through retrofit techniques. The study examined three case studies, comparing their results with a heat exchanger network composed solely of shell-and-tube heat exchangers. On the other hand, Angsutorn et al. (2021) introduced a novel mathematical model for industrial applications. This model leverages retrofit methodology concepts to optimise heat exchanger networks.
The review of the state-of-the-art reveals that there is limited information regarding the use of turbulence promoters in retrofit applications aimed at restoring network temperatures when a thermal load increase occurs. In this study, the use of turbulence promoters in shell-and-tube heat exchangers is analysed employing generalised correlations from two perspectives: first, to enhance heat transfer for a specific thermal load increase; and second, to mitigate adverse effects within a heat exchange network resulting from the thermal load increment.

2. Perforated twisted tape geometry

Perforated twisted tapes are thin, metallic strips with a thickness t_h and perforations on their surface with a diameter d_p. These tapes are placed inside tubes in the direction of flow, aiming to generate additional turbulence. The primary parameter characterizing a turbulence promoter is the Twist Ratio (TR), which is defined as:

$$TR = y/D \tag{1}$$

Where y represents the distance for a 360 ° twist, and D is the diameter of the strip (see Figure 1).

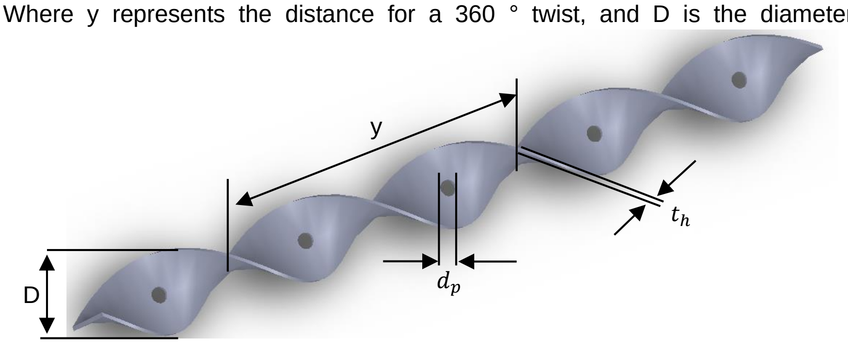


Figure 1: Main geometrical parameters of perforated twisted tapes

3. Thermohydraulic Performance

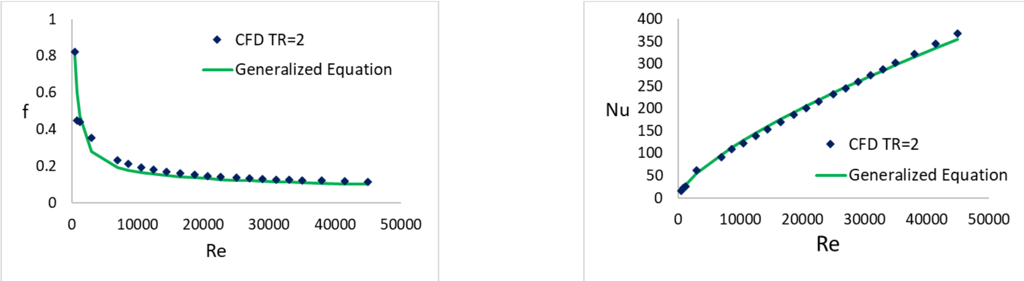
For the development of this study, thermohydraulic data for the Nusselt number and friction factor were generated using CFD. The base data were obtained from Dagdevir et al. (2021) for perforated Twisted Tape (PTT). The analysis was conducted for a Twist Ratio of 5.88 and a hole size of 0.008 m, covering Reynolds numbers from 6,961 to 22,654. Simulation results showed a global error of 8.43 % compared to Dagdevir et al. (2021) findings (Figure 2). Once the model was validated, the analysis was extended to the entire flow regime, from laminar to turbulent, and TR values from 2 to 7. The methodology proposed by Picón et al. (2023) was used to obtain generalised correlations. These expressions are valid for Reynolds numbers between 100 and 45,000, and Prandtl numbers between 2 and 20. The expression for the friction factor is:

$$f = \left[\left((2.0895TR + 30.493)Re^{(-0.0304TR-0.5461)}\right)^{10} + \left((0.0028TR + 3.8309)Re^{(-0.0137TR-0.312)}\right)^{10}\right]^{1/10} \tag{2}$$

The expression for the Nusselt number is:

$$Nu = \left[\left((-0.042TR + 0.4364)Re^{(0.0133TR + 0.5909)}\right)^{12} + \left((-0.02TR + 0.2319)Re^{(0.0103TR + 0.6811)}\right)^{12}\right]^{1/12} \quad (3)$$

When comparing predictions using Eq(2) and Eq(3) with numerically obtained values, the average absolute error is determined for each Reynolds value within the correlation's operational range. For the friction factor, the maximum and minimum average absolute errors are 13 % for TR = 4 and 4 % for TR = 5.88. In the case of the Nusselt number, the maximum and minimum errors are 7 % for TR = 4 and 2 % for TR = 2. Figure 2 illustrates the predictions for TR = 2, with errors of 12 % and 3 % for f and Nu.



a) b)

Figure 2: Comparison between experimental data and generalised expressions for TR=2. a) Friction factor, b) Nusselt number

3.1 Promoter selection methodology.

The generalised expressions shown in Eq(2) and Eq(3) operate within the range of $2 \leq TR \leq 7$, considering TR as a continuous function. For TR = 2, a denser turbulence promoter exhibits higher convective coefficient and elevated pressure drop. In contrast, for TR values of 7, lower convective coefficients and reduced pressure drops are obtained (Figure 3). The methodology proposed in this study allows for the use of turbulence promoters with two design purposes: firstly, to enhance energy recovery and reduce external heating consumption; and secondly, to restore thermal load when disturbances affect downstream-placed heat exchangers. Based on this, a low TR value of 2 significantly increases thermal load and the ability to restore temperatures within a network due to temperature perturbations. On the other hand, for larger TR values, the increase in thermal load is less pronounced, as is the temperature restoration.

Smooth tube | **Perforated Twisted Tape in tubes**

-Fixed thermal load
$Q, \Delta P$

Twist Ratio Range

TR=2
Greater: $h, Q, \Delta P$
Better counteracting effects

TR=7
Smaller: $h, Q, \Delta P$
Diminished counteracting effects

Figure 3: Relative performance of perforated twisted tapes for a range of TR

3.2 Temperature Disturbance

When thermal load is increased through the implementation of turbulence promoters, it leads to changes in the outlet temperatures of the equipment. If this equipment is part of a heat exchanger network, these changes can cause downstream disturbances within the network. To account for the propagation of such disturbances throughout the network, Delgado-García et al. (2022) proposed a model of thermal pathways and response equations to quantify the effects of temperature perturbations.

4. Case study

A heat exchanger network is analysed which includes a hot stream and two cold streams as shown in Figure 4. The equipment in this network consists of shell-and-tube heat exchangers, with design details, operating conditions, and physical properties of the streams specified in Table 1 and Table 2. The application of perforated twisted tapes occurs at two moments: First, the possibility of increasing the thermal load on exchanger E1 to reduce the load on heater ST1 is identified. Second, the effects of temperature disturbance on exchanger E2 are quantified. It is observed that the thermal load on E2 decreases, necessitating an increase in the thermal load to restore the temperature of stream C1 to 160 °C.

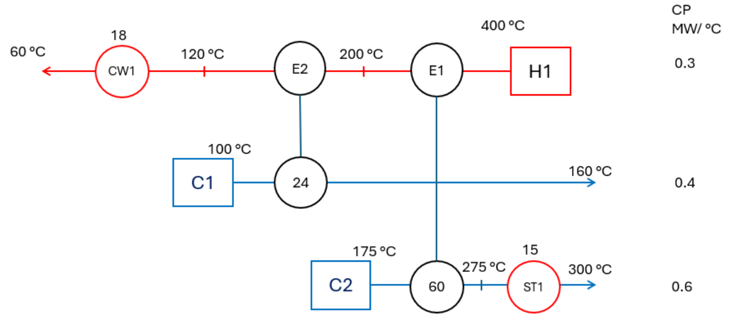
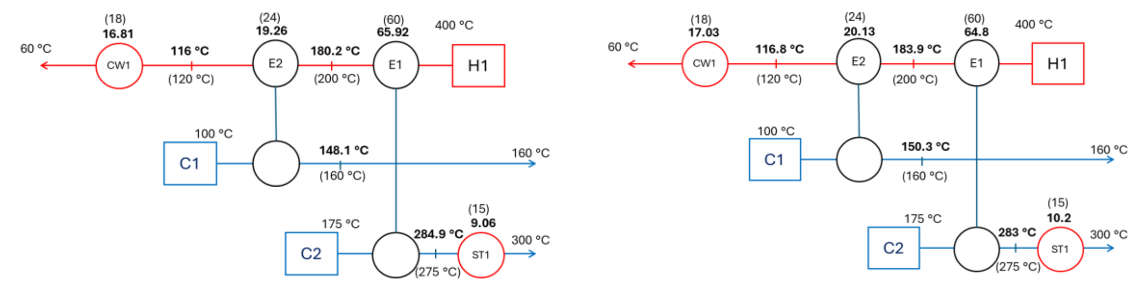


Figure 4: Heat exchanger network for case study

When turbulence promoters are incorporated into heat exchanger E1 with a TR of 2, there is a 9.91 % increase in heat recovery (equivalent to 5.92 MW), resulting in a total thermal load of 65.92 MW. For a TR of 7, an 8 % increase (equivalent to 4.8 MW) is achieved, with a total thermal load of 64.8 MW. However, pressure drop also rises. The original pressure drop within the tubes in exchanger E1 is 116.52 Pa. When using perforated twisted tapes (PTT) with TR=2, the pressure drop increases to 377 Pa, and for TR=7, the pressure drop is 199.1 Pa, representing an increase of 324 % and 70 %. Additionally, applying PTT with TR=2 reduces the thermal load on CW1 by 1.19 MW and on ST1 by 5.94 MW. With TR=7, there is a saving of 0.97 MW in CW1 and 4.8 MW in ST1. Figure 5 illustrates the resulting temperatures and thermal loads due to the use of turbulence promoters, with the original values indicated in parentheses.

Table 1: Operating conditions of streams in heat exchanger network

Stream No.	Stream type	T supply (◦C)	T target (◦C)	m (kg/s)	CP (MW/°C)
1	Hot	400	60	105.64	0.3
2	Cold	100	200	95.24	0.4
3	Cold	175	300	187.50	0.6



a) b)

Figure 5: Heat exchanger network using turbulence promoters in E1 a) TR=2, b) TR=7

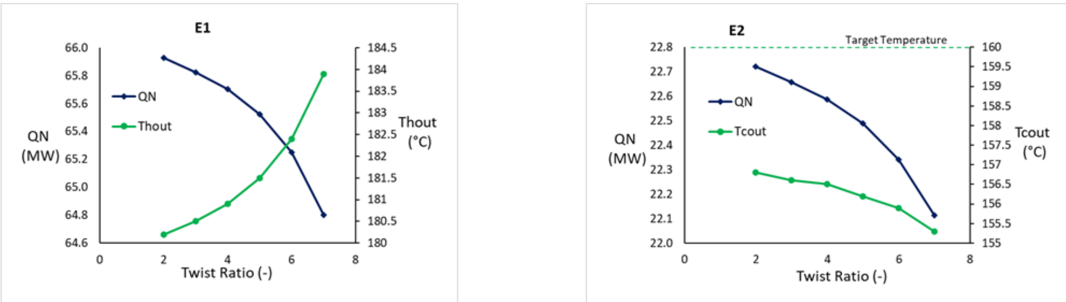
4.1 Disturbance analysis

Although significant energy savings are achieved in cooling and heating, there are also adverse effects observed in the network due to the propagation of disturbances. For instance, consider a scenario where the thermal load on heat exchanger E1 is increased with a TR of 2. As a result, the thermal load on E2 decreases from 24 MW to 19.26 MW. This reduction leads to a drop in the outlet temperature of the cold stream (C1) from 160 °C to 148.1 °C. Now, examining the case where a TR of 7 is implemented in E1, it is observed the heat load of exchanger E2 decreases from 24 MW to 20.13 MW, resulting in an outlet temperature of stream C1 at 150.3 °C. To restore the outlet temperature of stream C1 back to the desired 160 °C, the heat load on exchanger E2 must be increased. One effective approach to achieve this is by incorporating perforated twisted tapes within the unit.Figure 6a illustrates the behaviour of the heat load and the hot stream outlet temperature of heat exchanger E1 for different twist ratios. Figure 6b considers the disturbance caused by using a promoter with a TR of 7 in E1 while different twist ratios are used in E2. It is observed that even when using the denser promoter (TR=2) in E2, the outlet temperature of the cold stream (C1) from E2 does not return to its original value of 160

°C. To illustrate the thermal load and temperature results in the network, two cases are presented in Figure 7. Case A: Incorporating TR=2 promoters in both heat exchangers (Figure 7a); Case B: Using a TR=7 promoter in E1 and a TR=2 promoter in E2 (Figure 7b).

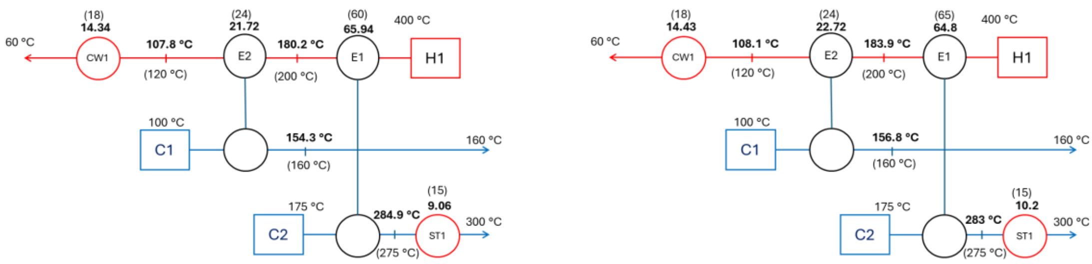
Table 2: Heat exchanger specifications of E1 and E2

	Exchanger E1		Exchanger E2	
	Tube side H1	Shell side C2	Tube side H1	Shell side C1
Heat load (MW)	60		24	
Heat transfer area (m^2)	7,315.31		4,607.37	
Heat capacity (kJ/kg °C)	2.84	3.2	2.84	4.2
Viscosity (kg/ms)	0.00034	0.00065	0.00034	0.0008
Conductivity (W/m °C)	0.19	0.45	0.19	0.59
Density (kg/m^3)	750	920	750	998
Pressure drop (Pa)	116.52	46,880.84	296.99	17,001.29
Heat transfer coefficient (W/m^2 °C)	236.97	462.46	357.11	520.31
Number of tubes (-)	11,696		7,005	
Tube passes (-)	1		1	
Thermal conductivity of material (W/m °C)	86		86	



a) b)

Figure 6. Heat exchanger performance of E1 and E2 with the incorporation of different TR



a) b)

Figure 7: Heat exchanger network using turbulence promoters in E2 a) Case A, b) Case B

In Case A, the thermal load in E2 is restored to 90.5 % of its original duty. The outlet temperatures for this heat exchanger are as follows: the hot stream (H1) exits at 107.8 °C, resulting in a reduction of the cooling load on CW1 of 3.66 MW. On the other hand, the cold stream C1 exits at 154.3 °C, necessitating the use of external steam services in C1 to restore it to its target temperature of 160 °C. The combined overall effect for Case A is a cooling load of 3.66 MW and a net heating saving of 3.66 MW. In Case B, the thermal load on E2 is restored to 94.6 % of its original value. The hot stream (H1) at the inlet of CW1 is 108.1 °C reducing the external cooling 1.28 MW, while the cold stream at the outlet reaches 156.8°C. Under these conditions, the net cooling savings amount to 3.52 MW, and the heating savings are 3.52 MW.

5. Conclusions

The conclusions of this study are as follows:

- In retrofit applications, the use of turbulence promoters must be approached with special attention to the generation and propagation of disturbances that could impact process operation.
- The network structure determines the need to counteract the adverse effects of introducing disturbances into other process streams.
- In the specific case study analysed, employing turbulence promoters with higher density (TR=2) in exchanger E1 results in a potential energy savings of 5.94 MW. However, this savings is diminished due to the adverse effects of disturbance propagation. Implementing promoters with TR=2 in E2 allows for an increase in its thermal load, but even so, restoring temperatures requires consuming an additional 2.28 MW of external energy, resulting in a net savings of 3.66 MW.
- If lower-density turbulence promoters (TR=7) are used in unit E1, it could result in a savings of 4.8 MW. However, to restore the temperature of unit C1 to 160 °C, an external energy input of 1.28 MW is required. This reduces the net savings to 3.52 MW.
- It is essential to complement the study with an economic feasibility analysis. This analysis must consider the increased pumping costs and expenses associated with incorporating additional equipment to restore process temperatures.

Nomenclature

CP – Heat capacity mass flow rate, MW/ °C
d_p – Perforation diameter of twisted tape, m
D – Twisted tape diameter, m
ΔP – Pressure drop, Pa
f – Friction factor, -
h – Heat transfer coefficient, W/m^2 °C
$\dot{m}$ – Mass flow rate, kg/s
Q – Original Thermal load, MW
QN – New Thermal load, MW
Re – Reynold number, -
t_h–Twisted tape thickness, m
y – Twist pitch , m

Abbreviations

CFD – Computational Fluid Dynamics
CP-TT – Centrally perforated twisted tape
PTT – Perforated Twisted tape
PTTT – Perforated Triple Twisted Tape
SC-TT – Square Cut Twisted Tape
TEF – Thermal Enhancement Factor
TR – Twist ratio
TT – Twisted tap

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