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Selection of Heat Transfer Promoters for Heat Exchanger Retrofit

Paola Durán-Plazas, Martín Picón-Núñez*, Jesús I. Minchaca-Mojica

Department of Chemical Engineering, Uiversity of Guanajuato, Naria Alta S/N, Guanajuato, Gto., Mexico, CP. 36050
picon@ugto.mx

From the thermal point of view, the main operational problems conventional heat exchangers of the tubular type face are the prevalence of the laminar sublayer on the surface immediately adjacent to the heat transfer surface that is responsible for low heat transfer coefficients. A passive means of disrupting this layer is using mechanical inserts known as turbulence promoters. The search for new geometries with better thermohydraulic performance has increased the number of designs available in the open literature. In sight of the large number of different options available for use in design, the selection of the right type of insert for a given application is the focus of this paper and introduces an alternative approach for the selection of turbulence promoters based on the thermal and hydraulic lengths. The approach allows for a quick comparison of the thermo-hydraulic performance in design and retrofit. It considers the effect of the magnitude of the heat transfer coefficient of the opposing stream as well as the effect of viscosity on the selection. Some turbulence promoters out of the many that have been published are used to demonstrate the methodology. This work confirms that the two conditions that result in a greater impact of the turbulence promoter on the overall heat transfer coefficient are when the tube-side presents the higher thermal resistance and when a highly viscous fluid flows through the tubes.

1. Introduction

Turbulence promoters can be used in the design of new shell and tube heat exchangers and in the retrofitting of existing ones. The thermo-hydraulic characteristics of a turbulence promoter can be such that, in design, it results in smaller heat transfer area and lower pressure drop compared to the use of smooth tubes. In addition, in retrofit situations of existing equipment, an increase in heat recovery with the same area could be achieved at the cost of increased pressure drop.

Amidst the performance comparison methods available to date, the performance criterion method (RC) relates the heat transfer obtained with inserts to the heat transfer obtained with a smooth pipe for the same pumping power and heat exchange surface area (García et al., 2007). This term is expressed as:

$$RC = Nu_a / Nu_o \tag{1}$$

Eiamsa-Ard et al. (2010), proposed a thermal performance factor (η) represented by Eq(2), where h_t is the heat transfer coefficient of the pipe with turbulence promoter, and h_p is the heat transfer coefficient of the smooth pipe. The subscript pp indicates constant pumping power. If η > 1 the system efficiency increases.

$$\eta = \left(h_t / h_p\right)_{pp} \tag{2}$$

The dimensionless number Fc, referred to as the field synergy number, represents the relation between the velocity and temperature gradient fields. A good synergy between the velocity and temperature gradient field is expected if Fc is close to 1.0.

$$Fc = Nu / Re\ Pr \tag{3}$$

Dang and Wang (2021) studied the enhancement mechanisms of convective heat transfer by means of the JF factor to evaluate the heat transfer performance and friction characteristics.

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$$JF = \frac{Nu/Nu_p}{f/f_p} \tag{4}$$

The subscript p represents the heat transfer and friction characteristics of the corresponding smooth pipe. JF is the thermal performance factor based on an identical flow rate. This definition indicates that the higher the JF factor, the better the heat transfer performance of the tube and the greater the heat transfer enhancement is relative to the smooth tube.

The PEC performance evaluation criterion considers both the change in the coefficient of friction and the Nusselt number, providing a picture of the trade-off between the heat transfer enhancement and the pressure drop penalty. If PEC > 1, the heat transfer enhancement is greater than the increase in pressure drop, making the tube enhancement favorable compared to the smooth tube design. The higher the η, the more favorable the enhancement is (Mousa et al., 2021).

$$PEC = \frac{Nu/Nu_p}{\left(f/f_p\right)^{1/3}} \tag{4}$$

Thermodynamic performance methods evaluate the performance of inserts by means of the Second Law of Thermodynamics using the concept of entropy. In a heat transfer process, entropy is generated due to the existence of a temperature driving force and due to the loss of energy in the form of pressure drop. Entropy generation is reduced with the incorporation of turbulence promoters since the temperature differences are smaller, but the existence of pressure drop causes the opposite effect, this is, entropy generation increases.

Keklikcioglu et al. (2017) in their experimental study, addressed the entropy generation in a circular tube with coiled wire inserts for a range of Reynolds numbers from 2,731 to 27,732. The experimental results revealed that the entropy generation number increases with the increase of Reynolds number.

Khanmohammadi and Mazaheri (2019), performed numerical modeling of heat transfer in a tube with single braided tape and double braided tape (coaxial) as enhancement elements. They concluded that low twist ratios in the two types of inserts lead to lower total entropy generation. In addition, the results of exergy destruction rate and Second Law efficiency show that the coaxial ribbon performs better than the single ribbon.

Chaurasia and Sarviya (2021) developed an experimental and numerical analysis study to evaluate the thermo-hydraulic performance using the entropy generation on a nanofluid flow in helical screw inserts in a tube with single and double strip and different torque ratios (TR) in laminar flow. The results showed that the entropy production number is at a lower value with double strip inserts than with single strip helical screw inserts with a low value of torque ratio. In their research Abu-Hamden et al. (2021), applied CFD (computational fluid dynamics) for three-dimensional simulation of fluid flow and turbulent thermal energy transferred through a circular duct with coiled finned wire inserts to evaluate the impact of the Prandtl and Reynolds numbers and geometric variables. The results show that if the Reynolds number increases by 100 % in the covered range, the thermal efficiency and the dimensionless exergy loss are reduced by 15.9 % and 9.4 %.

Two limitations of most of the comparison methods are that they only focus on the effect on heat transfer and friction characteristics, and they consider only the effect on the tube side of the heat exchanger. Direct comparison based on Nusselt and friction values provides a partial view of the effectiveness of a certain geometry. This is because many combinations of values give results that are not necessarily the best option for a certain application. A similar situation occurs when only one side is considered. The final choice of the promoter must be in terms of the actual effect on the heat duty, the pressure drop and the exchanger dimensions.

This paper presents an approach that bridges the gap between exchanger dimensions and thermohydraulic performance to guide the selection of turbulence promoters in design and retrofit applications. For a more accurate prediction of the benefits in terms of reduced surface area or increased heat duty right form the performance comparison stage, the effect of the heat transfer coefficient of the opposing stream must be taken into consideration as is demonstrated in this work.

2. Thermal and hydraulic length

For the design and selection of a turbulence promoter, this work uses the model based on the concept of thermal length and hydraulic length introduced by Picón and Melo (2020). The thermal length (L_T) represents the length of tube that a fluid requires to transmit or absorb a certain amount of heat and reach the target temperature. On the other hand, the hydraulic length represents the length of tube that would be required to fully absorb the specified pressure drop. A turbulence promoter is suitable for an application when the hydraulic length is larger than the thermal length. The thermohydraulic performance of turbulence promoters is generally reported in terms of the Nusselt number and the friction factor in the following form:

$$Nu = aRe^{b}Pr^{c}N^{d} \tag{5}$$

$$f = xRe^{-y}p^{-z} \tag{6}$$

Where a, b, c, d, x, y and z, are coefficients and exponents of the correlated equations; Nu, Re and Pr are the Nusselt, Reynolds, Prandtl numbers and f is the friction factor. N and p are expressions that depend on the geometrical features of the type of insert under consideration. To consider the effect of the heat transfer coefficient on both streams the overall heat transfer coefficient is calculated from its general expression:

$$1/U = 1/h_1A + 1/h_2 + R_k + R_f \tag{7}$$

The heat transfer area and the exchanger thermal length are calculated from:

$$A = \ Q/U\ F\Delta T_{LM} \tag{8}$$

$$L_T = \ A/\pi\ d_i\ N_t \tag{9}$$

The hydraulic length can be derived from the expression of the pressure drop across the core of the heat exchanger and is given as:

$$L_H = 2d_i\Delta P/\rho.f.v^2 \tag{10}$$

Where A is the heat transfer surface area, Q is the heat duty, h_1 and h_2 are the heat transfer coefficients, U is the overall heat transfer coefficient, F is the correction factor of the logarithmic mean temperature difference, d_i is the inner tube diameter, N_t is the number of tubes, R_k is the resistance due to conduction in the wall and R_f is the fouling factor.

3. Results

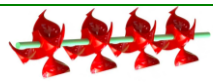
The first part of this section shows the results considering the thermal resistance of the opposing stream in five different scenarios: a) LT1: base case, no effect of opposing stream is considered; b) LT2: where h_{tube}<<h_{shell} with phase change; c) LT3: where h_{tube}<<h_{shell} without phase change; d) LT4: where $h_{tube} = h_{shell}$; e) LT5: where h_{tube}>>h_{shell}. Table 1 shows the operating and geometrical data for the study. For the sake of the analysis in this work, a set of 8 promoters were chosen at random from the open literature. Their thermohydraulic expressions and geometry are given in Table 2. Figure 1 shows the results using inserts numbered from 0 to 8, where the blue bar represents the hydraulic length and the rest of the bars represent the thermal length for each scenario. For interpretation, feasible designs are those where the hydraulic length is larger than the thermal length. The opposite condition means that the pressure drop is absorbed even before the thermal load has been fulfilled and this is unacceptable in terms of design.

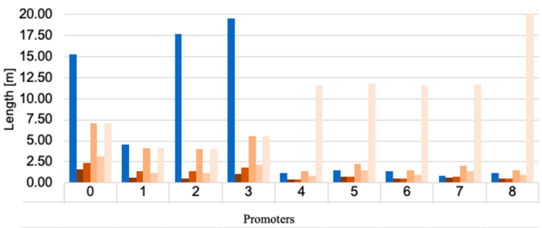
Table 1: Data for case study 1

Case study	Tube inner diameter (mm)	Tube outer diameter (mm)	Reynolds number	Pressured drop (kPa)	Viscosity (Pa·s)	Density (kg/m^3)	Heat capacity (J/kg °C)	Thermal conductivity (W/m °C)
	7.67	13.1	10,000	15	0.0005474	988.02	4,182	0.64
	T_{in} tube °C	T_{out} tube °C	Tw °C	T_{in} shell °C	T_{out} shelll °C	ΔT_{LM} °C	h_{shell}	
LT1	20	80	100	-	-	43.28	-	
LT2	20	80	-	100	100	43.28	10,000	
LT3	20	80	-	100	30	14.43	10,000	
LT4	20	80	-	100	100	43.28	h_{tube}	
LT5	20	81	-	100	30	14.43	500	

In the case of promoters No. 1, 2 and 3, Figure 1 shows that they give feasible designs as the hydraulic length is larger than the thermal length; however, it is notorious that for promoters 4, 5, 6, 7 and 8 the design are more constrained. Out of the five different inserts analysed in Figure 1, insert 2 gives the best results as the thermal lengths are the shortest; this has also another implication, at that same height the pressure drop absorbed is only a fraction of the allowed one resulting in an exchanger that absorbs only a fraction of its allowed pressure drop.

Table 2: Thermohydraulic performance and geometrical data for tube inserts

No.	Type	Geometrical Parameter	Nusselt number	Friction factor	Schematic
0	Smooth tube	-	$Nu = 0.023Re^{0.8}Pr^{0.4}$	$f = 0.184Re^{-0.2}$	
1	perforated twisted) M.M.K. Bhuiya et al. (2013)	Rp= 0.0450	$Nu = \left(0.0002R_p^3 - 0.0046R_p^2 + 0.3334R_p + 0.6569\right) * Re^{(.000005R_p^3 - 0.0013R_p^2 + 0.00734R_p + 0.6569} * Pr^{0.33}$ Error ± 4 %	$f = \left(-0.0027R_p^3 + 0.0583R_p^2 + 0.0455R_p + 24.536\right) * Re^{\left(0.00005R_p^3 - 0{,}0022R_p^2 + 0.012R_p - 0.6006\right)}$ Error ± 4 %	
2	V-Cut twisted tape P. Murugusen et al. (2010)	y=4, de/W=0.43, w/W=0.43	$Nu = 0.00296Re^{0.853}Pr^{0.33}y^{-0.222} + \left(1 + {}^{de}/_{W}\right)^{1.148} + \left(1 + \frac{w}{W}\right)^{0.751}$ Error ± 6 %	$f = 8.632Re^{-0.615}y^{-02.69} + \left(1 + \frac{de}{W}\right)^{2.447} + \left(1 + \frac{w}{w}\right)^{-1.914}$ Error ± 10 %	
3	Square-cut Murugusn et al. (2010)	y= 4.4	$Nu = 0.041Re^{0.826}Pr^{0.33}y^{-0.228}$ Error ± 6%	$f = 6.936Re^{-0.579}y^{-0.259}$ Error ± 8%	
4	Quadruple perforated- delta-winglet pairs (PW-XT). Skullong et al. (2016)	B=0.2 p=1.4	$Nu = 0.194Re^{0.777}Pr^{0.4}B^{0.317}p^{-0.373}$ Error ± 7.5 %	$f = 5.305Re^{-0.076}B^{0.976}p^{-0.989}$ Error ± 7.5 %	
5	Straight tape with center wings (T-W) and B-wing Eiamsa and Promvong (2011)	ep=1, ew=0.67	$Nu = 0.101Re^{0.733}Pr^{0.4}ep^{0.265}ew^{-0.287}$ Error ± 7 %	$f = 0.898Re^{-0.094}(e_p)^{-0.516}(e_w)^{0.655}$ Error ± 8 %	
6	Punch delta winglet vortex generator (PDWVG) Wijayanta et al. (2017)	∝= 50	$Nu = 0.013Re^{1.036}Pr^{0.3}\left(\frac{\propto}{90}\right)^{0.548}$ Error ± 6 %	$f = 37.74Re^{0.493}\left(\frac{\propto}{90}\right)^{0.37}$ Error ± 6 %	
7	Double-sided delta-winglet tape Skullong et al. (2016)	∝= 45	$Nu = 0.122Re^{0.777}Pr^{0.4}(1 + tan\alpha)^{0.427}(\mathrm{Pr}+1)^{-0.6}$ Error ± 10 %	$f = 1.546Re^{-0.0726}(1 + tan\alpha)^{1.605}(\mathrm{Pr}+1)^{-1.39}$ Error ± 10 %	
8	Twisted cross-baffles Nanan et al. (2016)	p/D=1.5	$Nu = 0.093Re^{0.797}Pr^{0.4}\left(\frac{P}{D}\right)^{-0.403}$ Error ± 3.8 %	$f = 1.414Re^{-0.06}\left(\frac{P}{D}\right)^{-1.036}$ Error ± 4 %	



	0	1	2	3	4	5	6	7	8
LH	15.31	4.57	17.70	19.55	1.22	1.54	1.38	0.85	1.16
LT1	1.57	0.60	0.56	1.06	0.44	0.74	0.49	0.68	0.47
LT2	2.36	1.39	1.36	1.85	0.46	0.76	0.51	0.69	0.48
LT3	7.08	4.17	4.07	5.56	1.38	2.27	1.53	2.08	1.45
LT4	3.13	1.19	1.13	2.12	0.89	1.48	0.99	1.35	0.94
LT5	7.08	4.17	4.07	5.56	11.56	11.77	11.60	11.73	49.01

Figure 1: Thermal and hydraulic length considering the effect of opposing stream on the selection of the turbulence promoter

The case of a new design is analysed using the data in Table 3. The tube geometry chosen for the study is: length, 4.4 m; tube inner diameter, 0.016 m; tube outer diameter, 0.02 m. Figure 2(a) shows the thermal and hydraulic length when the design is carried out using inserts. From the results it is seen that, promoters 2 and 3 give feasible designs since their hydraulic length is much larger than the thermal length. The hydraulic length in the case of all other promoters is shorter than the thermal length. Figure 2(b) shows the case of integrating the promoter directly in the heat exchanger for the case of retrofit. As expected, the thermal load is increased along with the pressure drop. In this case promoters 2 and 3 still give better results. The increase of pressure drop of the other inserts are unsuitable for the application.

Figure 3 shows a case where retrofit is performed in an existing exchanger that has four tube passes and whose tube dimensions are: length, 4.4 m; tube inner diameter, 0.016 m; tube outer diameter, 0.02 m. In this case, for the sake of analysis, a hypothetic viscosity of 0.008 Pa·s is considered. When retrofit involves a viscous fluid in the tube, the exchanger shows a considerable increase in pressure drop as shown in Figure 3(a) where it is seen that the pressure drop in all cases goes beyond the thermal length. This means that unless the pressure drop be increased, the system falls outside specifications. A design option available to release the pressure drop constraint, is to seek to modify the exchanger internal arrangement to reduce the velocity of the fluid in the tubes. This can be achieved by reducing the number of passes from 4 to 2. The results are shown in Figure 3(b). Under this new arrangement, despite the fact the fluid velocity is halved, the turbulence promoter still creates larger heat transfer coefficients; the benefit is seen in the pressure drop; only promoters 2 and 3 are suitable for the application.

Table 3. Data for case study 2

	Case study	
	Tube side (Cold fluid)	Shell side (Hot fluid)
Mass flow rate (kg/s)	68.8	28
Inlet temperature (K)	298	368
Outlet temperature (K)	313	313
Heat capacity (J/kg K)	4,200	2,800
Density (kg/m^3)	995	750
Viscosity (Pa·s)	0.0008	0.00034
Thermal conductivity (W/m K)	0.59	0.19
Pressure drop (Pa)	16,787	66,803

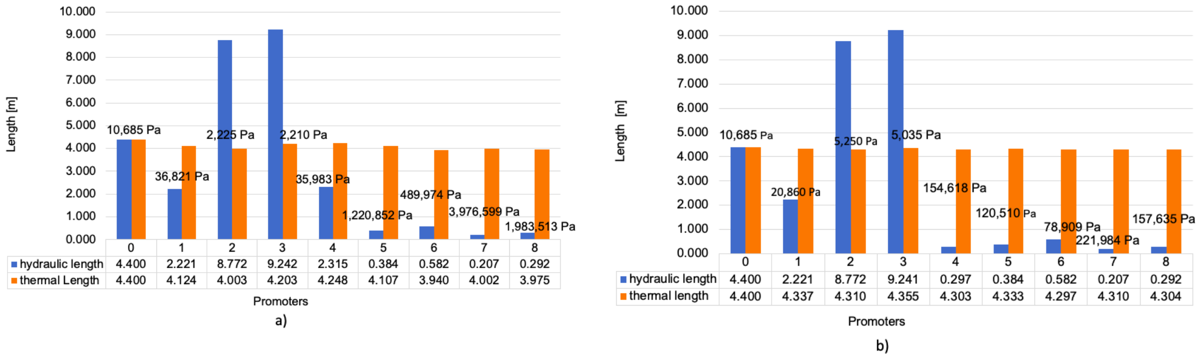


Figure 2: Thermal and hydraulic length for design with turbulence promoters: a) Design, b) Retrofit

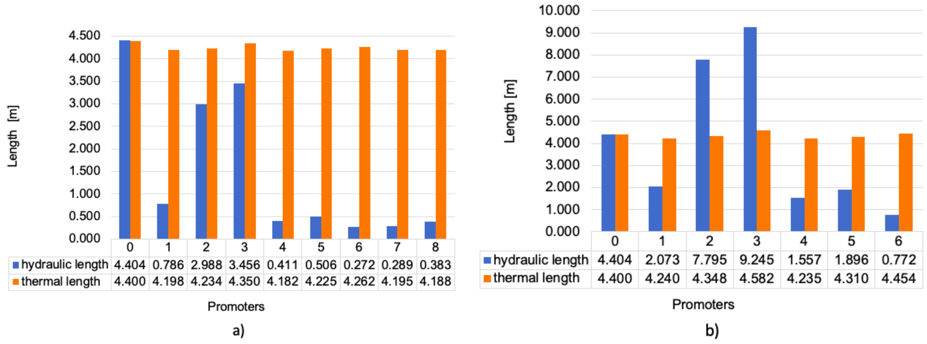


Figure 3: Heat exchanger retrofit considering a viscous fluid: a) Inserts without changing exchanger internals, b) Reduction of number of passes from 4 to 2

4. Conclusions

- The consideration of the two fluids in the evaluation of the thermal length-hydraulic length model does not alter the selection of the promoter but reduces the expected effect in terms of the heat transfer area to meet the heat duty. Slightly higher surface areas are required.
- This work confirms findings that the impact of the use of turbulence promoters on the overall heat transfer coefficient is when the tube side fluid has the highest resistance to heat transfer and when highly viscous fluids flow through the tubes.

- Performance comparison methods based only on the comparison of the Nusselt number and friction factor give only one part of the information for the appropriate selection of turbulence promoters in design and retrofit. The method presented here gives a clearer view and guides the designer to the most suitable device for the specific application.
- The appropriate selection of a turbulence promoter is important to achieve two objectives depending on the type of application: a) in design, reduce the heat transfer area for a certain thermal load, b) in retrofit, for the same heat transfer area installed, increase the thermal load without exceeding the allowed pressure drop.
- Proper selection of turbulence promoters depends on the availability of accurate expressions for their thermohydraulic performance. Future work includes the extension to proper selection in the retrofit of heat recovery networks for increased heat recovery considering pressure drop limitations.

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