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Globally optimal design of intensified shell and tube heat exchangers using complete set trimming

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ABSTRACT

Intensified shell and tube heat exchangers have been proposed for their use in the grassroots and retrofit designs of heat exchanger networks. Among them are those featuring helical baffles, externally and internally finned tubes, twisted-tape inserts, coiled-wire inserts, and twisted-tube exchangers. Some of these devices are suited for the tube side and others for the shell side, while the utilizations of different devices on both sides simultaneously are also possible. The optimal design of these exchangers has been attempted by several authors using different techniques with varied successes. In particular, finding local solutions require good initial points. Also, global solutions using global solvers are completely elusive. In the present work, we study the computational performance of the use of Complete Set Trimming to obtain the globally optimal design of intensified heat exchangers featuring the minimum capital cost and total annualized cost. The results of examples indicate the Set Trimming competes well with commercial global solvers that are highly time-consuming or sometimes incapable of solving the design problem.

1. Introduction

Shell and Tube (S&T) heat exchangers are the most common heat transfer devices widely used in chemical process industries. Therefore, the optimal design of S&T heat exchangers has considerable impacts on the capital and operational expenses of process plants, and because of that, their design attracts the attention of both practitioners and academic researchers.

Many textbooks (Kern, 1950; Cao, 2010), old and even new, proposed a trial-and-verification procedure to design S&T heat exchangers, while the nature of this approach was critically discussed in detail by Costa and Bagajewicz (2019). Essentially, this approach is a heuristics-driven method, which starts by assuming an overall heat transfer coefficient followed by an area estimation through the logarithmic mean temperature difference (LMTD) method. Then, it proposes the selection of a suitable geometry, with which the heat transfer coefficients and pressure drops are calculated. If the heat transfer area is higher than the required one (according to a given excess area) and the evaluated pressure drops are lower than the available ones, the procedure stops. Otherwise, the changes to the geometry are repeatedly made until all constraints are fulfilled. However, the guide as to how to pick the geometry and what to modify to arrive at an acceptable solution is rarely given and described. When it is, they are in a form of suggestions but not systematic approaches. The success of this traditional approach highly depends on the experience of skilled designers. In the trial and verification procedures described in previous literature, no mention is made of the need for optimal solutions, neither local nor global, to improve the first viable answer, an activity that is not even encouraged.

To depart from the trial and verification, many optimization approaches have been developed in the past few decades for the conventional S&T heat exchanger design, by minimizing either the heat transfer area (which has a direct relation with the capital cost) or the total annualized cost (TAC) including the capital cost of area and the operational cost of pumping power. These approaches are mainly of three classes: Heuristics and Enumeration procedures, Stochastic/Metaheuristics methods, and Mathematical Programming methods.

Heuristics and Enumeration procedures employ algorithms aided by graphs (Muralikrishna and Shenoy, 2000; Eryener, 2006). In this approach, the search regions of all feasible solution candidates are systematically analyzed using the operational and geometrical constraints, then the optimum values of design parame-

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ters are obtained on the proposed pressure drop graphs. However, the comparisons to trial and verification or other approaches are not made. Note that Heuristics and Enumeration procedures do not directly relate to the enumeration procedures which we use in this article.

Stochastic/Metaheuristics methods apply various types of optimization algorithms: Genetic Algorithms (Wildi-Tremblay and Gosselin, 2007; Ponce-Ortega et al., 2009), Simulated Annealing (Chaudhuri et al., 1997), Particle Swarm Optimization (Ravagnani et al., 2009; Patel and Rao, 2010), as well as some others (Asadi et al., 2014; Mohanty, 2016). One shortcoming of the stochastic/metaheuristics methods is that they usually demand parameter tuning, which strongly depends on human interventions and expertise, and for this reason, the optimal set of parameters can't be generalized to every problem. In addition, these algorithms are incapable of guaranteeing global optimality, although anecdotal comparisons show that they often come close.

Finally, Mathematical Programming uses mixed-integer nonlinear programming (MINLP) procedures since the models are nonlinear and contain binary variables (older attempts are based on nonlinear programming formulations, containing only continuous variables). Mizutani et al. (2003) applied a mixed-integer nonlinear model based on the Bell-Delaware method to optimize the capital and operational costs of S&T exchangers. The model considered discrete variables (diameter, length, etc.) as continuous variables. As a result, since heat transfer coefficients are highly nonlinear, the MINLP approaches used, have difficulties in finding the global optimum. Later, Ravagnani and Caballero (2007) stated that the solution of Mizutani et al. (2003) did not follow the manufacturing standards of the Tubular Exchanger Manufacturer Association (TEMA). They proposed a counting table to define the design variables of mechanical components according to TEMA standards. Ponce-Ortega et al. (2006) also used a mixed-integer nonlinear model to design several S&T exchangers assuming a constant overall heat transfer coefficient. In turn, Onishi et al. (2013) developed a new mixed-integer nonlinear model with partial objectives to achieve the optimum S&T heat exchanger design sequentially. The authors commented the model needed some manual initializations and tight bounds on variables, and so the optimization process was time-consuming. Aside from possible convergence problems associated with the local solvers used, these solvers did not guarantee global optimality. Departing from the above LMTD-based models, Kazi et al. (2021) proposed a differential-algebraic equation (DAE) distributed model where discretization schemes are applied to solve the differential equations. Unlike traditional works, the DAE model did not use an LMTD-type formula and correction factor. Their solution results showed the DAE model was computationally efficient making it applicable to the simultaneous heat exchanger and network synthesis (Chang et al., 2021). The approach departs from other works, because it is a discretized model making the comparisons to MINLP-based approaches difficult. The common feature of all the above mathematical programming models is that global optimality cannot be guaranteed especially when they were solved by local solvers.

More recently, reformulations exploiting the discrete nature of the design variables of S&T heat exchangers were proposed. Indeed, Gonçalves et al. (2017a) proposed a reformulation technique to convert the mixed-integer nonlinear model (MINLM) based on Kern equations for heat transfer and pressure drop into a mixed-integer linear model (MILM) that was solved using MILP procedure. The result was that the global optimum of the original model was obtained without any approximations and convergence problems. Their model made use of the standard values of several mechanical parts expressed as discrete choices. Later, Gonçalves et al. (2017b) gave several alternative formulations using a combinatorial representation of the search spaces to speed up the solution of the MILP. These alternatives were based on a binary-based combinatorial representation of the search space, while the computational efforts were reduced. Gonçalves et al. (2019) extended their approach to the use of the Bell-Delaware method. Pereira et al. (2021) formulated an integer-linear model (ILM) for the design of horizontal S&T condensers. Because of the linear nature of these models, the global optimum was always achieved without the need for any initialization.

Lemos et al. (2020) applied a new optimization procedure namely Complete Set Trimming, which uses a set-based combinatorial representation of the search space and uses the inequality constraints to eliminate portions of the search space. The inequalities are applied sequentially rendering smaller and smaller subsets of candidate solutions. At the end of Set Trimming, the optimum can be identified by sorting. Their test results indicate that Set Trimming is substantially faster than linear models.

The main purpose of heat transfer intensification is to increase the value of heat transfer coefficients for a given size exchanger. In addition, different technologies of heat transfer intensification have been applied to the retrofit designs of heat exchanger networks successfully (Wang et al., 2012; Pan et al., 2016). Based on different geometric characteristics, heat transfer intensification technologies could be generally classified into those applied to the tube side and those applied to the shell side, sometimes simultaneously. Some depart from the classical TEMA geometries, having a unique architecture (e.g. Twisted-Tube heat exchanger).

Many authors contributed to the detailed heat transfer coefficients and pressure drop calculations of the intensified S&T heat exchangers. Manglik and Bergles (1993) calculated the tube-side heat transfer coefficients and pressure drops of heat exchangers with twisted-tape inserts. Sarma et al. (2005) predicted the friction coefficients for the tube side with twisted-tape inserts, indicating how twisted ratios influence heat transfer characteristics inside tubes. In addition, Sethumadhavan and Raja Rao (1983) analyzed the effects of coiled-wire inserts on stream flows. Both laminar and turbulence flows were investigated within a wide range of Reynolds numbers. Ravigururajan and Bergles (1996) studied the roughness geometries and configurations of coiled wire. Empirical correlations were developed to estimate the pressure drops and convective heat transfer coefficient of the tube side. Huq et al. (1998) researched the heat transfer of flows with low Reynolds number in the internally finned tubes. The flow at high Reynolds number was investigated by Jensen and Vlakancic (1999) who provided some empirical correlations. To decrease error, Zdaniuk et al. (2008) conducted experiments and developed accurate correlations for internally finned tubes.

Moreover, twisted-tube heat exchangers enhance heat transfer on the tube side and reduce the pressure drop on the shell side. The tubes in this type of exchanger are shaped into an oval section with superimposed twists. Bishara et al. (2009) simulated the heat transfer of twisted-tube heat exchangers. Their examples showed the shell-side pressure drop of the twisted type heat exchanger was always lower than the exchanger using conventional segmental baffles. Gao et al. (2009) studied twisted tubes with machined porous walls to design an S&T heat exchanger. Yang et al. (2011) researched the performances of the twisted tubes from laminar to turbulent states. Their tests showed that the heat transfer coefficients and pressure drops of the twisted tubes were both higher than those of smooth round tubes. Pathade and Singh (2017) modeled twisted-tube heat exchangers and presented the equations to calculate the heat transfer coefficients and pressure drops. They revealed that the swirl flow inside the tube creates turbulence enhancing the thermal-hydraulic performances. Khoshvaght-Aliabadi and Feizabadi (2020) found that the application of the twisted tubes generated synergetic swirling effects and fluid mixings. As a result, the thermal boundary layers were disrupted efficiently, resulting in an effective heat transfer performance. Twisted-tube heat exchangers were also studied in depth by Tan et al. (2013) and Li et al. (2021) who calculated the pressure drops and heat transfer coefficients in detail.

The two most common technologies of shell-side heat transfer intensifications are helical baffles and externally finned tubes. Stehlik et al. (1994) showed helical baffles promoted heat transfers by 1.39 times and decreased shell-side pressure drops by 0.26–0.60 times compared with segmental baffles. Kral et al. (1996) analyzed the impacts of helical angle on shell-side pressure drop and investigated the conversions from pressure drops to heat transfer rates. Ganapathy (1996) highlighted that external fins could increase 2–4 times heat transfer surface areas compared to smooth tubes with dramatic increases of pressure drops. Mukherjee (1998) emphasized external fins increased the surface area for heat transfer compared to plain tubes with dramatic increases in pressure drops. Zhang et al. (2009) divided helical baffles into continuous and non-continuous ones according to geometry. They revealed helical baffle reduced the shell-side pressure drops compared with the uses of segmental baffles. Wang et al. (2010) proposed correlations of pressure drops and heat transfer coefficients to study the performance of external fins inside S&T heat exchangers. Serth and Lestina (2014) presented empirical correlations to calculate the heat transfer coefficient of externally finned tubes.

The shell- and tube-side intensification technologies can be used simultaneously. Pan et al. (2013) provided some new insights into the use of externally finned tubes and baffles as well as tube-side intensification technologies. The authors compared the costs of different technologies and concluded that tube inserts are cheaper especially when applied to network retrofit. Jiang et al. (2014) employed MINLP model to express the combinations of different intensification technologies. However, their proposed model had convergence problems when solved using a global solver, while the solution time was rather long or the problem is deemed infeasible in many cases. Indeed, Yang et al. (2020) proposed an initialization strategy and applied the outer-approximation algorithm (DICOPT) to solve the heat exchanger model successfully. They reported small times within a few seconds for this approach, although it is not clear if an initialization step (initial values in for the optimization; a usual maneuver) is included. Since the outer-approximation algorithm is used, global optimality can't be guaranteed. When using the branch and reduce algorithm (BARON), they reported very long times ($>10^5$ s) in some cases, and sometimes they could not solve the problem at all.

In this research, we apply the Complete Set Trimming to globally solve the design problem of S&T heat exchangers using different types of heat transfer intensification technologies on shell and tube sides. The primary contributions of this research are the following:

(1) Regular and different heat transfer intensification technologies are considered together in the design resulting in thirteen types of S&T heat exchangers. The solution results give the best combination of shell and tube side technologies automatically.
(2) We consider two possible allocations for the design of each heat exchanger candidate and pick the better one that has a smaller objective function.
(3) Complete Set Trimming procedure is applied to solve the design problem to global optimality in acceptable computational time.

This paper is structured as follows. First, the problem statement is defined. Then, the model equations of the intensification technologies are presented. Next, the design optimization procedure namely Complete Set Trimming is discussed. Following, results for the design of nine examples are presented. We finish with concluding remarks.

2. Problem statement

Given are a heat exchange task and different options of S&T heat exchanger, regular segmental baffle and plain tube, and intensification technologies. It is desired to obtain the globally optimal S&T heat exchanger type and its corresponding geometry minimizing the area capital cost or TAC. For the shell-side technologies, we consider Segmental Baffles (SB), Helical Baffles (HB), and Externally Finned Tubes (EF). For the tube-side technologies, we consider Plain Tubes (PT), Twisted-Tape Inserts (TI), Coiled-Wire Inserts (CI), Internally Finned Tubes (IF), and Twisted-Tubes (TT). Each intensified exchanger can be constructed using the combinations between SB, HB, EF and PT, TI, CI, and IF, while the TT heat exchanger is considered separately as it does not combine with other intensification options. Without loss of generality, the following assumptions are made:

(1) Only E-shell heat exchangers are considered.
(2) Both hot and cold streams are liquid, and no phase change takes place.
(3) The properties of the hot and cold streams, density, viscosity, heat capacity, and thermal conductivity, are based on average temperatures.
(4) The hot (cold) stream can be allocated at either the shell side or tube side.
(5) Fouling factors of shell and tube sides are constant parameters.

3. Model equations for intensified exchangers

In this section, we present the major equations for the regular and intensification technologies. These equations consist of geometric constraints, bounds on velocities, Reynolds numbers and pressure drops, the equations of heat transfer, heat exchanger area, and objective functions. The basic geometric variables defined to characterize the intensified S&T exchanger consist of the inner and outer diameters of tubes (dti and dte), shell diameter (Ds), tube length (L), number of baffles (Nb), number of tube passes (Npt), tube pitch ratio (rp), tube layout (lay), baffle cut ratio (Bc), twisted pitch (H) and the thickness of TI (δ), the helical pitch of CI (P_{cI}), CI diameter (E_{cI}), number of IF fins per unit length (N_{IF}), helical angle of IF fins (α_{IF}), the height of IF fins (e_{IF}), thickness of IF fins (t_{IF}), helical angle of HB (β_{HB}), number of EF fins per unit length (N_{EF}), height of EF fins (b_{EF}), thickness of EF fins (τ_{EF}), twisted pitch of TT (Tu). The definition of variables and parameters are given in Supplemental Material-Part A. The equations corresponding to the calculation of heat transfer coefficients and pressure drops are included in Supplemental Material-Part B. We follow with other important constraints, where the model parameters are presented with a symbol ^ on top.

3.1. Geometric constraints

The ratio between tube length L and shell diameter Ds has the following bounds for heat exchangers that use Segmental Baffles (Taborek, 2008):

$$3\ Ds \leq L \leq 15\ Ds \tag{1}$$

Moreover, the baffle spacing lbc of Segmental Baffles is bounded by fractions of shell diameter (Taborek, 2008). This is used only in exchangers with traditional baffles:

$$0.2\ Ds \leq lbc \leq 1.0\ Ds \tag{2}$$

For the Twisted-Tape Insert, the twist ratio ($y = \frac{H}{dti}$) and thickness (δ) have the following bounds (Jiang et al., 2014):

$$3.0 \leq y \leq 6.0 \tag{3}$$

$$0.002 \le \delta \le 0.004 \tag{4}$$

For the Coiled-Wire Insert, the helical pitch (P_{cI}) and helical angle (α_{CI}) has the following bounds (Jiang et al., 2014):

$$1.17 \le \frac{P_{cI}}{dti} \le 2.68 \tag{5}$$

$$32 \le \alpha_{cI} \le 61 \tag{6}$$

For Internally Finned Tube, the height (e_{IF}), thickness (t_{IF}) and the number (N_{IF}) of fins has the following bounds (Serth and Lestina, 2014):

$$0.0075 \le \frac{e_{IF}}{dti} \le 0.03 \tag{7}$$

$$0.00062 \le t_{IF} \le 0.00078 \tag{8}$$

$$N_{IF} \le N_{IF}^{max} \tag{9}$$

where N_{IF}^{max} is the maximum number of internal fins.

For Helical Baffle, the baffle spacing (lbc_{HB}) has bounds (Serth and Lestina, 2014):

$$0.3\ Ds \le lbc_{HB} \le 1.0\ Ds \tag{10}$$

For Externally Finned Tubes, the height (b_{EF}), thickness (τ_{EF}) and the number density (N_{EF}) of external fins have the following bounds (Zdaniuk et al., 2008).

$$0.000508 \le b_{EF} \le 0.003175 \tag{11}$$

$$0.000254 \le \tau_{EF} \le 0.000381 \tag{12}$$

$$N_{EF} \le N_{EF}^{max} \tag{13}$$

where N_{EF}^{max} is the maximum value of N_{EF}.

For Twisted-Tube heat exchanger, the twisted pitch Tu has the following bounds (Pathade and Singh, 2017):

$$0.00381 \le Tu \le 0.03175 \tag{14}$$

We note that the constraints (4), (8), (9), (11), (12), (13), and (14) are limits that are not tested during Set Trimming. They can be applied before the optimization to build the search space using the allowable discrete values of the design variables.

3.2. Bounds of velocities, Reynolds numbers, and pressure drops

The flow velocities (vt and vs) should have lower bounds to avoid fouling and upper bounds to avoid erosion and vibration:

$$\hat{v}tmin \le vt \le \hat{v}tmax \tag{15}$$

$$\hat{v}smin \le vs \le \hat{v}smax \tag{16}$$

where $\hat{v}tmin$ and $\hat{v}tmax$ respectively denote the lower and upper bounds of tube-side velocity (vt); $\hat{v}smin$ and $\hat{v}smax$ respectively are the bounds of shell-side velocity (vs).

The bounds of tube- and shell-side Reynolds numbers (Ret and Res) are used according to the validities of the thermo-fluid dynamic correlations:

$$\hat{R}etmin \le Ret \le \hat{R}etmax \tag{17}$$

$$\hat{R}esmin \le Res \le \hat{R}esmax \tag{18}$$

where $\hat{R}etmin$ and $\hat{R}etmax$ respectively denote the lower and upper bounds of Ret; $\hat{R}esmin$ and $\hat{R}esmax$ respectively denote the bounds of Res.

The pressure drops of tube side (ΔPt) and shell side (ΔPs) must be bounded by their available values ($\Delta\hat{P}tdisp$ and $\Delta\hat{P}sdisp$). They are indirectly related to the trade-offs between capital investments and operating expenses, or operational restrictions.

$$\Delta Pt \le \Delta\hat{P}tdisp \tag{19}$$

$$\Delta Ps \le \Delta\hat{P}sdisp \tag{20}$$

3.3. Heat transfer equation

Based on the LMTD method, the heat transfer equation related to the heat duty $\hat{Q}$ of the heat exchanger and the required heat transfer area $Areq$ is:

$$\hat{Q} = U\ Areq\ \Delta\hat{T}lm\ F \tag{21}$$

where U denotes the overall heat transfer coefficient; $\Delta\hat{T}lm$ denotes the LMTD for the counter-current configuration and F denotes the correction factor for the LMTD. The expression of U is written as follows, for an unfinned surface:

$$U = \frac{1}{\frac{dte}{dti\ ht} + \frac{\hat{R}ft\ dte}{dti} + \frac{dte\ ln\left(\frac{dte}{dti}\right)}{2\ \hat{k}tube} + \hat{R}fs + \frac{1}{hs}} \tag{22}$$

where ht denotes the convective heat transfer coefficient of tube side; hs denotes the convective heat transfer coefficient of shell side; $\hat{R}ft$ denotes the fouling factor of tube side; $\hat{R}fs$ denotes the fouling factor of shell side; $\hat{k}tube$ denotes the thermal conductivity of the tube wall. It should be pointed out that the detailed formulations for the convective coefficients are presented in Supplemental Material-Part B.

The LMTD ($\Delta\hat{T}lm$) for the counter-current configuration is:

$$\Delta\hat{T}lm = \frac{\left(\hat{T}hi - \hat{T}co\right) - \left(\hat{T}ho - \hat{T}ci\right)}{ln\left(\frac{\hat{T}hi - \hat{T}co}{\hat{T}ho - \hat{T}ci}\right)} \tag{23}$$

The correction factor of LMTD ($\hat{F}$) is equal to 1 for one tube pass and equal to the following expression for two or more even tube passes.

$$\hat{F} = \frac{\left(\hat{R}^2+1\right)^{0.5} - ln\left(\frac{1-\hat{P}}{1-\hat{R}\ \hat{P}}\right)}{\left(\hat{R}-1\right)\ ln\left[\frac{2-\hat{P}\ \left(\hat{R}+1-\left(\hat{R}^2+1\right)^{0.5}\right)}{2-\hat{P}\ \left(\hat{R}+1+\left(\hat{R}^2+1\right)^{0.5}\right)}\right]} \tag{24}$$

where $\hat{R}$ and $\hat{P}$ are expressed as:

$$\hat{P} = \frac{\hat{T}co - \hat{T}ci}{\hat{T}hi - \hat{T}ci} \tag{25}$$

$$\hat{R} = \frac{\hat{T}hi - \hat{T}ho}{\hat{T}co - \hat{T}ci} \tag{26}$$

3.4. Heat transfer area

The geometrical area of heat exchanger depends on the design technologies applied to shell and tube sides. For combinations of SB-PT, SB-TI, SB-CI, HB-PT, HB-TI, and HB-CI, the heat transfer area (GA) corresponds to the outer surface area of tubes:

$$GA = Ntt\ \pi\ dte\ L \tag{27}$$

For the combinations of SB-IF and HB-IF, the heat transfer area (GA) depends on the inner surface area of tubes and internal fins (Yang et al., 2020).

$$GA = Ntt\ L\ (\pi\ dti + 2\ N_{IF}\ e_{IF}) \tag{28}$$

For the combinations of EF-PT, EF-TI, and EF-CI, the heat transfer area (GA) depends on the outer surface areas of tubes and external fins (Yang et al., 2020).

$$GA = 2\ Ntt\ \pi\ r_1\ (L - N_{EF}\ L\ \tau_{EF}) + 2\ Ntt\ N_{EF}\ L\ \pi\ \left(r_{2c}^2 - r_1^2\right) \quad (29)$$

where r_1 and r_{2c} are respectively the radius of the bare tube and the modified fin radius, and they are calculated by:

$$r_1 = \frac{dte}{2} \quad (30)$$

$$r_2 = \frac{dte + 2\ b_{EF}}{2} \quad (31)$$

$$r_{2c} = r_2 + \frac{\tau_{EF}}{2} \quad (32)$$

where r_2 denotes the radius of the fin tube outside.

For the combination of EF-IF, the heat transfer area (GA) depends on the surface areas of tubes, external fins, and internal fins (Yang et al., 2020).

$$GA = 2Ntt\ \pi\ r_1\ (L - N_{EF}\ L\ \tau_{EF}) + 2Ntt\ N_{EF}\ L\ \pi\ \left(r_{2c}^2 - r_1^2\right) + 2Ntt\ L\ N_{IF}\ e_{IF} \quad (33)$$

Finally, the GA for Twisted tubes is:

$$GA = Ntt\ \pi\ dte_{TT}\ L \quad (34)$$

where dte_{TT} is the hydraulic diameter of the twisted tube.

The above heat transfer area (GA) must be larger than the required heat transfer area, according to a given design margin that is defined using an "excess area" ($\hat{A}exc$). This is mainly because process designers usually aim to employ the excess area ($\hat{A}exc$) to compensate for the uncertainties which are associated with fouling factors, physical properties, thermo-hydraulic correlations, etc. Therefore, it yields:

$$GA\ U \geq \left(1 + \frac{\hat{A}exc}{100}\right) \frac{\hat{Q}}{\hat{F}\ \Delta\hat{T}lm} \quad (35)$$

Note that the overall heat transfer coefficient (U) is calculated during Set Trimming.

3.5. Objective function

The usual objective functions used in the literature have been set as the minimization of heat transfer area and total annualized cost. Minimization of the area has been used as a substitute for minimizing capital costs because they correlate monotonically. However, this is no longer the case when one considers intensified S&T exchangers. Hence, we consider two alternative objective functions to be minimized.

The first one is the capital cost of the heat exchanger area (CA), subjected to the available pressure drops. The second one is the total annualized cost (TAC) including capital and operating costs.

$$min\ CA = \hat{a}cost\ EA^{\hat{b}cost} \quad (36)$$

$$min\ TAC = \hat{A}f\ \hat{a}cost\ EA^{\hat{b}cost} + \hat{c}cost\left(\frac{\Delta Pt\ \hat{m}t}{\hat{\rho}t} + \frac{\Delta Ps\ \hat{m}s}{\hat{\rho}s}\right) \quad (37)$$

where are $\hat{a}cost$, $\hat{b}cost$ and $\hat{c}cost$ are the coefficients of the correlation for the evaluation of the capital cost of a heat exchanger with area EA; $\hat{A}f$ denotes the annualizing factor for capital cost; EA denotes the heat exchanger area that is used in the cost correlations, which many times is different than the geometrical area. In other words, EA represents the area of the tubes only.

For SB-PT, SB-TI, SB-CI, SB-IF, HB-PT, HB-TI, HB-CI, HB-IF, the area EA is:

$$EA = Ntt\ \pi\ dte\ L \quad (38)$$

For EF-PT, EF-TI, EF-CI, and EF-IF, the area EA is:

$$EA = 2Ntt\ \pi\ r_1\ (L - N_{EF}\ L\ \tau_{EF}) \quad (39)$$

Finally, for Twisted-Tube heat exchanger (Pathade and Singh, 2017), the area EA is calculated by:

$$EA = Ntt\ \pi\ dte_{TT}\ L \quad (40)$$

where dte_{TT} is the hydraulic diameter of twisted tubes.

4. Design optimization procedure

In this research, we consider all the design variables of a regular S&T heat exchanger, including tube layout, the number of tube passes, the number of baffles, tube pitch ratio, tube length, shell diameter as well as tube inner and outer diameters. These variables are considered having discrete values. In principle, baffle cut ratio is, at first glance, a continuous optimization variable; however, the final fabrication dimensions require discrete values. Similarly, the design variables of all heat transfer intensification technologies are considered as discrete ones consisting of the twisted pitch of TI (H), the thickness of TI (δ), the helical pitch of CI (P_{cI}), CI diameter (E_{cI}), number of IF fins per unit length (N_{IF}), the helical angle of IF fins (α_{IF}), the height of IF fins (e_{IF}), the thickness of IF fins (t_{IF}), the helical angle of HB (β_{HB}), number of EF fins per unit length (N_{EF}), the height of EF fins (b_{EF}), the thickness of EF fins (τ_{EF}) and twisted pitch of TT (Tu).

The above design variables determine a discrete search space for the regular and intensified S&T heat exchanger design. As a consequence, the design problem involves solution candidates corresponding to a set of discrete values. Based on the set of the solution candidates associated with the search space, the core idea of the Complete Set Trimming procedure that we use as the design optimization procedure in this work is to make use of the constraints to cut the search space of the candidates step by step and finally pick the best one through sorting. The detailed steps of the Complete Set Trimming procedure were already described by Costa and Bagajewicz (2019) as well as Lemos et al. (2020), and they were first proposed by Gut and Pinto (2004) for a specific problem; Costa and Bagajewicz (2019) proposed to generalize them to other equipment. In addition, we consider two possible allocations for both hot and cold streams (tube side vs. shell side), that is, we apply the design optimization procedure to solve the problem twice for the two possible fluid allocations, but eliminating the heat exchanger candidates that have a larger objective function than the one found for the other allocation. Finally, the tube-side convective heat transfer coefficient of the Internally Finned Tube is obtained through solving a cubic equation in one unknown, which we present in Supplemental Material-Part B.

5. Examples

In the present section, nine examples taken from Gonçalves et al. (2017a) are solved for illustration purposes. The input data of hot and cold streams of the tested examples are provided in Supplemental Material-Part C. Table 1 lists all the discrete values of the design variables for the regular and intensification technologies which we consider in this research. Table 2 provides the corresponding cost coefficients of different types of S&T heat exchangers. The rest of the parameters used in the optimization are given in Supplemental Material-Part C. Examples 1–9 are solved in GAMS (Brooke et al., 2005) and are implemented through a server with Intel® Core^{TM} i7–3600 U processor which has 3.9 GHz speed and 32 GB memory and runs on Linux system.

As it was mentioned above, Yang et al. (2020) solved the models of the intensified S&T heat exchangers using DICOPT (outer-

Table 1
Discrete values of design variables.

Variable	Values
Shell diameter (Ds, m)	0.7874, 0.8382, 0.8890, 0.9398, 0.9906, 1.0668, 1.1430, 1.2192, 1.3716, 1.5240
Outer tube diameter (dte, m)	0.01905, 0.02540, 0.03175, 0.03810, 0.05080
Number of tube passes (Npt)	1, 2, 4, 6
Tube pitch ratio (rp)	1.25, 1.33, 1.50
Tube layout (lay)	1 (triangular layout), 2 (square layout)
Tube length (L, m)	1.2195, 1.8293, 2.4390, 3.0488, 3.6585, 4.8768, 6.0976
Number of baffles (Nb)	1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18, 19, 20
Twisted-Tapes pitch (H, m)	0.074700, 0.103275, 0.131850
Twisted-Tapes thickness (δ, m)	0.002, 0.003, 0.004
Coiled-Wire helical pitch (P_{cl}, m)	0.031955, 0.044179, 0.056402
Coiled-Wire diameter (E_{cl}, m)	0.001411, 0.001951, 0.002491
Number of internal fins per unit length (N_{IF})	30, 50
Helical angle of internal fins (α_{IF})	35°, 45°
Height of internal fins (e_{IF}, m)	0.000332, 0.000459, 0.000586
Thickness of internal fins (t_{IF}, m)	0.0007
Helical angle of helical baffle (β_{HB})	20°, 30°, 40°, 50°
Number of external fins per unit length (N_{EF})	50, 130, 210
Height of external fins (b_{EF}, m)	0.001841
Thickness of external fins (τ_{EF}, m)	0.000317
Twisted pitch of twisted tube (Tu, m)	0.00381, 0.00508, 0.00635, 0.00762, 0.01016, 0.01143, 0.01270, 0.01524, 0.01905, 0.02032, 0.02540, 0.03175

Table 2
Cost coefficients of different types of heat exchangers (Pan et al., 2013).

Exchanger	$\hat{a}cost$	$\hat{b}cost$	$\hat{c}cost$
SB-PT	20.0	0.8	1.31
SB-TI	21.0	0.8	1.31
SB-CI	21.5	0.8	1.31
SB-IF	30.0	0.8	1.31
HB-PT	22.0	0.8	1.31
HB-TI	23.0	0.8	1.31
HB-CI	23.5	0.8	1.31
HB -IF	32.0	0.8	1.31
EF-PT	30.0	0.8	1.31
EF-TI	31.0	0.8	1.31
EF-CI	31.5	0.8	1.31
EF-IF	40.0	0.8	1.31
TT*	30.0	0.8	1.31

*the cost coefficient is assumed by ourselves
SB = Segmental Baffle; HB = Helical Baffle;
EF = Externally Finned Tube; PT = Plain Tube;
TI = tube with Twisted-Tape Insert; CI = tube with Coiled-Wire Insert;
IF = Internally Finned Tube; TT= Twisted-Tube heat exchanger;

approximation algorithm) and BARON (branch and reduce algorithm). DICOPT runs fast (< 5.0 s), but the global optimality cannot be guaranteed and it is unclear whether the initialization step they used is included or not. In turn, BARON runs slowly (>10^5 s) and even has convergence problems/issues in some cases.

5.1. Capital cost minimization

The first objective of the design problem under study in this work is to obtain the optimal S&T heat exchangers with the minimum capital cost. Using the Complete Set Trimming procedure, we obtain the globally optimal solutions of S&T heat exchanger design for Examples 1–9, based on the discrete values of the design variables provided in Table 1 as well as the cost coefficients listed in Table 2. The cost comparisons among the expenses of the optimal solutions for different types of S&T heat exchangers are depicted in Fig. 1. From this figure, it could be seen that the capital cost of the twisted-tube heat exchanger (TT) is the lowest one in each example. However, the cost coefficient of a TT heat exchanger is difficult to find and we have to assume a value in the present research. Hence, we ignore TT heat exchangers and only compare the costs of the other S&T heat exchangers here.

The cost coefficients of the other S&T heat exchangers (Except TT heat exchanger) are taken from the work of Pan et al. (2013). Fig. 1 depicts the following observations: For Examples 1 and 6, the intensified S&T heat exchangers, that are designed by using Externally Finned and Plain Tubes (EF-PT), exhibit the lowest capital cost (we ignore TT heat exchangers in our comparison); For Examples 4 and 8, the intensified S&T heat exchangers designed by using Externally Finned and Internally Finned Tubes (EF-IF) exhibit the lowest capital investment; For Examples 5 and 9, the regular S&T heat exchangers designed by using Segmental Baffles and Plain Tubes (SB-PT) exhibit the lowest capital cost; For Examples 2, 3 and 7, the intensified S&T heat exchangers, designed respectively by using the Externally Finned Tubes with Coiled-Wire Inserts (EF-CI), the Externally Finned Tubes with Twisted-Tape Inserts (EF-TI), and Segmental Baffles and Internally Finned Tubes (SB-IF), exhibit the lowest capital cost. From the above result comparison, it could be concluded that the input data of the tested examples have a considerable impact on the optimum solution results. Finally, the design and thermo-fluid dynamic results of the optimal solutions for capital cost minimization are given in Supplemental Material-Part D.

Fig. 1 also indicates that the intensified S&T heat exchangers using Helical Baffles (HB) on the shell side feature higher capital costs compared to the intensified S&T heat exchangers using Segmental Baffles (SB) and Externally Finned Tubes (EF). Generally speaking, by using Helical Baffles on the shell side, the turbulent flow on the shell side declines. This could decrease the convective heat transfer coefficient and therefore increase the heat transfer area requirements and the corresponding capital investments. Moreover, it can be found from Fig. 1 that the intensified S&T heat exchangers designed by using Twisted-Tape Inserts (TI) and Coiled-Wire Inserts (CI) feature higher capital costs in most of our examples compared with the regular S&T heat exchangers designed by using Segmental Baffles and Plain Tubes (SB-PT). Such two tube inserts generally augment the turbulent flows on the tube sides, intensify the convective heat transfer coefficient of the tube-side stream and reduce the heat transfer area requirement. However, the cost coefficients of such two tube inserts are higher than those of the plain tubes. As a result, the capital costs of the intensified S&T heat exchangers designed by using tube inserts (TI and CI) are higher than those of the S&T heat exchangers using Plain Tubes in most cases. Table 3 provides the solution results of area capital cost minimization for Examples 1–9, respectively. The optimal so-

Fig. 1. The optimal capital cost of heat exchanger using different technologies for Examples 1–9.

Table 3
Solution results of capital cost minimization.

Example	*CA* ($)	Solution time (s)	S&T Type	Tube-side stream
1	2566.8	273.41	EF-PT	Hot
2	1489.3	267.39	EF-CI	Hot
3	1089.8	271.06	EF-TI	Hot
4	3165.4	269.36	EF-IF	Cold
5	872.2	268.37	SB-PT	Cold
6	1570.2	271.15	EF-PT	Hot
7	1191.4	270.26	SB-IF	Hot
8	3143.5	272.82	EF-IF	Cold
9	1491.7	270.72	SB-PT	Cold

Table 4
Minimum capital cost of each type of heat exchanger ($).

S&T Type	Examples								
	1	2	3	4	5	6	7	8	9
SB-PT	2995.2	1783.6	1149.1	3826.7	872.2	1760.2	1223.0	4373.3	1491.7
SB-TI	3121.1	1885.4	1230.0	4446.0	899.8	1867.2	1203.0	4368.7	1574.6
SB-CI	3032.5	1709.0	1259.2	4472.7	904.5	1911.7	1231.7	4391.5	1634.2
SB-IF	2749.3	1706.2	1167.6	3342.7	898.5	1789.9	1191.4	3366.9	1494.2
HB-PT	4009.0	2178.5	1376.6	5569.2	1092.4	2237.4	1533.4	5569.2	2098.5
HB-TI	3800.7	2166.2	1406.5	5822.3	1161.7	2339.1	1575.5	5189.0	2114.3
HB-CI	3594.4	2109.9	1437.0	5948.9	1187.0	2213.3	1609.7	5301.8	2160.3
HB-IF	3648.7	2066.3	1375.1	3937.5	1148.8	2162.6	1531.7	3937.5	2041.2
EF-PT	2566.8	1573.9	1104.9	3242.0	939.5	1570.2	1264.1	3242.0	1546.6
EF-TI	2667.3	1595.7	1089.8	3292.3	965.4	1622.5	1256.9	3292.3	1502.6
EF-CI	2672.5	1489.3	1107.4	3345.4	906.6	1621.4	1277.2	3345.4	1526.8
EF-IF	2785.0	1668.3	1171.3	3165.4	1047.8	1634.5	1273.3	3143.5	1621.8
TT	1424.3	873.5	632.5	1949.2	633.3	1172.3	652.3	1572.5	780.7

lution results of different types of S&T heat exchangers for capital cost minimization are presented in Table 4.

The computational time of each type of S&T heat exchanger is provided in Table 5. From this table, one could notice that the design problems of the Twisted-Tube (TT) heat exchangers are all solved within one second. This is primarily because the design variables for the TT heat exchangers are fewer than that of other S&T heat exchangers. Table 5 also indicates the design of the intensified S&T heat exchanger EF-IF consumes the longest computation time because there is a larger number of variables considered in the optimization.

5.2. TAC minimization

The second objective which we consider in this research is to minimize the total annualized cost (TAC) for different types of S&T heat exchanger design, including the exchanger capital cost and operational pumping cost related to the pressure drops. The TAC

Table 5
Computational time of each type of heat exchanger for capital cost minimization (s)

S&T Type	Examples								
	1	2	3	4	5	6	7	8	9
SB-PT	2.35	2.28	2.23	2.19	2.28	2.32	2.31	2.28	2.29
SB-TI	7.06	6.85	6.79	6.91	7.06	6.88	7.02	6.92	6.93
SB-CI	6.68	6.02	5.98	6.05	6.11	5.99	6.01	6.02	5.92
SB-IF	23.85	22.98	22.89	22.86	22.86	23.06	22.13	23.06	23.13
HB-PT	3.82	2.93	2.98	2.85	3.05	2.89	2.82	2.93	2.95
HB-TI	17.82	16.99	18.82	18.05	18.02	18.96	17.95	17.96	17.92
HB-CI	20.56	19.98	19.86	19.81	19.95	19.62	20.65	20.12	20.82
HB-IF	56.36	55.86	56.15	56.98	56.08	56.92	56.92	56.12	55.69
EF-PT	7.65	7.96	7.83	7.93	7.91	7.83	7.95	7.92	7.98
EF-TI	25.13	24.88	25.92	25.05	25.28	25.85	25.81	26.36	25.86
EF-CI	24.26	24.19	24.08	23.09	23.93	23.82	23.68	25.32	23.89
EF-IF	77.39	75.96	77.06	77.09	75.32	76.52	76.56	77.35	76.81
TT	0.48	0.51	0.47	0.50	0.52	0.49	0.45	0.46	0.53
Average	21.03	20.57	20.85	20.72	20.64	20.86	20.79	20.99	20.82

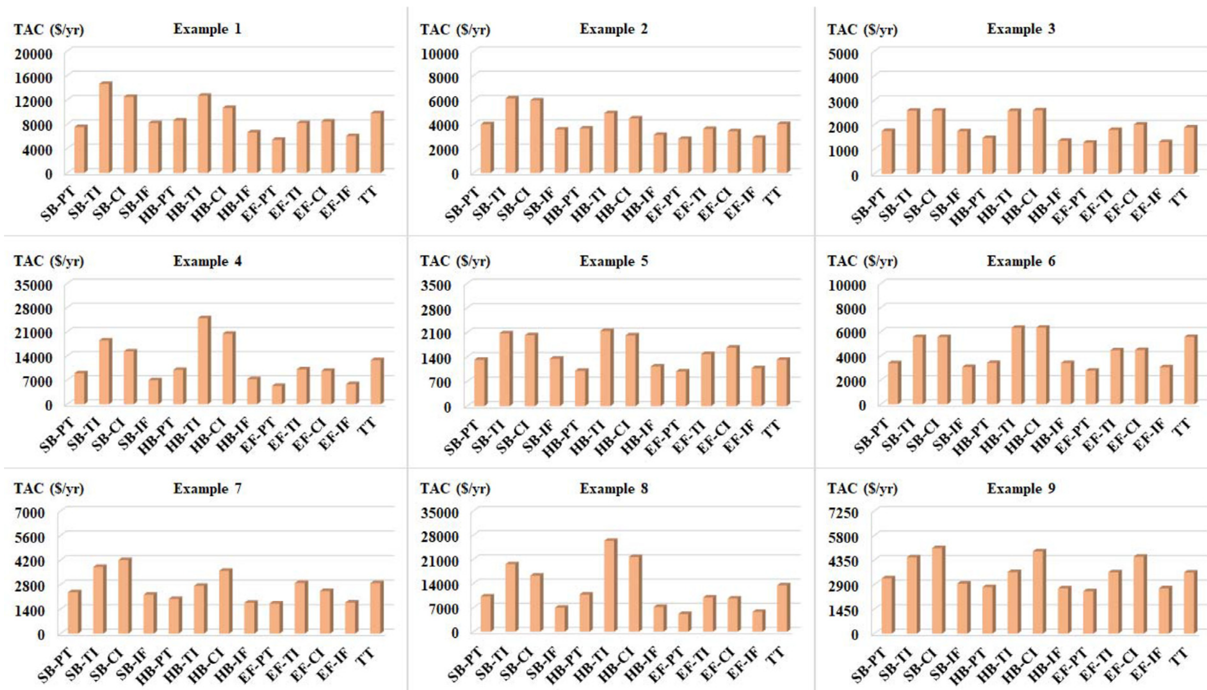

Fig. 2. The optimal TAC of heat exchanger using different technologies for Examples 1–9.

comparisons between the globally optimal solutions of the S&T heat exchangers designed by using different types of shell- and tube-side technologies are presented in Fig. 2. This figure illustrates that the intensified S&T heat exchangers designed by using Externally Finned and Plain Tubes (EF-PT) exhibit the lowest TACs for all the tested examples. In general, the Externally Finned Tube augments the turbulent flows on the shell side and hence the shell-side friction factor becomes large, increasing the pressure drops and the pumping expenses. However, the external fins could provide additional surface area for heat transfer and the convective heat transfer coefficient of the shell-side stream also increases accordingly. As a result, the saving in the capital cost of the heat exchanger area compensates for the increase in the operational expenses of pumping power. The optimum solution results of Examples 1–9 are presented in Table 6. The solution results of different types of S&T heat exchangers for TAC minimization are presented in Table 7. The detailed designs of the optimal solutions of TAC minimization are given in Supplemental Material-Part D.

Fig. 2 also clearly presents that the intensified S&T heat exchangers designed by using the tubes with Twisted-Tape Insert (TI) and Coiled-Wire insert (CI) exhibit higher TACs compared to the other heat transfer intensification technologies. This is primarily because such two heat transfer intensification technologies could generally increase the friction factors and the pressure drop of the tube-side stream. Therefore, the operational expense of the pumping power increases largely while it cannot be compensated by the savings in the heat transfer area. As a consequence, the TACs

Table 6
Solution results of TAC minimization.

Example	*TAC* ($/yr)	Solution time (s)	S&T Type	Tube-side stream
1	5460.0	276.69	EF-PT	Hot
2	2801.8	270.51	EF-PT	Hot
3	1276.1	273.12	EF-PT	Hot
4	5345.6	272.59	EF-PT	Cold
5	990.0	270.56	EF-PT	Cold
6	2792.7	272.36	EF-PT	Cold
7	1720.5	271.58	EF-PT	Cold
8	5147.7	273.92	EF-PT	Cold
9	2508.2	271.95	EF-PT	Cold

Table 7
Minimum TAC of each type of heat exchanger ($/yr).

S&T Type	Examples								
	1	2	3	4	5	6	7	8	9
SB-PT	7548.0	4003.4	1748.5	9013.3	1324.2	3408.9	2359.5	10,267.7	3274.3
SB-TI	14,646.8	6133.3	2580.9	18,498.1	2079.7	5568.6	3811.1	19,612.8	4513.2
SB-CI	12,508.1	5969.3	2582.7	15,292.8	2024.8	5570.7	4201.4	16,284.3	5055.2
SB-IF	8192.2	3574.6	1741.9	6992.6	1355.7	3096.2	2223.5	6990.8	2964.1
HB-PT	8619.0	3657.8	1464.7	9992.5	1011.1	3431.8	1984.7	10,788.8	2754.4
HB-TI	12,698.8	4919.2	2573.0	24,968.9	2147.5	6335.8	2727.1	26,355.1	3644.2
HB-CI	10,698.6	4484.1	2596.8	20,447.0	2018.8	6350.3	3595.2	21,659.8	4864.8
HB-IF	6663.0	3128.8	1347.0	7310.6	1134.1	3422.0	1761.9	7204.8	2677.1
EF-PT	5460.0	2801.8	1276.1	5345.6	990.0	2792.7	1720.5	5147.7	2508.2
EF-TI	8201.7	3620.8	1795.2	10,140.9	1483.4	4477.6	2891.2	9952.4	3631.9
EF-CI	8464.5	3433.8	2008.7	9716.7	1668.9	4494.1	2434.2	9642.9	4553.3
EF-IF	6070.6	2891.2	1297.6	5861.7	1081.2	3076.1	1774.7	5787.3	2682.6
TT	9820.4	4029.8	1892.0	12,793.0	1323.4	5583.8	2884.4	13,471.2	3618.0

Table 8
Computational time of each type of heat exchanger for TAC minimization (s).

S&T Type	Examples								
	1	2	3	4	5	6	7	8	9
SB-PT	2.36	2.29	2.25	2.21	2.29	2.33	2.32	2.29	2.31
SB-TI	7.07	6.79	6.82	6.96	7.08	6.89	7.03	6.97	6.96
SB-CI	6.69	6.38	6.58	6.06	6.12	6.02	6.11	6.08	6.03
SB-IF	23.98	23.02	22.92	22.89	22.73	23.19	22.36	23.16	23.12
HB-PT	3.83	3.85	3.68	3.56	3.62	3.83	2.85	2.98	2.98
HB-TI	17.96	17.75	18.25	18.72	18.65	18.38	18.36	17.98	17.95
HB-CI	20.96	20.18	19.92	19.96	19.98	19.69	20.68	20.36	20.87
HB-IF	56.87	55.36	56.19	56.99	56.68	56.98	56.97	56.52	56.72
EF-PT	7.96	7.85	7.86	7.95	7.98	7.93	7.98	7.97	7.99
EF-TI	25.65	25.28	25.89	25.36	25.39	25.89	25.96	26.39	25.87
EF-CI	24.89	24.19	25.12	24.26	23.98	23.92	23.72	25.37	23.78
EF-IF	77.98	77.06	77.16	77.16	75.52	76.82	76.78	77.38	76.82
TT	0.49	0.51	0.48	0.51	0.54	0.49	0.46	0.47	0.55
Average	21.28	20.81	21.01	20.97	20.81	20.95	20.89	21.07	20.92

of the intensified S&T heat exchanger designed using the tubes with Twisted-Tape Insert (TI) or Coiled-Wire Inserts (CI) is generally higher than the intensified S&T heat exchangers designed by using Plain Tubes (PT) and Internally Finned Tubes (IF).

The computational time of each type of heat exchanger is provided in Table 8. This table shows the design of TT heat exchanger is solved in less than one second. Similar to the captical cost minimization, this is mainly due to thet fact that the fewest types of design variables are considered in the design optimization of TT heat exchanger. Also, the design of EF-IF heat exchanger involves many more optimization variables and thus demands the longest computational time.

6. Conclusion

This research concentrates on the optimal design of shell and tube heat exchangers using different types of shell- and tube-side technologies. The shell-side technologies consist of Segmental Baffles, Helical Baffles, and Externally Finned Tubes, while the tube-side ones consist of Twisted Tubes, Plain Tubes, tubes with Twisted-Tape Inserts, tubes with Coiled-Wire Inserts, and Internally Finned Tubes. Due to the physically discrete nature and/or the manufacturing standards, the design variables of the regular and heat transfer intensification technologies can be considered as the ones that have discrete values. Based on this, we use Complete Set Trimming procedures to obtain the globally optimal solutions for the design problem of intensified S&T heat exchangers.

Nine examples from the literature are tested for illustrative purposes and the optimal results indicate: The intensified heat exchangers designed using Twisted Tubes require the minimum capital expenses; The input data and the related cost parameters have great impacts on the final solution results of capital cost minimization; The intensified S&T heat exchangers designed using Externally Finned Tube and Plain Tube exhibit the lowest total annualized cost. One of our future works is to consider more discrete values of the design variables for intensified S&T heat exchangers. Another future work is to integrate intensified heat exchanger design into the synthesis/retrofit design of heat exchanger networks.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Supplementary materials

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