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Globally optimal design optimization of conventional and intensified shell and tube condensers using complete set trimming

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ABSTRACT

To this date, condenser design optimization has predominantly relied on empirical trial-and-verification approaches, inevitably yielding suboptimal solutions. In this article, we propose an optimal design method for single-component condensers, using an emerging global optimization technique known as Complete Set Trimming. This approach not only selects the most efficient type of condenser (vertical or horizontal) but also determines the optimal geometry. We also introduce the addition of intensification to condensers, an item lacking in the literature of the design optimization of this kind of equipment. Finally, we inspect the effect of pressure drop limitations on the optimal design, another issue lacking in the literature and we consider alternative objective functions to the current practice of minimizing area. The method is non-iterative, gradient-free, and guarantees global optimality. Instead of the current suggested practice of running each case separately, we introduce a new variant of Set Trimming, where all combinations of intensification (twisted tapes, coils, and fins) and conventional configurations (horizontal or vertical condensers, condensation on the shell side or tube side) are inspected sequentially and automatically. Also, departing from the current practice of seeking minimum area, we also consider the minimization of capital cost and total annualized cost. The limitations on pressure drop force more costly condensers as well as different condenser types being optimal, a feature that has not been covered in the literature. The results of two design examples showed that cost values for vertical configurations were consistently higher, with a minimum increase of 11% and a maximum increase exceeding 100%.

1. Introduction

Condensers are widely used in various industrial operations, such as distillation columns, refrigeration cycles, power generation, and other applications. As the most widely used heat exchanger, the design optimization of shell and tube condensers has evolved into a fundamental topic in the research field. Some design procedures are provided in textbooks, such as Kakaç and Liu (2002), Serth (2007), Towler and Sinnott (2008), and Cao (2010). The condenser design methods recommended and used in these textbooks are all founded on heuristic rules. For the design of heat transfer equipment, trial-and-verification methods are typically utilized. Initially, an estimate of the total heat transfer coefficient is made to obtain a first estimate of the heat transfer area. Subsequently, suitable geometric parameters are selected to enable the calculation of the film heat transfer coefficients. A comparison is then made between the calculated overall heat transfer coefficient and the estimated value or between the required area and the actual area (or heat load) to ascertain the reasonableness of the estimate and determine if further trials are required. In certain instances, new estimations may need to be generated following program verification to pursue enhancements (Costa and Bagajewicz, 2019). This practice was proposed in the last century, and related algorithms became mainstream. At present, commercial software can perform the design of heat exchangers and generate a design solution tailored to the input fluid streams (Sahajpal and Shah, 2013). However, these procedures cannot guarantee local optimality when designing conventional heat exchangers, let alone global optimality, or automatically designing intensified heat exchangers.

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In addition to the procedures outlined in the previously mentioned textbooks, the literature considers two main models for the design optimization of condensers: Nonlinear Optimization Models (NLOM), or Mixed-Integer Nonlinear Optimization Model (MINLOM), the former considering the geometric parameters (diameters, lengths, etc.) using continuous variables, and the latter considering discrete options of these parameters. In turn, these models can be solved using three main approaches: metaheuristic algorithms, mathematical programming methods, and Set Trimming, the method we use for the first time.

For metaheuristics algorithms, Allen and Gosselin (2008) employed genetic algorithms to minimize the total annualized cost of shell and tube heat exchangers for condensation on the shell side or tube side. Hajabdollahi et al. (2011) demonstrated that genetic algorithms outperform particle swarm algorithms in the design of shell and tube condensers from a computational perspective. Chandrakanth et al. (2018) utilized a design of experiment approach using the Taguchi method in optimizing condenser design, incorporating second law analysis to determine the optimal heat transfer area and pressure drop. Xiao et al. (2019) combined genetic algorithms with simulated annealing for the design optimization of heaters using saturated steam. Gürses et al. (2022a) used the African Vulture optimization algorithm to achieve cost minimization in shell-and-tube heat exchangers, and they also demonstrated that this algorithm can be applied to the design of plate-fin and tube-fin heat exchangers. Lastly, Gürses et al. (2022b) applied the Artificial Gorilla Troops algorithm to optimize fine plate heat exchangers, proving that the algorithm has robustness and feasibility for application in engineering design optimization.

While metaheuristic algorithms have the potential to find global optima, they may frequently converge towards local minima, necessitating control parameter adjustments to attain a satisfactory performance. Furthermore, these parameters are often case-specific and may not be effective for other scenarios. Due to these limitations, researchers have explored the application of mathematical programming methods for heat exchanger design (Jegede and Polley, 1992; Mizutani et al., 2003; Ponce-Ortega et al., 2006). In recent years, papers have been focusing on the conversion of nonlinear equations to linear expressions present in condenser models. Gonçalves et al. (2017a,b) proposed to transform the mixed integer nonlinear model of shell and tube heat exchangers into a linear model, sometimes an integer linear model. By rigorously reformulating the model to a linear form, they ensure global optimality and robustness without the need for initial values. It is worth pointing out that these reformulations are rigorous in the sense that no linearizations or approximations are being introduced, such that the solution to the reformulated problem is exactly the same as the solution to the original problem. Pereira et al. (2021) applied this approach to horizontal condensers. However, mathematical programming methods also have drawbacks, such as reliance on commercial solvers.

In response to the limitations of metaheuristic algorithms and mathematical optimization methods, in recent years, an emerging algorithm known as Set Trimming has been applied to the design of heat exchangers. Set Trimming represents a generalization of the concept introduced by Gut and Pinto (2004) and is completely different from metaheuristic algorithms and mathematical optimization methods. In the design of chemical process equipment, a design solution comprises a combination of discrete values of the design variables associated with a given value of the objective function. The complete set of combinations of design variables form the search space. In Complete Set Trimming, inequality constraints are sequentially applied to test the candidates of the search space, leading to a fully trimmed set containing the feasible design candidates, from which the optimal design is derived through sorting. This technique is particularly suited for scenarios where all variables are discrete. This is the case of the design of heat exchangers and condensers in particular. Indeed, all independent geometrical variables are discrete because manufacturers have standardized them. In addition, we point out that there are no other continuous independent variables, so contrary to first impressions, the technique presented here does not rely on any discretization of any continuous variable (velocities, Reynolds numbers, overall heat transfer coefficients, film coefficients, etc.) In various heat exchanger design applications, Set Trimming has demonstrated computational advantages in areas, such as shell and tube heat exchangers without phase change (Lemos et al., 2020) and gasketed plate heat exchangers (Nahes et al., 2023a), as well as other equipment including distillation trays (da Cruz et al., 2023). Certain variants of Set Trimming necessitate Smart Enumeration, as not all candidates remain feasible after Set Trimming as discussed by Costa and Bagajewicz (2019) and explored in the design of fired heaters (Kim et al., 2023), distillation columns (Peccini et al., 2023), and catalytic reactors (Nahes et al., 2023b). However, these variants are not needed for the condenser design discussed in this article, as long as overall heat transfer coefficients can be calculated using the properties of the inlet and outlet streams as well as the exchanger geometry, but are needed when properties of the fluid are strongly dependent on the temperature, or the modeling of the condensation requires discretized models.

On the other hand, there appears to be a scarcity of research focusing on the optimization of heat exchanger configurations, including the decision of whether the heat exchangers should be arranged horizontally or vertically. This issue can also be extended to determine the location of hot stream condensation in condensers, whether it occurs on the tube side or the shell side. While optimizing the configuration of heat exchangers may not be overly challenging—by running each optimization option independently and comparing the outcomes—automation of the program becomes essential in scenarios demanding repetitive tasks. This aspect also serves as a partial advantage of this study.

When considering safety and corrosion, the configuration of the condenser is mostly determined by empirical rules. For example, Serth (2007) discussed factors such as selection criteria for stream allocation, direction of fluid circulation in vertical condensers, and the use of baffles. According to Mueller (2008), the choice of geometric configuration (vertical or horizontal) and stream allocation depends on operating pressure, temperature, available pressure drop, corrosion, scaling, and other factors. In many of the references mentioned above (Kakaç and Liu, 2002), there have also been discussions on the selection of shell types, with a common view favoring the use of shell side pressure drop as a selection parameter. In summary, existing literature often provides vague selection recommendations without providing detailed data explanations for obtaining the best choice from an economic perspective.

The optimization of enhanced heat transfer in heat exchangers has emerged as a significant research focus amid ongoing industrial advancements. Yang et al. (2020) employed mathematical programming to optimize the combination of various heat transfer enhancement techniques in shell-and-tube heat exchangers, resulting in better design solutions compared to traditional design procedures. Chang et al. (2022) employed Complete Set Trimming to design intensified shell and tube heat exchangers without phase change. Experimental testing methods are predominant in the research on enhanced heat transfer in condensers, and there appears to be a lack of literature focusing on their design optimization. For example, Yin et al. (2017) investigated the effects of various tube types on condenser performance, revealing that the heat transfer coefficient of helical tubes is 14.6 % to 28.9 % higher than that of round tubes. Additionally, the heat transfer coefficient of elliptical tubes exceeds that of round tubes by 10.2 % to 18 %. Additionally, Gu et al. (2020) performed a theoretical analysis of a condenser consisting of round inner tubes and elliptical outer fins, finding an enhancement in heat transfer capability ranging from 2.34 % to 9.28 %.

Aiming at overcoming the limitations and research gaps identified in existing condenser optimization approaches, the following specific objectives are a direct contribution of this paper to the area:

- Optimization of condensers using Complete Set Trimming.
- Selection of condenser configurations, including four variants (horizontal and vertical condensers with condensation inside and outside the tubes), in addition to conventional design variables.

- Optimal design of heat transfer enhancement techniques for condensers (tape inserts, finned tubes, etc.).
- Utilization of alternative objective functions.
- Automatic design of all types of condensers.

This article is organized as follows. In the second section, we elaborate on the thermal and hydraulic models of condensers without enhanced heat transfer, as well as the optimization constraints to ensure feasible solutions. In addition, we also present key models of five intensification technologies, including twisted-tape insert, internally finned tube, coiled-wire insert, externally finned tube, and helical baffle. In the third section, a detailed explanation of the proposed design approach is provided. In the fourth and fifth sections, an analysis of examples from the literature is performed, the performance of the proposed approach and the computational efficiency of the algorithm is evaluated, and finally, conclusions are presented.

2. Mathematical model

The optimization is based on the following assumptions:

(1) The condensing stream is a single component.
(2) Only E-shell condensers are considered.
(3) The density, viscosity, heat capacity, and thermal conductivity of hot and cold streams are assumed constant and evaluated at the average temperature.

Multicomponent condensing streams present temperature and phase changes simultaneously inside the condensers, the type of shells may require some other correlations than the ones used, and the variation of properties along the condenser is an issue that requires a different model than the LMTD model. To deal with these cases, Complete Set Trimming requires some revision, which is not done in this article.

In the description of the optimization model, the problem parameters, which are fixed before the optimization, are represented with the symbol "^" on top.

In this paper, two alternative objective functions are employed:

- Objective 1: Minimum Capital Cost

Capital cost is the most common indicator in heat exchanger evaluation, calculated as shown in equation (1). Generally, it is monotonically positively correlated with the heat transfer area. Therefore, we choose capital cost as one of the objective functions for optimization:

$$C_A = \widehat{\alpha} A_{tube}{}^{\widehat{\beta}} \tag{1}$$

where $\widehat{\alpha}$ and $\widehat{\beta}$ are the cost coefficients for condensers. The value of the constant $\widehat{\alpha}$ depends on the intensification, while $\widehat{\beta}$ is a constant that does not. Neither $\widehat{\alpha}$ nor $\widehat{\beta}$ depend on the type of exchanger (vertical or horizontal).

The area used when twisted tapes, coil inserts, and internally finned tubes are used is the area of the bare tube:

$$A_{tube} = Ntt\pi d_{t,e} L \tag{2}$$

where Ntt is the total number of tubes, $d_{t,e}$ is the external tube diameter, and L is the tube length. For the externally finned tubes, we use:

$$A_{tube} = Ntt\pi d_{t,e}(L - N_{EF}\widehat{t}_{EF}) \tag{3}$$

where N_{EF} is the number of external fins per unit length, and $\widehat{t}_{EF}$ is the thickness of the external fins. The above expression was used by Pan et al. (2013) and Yang et al. (2020) and is based on an exposed naked tube without counting the fin area.

- Objective 2: Minimum Total Annualized Cost:

The total annualized cost (TAC) consists of the annualized equipment cost (C_A) plus the pumping costs of the heat exchanger (C_P):

$$TAC = \widehat{af} C_A + C_p \tag{4}$$

where $\widehat{af}$ is the annualization factor given by:

$$\widehat{af} = \frac{\widehat{i}(1+\widehat{i})^{\widehat{ny}}}{(1+\widehat{i})^{\widehat{ny}} - 1} \tag{5}$$

where $\widehat{i}$ is the interest rate and $\widehat{ny}$ is the number of years used for annualization. In turn, C_P is given by:

$$C_P = \widehat{EC}\widehat{Hour}\left(\frac{\widehat{m}_{coolant}\Delta p_{coolant}}{\widehat{\rho}_{coolant}\widehat{\eta}}\right) \tag{6}$$

where $\widehat{EC}$ is the electricity cost, $\widehat{Hour}$ is the operating time of the equipment for one year, and $\widehat{m}_{coolant}$, $\Delta p_{coolant}$, $\widehat{\rho}_{coolant}$ and $\widehat{\eta}$ are the flow rate, pressure drop, density of the cooling fluid, and the overall efficiency of the pump, respectively.

One important constraint is on heat exchange area:

$$A \geq (1 + \widehat{\%A}_{exc})A_{req} \tag{7}$$

where $\widehat{\%A}_{exc}$ is the percentage of excess area imposed by the designer (usually to cover uncertainty), A_{req} is the required area to accomplish the heat transfer task, and A is the heat transfer area based on the outer surface of the tubes, as in Eq. (2). The required area is based on the well-known $\widehat{LMTD}$ model:

$$A_{req} = \frac{\widehat{Q}}{U\widehat{LMTD}} \tag{8}$$

where $\widehat{Q}$ is the heat load and U is the overall heat transfer coefficient, a value that depends on the final condenser design. Thus, any feasible design needs to have an area larger than the required area. In turn, the logarithmic mean temperature difference ($\widehat{LMTD}$) is given by:

$$\widehat{LMTD} = \frac{\widehat{T}_{c,out} - \widehat{T}_{c,in}}{\ln\left(\dfrac{\widehat{T}_h - \widehat{T}_{c,in}}{\widehat{T}_h - \widehat{T}_{c,out}}\right)} \tag{9}$$

where $\widehat{T}_h$ is the saturation temperature of the hot stream, and $\widehat{T}_{c,in}$ and $\widehat{T}_{c,out}$ are the inlet and outlet temperatures of the cold stream. The overall heat transfer coefficient (U) for all options, except externally finned tubes, is given by:

$$U = \frac{1}{\dfrac{d_{t,e}}{d_{t,i}h_t} + \dfrac{\widehat{Rf}_t d_{t,e}}{d_{t,i}} + \dfrac{d_{t,e}\ln\left(\dfrac{d_{t,e}}{d_{t,i}}\right)}{2\widehat{k}_{tube}} + \widehat{Rf}_s + \dfrac{1}{h_s}} \tag{10}$$

where $d_{t,e}$ and $d_{t,i}$ are the external and internal tube diameters, respectively, h_t and h_s are the tube side and shell side heat transfer coefficients, $\widehat{k}_{tube}$ is the thermal conductivity of the tube wall, and $\widehat{Rf}_t$ and $\widehat{Rf}_s$ are the fouling factors of the tube side and shell side streams, respectively.

The corresponding expression of the overall heat transfer coefficient for externally finned heat exchangers is given by:

$$U = \frac{1}{\dfrac{d_{t,e}}{d_{t,i}h_t} + \dfrac{\widehat{Rf}_t d_{t,e}}{d_{t,i}} + \dfrac{d_{t,e}\ln\left(\dfrac{d_{t,e}}{d_{t,i}}\right)}{2\widehat{k}_{tube}} + \dfrac{\pi d_{t,e}\widehat{Rf}_s}{\eta_{tot}A_{tot,e}} + \dfrac{\pi d_{t,e}}{\eta_{tot}A_{tot,e}h_s}} \tag{11}$$

where $A_{tot,e}$ is the externally finned surface area per unit length and η_{tot} is the overall efficiency of the finned surface. The reference area of the expression of the overall heat transfer coefficients corresponds to the bare external surface (i.e. the area of the tubes, as though the fins do not exist). We include the equations for the finned surface efficiency in the Supplemental Material -Part B.

The pressure drops in the tube and shell sides (Δp_t and Δp_s) can be optionally limited to a maximum for the tube side and the shell side, respectively ($\widehat{\Delta p}_{t,max}$and $\widehat{\Delta p}_{s,max}$). Likewise, lower and upper bounds on flow velocities on the tube-side velocity and shell-side (vt and vs) are usually limited by minimum and maximum values ($\widehat{v}_{t,min}, \widehat{v}_{t,max}$ and $\widehat{v}_{s,min}, \widehat{v}_{s,max}$). Finally, to fall in the validity range of the correlations for the convective heat transfer coefficients, lower bounds on the Reynolds numbers on the tube and shell sides (Re_t and Re_s) are imposed ($\widehat{Re}_{t,min}$ and $\widehat{Re}_{s,min}$). Pressure drops, Reynolds number, and velocity limits sometimes vary with fluid allocation.

The baffle spacing bounds are (Taborek, 2008a):

$$0.2d_s \le lbc \le 1.0\,d_s \tag{28}$$

where d_s is the shell diameter and lbc is the baffle spacing. The bounds of the ratio between tube length (L) and shell diameter (d_s) are (Taborek, 2008b):

$$3\,d_s \le L \le 15\,d_s \tag{29}$$

The twist ratio (y) of the twisted-tape insert bounds are (Jiang et al., 2014):

$$3.0 \le y \le 6.0 \tag{30}$$

The helical pitch (P_{CI}) and helical angle (α_{CI}) of the coiled-wire insert bounds are (Jiang et al., 2014):

$$1.17 \le \frac{P_{CI}}{d_{t,i}} \le 2.68 \tag{31}$$

$$32 \le \alpha_{CI} \le 61 \tag{32}$$

The height (e_{IF}) of the internally finned tube bounds are (Serth and Lestina, 2014):

$$0.0075 \le \frac{e_{IF}}{d_{t,i}} \le 0.03 \tag{33}$$

The helical baffle spacing (lbc_{HB}) bounds are (Serth and Lestina, 2014):

$$0.3d_s \le lbc_{HB} \le 1.0d_s \tag{34}$$

The models for heat transfer coefficients for the tube side and the shell side, for regular and intensified exchangers, as well as the pressure drop models, are shown in Supplemental Material-Part B.

3. Global optimization

The condenser design variables of regular heat exchangers are the tube inner and outer diameters ($d_{t,i}$, $d_{t,e}$), tube length (L), tube pitch ratio (rp), tube layout ($Klay$), number of tube passes (Npt), and shell diameter (d_s). Additionally, the design variables for intensification technologies are: the pitch (H) and thickness (δ) of Twisted-tape Insert (TI), the helical pitch (PCI) and diameter (ECI) of Coiled-wire Insert (CI), the number of fins per unit length (N_{IF}), the helical angle (α_{IF}) and height (e_{IF}) of Internally Finned Tubes (IF), the helical angle (β_{HB}) of helical baffle (HB), the number of fins per unit length (N_{EF}), the height of fins ($\widehat{b}_{EF}$), and the thickness of the fins ($\widehat{t}_{EF}$) for the Externally Finned Tubes (EF). These variables are discrete, the available values being parameters given in advance.

Instead of describing the feasible space traditionally, that is, introducing binary variables for each parameter choice, a set of discrete candidate solutions is built by making all combinations of discrete values of the variables. Thus, each candidate is a specific condenser, which might be feasible or not. Based on this, Set Trimming (Costa and Bagajewicz, 2019) is employed to trim the search space. By incorporating inequality constraints sequentially, infeasible design candidates are gradually removed. After applying all inequality constraints, the final set contains all feasible candidates. Subsequently, through a simple sorting procedure, the optimal alternative can be determined.

The concept of Set Trimming was introduced by Costa and Bagajewicz (2019) by formalizing and generalizing an algorithm proposed by Gut and Pinto (2004) for a particular scenario. Set Trimming has been demonstrated to apply to the global optimization of various engineering devices. For instance, Lemos et al. (2020) applied Set Trimming to optimize shell-and-tube heat exchangers with no phase change. Chang et al. (2022) first used Set Trimming for the design of intensified heat exchangers without phase change and with a fixed configuration option. Chang and Shen (2024) followed with a similar article. It has also been demonstrated that global optimal solutions can be obtained for distillation column trays (da Cruz et al., 2023), distillation columns (Peccini et al., 2023), fired-heaters (Kim et al., 2023), reactors (Nahes et al., 2023b), and other equipment. While some kind of exhaustive enumeration could be performed (by for example creating a table, and subsequently testing all candidates one by one), Set Trimming is faster and more amenable to repeated runs and larger sets, regardless of the computational platform where such enumeration is implemented,

In this paper, we utilize Set Trimming to automatically obtain the optimal design among all condenser alternatives, as illustrated in Fig. 1. Supplemental Material-Part H provides additional details.

During the optimization, the program initially computes the optimal objective function for a certain type of condenser, such as horizontal shell side condensation. Subsequently, when calculating the second type of condenser, the optimal objective function from the first type is used as an additional constraint to eliminate candidates with higher objective function values. If the second type of condenser fails to achieve a smaller objective function, the program retains the computed objective function values from the previous type of condenser (i.e., the incumbent is not updated). It then proceeds to trim the search space for the next type of condenser. However, if the second type of condenser yields a smaller objective function, its optimal objective function value is employed for trimming the search space of the next type of condenser (i.e., the incumbent is updated). Therefore, the program proceeds to optimize each type of condenser in succession, ultimately obtaining the optimal condenser configuration and design variables.

The method does not need any parameter tuning, or multiple starting points, which is an advantage over using Metaheuristics Methods, which aside from not guaranteeing global optimality, suffer from these drawbacks. Starting points are also often needed by mathematical programming approaches.

4. Results

Two examples are used: the design of an overhead condenser and a heater using condensing saturated steam. The Complete Set Trimming was implemented using GAMS and runs on a server with Intel® Core i7–8700 U processor, which has 3.20 GHz speed and 16 GB memory.

For each case, we have two different optimization objectives, namely, Capital Cost (C_A, $\$$) and Total Annualized Cost (TAC, $\$/yr$). We use the following parameters: $\widehat{i} = 10\ \%$; $\widehat{ny} = 5\ yr$, $\widehat{EC} = 0.12\ \$/kWh$, $\widehat{Hour} = 8{,}000\ h$, $\widehat{\beta}_{IF} = 0.000208\ K^{-1}$, $\widehat{k}_{fin} = 103.84\ W/(m{\cdot}K)$, $\widehat{b}_{EF} = 1.841\ mm$ and $\widehat{t}_{EF} = 0.317\ mm$, $\widehat{t}_{IF} = 0.7\ mm$. The tube thickness is 2 mm, and $k_{tube} = 45\ W/(m{\cdot}K)$. For equation (1), we use $\widehat{\beta} = 0.8$ and values of $\widehat{\alpha}$ in Table 1. These values are taken from Pan et al. (2013) and Chang et al. (2022). However, with these parameters, the cost of a condenser is unrealistically small. For example, a 100 m^2 horizontal conventional condenser would cost around US\$800. To use more realistic values and to preserve the concavity of the cost function, we multiply these values by 50. If this correction is not made, the value of capital, when used to obtain the TAC, unrealistically favors condensers with small pressure

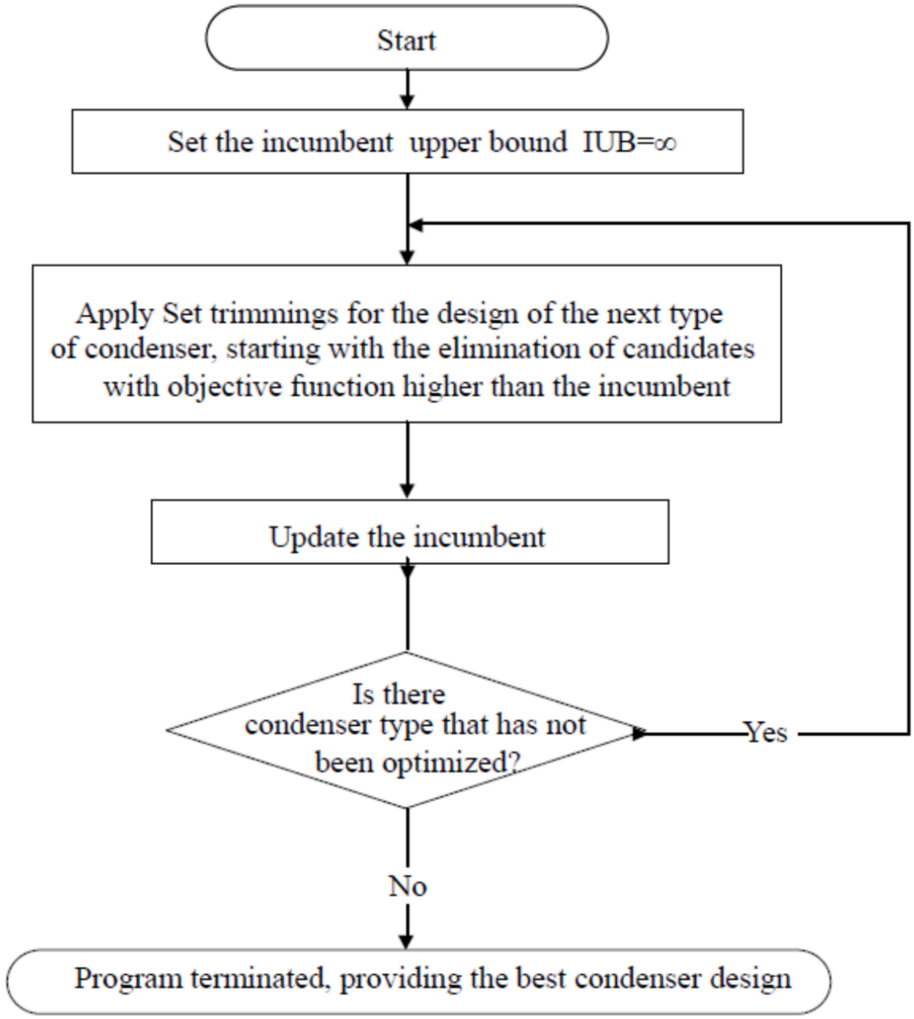

Fig. 1. Flowchart for global optimization.

Table 1
Cost coefficient..$\widehat{\alpha}$

Intensification Technique	$\widehat{\alpha}$
Twisted-tape Insert (TI)	1050
Coiled-wire Insert (CI)	1075
Internally Finned tube (IF)	1500
Helical Baffle (HB)	1100
Externally Finned tube (EF)	1500
No intensification	1000

Table 2
Design variables alternatives for general models.

Variable	Values
Outer tube diameter (m)	0.016, 0.018, 0.020, 0.022, 0.025, 0.030, 0.032, 0.038, 0.040, 0.070
Tube length (m)	1.219, 1.829, 2.439, 3.049, 3.659, 4.877, 6.098
Tube pitch ratio	1.25, 1.33, 1.50
Tube layout	1 = Square, 2 = Triangular
Number of tube passes	1, 2, 4, 6 (*)
Number of baffles	1, 2, …, 20
Shell diameter (m)	0.438, 0.489, 0.540, 0.591, 0.635, 0.686, 0.737, 0.787, 0.838, 0.889, 0.940, 0.991

* Vertical exchangers with condensation in the tubes have one tube pass.

drops.

We simplify the model names in the results presentation: horizontal shell side condensation (HS), horizontal tube side condensation (HT), vertical shell side condensation (VS), and vertical tube side condensation (VT). Tables 2 and 3 show the input data of this design problem and the physical properties of cold and hot streams. The two sets are from Pereira et al. (2021) and Chang et al. (2022), respectively.

We tested the Set Trimming results for the minimization of heat transfer area with the floating head with literature values obtaining the same results as Pereira et al. (2021). The detailed solutions of the optimal condensers with the floating head are shown in Supplemental Material-Part C.

Example 1. (*This example corresponds to a condenser of pure acetone associated with a distillation tower. This problem (Table 4) was first proposed by Smith (2005) and Smith (2016) and designed globally by Pereira et al. (2021), using rigorous reformulation into a linear model and solved using MILP procedures. We get the same results for the type of condensers they investigated and we present new results for other types. The maximum pressure drops are from Pereira et al. (2021). To make a comparison with the results of Pereira et al. (2021), no extra area was considered during the solution process ($\widehat{\%A}_{exc} = 0$).)* Fig. 2 *presents the capital cost design results for both vertical and horizontal condensers using the tube-side and shell-side intensification models. For this figure, all condensers were run separately. Intensification technologies can reduce the heat transfer area* (*we assume a fixed tubesheet for these*), *but they have additional costs* (*the clearance between the tube bundle and the shell is different from the floating*

Table 3
Design variables alternatives for intensification models.

Variable	Values
Twisted-Tape pitch (m)	0.074700, 0.103275, 0.131850
Twisted-Tape thickness (m)	0.002, 0.003, 0.004
Coiled-Wire helical pitch (m)	0.031955, 0.044179, 0.056402
Coiled-Wire diameter (m)	0.001411, 0.001951, 0.002491
Number of internal fins per unit length	30, 50
Helical angle of internal fins	35°, 45°
Height of internal fins (m)	0.000332, 0.000459, 0.000586
Helical baffle angle	20°, 30°, 40°, 50°
Number of external fins per unit length	50, 130, 210

head case).

When optimizing the capital cost (C_A), horizontal tube-side condensation with Externally Finned Tubes is the best condenser design for Example 1. When considering only conventional condensers, the cheapest is the horizontal condenser with tube-side condensation. The difference in cost between these two exchangers is relatively small.

Although several intensification techniques lead to a reduction in heat transfer area (HS-CI: −3.6 %, HS-IF: –32.6 %, VS-CI: −2.1 %, VS-IF: −36.0 %, VS-TI: −0.3 %, HT-HB: −3.6 %), their cost increases because of the cost coefficients. In contrast, the Externally Finned condensers reduce the heat transfer area more significantly compared to the conventional condensers (compared to HT, HT-EF: −43.3 %, compared to VT, VT-EF: −42.7 %), reducing the capital cost.

We remind the user that the areas of these externally finned tubes are not the real area of the finned surface; rather, it is the exposed area of the finned tube, without adding the area of the fins, which is employed for the evaluation of the objective function, according to Pan et al. (2013) approach (see Eq. (3). Finally, we remark that given all possible uncertainties, the following condensers are competitive in cost: HT, HT-HB, HT-EF, HS, HS-CI, HS-TI and HS-IF.

The heat transfer area of the HS-TI is slightly larger than that of the unenhanced condensers, which is due to the characteristics of the optimal heat exchanger design. The use of intensification technologies increases the heat transfer coefficient and reduces the heat transfer area to accomplish the heat transfer task. The increase in heat transfer area for VT-HB is due to the usual use of single tube pass in vertical condensers.

Fig. 3 illustrates the results when TAC is minimized. For this figure, all condensers were run separately.

When optimizing TAC, a horizontal condenser with tube-side condensation with Externally Finned tubes (HT-EF) is the best design for Example 1. When considering only conventional condensers, the cheapest is a horizontal condenser with shell side condensation (HS), although the difference with the horizontal condenser with tube side condensation is small.

The heat exchange area for the optimal Internally Finned condensers is smaller than the conventional ones, but adding fins increases the friction coefficient of the tube by increasing the turbulent flow within the tube side, resulting in higher pressure drop and consequently increasing pump operating costs. The reduced area fails to provide adequate compensation, thereby resulting in larger TAC values for both horizontal and vertical condensers in these conditions.

Similarly to the Coiled-wire Insert case, the Twisted-tape Insert slightly reduces the optimal heat transfer area. However, unlike the Coiled-wire Insert, the Twisted-tape Insert increased the optimal

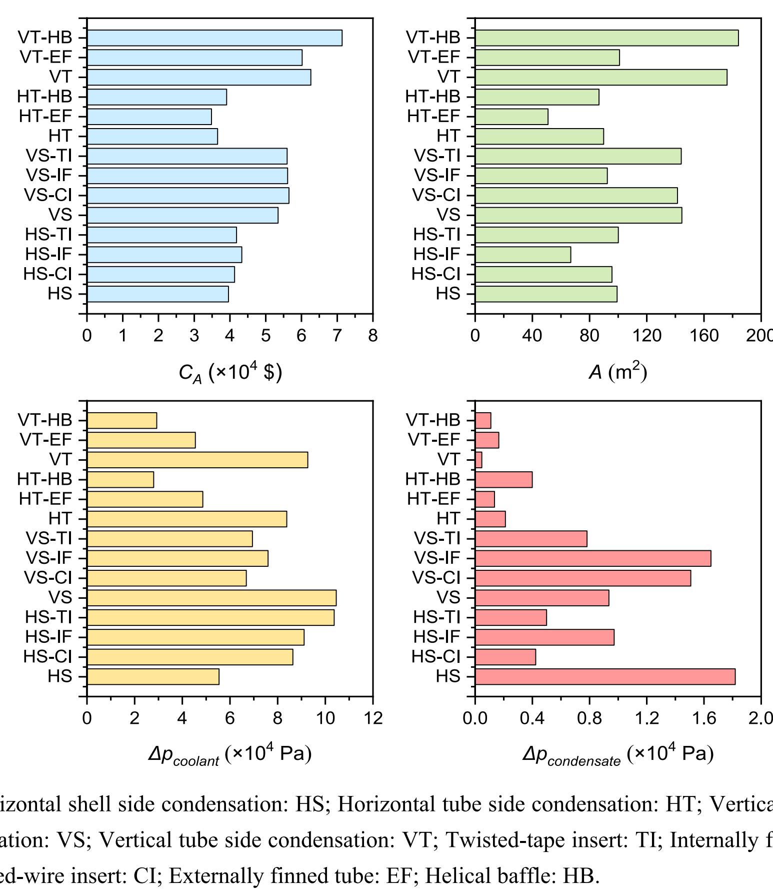

Fig. 2. Optimal results of the C_A minimization for Example 1. * Horizontal shell side condensation: HS; Horizontal tube side condensation: HT; Vertical shell side condensation: VS; Vertical tube side condensation: VT; Twisted-tape insert: TI; Internally finned tube: IF; Coiled-wire insert: CI; Externally finned tube: EF; Helical baffle: HB.

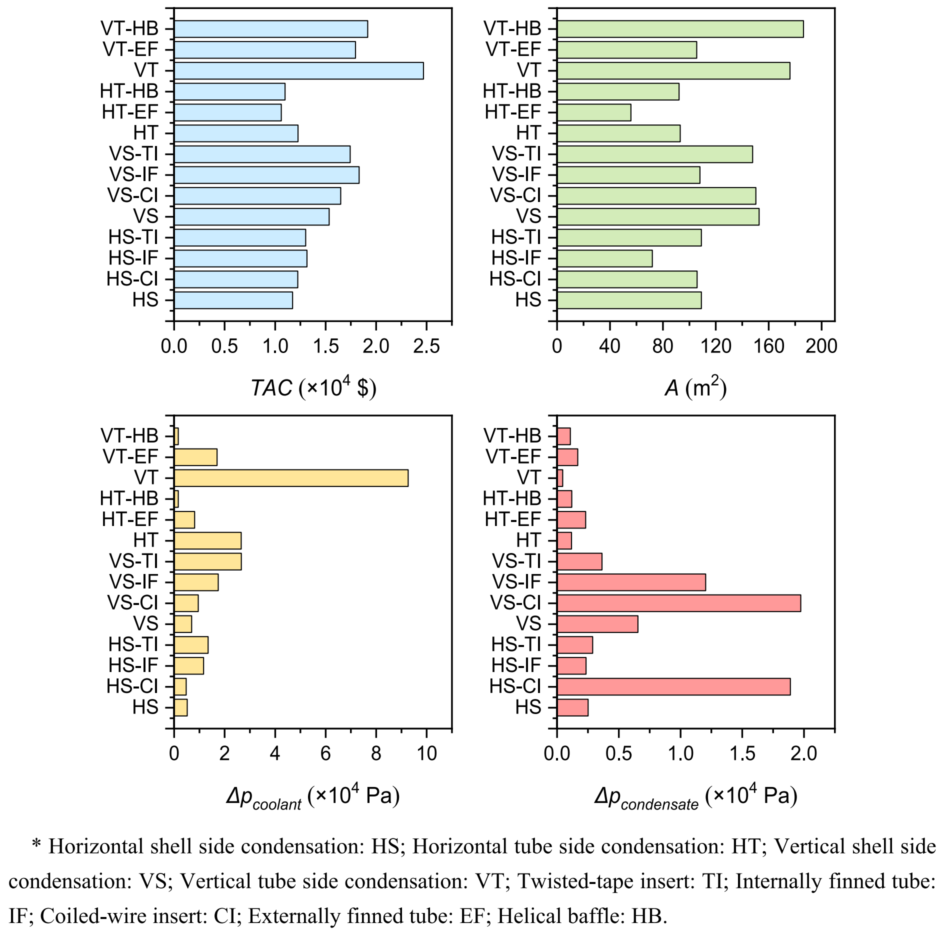

Fig. 3. Optimal results of the *TAC* minimization for Example 1. * Horizontal shell side condensation: HS; Horizontal tube side condensation: HT; Vertical shell side condensation: VS; Vertical tube side condensation: VT; Twisted-tape insert: TI; Internally finned tube: IF; Coiled-wire insert: CI; Externally finned tube: EF; Helical baffle: HB.

pressure drop on the tube side under all conditions, then increasing the operating costs, outweighing the compensation provided by the reduced area.

In both vertical and horizontal configurations, the TAC of the Externally Finned condenser is the lowest. Although the optimal operating costs are the lowest ones using Helical Baffles, the final TACs are higher.

All detailed results, including design solutions and thermofluid dynamic results of all different optimization objectives, are included in Supplemental Material-Part C.

Example 2. This example is a heater that uses condensing saturated steam to heat methanol. See Table 5 for design problem data and physical properties of the stream, respectively. This example is from Pereira et al. (2021).

The results of Example 2 *when using intensification using fixed tube-sheet for minimizing C_A are shown in* Fig. 4. *As in* Example 1, *all the condensers were optimized independently for comparison.*

As in Example 1, when optimizing C_A, horizontal condensers with tube-side condensation with Externally Finned are the optimal design for Example 2. This is because the heat transfer coefficient is the largest. When considering only conventional condensers, the cheapest is horizontal with shell-side condensation.

The optimal area of the horizontal condensers decreases by 13.5 % and 29.4 % for shell-side condensation and tube-side condensation, respectively, when compared to the vertical condenser. At the same time, when Coiled-Wire Inserts are used, the area decreases by 15.2 % and 11.1 % for conventional horizontal and vertical condensers, respectively, and when Twisted-tape Inserts are used, it decreases by 5.5 % and 3.2 %. The capital cost of the vertical condenser with shell side intensification remains relatively high, and the capital cost of Externally Finned intensification is the smallest. For horizontal and vertical condensers with Helical Baffles, compared with the conventional condensers, the tube area only decreases by 3.6 % and 14.7 %.

Fig. 5 shows the design solutions for minimizing TAC with intensification models. Again, all these condensers were optimized separately.

When optimizing *TAC*, horizontal condensers with shell-side condensation and Internally Finned are the optimal designs for this Example. When considering only conventional condensers, the cheapest is horizontal with shell-side condensation.

Detailed results can be found in Supplemental Material-Part C.

Comments on the results of both examples: Without the intention of generalizing the conclusions, we observe some trends in the results of both examples.

The optimal intensified condensers tend to exhibit larger pressure drops, but not always. For example, in the vertical condenser with shell-side condensation, pressure drops for intensified condensers are lower compared to conventional condensers.

For the optimal condensers with tube side condensation, despite the higher number of baffles and larger shell diameter in intensified condensers, its higher equivalent diameter leads to smaller pressure drops.

When minimizing capital cost, the optimal vertical condensers have a larger capital cost. This is because the model for the vertical condenser leads to a smaller overall heat transfer coefficient, which in turn increases the heat transfer area. When minimizing total annual cost, we find that under the same intensification technology conditions, the

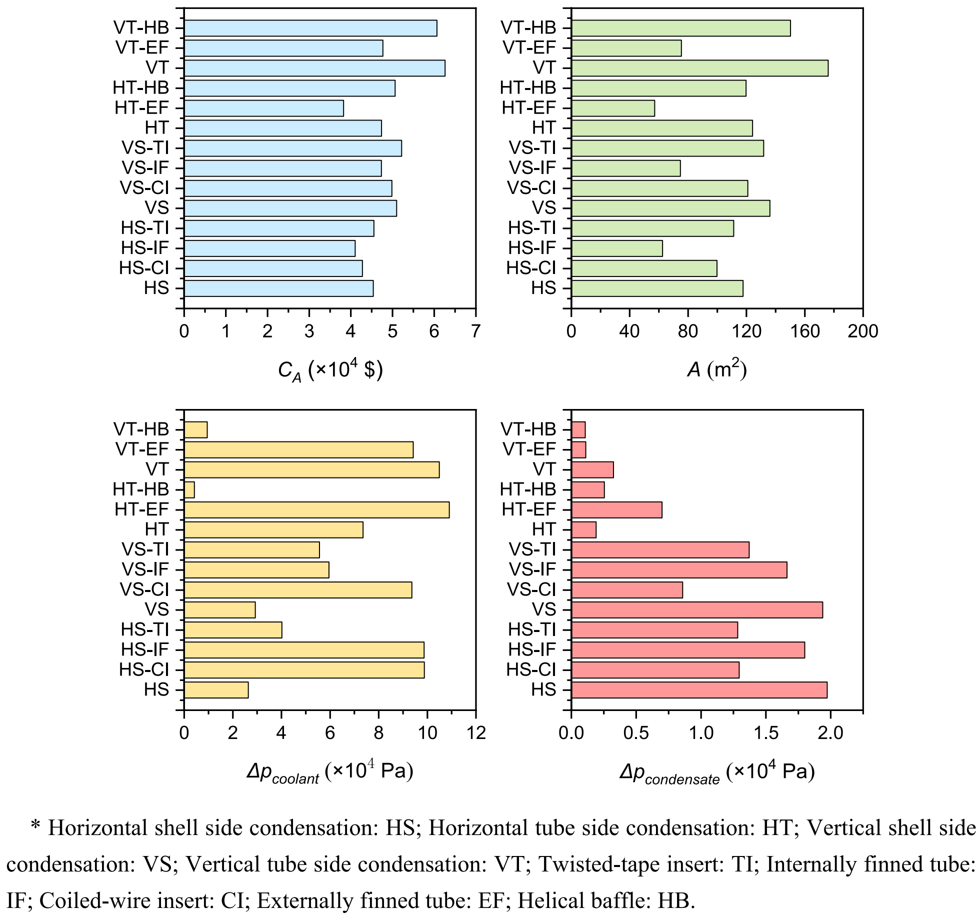

Fig. 4. Optimal results of the C_A minimization for Example 2. * Horizontal shell side condensation: HS; Horizontal tube side condensation: HT; Vertical shell side condensation: VS; Vertical tube side condensation: VT; Twisted-tape insert: TI; Internally finned tube: IF; Coiled-wire insert: CI; Externally finned tube: EF; Helical baffle: HB.

optimal TAC of a vertical condenser is larger than that of a horizontal condenser. Also, intensification results in only a marginal reduction in the heat transfer area, so it could potentially lead to a TAC larger than that of conventional condensers. The pressure drops and heat transfer area of the vertically oriented configuration are both larger than those of the horizontal ones. The Coiled-wire Insert reduces both the optimal heat transfer area and the pressure drop in the tube side, whereas the Internally Finned significantly reduces the heat transfer area with a smaller increase in pressure drop, resulting in a lower TAC. Twisted-tape Insert technology exhibits a higher TAC, primarily because the reduction in heat transfer area is not significant in the horizontal configuration and because it increases pressure drop in the vertical configuration.

For tube-side condensation, the pressure drops and heat transfer area obtained by the shell-side intensification models are still small, resulting in a lower TAC compared to those obtained without intensification.

Effect of limitations on maximum pressure drop: We re-run the examples, limiting the pressure drop of the coolant to 20 kPa, 40 kPa, and 60 kPa, respectively. Detailed results are provided in the Supplemental Material-Part C, and part of the optimization results are shown in Tables 6 and 7. As expected, lower pressure drop limits lead to poorer C_A and *TAC* solutions and, in some cases, infeasible solutions for the set of geometric options considered. Also, it is expected that as the maximum pressure drop decreases, some switch of optimal configuration may take place. Indeed, in all cases, the cheapest condenser remains is HT-EF, except in Example 2, where it remains the best for 40 and 60 kPa in minimizing C_A, but it switches to HS-IF for 20 kPa; HS-IF remains the best in minimizing *TAC* in Example 2. On average, the capital cost of the condensers increases by 2.3 %, 3.7 %, and 6.0 % for the 60, 40, and 20 kPa, respectively, when the capital cost is minimized and no changes when TAC is minimized.

From the above results, without the intention of generalizing, we conclude that the optimal cost condenser was not significantly affected by pressure drop limitations in these examples. Indeed, while a trade-off in costs is not a factor when the capital cost is minimized, the limitations on pressure drop present a different trade-off.

Solution time: Due to the lack of previous work on the design of condensers using optimization (metaheuristics or mathematical programming) including intensification and multiple configurations simultaneously, comparisons are impossible. At the same time, comparisons of solution time for Trial and Verification procedures are impossible because they are not computer-based.

For the existing set of variable choices, the Set Trimmings are solved according to the logic shown in Fig. 1 for the set of variable choices shown in Tables 2 and 3. These sets of parameters were enlarged to illustrate the impact of larger sets on time: two additional options for $d_{t,e}$ (0.019 m and 0.051 m), and five additional options for d_s (1.067 m, 1.143 m, 1.219 m, 1.372 m, and 1.524 m), were added. The expanded set of variables can be found in Supplemental Material – Part G. The solving time for each condenser for these two parameter sets are summarized in Table 8 and Table 9.

The computational times employed by each Set Trimming are very small. In our experience, they are orders of magnitude smaller for shell and tube exchangers without phase change solved using mathematical programming (Lemos et al., 2020), which are only globally optimal if a global solver is used, or they are rigorously reformulated as linear and solved using linear programming. Our experience in using GAMS and

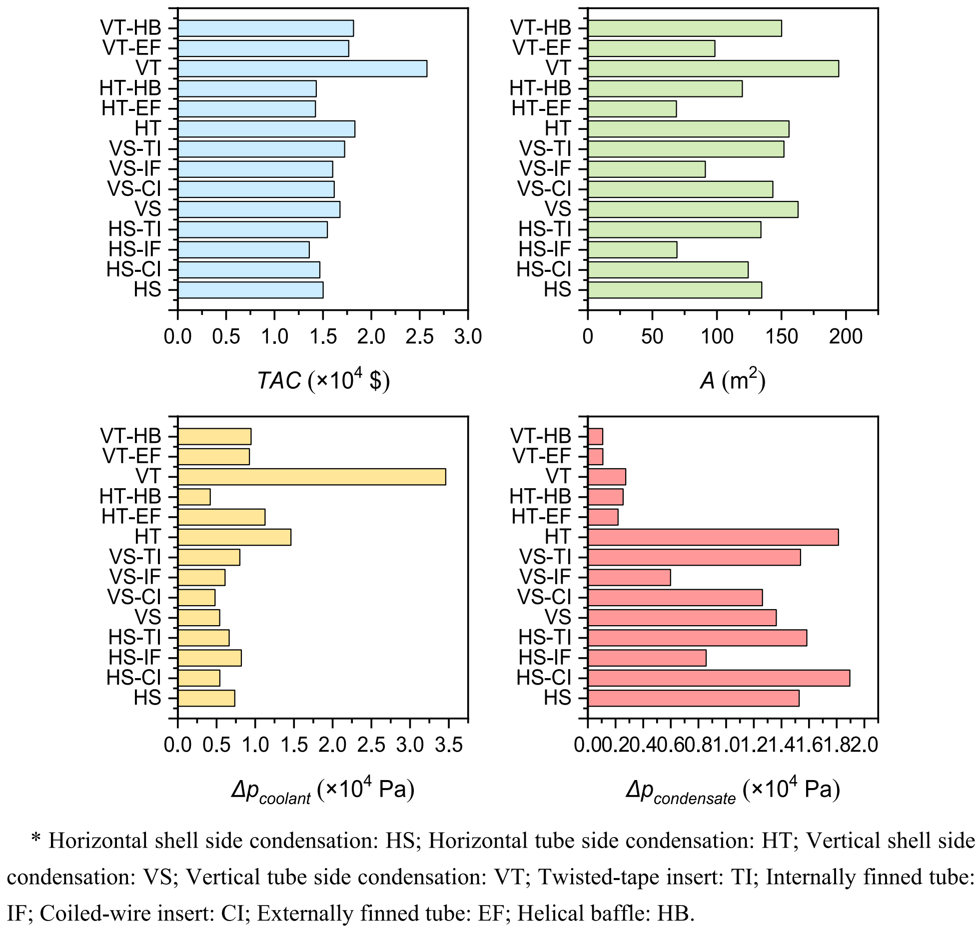

Fig. 5. Optimal results of *TAC* minimization for Example 2. * Horizontal shell side condensation: HS; Horizontal tube side condensation: HT; Vertical shell side condensation: VS; Vertical tube side condensation: VT; Twisted-tape insert: TI; Internally finned tube: IF; Coiled-wire insert: CI; Externally finned tube: EF; Helical baffle: HB.

Table 4
Example 1: Design problem data and physical properties.

	Condensing Stream (Acetone)	Coolant Stream (Water)
Mass flow rate (kg/s)	5.8	68.55
Inlet temperature (°C)	67	25
Outlet temperature (°C)	67	35
Fouling factor (m^2K/W)	0.00009	0.0002
Liquid density (kg/m^3)	736	996
Vapor density (kg/m^3)	3.12	–
Liquid heat capacity (J/(kg·K))	2,320	4,180
Liquid viscosity (Pa·s)	$0.213{\cdot}10^{-3}$	$0.797{\cdot}10^{-3}$
Vapor viscosity (Pa·s)	$1.4{\cdot}10^{-6}$	–
Liquid thermal conductivity (W/(m·K))	0.137	0.618
Heat of Vaporization (J/kg)	494,000	–
Flow velocity bounds (m/s)	10—30	1—3
Maximum pressure drop (kPa)	20	110

Table 5
Example 2: Design problem data and Physical properties.

	Condensing stream (Saturated steam)	Coolant stream (Methanol)
Mass flow rate (kg/s)	9.35	133.3
Inlet temperature (°C)	177	30
Outlet temperature (°C)	177	80
Fouling factor (m^2K/W)	0.0002	0.0001
Liquid density (kg/m^3)	890	750
Vapor density (kg/m^3)	4.83	–
Liquid heat capacity (J/(kg·K))	4,393	2,840
Liquid viscosity (Pa·s)	$0.153{\cdot}10^{-3}$	$0.340{\cdot}10^{-3}$
Vapor viscosity (Pa·s)	$1.6{\cdot}10^{-6}$	–
Liquid thermal conductivity (W/(m·K))	0.674	0.19
Heat of Vaporization (J/kg)	2025,000	–
Flow velocity bounds (m/s)	10 – 30	1 – 3
Maximum pressure drop (kPa)	20	110

Python is that GAMS is much slower because it spends considerable time building the initial sets. Intensified condensers have longer solving times due to their multiple variables related to intensification technologies. The solving time for the four common types of heat exchangers is less than 1 s, and solving for all 14 types of heat exchangers takes approximately 40–50 s. If each condenser design is run separately to further compare results and choose the best, the times are 76–82 s. For the expanded sets of parameters, the total time is 65–70 s, even though the size of the initial sets increases to values ranging from 685,440 to 8.2 million. This proves that the method can handle large sets of parameters without a large penalty in computational time.

Details of Set Trimming: The details of the number of candidates surviving in the optimization of each exchanger type separately are shown in the Supplemental Mmterial Part E. Roughly, the initial number of candidates for conventional condensers is 403,200, whereas for intensified exchangers, it varies from 1.2 to 4.8 million. The conventional condenser performs geometric trimming based on equations (28) and (29), reducing the number of candidates to 207,600 (with VT being 51,900). In addition to employing equations (28) and (29), the intensified condenser, depending on the type of intensification technology,

Table 6
Optimal results under different pressure drop limitations in Example 1.

Limitations (kPa)	A (m^2, for Min C_A)			A (m^2, for Min TAC)			TAC ($/yr)		
	20	40	60	20	40	60	20	40	60
HS	104.2	103.1	99.3	109.2	109.2	109.2	11,727	11,727	11,727
HS-CI	101.2	100.3	97.1	105.9	105.9	105.9	12,238	12,238	12,238
HS-IF	72.1	69.3	67.7	72.1	72.1	72.1	13,150	13,150	13,150
HS-TI	104.5	102.0	100.3	109.2	109.2	109.2	13,018	13,018	13,018
VS	148.9	148.9	147.0	152.8	152.8	152.8	15,350	15,350	15,350
VS-CI	144.2	144.2	143.3	150.4	150.4	150.4	16,495	16,495	16,495
VS-IF	108.1	107.7	102.3	108.1	108.1	108.1	18,310	18,310	18,310
VS-TI	–	147.0	144.2	–	147.9	147.9	–	17,427	17,427
HT	–	93.2	91.7	–	93.2	93.2	–	12,269	12,269
HT-EF	51.0	51.0	51.0	55.9	55.9	55.9	10,608	10,608	10,608
HT-HB	88.0	86.7	86.7	92.3	92.3	92.3	10,976	10,976	10,976
VT	–	–	–	–	–	–	–	–	–
VT-EF	105.6	101.6	101.0	105.6	105.6	105.6	17,952	17,952	17,952
VT-HB	184.4	184.2	184.2	186.4	186.4	186.4	19,160	19,160	19,160

* Horizontal shell side condensation: HS; Horizontal tube side condensation: HT; Vertical shell side condensation: VS; Vertical tube side condensation: VT; Twisted-tape insert: TI; Internally finned tube: IF; Coiled-wire insert: CI; Externally finned tube: EF; Helical baffle: HB.

Table 7
Optimal results under different pressure drop limitations in Example 2.

Limitations (kPa)	A (m^2, for Min C_A)			A (m^2, for Min TAC)			TAC ($/yr)		
	20	40	60	20	40	60	20	40	60
HS	124.5	117.8	117.8	134.8	134.8	134.8	15,014	15,014	15,014
HS-CI	112.1	104.8	103.5	124.3	124.3	124.3	14,676	14,676	14,676
HS-IF	66.0	64.7	63.5	69.1	69.1	69.1	13,596	13,596	13,596
HS-TI	118.6	114.7	111.3	134.2	134.2	134.2	15,464	15,464	15,464
VS	147.9	136.2	136.2	163.0	163.0	163.0	16,758	16,758	16,759
VS-CI	135.5	127.9	122.2	143.5	143.5	143.5	16,168	16,168	16,168
VS-IF	80.6	80.6	74.8	91.1	91.1	91.1	16,010	16,010	16,011
VS-TI	141.2	135.5	131.9	152.1	152.1	152.1	17,248	17,248	17,248
HT	152.5	140.7	133.8	156.0	156.0	156.0	18,312	18,312	18,312
HT-EF	66.5	62.7	60.5	68.7	68.7	68.7	14,230	14,230	14,230
HT-HB	119.8	119.8	119.8	119.8	119.8	119.8	14,307	14,307	14,307
VT	–	194.5	189.9	–	194.5	194.5	–	25,757	25,757
VT-EF	90.1	83.6	81.6	98.5	98.5	98.5	17,674	17,674	17,674
VT-HB	150.3	150.3	150.3	150.3	150.3	150.3	18,160	18,160	18,160

* Horizontal shell side condensation: HS; Horizontal tube side condensation: HT; Vertical shell side condensation: VS; Vertical tube side condensation: VT; Twisted-tape insert: TI; Internally finned tube: IF; Coiled-wire insert: CI; Externally finned tube: EF; Helical baffle: HB.

Table 8
The computational time (s) using the flowchart in Fig. 1.

Example 1			Example 2			Example 1 (larger variables)			Example 2 (larger variables)		
Type	Min C_A	Min TAC	Type	Min C_A	Min TAC	Type	Min C_A	Min TAC	Type	Min C_A	Min TAC
HS	0.8	0.8	HS	0.8	0.8	HS	1.2	1.2	HS	1.4	1.3
HS-CI	4.6	7.5	HS-CI	4.7	7.7	HS-CI	7.1	12.8	HS-CI	7.3	12.3
HS-IF	7.5	7.1	HS-IF	7.4	7.1	HS-IF	12.0	11.4	HS-IF	11.9	11.2
HS-TI	4.9	4.3	HS-TI	4.9	4.1	HS-TI	7.6	6.6	HS-TI	7.7	7.0
HT	0.7	0.7	HT	0.6	0.6	HT	1.0	0.9	HT	0.9	1.1
HT-EF	2.5	3.0	HT-EF	2.3	2.6	HT-EF	3.7	4.4	HT-EF	3.5	5.0
HT-HB	2.5	3.2	HT-HB	2.3	3.0	HT-HB	3.8	4.6	HT-HB	3.6	4.8
VS	0.6	0.6	VS	0.6	0.6	VS	0.9	0.9	VS	0.9	0.9
VS-CI	4.4	7.5	VS-CI	4.5	7.6	VS-CI	6.8	12.8	VS-CI	7.0	12.3
VS-IF	7.2	6.9	VS-IF	7.1	7.0	VS-IF	11.2	11.2	VS-IF	11.4	11.1
VS-TI	4.9	4.3	VS-TI	4.9	4.1	VS-TI	7.5	6.7	VS-TI	7.6	6.9
VT	0.2	0.4	VT	0.3	0.4	VT	0.3	0.6	VT	0.3	0.7
VT-EF	0.7	1.0	VT-EF	0.8	1.0	VT-EF	1.0	1.6	VT-EF	0.9	1.7
VT-HB	0.7	0.8	VT-HB	0.7	0.9	VT-HB	1.0	1.2	VT-HB	0.9	1.3
Total	41.5	48.1	**Total**	41.9	47.5	**Total**	65.1	77.1	**Total**	65.3	77.6

* Horizontal shell side condensation: HS; Horizontal tube side condensation: HT; Vertical shell side condensation: VS; Vertical tube side condensation: VT; Twisted-tape insert: TI; Internally finned tube: IF; Coiled-wire insert: CI; Externally finned tube: EF; Helical baffle: HB.

can utilize equations (30) to (34) for geometric trimming. This approach reduces the number of candidates of intensified condensers to 747,360–103,800. Next, the velocity and Reynolds-based trimming reduces those numbers to 13,663–32,285 for conventional condensers and 16,374–294,008 for intensified exchangers. Then, the pressure drop-based trimming reduces those numbers to 5,860–24,832 for

Table 9
Computational time (s) using individual runs.

Example 1			Example 2			Example 1 (larger variables)			Example 2 (larger variables)		
Type	Min C_A	Min *TAC*	Type	Min C_A	Min *TAC*	Type	Min C_A	Min *TAC*	Type	Min C_A	Min *TAC*
HS	0.8	0.8	HS	0.7	0.8	HS	1.2	1.3	HS	1.2	1.2
HS-CI	7.1	7.1	HS-CI	6.8	7.2	HS-CI	11.4	11.2	HS-CI	11.2	11.1
HS-IF	16.2	16.3	HS-IF	14.6	14.6	HS-IF	25.6	25.9	HS-IF	25.0	25.1
HS-TI	7.5	7.5	HS-TI	7.3	7.2	HS-TI	11.9	12.0	HS-TI	11.9	12.0
HT	0.9	0.8	HT	0.8	0.8	HT	1.3	1.3	HT	1.3	1.3
HT-EF	4.2	4.1	HT-EF	3.8	3.8	HT-EF	6.1	6.0	HT-EF	6.3	6.2
HT-HB	4.2	4.2	HT-HB	3.7	3.7	HT-HB	6.2	6.2	HT-HB	6.2	6.0
VS	0.8	0.8	VS	0.8	0.8	VS	1.3	1.3	VS	1.3	1.3
VS-CI	7.3	7.3	VS-CI	7.0	7.0	VS-CI	11.7	11.6	VS-CI	11.4	11.4
VS-IF	16.7	16.8	VS-IF	15.1	15.2	VS-IF	26.3	26.8	VS-IF	25.7	25.8
VS-TI	8.0	7.9	VS-TI	7.6	7.5	VS-TI	12.4	12.7	VS-TI	12.4	12.6
VT	0.9	0.9	VT	0.8	0.8	VT	0.5	0.5	VT	0.5	0.4
VT-EF	3.6	3.6	VT-EF	3.5	3.5	VT-EF	1.4	1.4	VT-EF	1.5	1.5
VT-HB	4.1	4.1	VT-HB	3.6	3.6	VT-HB	1.7	1.6	VT-HB	1.5	1.5
Total	82.3	82.3	**Total**	76.2	76.5	**Total**	119.2	119.8	**Total**	117.2	117.3

* Horizontal shell side condensation: HS; Horizontal tube side condensation: HT; Vertical shell side condensation: VS; Vertical tube side condensation: VT; Twisted-tape insert: TI; Internally finned tube: IF; Coiled-wire insert: CI; Externally finned tube: EF; Helical baffle: HB.

conventional condensers and 7,080–141,723 for intensified exchangers. Finally, after the area trimming, the surviving candidates are 23–11,319 for conventional condensers and 1,197–37,537 for intensified exchangers. From these feasible candidates, the best condenser is obtained by sorting.

The algorithm in Fig. 1 runs all trimmings sequentially, with one additional trimming for both objective functions, which requires keeping the best solution (Incumbent upper bound), where all feasible and infeasible candidates that exhibit a cost higher than the incumbent from the initial set for the set of candidates under consideration, are eliminated. The details of these trimmings are displayed in Supplemental Material Part F.

5. Conclusions

This paper primarily investigates the design optimization of different types of shell and tube condensers using Set Trimming. It covers four conventional condenser configurations and five intensification technologies applied to the cold stream.

The conventional condenser configurations include horizontal shell-side condensation, horizontal tube-side condensation, vertical shell-side condensation, and vertical tube-side condensation. Tube-side intensification techniques include Twisted-tape Insert, Coiled-wire Insert, Internally Finned. Shell-side intensification techniques include Helical Baffle, Externally Finned. During optimization, the design variables are discrete, and globally optimal solutions for different types of condenser design problems are obtained using Set Trimming. The Set Trimming results of our models were compared with literature results and validated.

The optimization results indicate that vertical condensers generally have larger heat transfer areas, which is related to the vertical configuration model. When minimizing capital costs, externally finned technology is more likely to be chosen because this intensification technology can significantly reduce the heat exchange area, compensating for the increase in intensification costs. When minimizing TAC, intensification techniques increase the heat transfer coefficient and thus reduce the heat exchange area (the intended purpose), but they increase the pressure drop on the enhanced side, possibly leading to higher costs. Finally, we explore the impact of pressure drop limits, finding that they significantly affect the area and sometimes the optimal type of condenser.

In addition, the sequential Set Trimming exhibits a small optimization time, even when run separately. The integration of the individual optimization further shortened the solution time.

6. Nomenclature

The nomenclature is given in Supplemental Material Part A.

CRediT authorship contribution statement

Yi Cui: Writing – original draft, Validation, Investigation, Data curation. **Chenglin Chang:** Methodology. **Yufei Wang:** Writing – review & editing, Supervision, Methodology, Funding acquisition. **André L.H. Costa:** Writing – review & editing, Methodology. **Miguel J. Bagajewicz:** Writing – review & editing, Supervision, Methodology, Conceptualization.

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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Appendix A. Supplementary data

Supplementary data to this article can be found online at https://doi.org/10.1016/j.ces.2025.121776.

Data availability

No data was used for the research described in the article.

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