HEAT EXCHANGER OPTIMIZED DESIGN
COMPARED WITH INSTALLED INDUSTRIAL SOLUTIONS

Antonio C. Caputo
University of Roma Tre, Department of Engineering,
Via della Vasca Navale, 79, Roma, 00146, Italy.
acaputo@uniroma3.it

Pacifico M. Pelagagge & Paolo Salini
University of L'Aquila, Department of Industrial Engineering, Information and Economics,
Zona industriale di Pile, L'Aquila, 67100, Italy
pacifico.pelagagge@univaq.it
paolo.salini@univaq.it

ABSTRACT
Commercial software tools for computer aided design of shell and tube heat exchangers are widely used in engineering departments of process plant equipment manufacturers. In this paper a comparison is carried out between actual installed heat exchangers, designed resorting to a leading commercial software tool, and the corresponding equipment configurations obtained by a genetic algorithm-based software tool, developed by the authors for optimal heat exchangers design. Reference is made to a set of four case studies representing exchangers built by a firm operating in the process plant construction sector and designed utilizing the commercial software tool. The corresponding design specifications are then used to redesign the heat exchangers resorting to the above mentioned research tool, and the resulting architectures are compared on the basis of equipment weight, assuming that this is the parameter used by manufacturers to estimate cost. Results show that the research tool, although characterized by a simpler user interface and reduced set of features, consistently delivers superior equipment architectures with significant weight reduction respect commercial solutions, allowing at the same time the compliance with thermal duty specifications. This case study analysis against installed benchmark equipment contributes to validate the developed optimization software tool and shows its capabilities of delivering less expensive heat exchanger designs.

Keywords
Heat exchanger design, genetic algorithm, optimization

1. INTRODUCTION

Design of a heat exchanger is an iterative process relying heavily on designer's experience. Usually a reference geometric configuration is chosen at first, and an allowable pressure drop value is fixed. Then heat exchanger structural features, i.e. the design variables, are chosen based on the design specifications and on assumption of several mechanical and thermodynamic parameters, in order to obtain a satisfactory heat exchange coefficient leading to a suitable utilization of the heat transfer surface. The procedure is then iterated until a reasonable design is obtained which meets specifications with a satisfying compromise between pressure drop and thermal exchange performances. A number of textbooks (i.e. Kern, 1950) or reference handbooks (Hewitt, 1998) are available to guide the designer in this process, while numerical examples are provided for instance by Kakac et al. (2012) for a range of exchanger types, or by Mukherjee (1998) for shell and tube equipment.

Although well proven, this kind of approach is time-consuming and may not lead to a cost-effective design as no economic criteria are explicitly accounted for and no guarantee of the solution optimality is given. Considering the functional importance and widespread utilization of heat exchangers in process plants, their minimum cost design is, instead, an important goal. In particular, the minimization of energy related expenses is critical in the optic of energy savings and resources conservation, as well as in a life-cycle cost perspective. On the other hand, weight or surface area minimization is important when capital investment is to be reduced. In the literature, attempts to automate and optimise the heat exchanger design process have been proposed from a long time, and the problem is still the subject of ongoing research. The suggested approaches mainly vary in the choice of the objective function, in the number and kind of sizing parameters utilized, and in the numerical or analytical optimisation method employed. Software packages are also available on the market to assist designers in developing satisfying equipment architectures in reasonable time, while some of them also include optimization features aimed at minimizing cost. Designers of engineering departments frequently turn to these commercial software producing one or more alternative heat exchanger design, and the final configuration is chosen on the basis of designer's experience.

On the other side, academic literature in recent times has shown a renewed interest in developing tools which automate the design process while seeking an optimized design. This corresponds to the availability of new optimization techniques, such as genetic algorithms (GA) and other evolutionary algorithms, able to handle a large number of design parameters including both discrete and continuous variables, without resorting to gradient-based methods, to efficiently explore a large solution space.

Tayal et al. (1999) were among the first to suggest using GA in heat exchanger design optimization. However, they did not develop a design tool but rather a methodology based on a command procedure to run the HTRI commercial design program iteratively coupled to a GA or a Simulated Annealing optimization engine. Caputo et al. (2008) developed a GA-based design optimization tool which is the basis of the one utilized in this work. Ponce-Ortega et al. (2009) use a GA and the Bell- Delaware sizing method to minimize the total annual cost of shell-and-tube heat exchangers. Amini and Bazargan (2014) as well as Sanaye and Hajabdollahi (2010) adopt the ε-NTU approach for computing heat transfer rates and the Bell-Delaware procedure to size the heat exchanger, choosing design variables values resorting to a GA and exploring the Pareto frontier of efficiency vs total cost. They also perform parametric analysis to assess the role played by relevant design variables. Fettaka et al. (2013) frame the GA-based design problem as a multiobjective optimization one, attempting to minimize simultaneously surface area and pumping power. Azad and Amidpour (2011) and Yang et al. (2014) utilize constructal theory to define an objective function which is then optimized resorting to GA. Guo et al. (2009a) use GA coupled with an objective function represented by the field synergy number which is defined as the indicator of the synergy between the velocity field and the heat flow. Guo et al. (2009b) use a GA to minimize an objective function representing the dimensionless entropy generation rate.

Apart from GA, a number of other evolutionary optimization techniques have been suggested to solve the shell-and-tube exchanger design problem. Babu and Munawar (2007) use Differential Evolution (DE) algorithm to minimize the heat transfer area of shell-and-tube heat exchangers. Ravagnani al. (2009), Patel and Rao (2010), and Lahiri et al. (2012) adopt a Particle Swarm Optimization (PSO) method showing that it can be as effective as a GA. Mariani et al. (2012) adopt a modified quantum particle swarm optimization (QPSO) method, named Zaslavskii chaotic map sequences (QPSOZ) to shell-and-tube heat exchanger optimization, showing that it could be superior to GA, PSO, and classical QPSO. Asadi et al. (2014) approach the design optimization problem resorting to a so called Cuckoo-search-algorithm, which is shown to provide improved results respect a GA and PSO. Şahin et al. (2011) instead use the Artificial Bee Colony (ABC) algorithm to minimize the total discounted cost of the equipment. Hadidi and Nazari (2013) adopt the biogeography-based (BBO) algorithm which attempts to mimic population migration across diverse habitats and compare it to other evolutionary optimization techniques such as GA, PSO and ABC. Fesanghary et al. (2009) use global sensitivity analysis (GSA) and harmony search algorithm (HSA) comparing the effectiveness of their approach to GA. Hadidi et al. (2013) develop an economic optimization model based on imperialist competitive algorithm (ICA). Lahiri and Khalfe (2014) instead adopt both hybrid DE and Ant Colony Optimization techniques. Rao and Patel (2013) suggest using a Teaching-learning-based optimization (TLBO) method, which is an heuristic algorithm based on the natural phenomenon of teaching-learning process. They compare obtained results with those of GA.

Costa and Queiroz (2008) develop a design algorithm using an iterative procedure to explore the design space where search is carried out along the tube count table where the established constraints and the investigated design candidates are employed to eliminate nonoptimal alternatives, thus reducing the number of rating runs executed. Surface area minimization was the stated design objective. Serna and Jiménez (2005) develop an analytical procedure for heat exchanger optimization based on Bell–Delaware design method and a compact formulation that relates the shell-side pressure drop with the heat exchanger area and the heat transfer coefficient. Ravagnani and Caballero (2007) as well as Onishi et al. (2013) develop mixed-integer non-linear programming models to optimize shell-and-tube exchangers.

However, some scholars are skeptical about the use of precise optimization methods when applied to heat exchanger design, owing to the inherent fuzziness of the problem given the uncertainty in operating conditions and in the adopted design correlations (Bell, 2000). Nevertheless, while from this point of view some studies considering heat exchangers operating under variable stochastic conditions have been developed (Caputo et al., 2010 and 2012), the utilization in industrial environment of design tools developed in research institutions for academic purposes is still limited.

Therefore, the aim of this paper is to explore in a realistic context the performances of a heat exchanger design optimization tool, developed for research purposes and available in the literature (Caputo et al., 2008), by comparing the architecture of representative heat exchangers designed with this tool with those of corresponding heat exchangers designed by the engineering department of a process plant construction contractor resorting to a state of the art commercial software, and currently operating in process plants. In this manner the research tool can be validated utilizing actual equipment installed in process plants as a benchmark, and its capabilities in providing lower cost design can be assessed. The paper is organized as follows. At first a description of the adopted research tool for optimal design of shell-and-tube heat exchangers is described in brief. Then the design procedure utilized by commercial heat exchangers design packages is reviewed. Afterwards, a methodology for comparing in a consistent manner the results of the two design approaches is stated. Finally, four distinct case studies are examined and their results are discussed.

2. REFERENCE RESEARCH SOFTWARE TOOL

The design procedure used in this paper is described extensively in a previous work (Caputo et al. 2008) where a detailed computer model has been developed for optimal design of shell-and-tube heat exchangers operating in stationary conditions and without uncertainties in heat transfer estimation. The tool is built on an optimization procedure based on GA and relies on equipment design procedures based on proven and widely accepted literature methods. The original model utilized the procedure developed by Kern (Kern, 1950) , while subsequent versions adopt the Bell-Delaware design method (Hewitt, 1998). The tool, which is the result of an ongoing research effort, has been at first extended to take into account constructive details in the capital cost estimation (Caputo et al. 2009), and to allow a joint optimization of exchanger design and cleaning schedule (Caputo et al. 2011). The model has been also upgraded to allow optimal design under deterministically or stochastically variable operating conditions (Caputo et al. 2010, 2012). Here only a short description is provided, whereas the reader may refer to the above papers for details on the GA implementation and the exchanger sizing procedure.
In this paper no uncertainty in heat transfer estimation is accounted for and stationary and known operating condition are assumed for design purposes. This choice is dictated by the need to operate in a context similar to the one encountered by those designing an exchanger using commercial software tools.

The procedure for optimal heat exchanger design includes the following steps:

- estimation of the exchanger heat transfer area based on the required duty and other design specification, assuming a set of design variables values;
- evaluation of the objective function (in the original software version the total life-cycle cost, but here the equipment weight);
- utilization of the optimisation algorithm to select a new set of values for the design parameters;
- iteration of the previous steps until a minimum of the objective function is found.

The entire process is schematised in Figure 1. Design specification indicate the heat duty of the exchanger, and are given by imposing five of the following six parameters: the mass flow rates of the two fluids (m_h and m_c), as well as the inlet and outlet temperatures of the hot (t_{hi}, t_{ho}), and cold (t_{ci}, t_{co}) fluids. The remaining parameter being determined by an energy balance.

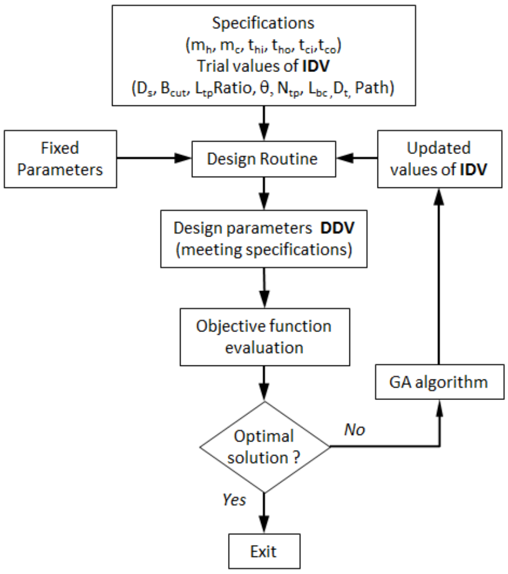


Figure 1. Optimized design procedure.

A set of thermo physical properties, process data, fouling resistances, and fixed equipment characteristics (heat exchanger TEMA type) are assigned by designers. Starting from such input data a random starting value is given to a set of independent design variables (IDV). The IDV number and meaning depends on the equations used to size the equipment. In this work the Bell-Delaware method is used, and the selected IDV are the inside shell diameter D_s (m), the tube outside diameter D_t (m), the central baffle spacing L_{bc} (m), the extremal baffle spacings L_{bi} and L_{bo}, the pitch ratio $L_{tp}Ratio$, the baffle cut B_{cut}, the sealing strips number N_{ss}, the tube layout angle θ, the tube pass number N_{tp}, and the fluid path (hot stream inside tubes or shell). The other heat exchanger's characteristics (i.e. the dependent design variables, DDV) are then directly computed from the IDV. Using empirical rules of thumbs it is possible to determine the tubes number N_{tt}, whereas using Bell-Delaware's design equations it is possible to evaluate all the others geometrical DDV and fluid dynamical equipment characteristic (i.e. flow velocity, pressure loss etc.). Once the DDV are computed from the IDV the overall heat transfer coefficient (U) is estimated on the basis of shell-side and tube-side heat exchange coefficients h_s, h_t and fouling coefficients

$$U = \frac{1}{\frac{1}{h_s} + R_{f,s} + \frac{D_t \ln\left(D_t / D_{ti}\right)}{2\lambda} + \frac{D_t}{D_{ti}}\left(R_{f,t} + \frac{1}{h_t}\right)} \tag{1}$$

where $R_{f,s}$ and $R_{f,t}$ are the shell-side and tube-side fouling resistances, λ the thermal conductivity of tube walls, and D_{ti} the internal tubes diameter.

This allows to determine the minimum total heat exchanger's heat transfer area

$$S_{\min} = \frac{Q}{U\,\Delta T_{ML}\,F} \tag{2}$$

being Q the heat duty, ΔT_{ML} the logarithmic mean temperature difference, and F the temperature difference corrective factor according to TEMA rules for shell and tube exchangers according to the equipment architecture. By knowing the required S_{min} and computing the heat transfer area per unit length of the shell, on the basis of tubes number and diameter, the minimum tube length L_{tt} (m) follows.

An integer baffles number is then computed as

$$N_b = \sup \mathrm{int}\left[\frac{L_{tt}}{L_{bc}}\right] \tag{3}$$

allowing to update the tubes length as

$$L_{tt} = (N_b - 1)L_{bc} + L_{bi} + L_{bo} \tag{4}$$

The definitive tubes length is then assumed as that of the nearest greater commercial tube length, and the difference between computed and commercial tubes length is evenly distributed by increasing L_{bi} and L_{bo}. This determines the actual surface area (S) of the exchanger, which can result greater than S_{min}, thus defining all constructive details of the exchanger satisfying the assigned thermal duty specifications.

Constructive details of the exchanger structure are then used to compute the objective function. The optimisation algorithm, based on the value of the objective function, updates the trial values of the optimisation variables (IDV) which are then passed to the design routine to define a new architecture of the equipment. This process is iterated until a minimum of the objective function is found or a prescribed convergence criterion is met. While genetic algorithms, being a stochastic optimization technique, can not guarantee that the absolute optimum is found, the presence of random mutations in the generated population reduces the risk of being caught in local minima. Moreover, genetic algorithms are a frequently used and generally accepted optimization technique in heat exchanger design. However, it should be pointed out that the scope of this paper is not to design the "absolute best" heat exchanger for each case study, but rather to find a good engineering design which is superior to those obtainable by commercial design software tools and the associated computer aided manual design routines.

In this work the objective function to be minimized is the equipment weight; this is due to the widespread use in engineering departments of weight as cost driver for equipment investment cost estimation. Furthermore, a lower equipment weight is beneficial to transportation and handling, also making installation and maintenance operations easier. In order to avoid the optimization routine to seek a minimum weight by excessively increasing flow velocities in order to increase the overall heat transfer coefficient and reduce S_{min}, a constraint on maximum allowable pressure drop has been included. A threshold of 70 kPa has been set consistent to commonly accepted design guidelines. A dedicated algorithm allowing equipment weight estimation (Caputo et al., 2009) has been coded in the software tool to compute the value of the objective function. Starting from the equipment geometric features and chosen construction materials the routine computes the weight considering the main heat exchanger's parts (front and rear end, shell, tube bundle, baffles, nozzles, tie rods and spacers, flanges and others ancillaries).

3. COMMERCIAL HEAT EXCHANGERS DESIGN PACKAGES

As previously mentioned, the design procedure of a shell-and-tube heat exchanger is very time consuming, and in industrial practice it is common practice to rely on dedicated commercial software. A number of software packages are available to assist designers in producing a satisfactory equipment design. The procedure is usually iterative and the output is not deterministically defined as a number of design parameters have to be chosen by the user, who may interact with the software multiple times during the design process. As a consequence the quality of the overall design depends largely on designer's experience. For this reason, usually, such tools do not attempt an equipment optimization, but rather provide a check about whether the produced design meets commonly accepted good design values for some critical parameters, such as ratio of thermal resistances, ratio of cross to window velocity, shell-side and tube-side velocity (i.e. maximum pressure drop), flow induced tubes vibration and so on. In case a design rule is violated a warning is issued and the designer can change his choices until a satisfying design is obtained. Notable examples of state of the art commercial software tools for shell-and-tube heat exchanger design are HYSYS, AspenTech's Aspen Shell & Tube Exchanger, or Heat Transfer Research Institute's HTRI Xist of the Xchanger suite. Some of these tools include some sort of design optimization capability, but details of the proprietary optimization routines usually are not disclosed.

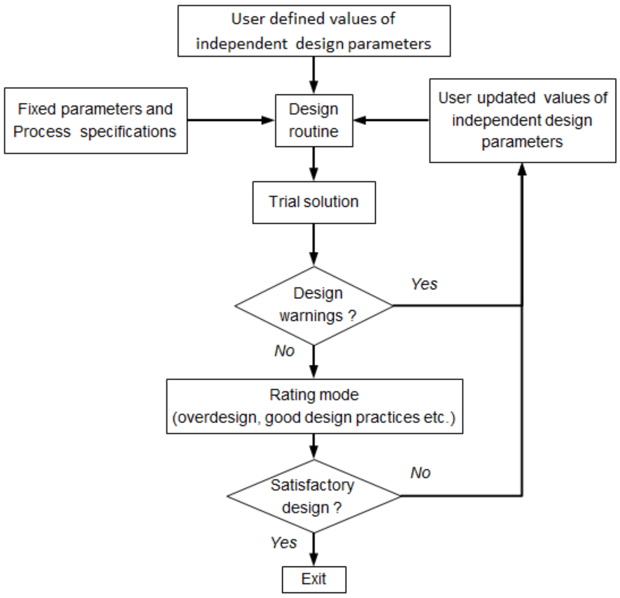


Figure 2. Commercial software utilization procedure

Overall, the general operating scheme of the above tools, when optimization routines are not included, is depicted in Figure 2. In the *design mode* the user inputs process specifications, defines some fixed requirements, i.e. maximum pressure drop or equipment arrangement, then he chooses a set of trial values for some design parameters (i.e. tubes pitch, tubes diameter, tubes length etc.). The software delivers a trial solution based on proprietary design routines, and possibly issues a series of warnings when some threshold values of sensitive design parameters are exceeded. The user then updates his design choices and the entire procedure is iterated until no warning is issued. Then this tentative design is used as a basis for running the software in the *rating mode*, where the actual performances of the equipment as designed are evaluated. A percentage of overdesign as well as other performance indicators are computed in order to verify that good design practices are observed. In case the user is not satisfied with the current equipment architecture, or with the results of the rating calculations, the entire design process may be started again.

4. COMPARISON METHODOLOGY

To show the capabilities of the developed research tool in comparison to standard commercial packages, a set of actual heat exchangers designed by an Italian firm operating worldwide as engineering contractor for process plants construction will be considered. The four heat exchangers analyzed in this paper are all installed and operational, and are intended to represent the typical level of equipment design quality likely to be encountered in the process plant construction industry. All exchanger were designed by expert operators resorting to one of the previously cited commercial tools, the name of which will not be disclosed. However, there is no claim in this work that the reference design is the best design that could be obtained by the utilized commercial software. In fact, the final design is the result of:

a) the interaction between the built-in design rules and the correlations coded in the software tool;
b) the expert choices made by the designers when setting values of the free design parameters.

Therefore, given some initial specification, the same software package will generate different designs when utilized by different designers. Nevertheless, it can be reasonably stated that examined designs represents at least the average performance that a traditional design process utilizing state of the art commercial software tools allows. In fact, the considered heat exchangers have been manufactured by a specialized firm competitive in its business and are currently operating in process plants across the world.

The same specifications utilized for designing the benchmark exchangers have been fed to the previously described GA-based research software tool utilizing equipment weight as the objective function to be minimized. The benchmark equipment and the proposed design are then compared referring to weight.

Note that the equipment weight estimated by the research software tool when searching for the optimal solution is not directly comparable with the weight declared for the exchanger generated by the commercial software package. In fact, the heat exchanger components considered when estimating the equipment total weight may differ among commercial software packages and between a specific commercial software and the proposed research tool. Moreover, the two tools utilized in this work adopt different weight estimation algorithms, and the commercial software weight estimation model is generally not known. Therefore, in order to allow a consistent comparison between the two candidate solutions of each case study (one generated using commercial software and one obtained using the considered research tool) the weight of both equipment has been computed resorting to the commercial software routine, as depicted in Figure 3. In greater detail when an optimal equipment architecture has been generated by the research software tool, its constructive details are fed to the commercial software package which is then run in the *rating mode* instead of the *design mode*. This is made to check that the *optimized* design satisfies the design specifications on the basis of the heat transfer correlations used by the commercial tool, and to compute the equipment weight using the same routines used to compute the weight of the original design. In this manner we compare weights computed by the same commercial software package, which is also used to verify the thermal duty, for equipments designed by two different software packages.

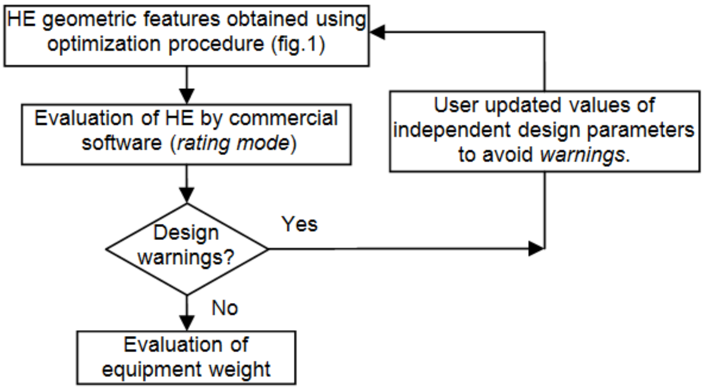


Figure 3. Optimized HE weight evaluation procedure

This sort of cross-check is also useful to verify that design *warnings* are not issued by the commercial software when rating the exchanger designed by the other method. In case a *warning* appears then the operator makes some slight changes to the equipment architecture (usually minor changes of baffles spacing or tubes length is enough) until the *warning* disappears. In this way we are certain that alternative designs, passing all checks about correctness of constructive mechanical details, are compared in a consistent manner on the basis of weight.

5. HEAT EXCHANGERS DESIGN COMPARISON

As previously mentioned, in order to test the effectiveness of alternative tools for heat exchanger design, four representative reference exchangers designed by the engineering department of an Italian contractor and installed in chemical plants worldwide were chosen. In the following each case study is examined separately while complete constructive details of all case studies equipment are reported in a Table included in Appendix A.

5.1 Soda-Water heat exchanger

This is a split ring floating head type (TEMA classification AES) exchanging heat between a 20% soda solution in water and a cooling water stream. It is a small sized equipment having a duty of 413 kW. Thermal specification and streams properties are detailed in Table 1. In this case the customer did not specify any design characteristic.

Table 1. Soda-Water heat exchanger design specification.

Hot stream (soda 20%)		**Cold stream (water)**	
Inlet temperature [°C]	55.0	Inlet temperature [°C]	34.0
Outlet temperature [°C]	45.0	Outlet temperature [°C]	44.0
Mass flow rate [kg/s]	11.2	Mass flow rate [kg/s]	9.88
Heat duty [kW]	413.0		

A sketch of the built heat exchanger and of the proposed one is shown in Figure 4.

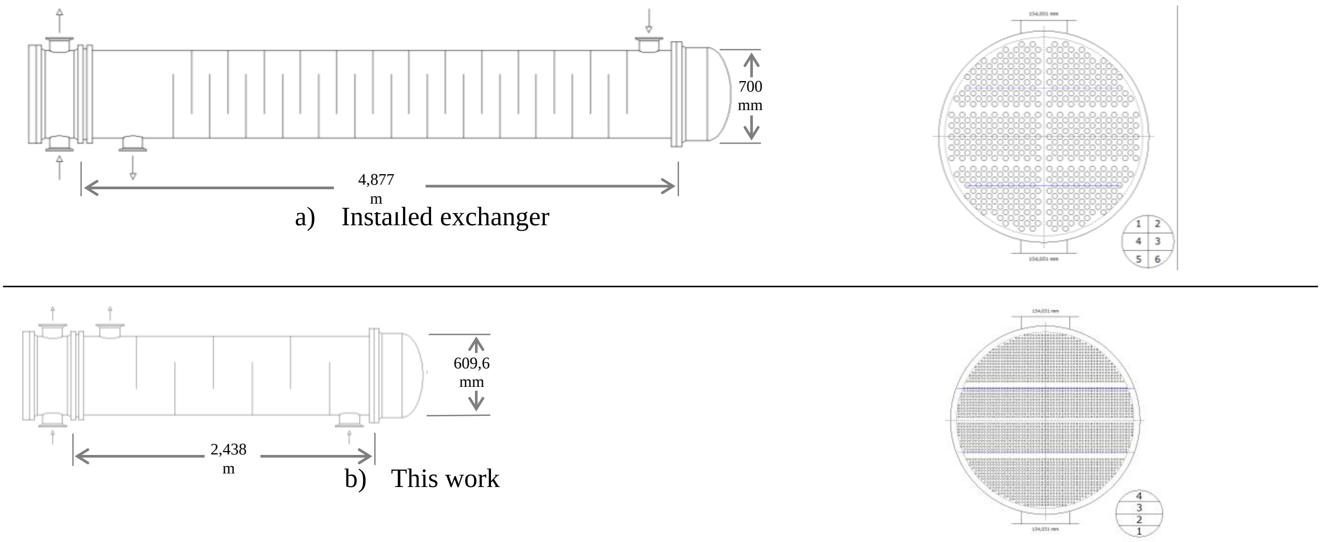


Figure 4. Original solution (a) vs optimized (b) heat exchanger

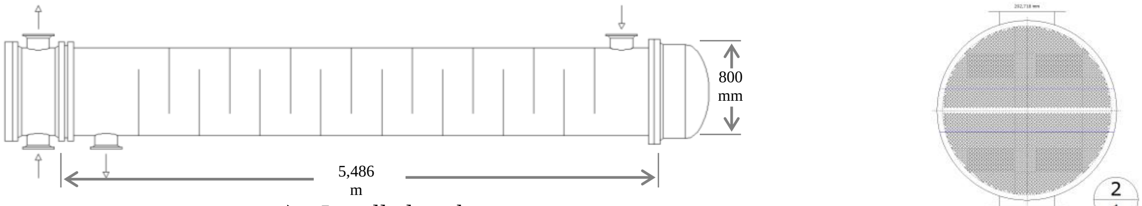
According to the proposed design a reduction of weight from 5,287 kg to 2,697 kg (-48,9%) has been obtained. This is due mainly to the use of smaller tubes diameter allowing the insertion of a greater number of tubes within a shell similar to the original. As a consequence, the equipment became shorter and the proposed heat exchanger is less slender respect to the commercial solution. However the shell diameter-to-length ratio remains in the usual 3 to 15 range. This is the only case study in which the research software chooses fluid paths different from those chosen by the designer of the original solution. The overdesign of installed solution is of about 27% whereas the proposed solution has an overdesign of 36%. From an energy consumption point of view it must be pointed out that the installed solution has pressure drops of 5.5 and 27.6 kPa for shell and tube-side, whereas the proposed solution shows pressure drops of 1.7 kPa shell-side and 28 kPa tube-side, meaning that the proposed solution presents an energy saving, and consequently a reduced operating cost, respect to the installed solution.

5.2 KHO-Water heat exchanger

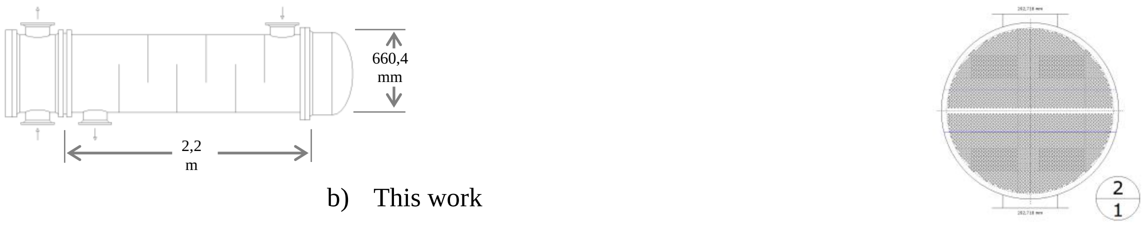
This is a small power (868 kW) split ring floating head type (TEMA classification AES) exchanging heat between a 49% solution of KHO (potassium hydroxide) and a cooling water stream. Thermal specification and streams properties are detailed in Table 2. Figure 5 shows the obtained alternative configurations.

Table 2. Solution of KHO-Water design specification.

Hot stream (KHO 49%)		**Cold stream (water)**	
Inlet temperature [°C]	65.0	Inlet temperature [°C]	34.0
Outlet temperature [°C]	46.0	Outlet temperature [°C]	44.0
Mass flow rate [kg/s]	17.8	Mass flow rate [kg/s]	20.8
Heat duty [kW]	868.0		



a) Installed exchanger



b) This work

Figure 5. Original solution (a) vs optimized (b) heat exchanger

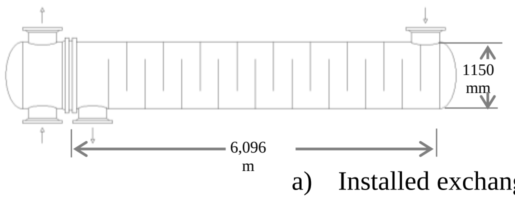
The design proposed in this work results less heavy than the realized one, with a reduction of weight from 6,762 kg to 2,842 kg (58% reduction). Similarly to the previous case study, the lighter configuration is reached using a greater number of smaller tubes respect to the original configuration. Moreover, the tube passes is reduced from the original 6 to 2. The proposed configuration is quite "squat" being at the lower limit of the usual slenderness range, with precisely 3.2 shell length-to-diameter ratio. The overdesign of both installed and proposed solutions is the same (20%). In this case the shell-side pressure drop is greater in the proposed solution (8 kPa) respect to the installed equipment (5 kPa). However, pressure drop is within the allowed limit and, moreover, the tube-side pressure drop in the proposed solution is drastically reduced (7 kPa) being one seventh of pressure drop in industrial solution (48.4 kPa).

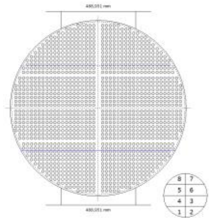
5.3 HKGO-Water heat exchanger.

This medium size unit (2,535 kW) is an U-Tube heat exchanger (TEMA classification BEU) exchanging heat between cooling water and a HKGO (heavy cocker gas oil) fluid stream. Thermal specification and streams properties are detailed in Table 3. In this case the customer imposed the following additional specifications: minimum tube diameter of 19.05 mm, tube thickness of 2.108 mm (14 BWG), cold fluid must flow tube side whereas the layout angle is 90 (square pattern). The above constraint s were forced to the research software tool. Figure 6 shows the main heat exchanger characteristics for original and optimized equipment.

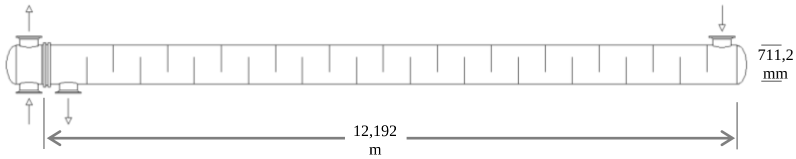
Table 3. HKGO-Water heat exchanger design specification.

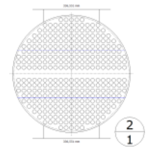
Hot stream (HKGO)		**Cold stream (water)**	
Inlet temperature [°C]	186.0	Inlet temperature [°C]	60.0
Outlet temperature [°C]	90.0	Outlet temperature [°C]	80.0
Mass flow rate [kg/s]	11.94	Mass flow rate [kg/s]	30.32
Heat duty [kW]	2,535.0		





a) Installed exchanger





b) This work

Figure 6. Original solution (a) vs optimized (b) heat exchanger

In this case the weight reduction is about 36% (the original weight of 15,618 kg has been reduced to 9,999 kg). The minimum dimension imposed to tubes diameter prevents a weight reduction using smaller tubes, so the optimization algorithm suggests a slender design for the entire equipment with a reduced tube passes number. Note that the proposed design also assures a significant pressure drop reduction tube-side. The overdesign of installed solution is 56% (very high) whereas the proposed solution has an overdesign of 24%. The pressure drop shell side is 3.1 kPa for the original solution and 4.5 kPa for the optimized solution, but tube-side pressure drop of the installed solution is much higher (63.1 kPa) than in the proposed solution (5.6 kPa).

5.4 Lean Flexsorb SE-Water heat exchanger.

This is a medium size heat exchanger (2,120 kW) split ring floating head equipment. Thermal specifications and streams properties are detailed in Table 4. In this case too the customer imposed additional specifications: minimum tube diameter of 19.05 mm, tube thickness of 2.77 mm, cold fluid flowing tube-side, and 90° layout angle (square pattern). Figure 7 shows the main heat exchanger characteristics for original and optimized equipment.

Table 4. Heavy organics-Water design specification.

Hot stream (Lean Flexsorb SE)		**Cold stream (water)**	
Inlet temperature [°C]	50.0	Inlet temperature [°C]	32.0
Outlet temperature [°C]	45.0	Outlet temperature [°C]	41.0
Mass flow rate [kg/s]	111.79	Mass flow rate [kg/s]	56.34
Heat duty [kW]	2,120.0		

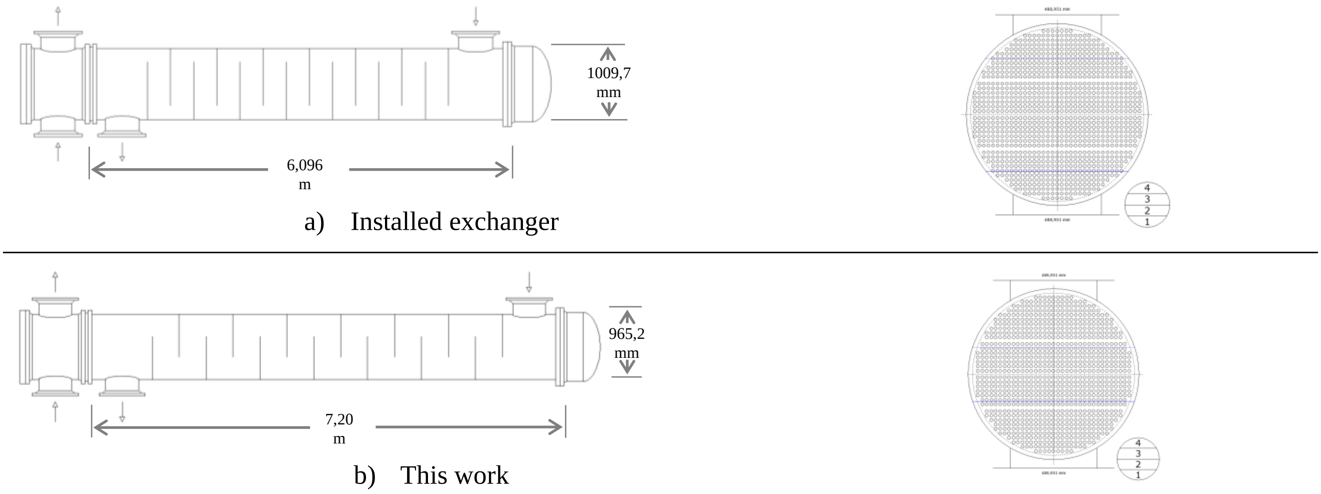


Figure 7. Original solution (a) vs optimized (b) heat exchanger

The design proposed in this paper has a weight of 13,029 kg versus the 14,037 kg of the originally built solution and is much more slender. The 7% weight reduction is much lower than that obtained in previous case studies owing to the customer imposed constraints which reduce optimization possibilities. The overdesign of installed solution is about 11% whereas the proposed solution has an overdesign of 15%. In this case pressure drops are quite similar, being tube-side 75 kPa (installed solution) versus 60.9 kPa (proposed solution), whereas shell-side is 60.7 kPa for installed solution and 53.4 kPa in the proposed solution.

Overall, the obtained weight savings obtained resorting to the optimization procedure are compared in Figure 8.

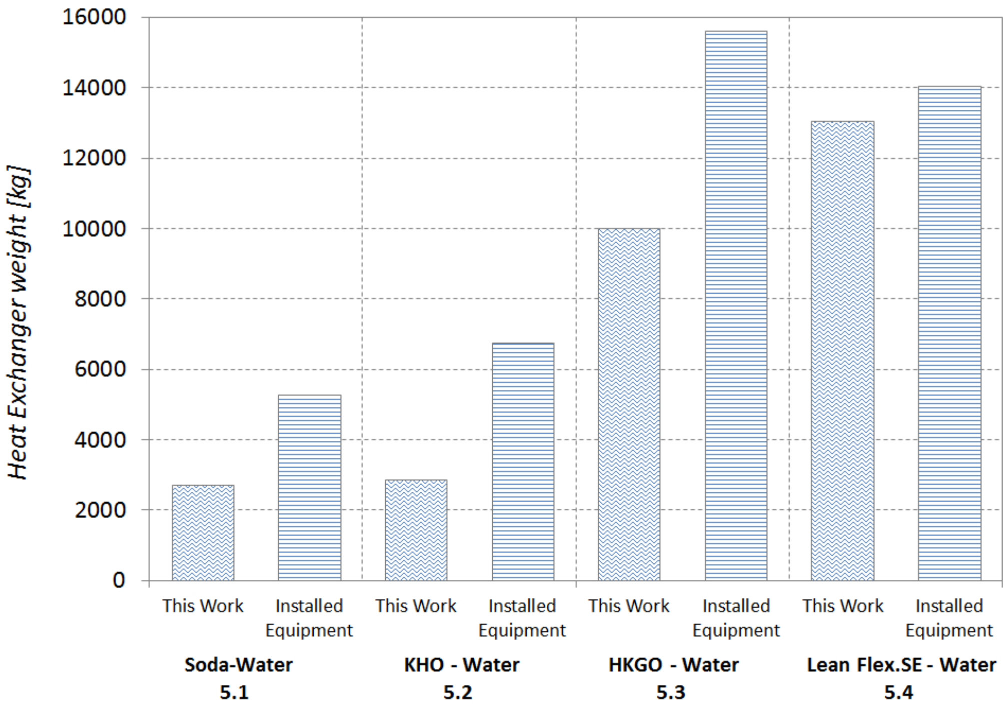


Figure 8. Comparison of weight savings.

In order to better comprehend obtained results, a few comments may be useful. At first it should be pointed out that the optimization routine does not attempt to reduce the equipment weight by reducing the heat transfer surface area through an increase of U, but rather by pursuing radically different equipment architectures. This is a feature allowed by the capability of better exploring the design space. In fact, values of U, exception made for Case # 3, are roughly similar. Similarity of U values between reference operational heat exchangers and the corresponding optimized solutions derives from the fact that the fouling factors act as a limiting heat transfer coefficient. Nevertheless, by comparing values of heat transfer coefficient and surface area, as reported in Table 5, one notes that there is not a strict correlation between U and S as would be expected. This apparent inconsistency is perfectly reasonable and can be justified as follows.

Table 5. Comparison of U and S values.

	U (installed exchanger) W/m^2 K	U (optimized exchanger) W/m^2 K	U variation	S (installed exchanger), m^2	S (optimized exchanger), m^2	S variation	NOTES
Case 1	435.7	493	11.6%	120.2	127.5	5.7%	Both U and S increase
Case 2	409.7	445	7.9%	186.4	177.5	-5.0%	U increases and S decreases less than linearly
Case 3	132.7	191.3	30.6%	487.4	314.3	-55.1%	U increases and S decreases more than linearly
Case 4	616.3	599.8	-2.8%	383.1	401.3	4.5%	U decreases and S increases more than linearly

While from the theoretical point of view the minimum heat transfer area is inversely proportional to U, according to Eq. (2), the actual heat transfer surface area S of the equipment is constrained by constructive requirements, the only condition to be met in the final design is that $S \geq S_{min}$. The possible increase of actual surface area irrespective of changes in U can be understood by following the logical design steps described in Section 2. At first the design routine determines tubes number and fluid velocities based on the assumed values of independent design variables (i.e. shell and tubes diameters, etc.), then heat transfer coefficients tube-side and shell-side are computed, thus determining the available heat transfer area per unit shell length. The value of overall heat transfer coefficient U is then computed, allowing to determine the minimum heat transfer area. This allows to compute the minimum tubes length and the number of baffles according to the specified baffles spacing. However, for constructive reasons the baffles number has to be rounded up to the next integer and the required tubes length has to be updated to account for the increased baffles number and the inlet and outlet baffles spacing. Then this increased tubes length is further rounded to the next commercial tubes length. This procedure may determine an increase of the actual heat transfer area even if a slight increase of U respect the reference exchanger was obtained. Therefore, a reduction in equipment size is not always related to an augmentation of overall heat transfer coefficient. Moreover, equations defining structural parameters are often non linear, so that a proportionality between U and S does not hold strictly. This is what happens in Case #1, while in other cases the expected inverse variation of S respect U is observed, although not in a linear manner owing to the above stated reasons.

No further structural constraints were requested by customers and applied to the design process, apart from those already indicated. However, to avoid excessive values of flow velocities, which could penalize operational cost, in the optimization routine the previously cited upper pressure drop limit of 70 kPa was included, according to accepted good design practice. Nevertheless, this constraint was never enforced during the optimization process as the optimizer was always able to find better architectures which did not require an excessive increase of fluid velocity. The fact that both lighter exchangers and lower pressure drop values were obtained may be considered a "coincidence" but this is the result of better exploring the design space, which is usually not allowed by human designers who often get stuck into consolidated habits and preferred design configurations.

6. CONCLUSIONS

In this paper heat exchanger configurations obtained resorting to a research software package allowing optimized design have been compared to configurations obtained resorting to commercial heat exchanger design packages commonly used by engineering departments of industrial contractors participating to process plant construction projects. Four different case studies were investigated considering equipment of different sizes compared on the basis of overall weight. Even if the shell and tube heat exchangers design procedure is established, and a number of commercial software tools are available on the market, the final design is strongly dependent on the designer's experience. In all examined cases the numerical optimization procedure outperformed solutions obtained by commercial design software. This is a consequence of the capability of exploring a much wider set of alternative configuration by a genetic algorithm, and eliminating the subjective role played by the designer on the basis of his own experience. Obtained results, even if non generalizable, show that a lower weight was reached thanks to a reduction of shell diameter and length, as well as tube diameter. Weight reductions between 8% and 58% were obtained over a wide range of equipment sizes. A significant weight reduction respect original design was also attained when constraints imposed by customer restricted the design space by reducing the number of optimization variables.
The proposed design perform better even from an operating cost point of view, given that pressure drop is always lower (at least quite similar) than that of commercial solutions. The obtained weight reduction of the proposed solutions is high enough to ensure that a significant performance improvement is maintained even if the obtained solution if slightly modified for instance to increase the mechanical robustness of the exchanger (i.e. using higher tube thickness for instance).

This experiment, although limited to a small set of case studies, contributes to the validation of the proposed heat exchanger design optimization algorithm, and shows that heat exchangers design tools developed for research purposes may provide better solutions than widespread commercial software packages irrespective of the ability of the operator using them. This may help to push research in this field and reduce the skepticism of industry designers towards tools developed in academic setting.

7. REFERENCES

Amini, M., Bazargan, M., 2014. Two objective optimization in shell-and-tube heat exchangers using genetic algorithm, *Applied Thermal Engineering*, vol. 69, n. 1-2, pp. 278-285.

Asadi, M., Song. Y., Sunden, B., Xie, G., 2014. Economic optimization design of shell-and-tube heat exchangers by a cuckoo-search-algorithm, *Applied Thermal Engineering*, 73 , pp. 1032–1040.

Azad, A.V., Amidpour, M., 2011. Economic optimization of shell and tube heat exchanger based on constructal theory, *Energy*, vol. 36, n. 2, pp. 1087-1096.

Babu, B.V., Munawar, S.A., 2007. Differential evolution strategies for optimum design of shell and tube heat exchangers, C*hemical Engineering Science*, Vol. 62, pp. 3720-3739.

Bell, K.J., 2000. On the pessimization of heat exchangers, *Heat Transfer Engineering*, Vol. 21, pp. 1-2.

Caputo, A.C., Pelagagge, M.P., Salini, P., 2008. Heat exchangers design based on economic optimization, *Applied Thermal Engineering*, 28 , pp. 1151–1159.

Caputo, A.C., Pelagagge, P.M., Salini, P., 2009. A manufacturing-based cost model for shell and tube heat exchangers. *Proceedings of the 7th International Conference on Manufacturing Research (ICMR09)*, University of Warwick (UK), September 8-10, pp. 134-142.

Caputo, A.C., Pelagagge, P.M., Salini, P., 2010. Heat exchanger design optimization under stochastic operating conditions. *Proc. 13th Brazilian Congress of Thermal Sciences and Engineering ENCIT 2010*, Uberlandia, Minas Gerais, Brasil, December 5-10, Paper ENC10-0098. Available at the url http://www.abcm.org.br/anais/encit/2010/PDF/ENC10-0098.pdf

Caputo, A.C., Pelagagge, P.M., Salini, P., 2011. Joint economic optimization of heat exchanger design and maintenance policy, *Applied Thermal Engineering*, Vol. 31, pp. 1381-1392.

Caputo, A.C., Pelagagge, P.M., Salini, P., 2012. Robust approach for shell and tubes exchangers optimization under uncertain heat transfer estimation. *Proc. 14th Brazilian Congress on Thermal Sciences and Engineering ENCIT 2012*, Paper ENCIT2012-0107, Rio de Janeiro, Brazil, October 18-22.

Costa, A.L.H., Queiroz, E.M., 2008. Design optimization of shell-and-tube heat exchangers, *Applied Thermal Engineering*, vol. 28, n. 14-15,pp. 1798-1805.

Fesanghary, M., Damangir, E., Soleimani, I., 2009. Design optimization of shell and tube heat exchangers using global sensitivity analysis and harmony search algorithm, *Applied Thermal Engineering*, vol. 29, n. 5-6, pp. 1026-1031.

Fettaka, S.,Thibault, J., Gupta, Y., 2013. Design of shell-and-tube heat exchangers using multiobjective optimization, *International Journal of Heat and Mass Transfer*, vol. 60, n. 1, pp. 343-354.

Guo, J., Xu, M., Cheng, L., 2009a. The application of field synergy number in shell-and-tube heat exchanger optimization design, *Applied Energy*, vol. 86, n. 10, pp. 2079-2087.

Guo, J., Cheng, L., Xu, M., 2009b. Optimization design of shell-and-tube heat exchanger by entropy generation minimization and genetic algorithm, *Applied Thermal Engineering*, vol. 29, n. 14-15, pp. 2954-2960.

Hadidi, A., Hadidi, M., Nazari, A., 2013. A new design approach for shell-and-tube heat exchangers using imperialist competitive algorithm (ICA) from economic point of view, *Energy Conversion and Management*, vol. 67 ,pp. 66-74.

Hadidi, A., Nazari, A., 2013. Design and economic optimization of shell-and-tube heat exchangers using biogeography-based (BBO) algorithm, *Applied Thermal Engineering*, vol. 51, n. 1-2, pp. 1263-1272.

Hewitt, G.F. (Ed.), 1998. Heat Exchanger Design Handbook, Begell House, New York, USA.

Kacak, S., Liu, H., Pramuanjaroenkij, A., 2012. Heat Exchangers. Selection, Rating and Thermal Design, CRC Press, Boca Raton, Florida, USA.

Kern, D.Q., 1950. Process Heat Transfer, McGraw-Hill. New York, USA.

Lahiri, S.K., Khalfe, N., Wadhwa, S.K., 2012. Particle swarm optimization technique for the optimal design of shell and tube heat exchangers, *Chemical Product and Process Modeling*, vol. 7, n. 1, Article 14.

Lahiri, S.K.,Khalfe, N., 2014. Improve shell and tube heat exchangers design by hybrid differential evolution and ant colony optimization technique, *Asia-Pacific Journal of Chemical Engineering*, vol. 9, n. 3, pp. 431-448.

Mariani, V.C., Duck, A.R.K., Guerra, F.A., Coelho, L.D.S.,Rao, R.V., 2012. A chaotic quantum-behaved particle swarm approach applied to optimization of heat exchangers, *Applied Thermal Engineering*, vol. 42, pp. 119-128.

Mukherjee, R., 1998. Effectively design shell-and-tube heat exchangers, *Chemical Engineering Progress*, Vol. 94, No. 2, pp. 21-37.

Onishi, V.C., Ravagnani, M.A.S.S., Caballero, J.A., 2013. Mathematical programming model for heat exchanger design through optimization of partial objectives, *Energy Conversion and Management*, vol. 74, pp. 60-69.

Patel, V.K., Rao, R.V., 2010. Design optimization of shell-and-tube heat exchanger using particle swarm optimization technique, *Applied Thermal Engineering*, vol. 30, n. 11-12, pp. 1417-1425.

Ponce-Ortega, J.M., Serna-Gonzalez, M., Jimenez-Gutierrez, A., 2009. Use of genetic algorithm for the optimal design of shell-and-tube heat exchangers, *Applied Thermal Engineering*, Vol. 29, pp. 203-209.

Rao, R.V., Patel, V., 2013. Multi-objective optimization of heat exchangers using a modified teaching learning-based optimization algorithm, *Applied Mathematical Modelling*, vol. 37, n. 3, pp. 1147-1162.

Ravagnani, M.A.S.S., Caballero, J.A., 2007. A MINLP model for the rigorous design of shell and tube heat exchangers using the TEMA standards, *Chemical Engineering Research and Design*, vol. 85, n. 10 A, pp. 1423-1435.

Ravagnani, M.A.S.S., Silva, A.P., Biscaia Jr., E.C., Caballero, J.A., 2009. Optimal design of shell-and-tube heat exchangers using particle swarm optimization, *Industrial and Engineering Chemistry Research*, vol. 48, n. 6 ,pp. 2927-2935.

Sanaye, S., Hajabdollahi, H., 2010. Multi-objective optimization of shell and tube heat exchangers, *Applied Thermal Engineering*, vol. 30, n. 14-15, pp. 1937-1945.

Şahin, S. A., Kiliç, B., Kiliç, U., 2011. Design and economic optimization of shell and tube heat exchangers using Artificial Bee Colony (ABC) algorithm, *Energy Conversion and Management*, vol. 52, n. 11, pp. 3356-3362.

Serna, M., Jiménez, A., 2005. A compact formulation of the Bell-Delaware method for heat exchanger design and optimization, *Chemical Engineering Research and Design*, vol. 83, n. 5 A ,pp. 539-550.

Tayal, M.C., Fu,Y., Diwekar, U.M., 1999. Optimal design of heat exchangers: A genetic algorithm framework, *Industrial and Engineering Chemistry Research*, Vol. 38, pp. 456-467.

Yang, J., Fan, A., Liu, W., Jacobi, A.M., 2014. Optimization of shell-and-tube heat exchangers conforming to TEMA standards with designs motivated by constructal theory, *Energy Conversion and Management*, Vol. 78, pp. 468-476.

Appendix A. Heat exchangers constructive details

		Soda-Water		KOH Solution -Water		HKGO-Water		Lean Flexsorb SE-Water	
		This work	Installed equipment	This work	Installed equipment	This work	Installed equipment	This work	Installed equipment
TubeSide	[-]	cold fluid	hot fluid	cold fluid	cold fluid	cold fluid	cold fluid	cold fluid	cold fluid
D_s	[mm]	609.6	700	660.4	800	711.2	1,150	965.2	1,016
B_{cut}	[%]	33	27	38	25	30	26	34	19
L_{bc}	[mm]	300	145	250	275.5	497	317.5	400	330
L_{bi}	[mm]	445	394.1	447	402.1	546	317.5	959	874.2
L_{bo}	[mm]	445	550.26	447	503.69	713	603.1	959	728
L_{tp}Ratio	[-]	1.46	1.33	1.32	1.33	1.26	1.33	1.32	1.33
D_t	[mm]	6.35	19.05	6.35	19.05	25.4	19.05	19.05	19.05
θ	[deg]	90	45	45	45	90	90	90	90
N_{tp}	[-]	4	6	2	6	2	8	2	4
N_{tt}	[-]	2,674	426	4,044	586	318	1,336	942	1,050
L_{tt}	[m]	2.44	4.88	2.2	5.49	12.19	6.10	7.20	6.00
v_s	[m/s]	0.14	0.16	0.16	0.09	0.19	0.09	1.06	0.78
v_t	[m/s]	0.80	0.67	0.54	1.1	0.55	1.1	1.38	1.49
h_s	[W/m^{2}K]	2,259	2,738.3	2,208	1,107.4	237.2	148.5	3,107.9	3,322.3
h_t	[W/m^{2}K]	4,210	2,509.9	2,488	5,737.7	3,762	6,861.9	6,818.4	7,465.9
$R_{f,t}$	[m^{2}K/W]	0.000289	0.000289	0.000290	0.000290	0.000170	0.000170	0.000516	0.000516
$R_{f,s}$	[m^{2}K/W]	0.000858	0.000858	0.000850	0.000850	0.000350	0.000350	0.000350	0.000350
U_{actual}	[W/m^{2}K]	493	435.7	445	409.7	191.3	132.7	599.8	616.3
S	[m2]	127.5	120.2	177.5	186.4	314.3	487.4	401.3	383.1
L/D	[-]	4	6.7	3.3	6.6	18.7	5.3	7.5	5.9
Overdesign	[%]	37	26	20	32	24	55.9	15	11
Δp_S	[kPa]	1.7	5.5	8	5	4.5	3.1	53.4	60.7
Δp_T	[kPa]	28	27.6	7	48.8	5.6	63.1	60.9	75
Weight	[kg]	2,697	5,287	2,842	6,762	9,999	15,618	13,029	14,037

Nomenclature

TubeSide	*[-]*	*Fluid passing tube-side*
B_{cut}	*[%]*	*Baffle cut*
D_s	*[mm]*	*Shell inside diameter*
D_t	*[mm]*	*Outer tube diameter*
D_{ti}	*[mm]*	*Internal tube diameter*
F	*[-]*	*Temperature difference correction factor*
h_S	*[W/m²K]*	*Shell side convective heat transfer coefficient*
h_T	*[W/m²K]*	*Tube side convective heat transfer coefficient*
L/D	*[-]*	*Length to diameter ratio*
L_{bc}	*[mm]*	*Central baffle spacing*
L_{bi}	*[mm]*	*Inlet baffle spacing*
L_{bo}	*[mm]*	*Outlet baffle spacing*
L_{tp}Ratio	*[-]*	*Pitch ratio*
L_{tt}	*[m]*	*Total tube length*
m_h	*[kg/s]*	*Mass flow rate of hot fluid*
m_c	*[kg/s]*	*Mass flow rate of cold fluid*
N_{ss}	*[-]*	*Numbero of sealing strips*
N_{tp}	*[-]*	*Tube passes number*
N_{tt}	*[-]*	*Total tube number*
Overdesign	*[%]*	*Surface overdesign percentage*
Path	*[-]*	*Variable to assign a side to each fluid (shell/tube side)*
Q	*[W]*	*Heat duty*
$R_{f,s}$	*[m²K/W]*	*Fouling factor shell-side*
$R_{f,t}$	*[m²K/W]*	*Fouling factor tube-side*
S	*[m²]*	*Heat exchanger surface area*
t_{hi}	*[K]*	*Inlet hot fluid temperature*
t_{ho}	*[K]*	*Outlet hot fluid temperature*
t_{ci}	*[K]*	*Inlet cold fluid temperature*
t_{co}	*[K]*	*Outlet cold fluid temperature*
U_{actual}	*[W/m²K]*	*Ovarall heat transfer coefficient*
v_s	*[m/s]*	*Shell side flow velocity*
v_t	*[m/s]*	*Tube side flow velocity*
Weight	*[kg]*	*Equipment dry weight*
θ	*[deg]*	*Pitch pattern*
Δp_S	*[Pa]*	*Pressure drop shell side*
Δp_T	*[Pa]*	*Pressure drop tube side*
ΔT_{ML}	*[K]*	*Mean logarithmic temperature difference*
λ	*[W/m K]*	*Thermal conductivity of tubes walls*