

# Heat exchanger design based on economic optimisation

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## Abstract

Owing to the wide utilization of heat exchangers in industrial processes, their cost minimization is an important target for both designers and users. Traditional design approaches are based on iterative procedures which gradually change design parameters until a satisfying solution, which meets the design specifications, is reached. However, such methods, besides being time consuming, do not guarantee the reach of an economically optimal solution. In this paper a procedure for optimal design of shell and tube heat exchangers is proposed, which utilizes a genetic algorithm to minimize the total cost of the equipment including capital investment and the sum of discounted annual energy expenditures related to pumping. In order to verify the capability of the proposed method, three case studies are also presented showing that significant cost reductions are feasible with respect to traditionally designed exchangers. In particular, in the examined cases a reduction of total costs up to more than 50% was observed.

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**Keywords:** Heat exchangers; Economic optimisation; Genetic algorithm

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## 1. Introduction

The design of a heat exchanger is an iterative process relying heavily on the designer's experience. Usually a reference geometric configuration of the equipment is chosen at first and an allowable pressure drop value is fixed. Then, the values of the design variables are defined based on the design specifications and the assumption of several mechanical and thermodynamic parameters in order to have a satisfactory heat transfer coefficient leading to a suitable utilization of the heat exchange surface. The designer's choices are then verified based on iterative procedures involving many trials until a reasonable design is obtained which meets design specifications with a satisfying compromise between pressure drops and thermal exchange performances [1–4].

Although well proven, this kind of approach is time consuming and may not lead to cost-effective designs as no

cost criteria are explicitly accounted for. Considering the functional importance and widespread utilization of heat exchangers in process plants, their minimum cost design is thus an important goal. In particular, the minimization of energy related expenses is critical in the optic of energy savings and resources conservation.

In the literature, attempts to automate and optimise the heat exchanger design process have been proposed for 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 optimisation method employed.

As far as the objective function is considered, most authors consider the sum of capital investment related to the heat transfer area, and energy related costs connected to overcoming friction losses in the fluid flow (pumping losses). However, some authors consider only pumping costs [5] or capital investment [6], while others assume entropy generation as the objective function to be minimized [7–10]. Even the ratio of performances to cost was analysed [11].

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## Nomenclature

$a_s$	shellside pass area ( $\text{m}^2$ )	$P_t$	tube pitch (m)
$a_1$	numerical constant (€)	$Q$	heat duty (W)
$a_2$	numerical constant ( $\text{€}/\text{m}^2$ )	$Re_s$	Reynolds number (shellside)
$a_3$	numerical constant	$Re_t$	Reynolds number (tubeside)
$B$	baffles spacing (m)	$R_{\text{foul},\text{shell}}$	conductive fouling resistance shellside ( $\text{m}^2 \text{K}/\text{W}$ )
$Cl$	clearance (m)	$R_{\text{foul},\text{tube}}$	conductive fouling resistance tubeside ( $\text{m}^2 \text{K}/\text{W}$ )
$C_i$	capital investment (€)	$S$	heat exchange surface area ( $\text{m}^2$ )
$C_E$	energy cost ( $\text{€}/\text{kW h}$ )	$T_{is}$	inlet fluid temperature shellside (K)
$C_o$	annual operating cost (€/yr)	$T_{os}$	outlet fluid temperature shellside (K)
$C_{oD}$	total discounted operating cost (€)	$T_{it}$	inlet fluid temperature tubeside (K)
$C_{\text{tot}}$	total annual cost (€)	$T_{ot}$	outlet fluid temperature shellside (K)
$d_o$	tube outside diameter (m)	$U$	overall heat transfer coefficient ( $\text{W}/\text{m}^2 \text{K}$ )
$d_i$	tube inside diameter (m)	$v_t$	fluid velocity tubeside (m/s)
$D_e$	equivalent shell diameter (m)	$v_s$	fluid velocity shellside (m/s)
$D_s$	shell inside diameter (m)		
$F$	temperature difference corrective factor		
$f_t$	Darcy friction factor tube side		
$f_s$	friction factor shellside		
$h_s$	convective coefficient shellside ( $\text{W}/\text{m}^2 \text{K}$ )		
$h_t$	convective coefficient tubeside ( $\text{W}/\text{m}^2 \text{K}$ )		
$H$	annual operating time (h/yr)		
$i$	annual discount rate (%)		
$K_1$	numerical constant		
$L$	tubes length (m)		
$m_s$	shellside mass flow rate ( $\text{kg}/\text{s}$ )		
$m_t$	tubeside mass flow rate ( $\text{kg}/\text{s}$ )		
$n$	number of tubes passages		
$n_1$	numerical constant		
$ny$	equipment life (yr)		
$N_t$	tubes number		
$P$	pumping power (W)		
$Pr_s$	Prandtl number (shellside)		
$Pr_t$	Prandtl number (tubeside)		

### Greek symbols

$\Delta h$	$(h_s - h_t)$ ( $\text{W}/\text{m}^2 \text{K}$ )
$\Delta P_s$	shellside pressure drop (Pa)
$\Delta P_t$	tubeside pressure drop (Pa)
$\Delta P_{\text{tube length}}$	tube length pressure drop (Pa)
$\Delta P_{\text{tube elbow}}$	tube elbows pressure drop (Pa)
$\Delta T_{ML}$	mean logarithmic temperature difference ( $^{\circ}\text{C}$ )
$\lambda$	thermal conductivity ( $\text{W}/\text{m K}$ )
$\eta$	overall pumping efficiency
$\mu_t$	viscosity at tube wall temperature ( $\text{Pa s}$ )
$\mu_w$	viscosity at core flow temperature ( $\text{Pa s}$ )
$\pi$	numerical constant
$\rho_t$	fluid density tubeside ( $\text{kg}/\text{m}^3$ )
$\rho_s$	fluid density shellside ( $\text{kg}/\text{m}^3$ )
$\phi_t$	$\mu/\mu_w$ tubeside
$\phi_s$	$\mu/\mu_w$ shellside

Passing to the optimisation variables, most works aim towards the simultaneous selection of several design parameters, while some authors focus on the effects of changing a single parameter. As an example of the latter case, Saffar-Aval and Damangir [12] developed general correlations for determining only optimum baffle spacing. However, parametric analysis methods were often adopted [6,13–16], involving the effects of multiple parameters. With reference to the employed numerical optimisation method, Lagrange multipliers were widely adopted [11,17,18]. Fontein and Wassink [19] adopted the simplex method, while Palen et al. [20] utilized the so-called Complex method. Afimiwala [21] instead utilized various non-linear programming methods, while geometric programming was also proposed [22,23]. Passing on to newer optimisation techniques, Chauduri et al. [24] also adopted simulated annealing including vibration constraints. In more recent times genetic algorithms (GA) have demonstrated to be an effective approach utilized by several researchers [10,25–28].

In order to contribute to a solution of the optimised heat exchanger design problem, in this paper a methodological approach aimed at defining an optimal heat exchanger design through the minimization of total heat exchange-related costs is proposed resorting to evolutionary computation techniques based on GA. The proposed method, starting from the user defined specifications, enables to directly define the heat exchanger configuration and sizing details by concurrently determining the values of the optimal design parameters that are able to meet the specification at the minimum total discounted cost. In the paper, after briefly describing the exchanger design procedure as well as the choice of optimisation variables, a proper cost function is selected and the optimisation algorithm is described. Some application examples are also presented in order to assess the capabilities of the method and a parametric analysis is carried out to evaluate the sensitivity to relevant cost parameters and changes in the objective function.

## 2. Proposed approach

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 specifications assuming a set of design variables values;
- evaluation of the capital investment, operating cost, and the objective function;
- utilization of the optimisation algorithm to select a new set of values for the design variables;
- iteration of the previous steps until a minimum of the objective function is found.

The entire process is schematised in Fig. 1. At present, the procedure excludes heat transfer with phase change.

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, as well as the inlet and outlet temperatures of the fluids shellside  $T_{is}$ ,  $T_{os}$ , and tubeside,  $T_{it}$ ,  $T_{ot}$ , the remaining parameter being determined by an energy balance.

Fixed parameters assigned by the user are the tubesheet patterns (triangular or square) and pitch, the number of tubeside passages (1, 2, 4, ...), the fouling resistances  $R_{foul,shell}$  and  $R_{foul,tube}$ , and the thermophysical properties of both fluids.

The optimisation variables, with values assigned iteratively by the optimisation algorithm, are the shell inside diameter  $D_s$ , tube outside diameter  $d_o$ , and baffles spacing  $B$ .

Based on the actual values of the design specifications and the fixed parameters, and on the current values of the optimisation variables, the exchanger design routine determines the values of the shellside and tubeside heat exchange coefficients  $h_s$ ,  $h_t$ , the overall heat exchange area  $S$ , the number of tubes  $N_t$ , the shell and tube length  $L$  and the tubeside and shellside flow velocities  $v_s$  and  $v_t$ , thus defining all constructive details of the exchanger satisfying the assigned thermal duty specifications. The computed values of flow velocities and the constructive details of the exchanger structure are then used to evaluate the objective function. The optimisation algorithm, based on the

value of the objective function, updates the trial values of the optimisation variables which are then passed on to the design routine to define a new architecture of the heat exchanger. The process is iterated until a minimum of the objective function is found or a prescribed convergence criterion is met.

### 2.1. Heat exchanger design procedure

The adopted design procedure is based on computing the heat exchange surface area through the mean logarithm temperature difference approach

$$S = \frac{Q}{U \Delta T_{MLF}}, \quad (1)$$

$F$  being the temperature difference corrective factor according to TEMA rules for shell and tube exchangers functionally to the equipment architecture [1].

The heat transfer coefficient is computed through the following equations [1,29]. Fouling resistances are assigned from the literature data based on fluid type and operating temperature

$$U = \frac{1}{\frac{1}{h_s} + R_{foul,shell} + \frac{d_o}{d_i} \cdot \left( R_{foul,tube} + \frac{1}{h_t} \right)}, \quad (2)$$

$$d_i = 0.8d_o. \quad (3)$$

The tubeside heat transfer coefficient  $h_t$  is computed, according to the flow regime, resorting to the following correlations:

$$h_t = \frac{\lambda}{d_i} \cdot \left[ 3.657 + \frac{0.0677 \cdot (Re_t \cdot Pr_t \cdot \frac{d}{L})^{1.33}}{1 + 0.1 \cdot Pr_t \cdot (Re_t \cdot \frac{d_i}{L})^{0.3}} \right] \quad (4)$$

$$h_t = \frac{\lambda}{d_i} \cdot \left\{ \frac{\frac{f_t}{8} \cdot (Re_t - 1000) \cdot Pr_t}{1 + 12.7 \cdot \sqrt{\frac{f_t}{8} \cdot (Pr_t^{0.67} - 1)}} \cdot \left[ 1 + \left( \frac{d_i}{L} \right)^{0.67} \right] \right\} \quad (5)$$

$$(2.300 < Re_t < 10.000; Gnielinski [30]), \quad (5)$$

$$h_t = \frac{\lambda}{d_o} \cdot 0.027 \cdot Re_t^{0.8} \cdot Pr_t^{1/3} \cdot \left( \frac{\mu_t}{\mu_w} \right)^{0.14} \quad (6)$$

$$(Re_t > 10.000, Sieder and Tate [1]), \quad (6)$$

where  $f_t$  is the Darcy friction factor [4] given as

$$f_t = (1.82 \log_{10} Re_t - 1.64)^{-2}. \quad (7)$$

The shellside heat transfer coefficient  $h_s$  is instead computed resorting to Kern's formulation referring to segmental baffled shell and tube exchangers [1]

$$h_s = \frac{\lambda}{D_e} \cdot 0.36 \cdot Re_s^{0.55} \cdot Pr_s^{1/3} \cdot \left( \frac{\mu_t}{\mu_w} \right)^{0.14}, \quad (8)$$

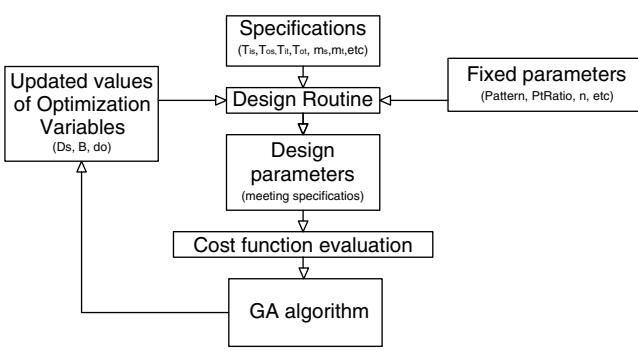


Fig. 1. Proposed optimisation algorithm.

$D_e$  being the shell hydraulic diameter computed as [1,29]

$$D_e = \frac{4 \cdot (\text{Pt}^2 - \pi d_o^2/4)}{\pi d_o}; \quad D_e = \frac{4 \cdot (0.43 \cdot \text{Pt}^2 - 0.5 \cdot \pi \cdot d_o^2)}{0.5 \cdot \pi \cdot d_o} \quad (9)$$

for square and triangular patterns, respectively, and also utilized for calculating Reynolds number.

To evaluate Reynolds numbers the flow velocities are computed as follows [1]:

$$v_t = \frac{m_t}{\frac{\pi d_o^2}{4} \cdot \rho_t} \cdot \frac{n}{N_t}, \quad (10)$$

$$v_s = \frac{m_s}{a_s \cdot \rho_s}; \quad a_s = \frac{D_s \cdot B \cdot \text{Cl}}{\text{Pt}}; \quad \text{Cl} = \text{Pt} - d_o. \quad (11)$$

The number of tubes is then computed from

$$N_t = K_1 \cdot \left( \frac{D_s}{d_o} \right)^{n_1}, \quad (12)$$

where constants  $K_1$  and  $n_1$  are defined according to the number of passes and tubes arrangement as described in detail in [29].

Knowing the total heat exchange surface  $S$  from Eq. (1), the necessary tube length  $L$  follows as

$$L = \frac{S}{\pi \cdot d_o \cdot N_t}. \quad (13)$$

Therefore, starting from the design specifications, from the user-defined fixed parameters, and from trial values of the optimisation variables  $D_s$ ,  $d_o$ ,  $B$ , at first the number of tubes  $N_t$  and the shell hydraulic diameter  $D_e$  are computed resorting to Eqs. (12) and (9), respectively. The fluid velocities are then evaluated through Eqs. (10) and (11). The shellside and tubeside heat transfer coefficients are then computed from Eqs. (4)–(8) as well as the overall coefficient  $U$  from Eq. (2). The required heat exchange area  $S$  follows from Eq. (1). Finally, the tubes length  $L$  is defined through Eq. (13).

The procedure is repeated computing new values of  $S$  and  $L$ , and a corresponding exchanger architecture meeting specifications, each time the optimisation algorithm changes the values of the design variables  $D_s$ ,  $d_o$ ,  $B$  in the attempt of minimizing the objective function.

## 2.2. Objective function calculation

The objective function has been assumed as the total present cost  $C_{\text{tot}}$ ,

$$C_{\text{tot}} = C_i + C_{\text{oD}}. \quad (14)$$

The capital investment  $C_i$  is computed as a function of the exchanger surface adopting Hall's correlation [31]

$$C_i = a_1 + a_2 S^{a_3}, \quad (15)$$

where  $a_1 = 8000$ ,  $a_2 = 259.2$  and  $a_3 = 0.91$  for exchangers made with stainless steel for both shells and tubes.

The total discounted operating cost related to pumping power to overcome friction losses is instead computed from the following equations:

$$C_{\text{oD}} = \sum_{k=1}^{\text{ny}} \frac{C_o}{(1+i)^k}, \quad (16)$$

$$C_o = P \cdot C_E \cdot H, \quad (17)$$

$$P = \frac{1}{\eta} \left( \frac{m_t}{\rho_t} \cdot \Delta P_t + \frac{m_s}{\rho_s} \cdot \Delta P_s \right). \quad (18)$$

The tubeside pressure drop is computed as the sum of distributed pressure drop along the tubes length and concentrated pressure losses in elbows and in the inlet and outlet nozzles [1]

$$\Delta P_t = \Delta P_{\text{tube length}} + \Delta P_{\text{tube elbow}} \\ = \frac{\rho \cdot v_t^2}{2} \cdot \left( \frac{L}{d_i} f_t + p \right) \cdot n. \quad (19)$$

Different values for constant  $p$  are given in the literature. Sinnott [29] assumes  $p = 2.5$ , while Kern [1] assumes  $p = 4$ . When referring to these two authors, in the case studies section, the proper value of  $p$  will be utilized.

The shellside pressure drop is instead

$$\Delta P_s = \frac{\rho_s \cdot v_s^2}{2} \cdot f_s \cdot \frac{L}{B} \cdot \frac{D_s}{D_e}, \quad (20)$$

where  $f_s = 2 \cdot b_0 \cdot Re_s^{-0.15}$  is the friction factor (with  $b_0 = 0.72$  from Peters and Timmerhaus [32], valid for  $Re < 40,000$ ).

## 3. Genetic algorithm implementation

The optimisation procedure was implemented resorting to a genetic algorithm (GA) [33]. Starting from an initial population of randomly created individuals representing candidate solutions, in this case a heat exchanger of specific configuration and meeting the design specifications, the GA applies the principle of survival of the fittest to produce better performing individuals in subsequent evolutionary generations of the examined population. The total cost of each candidate solution represents the fitness function of the individual which is a measure of its quality with respect to the entire population. According to GA theory, each new generation is composed of

1. The best individual(s) copied from the previous generation (the so called Elite Count).
2. New child individuals obtained by crossover recombination of the genes from a pair of selected parents of the current generation.
3. Mutant individuals.
4. Migrant individuals from past generations.

The described design optimisation procedure was implemented on a personal computer resorting to the Genetic Algorithm toolbox of the scientific computing environment MATLAB [34]. Following an experimental campaign, the following setting parameters for the GA were chosen.

Each generation was made of 20 individuals as the added computational time required to analyse larger popu-

lations was not offset by the smaller number of generations required to attain convergence considering that always the same best individual was obtained.

The maximum number of generations was set at 100. However, in the tests convergence was always obtained within about 50 generations. The number of best performing individuals of a generation which are transferred to the next one (Elite count) was set at 2.

The adopted “Crossover Fraction” parameter, i.e. the percentage of individuals of each generation, excluding the Elite Count individuals, which are generated through a crossover recombination of selected individuals of the previous generation was 0.5. The selection algorithm used for picking the parent individuals was the *selectionroulette* in which parents are picked with a probability proportional to their fitness function. The crossover method utilized is the so-called *crossoverscattered*, where a random binary vector is created having a number of bits equal to the number of genes of an individual. Then, the genes where the value is 1 are copied from the first parent, while the genes where the value is 0 are copied from the second parent. The obtained genes are then combined to form the child.

The percentage of individuals who undergo a mutation derives instead from the previously defined parameters in that all individuals who are not bred and are not part of the elite count are subject to mutation.

Finally, migration of individuals from previous generations is allowed each third generation. The number of transferred individuals is (*PopulationSize* – *EliteCount*) \* *MigrationFraction* where the migration fraction was set at 0.5. Those migrant individuals substitute the worst individuals of the current generation.

## 4. Results

### 4.1. Analysis of case studies

The effectiveness of the described approach was assessed by analysing some relevant case studies taken from the literature, in order to have reliable reference sizing data for the sake of comparison. The following three different test cases, representative of a wide range of possible applications, were considered:

- Case #1: methanol – brackish water exchanger, duty 4.34 (MW).
- Case #2: kerosene – crude oil exchanger, duty 1.44 (MW).
- Case #3: distilled water – raw water exchanger, duty 0.46 (MW).

For each case the original design specifications, shown in Table 1, were supplied as input to the optimisation algorithm and the resulting optimal exchangers architectures given by the GA method were compared with the original design solution given by the referenced author (Table 2). Costs of both the original and the GA solutions were computed as shown in Section 2.2 in order to allow a consistent comparison.

The following upper and lower bounds for the optimisation variables were imposed: shell internal diameter  $D_s$  ranging between 0.1 m and 1.5 m; tubes outside diameter  $d_o$  ranging from 0.015 m (5/8 in.) to 0.051 m (2 in.); baffles spacing  $B$  ranging from 0.05 m to 0.5 m.

All values of discounted operating costs were computed with  $n_y = 10$  yr, annual discount rate  $i = 10\%$ , energy cost  $C_E = 0.12 \text{ €/kW h}$  and an annual amount of work hours  $H = 7000 \text{ h/yr}$ .

*Case 1: methanol–brackish water exchanger.* This case was taken from Sinnott [29]. The original design assumed an exchanger with two tubeside passages and one shellside passage. The same architecture was retained in the GA approach.

A slight reduction of heat exchange area resulted thanks to a reduction of the exchanger length, even if the number of tubes increased significantly and the tubes diameter was decreased. The capital investment decreased correspondingly (−4.4%). However, the higher number of tubes and the shorter shell enabled to reduce both the shellside and tubeside flow velocity leading to a marked decrease of pressure losses. Therefore, the annual pumping cost decreased markedly (−55.14%). Overall, the combined reduction of capital investment and operating costs led to a reduction of the total cost of about 14.5%.

*Case 2: kerosene–crude oil exchanger.* This case study was taken from Kern [1]. The original design assumed an exchanger with four tubeside passages (with square pitch

Table 1  
Case studies specifications

	Mass flow (kg/s)	T input (°C)	T output (°C)	$\rho$ (kg/m <sup>3</sup> )	Cp (kJ/kg K)	$\mu$ (Pa s)	$\lambda$ (W/m K)	$R_{fouling}$ (m <sup>2</sup> K/W)
<i>Case 1</i>								
Shellside: methanol	27.80	95.0	40.0	750	2.84	0.00034	0.19	0.00033
Tubeside: sea water	68.90	25.0	40.0	995	4.20	0.00080	0.59	0.00020
<i>Case 2</i>								
Shellside: kerosene	5.52	199.0	93.3	850	2.47	0.00040	0.13	0.00061
Tubeside: crude oil	18.80	37.8	76.7	995	2.05	0.00358	0.13	0.00061
<i>Case 3</i>								
Shellside: distilled water	22.07	33.9	29.4	995	4.18	0.00080	0.62	0.00017
Tubeside: raw water	35.31	23.9	26.7	999	4.18	0.00092	0.62	0.00017

Table 2

A comparison of the heat exchanger design

	Case study #1		Case study #2		Case study #3	
	Literature values	This work	Literature values	This work	Literature values	This work
$D_s$ (m)	0.894	0.830	0.539	0.630	0.387	0.620
$L$ (m)	4.830	3.379	4.880	2.153	4.880	1.548
$B$ (m)	0.356	0.500	0.127	0.120	0.305	0.440
$d_o$ (m)	0.020	0.016	0.025	0.020	0.019	0.016
$P_t$ (m)	0.025	0.020	0.031	0.025	0.023	0.020
$C_l$ (m)	0.005	0.004	0.006	0.005	0.004	0.004
$N_t$	918	1567	158	391	160	803
$v_t$ (m/s)	0.75	0.69	1.44	0.87	1.76	0.68
$Re_t$	14,925	10,936	8227	4068	36,400	9487
$Pr_t$	5.7	5.7	55.2	55.2	6.2	6.2
$h_t$ (W/m <sup>2</sup> K)	3812	3762	619	1168	6558	6043
$f_t$	0.028	0.031	0.033	0.041	0.023	0.031
$\Delta P_t$ (Pa)	6251	4298	49,245	14,009	62,812	3673
$a_s$ (m <sup>2</sup> )	0.0320	0.0831	0.0137	0.0148	0.0236	0.0541
$D_e$ (m)	0.014	0.011	0.025	0.019	0.013	0.015
$v_s$ (m/s)	0.58	0.44	0.47	0.43	0.94	0.41
$Re_s$	18,381	11,075	25,281	18,327	16,200	8039
$Pr_s$	5.1	5.1	7.5	7.5	5.4	5.4
$h_s$ (W/m <sup>2</sup> K)	1573	1740	920	1034	5735	3476
$f_s$	0.330	0.357	0.315	0.331	0.337	0.374
$\Delta P_s$ (Pa)	35,789	13,267	24,909	15,717	67,684	4365
$U$ (W/m <sup>2</sup> K)	615	660	317	376	1471	1121
$S$ (m <sup>2</sup> )	278.6	262.8	61.5	52.9	46.6	62.5
$C_i$ (€)	51,507	49,259	19,007	17,599	16,549	19,163
$C_o$ (€/yr)	2111	947	1304	440	4466	272
$C_{oD}$ (€)	12,973	5818	8012	2704	27,440	1671
$C_{tot}$ (€)	64,480	55,077	27,020	20,303	43,989	20,834

pattern) and one shellside passage. The same architecture was retained in the GA approach.

In this case too a moderate decrease of the heat exchange area was observed (about 13%). This was the combined result of an increase of shell diameter coupled with a strong increase of the number of tubes, paired with a significant decrease of both diameter and length of the tubes. The capital investment therefore decreased by 7.4%. However, the marked reduction of flow velocities enabled a saving of 66.2% in the annual operating expenses, leading to a net reduction of the total cost which decreased by about 24.8% with respect to the original solution.

*Case 3: distilled water–raw water exchanger.* This case study was taken from Kern [1]. The exchanger structure is characterized by two tubesides passages (triangular pitch pattern) and one shellside passage.

In the latter case, a marked increase of about 34% of the heat exchange area was observed with a corresponding increase of capital investment (+15.8%). This was caused by a significant increase of shell diameter as well as the number of tubes, not compensated by the strong reduction of the length of the tubes and by a minor decrease of the diameter of the tubes. Conversely, a very high reduction of flow velocities and pressure drops allowed to drastically cut about 93.9% of annual operating costs. This saving wholly offset the higher capital investment allowing a marked reduction of about

52.6% of total costs. Overall if we consider the differential investment comparing the optimised exchanger with the original design, we get an increase of capital investment of 2614 € and a decrease of annual operating costs of 4194 €. Therefore, the payback period required to offset the incremental investment through annual savings is 0.62 years.

An overall cost comparison for the three case studies is provided in Fig. 2.

For the sake of completeness, trials were also made by parametrically changing electricity cost in the total cost function in order to assess the sensitivity of the GA to variation in the economic parameters. The effect of ±50% variation of electricity price with respect to the nominal value was examined. The results shown in Table 3 describe that the GA responds correctly by trying to reduce pressure losses when electricity price increases at the expense of an increased exchanger surface area, and making the opposite when  $C_E$  decreases. In particular, when  $C_E$  increased by 50%  $C_i$  increased by 4.1% and  $C_{oD}$  by 22.8%, but the total cost increased by only 6%. When instead  $C_E$  decreased by 50%  $C_i$  decreased by 5% and  $C_{oD}$  by 12%, while  $C_{tot}$  lowered by 5.8%.

Finally, it was also attempted to minimize the difference between the shellside and tubeside heat transfer coefficient by assuming the following objective function:

$$\Delta h = h_s - h_t, \quad (21)$$

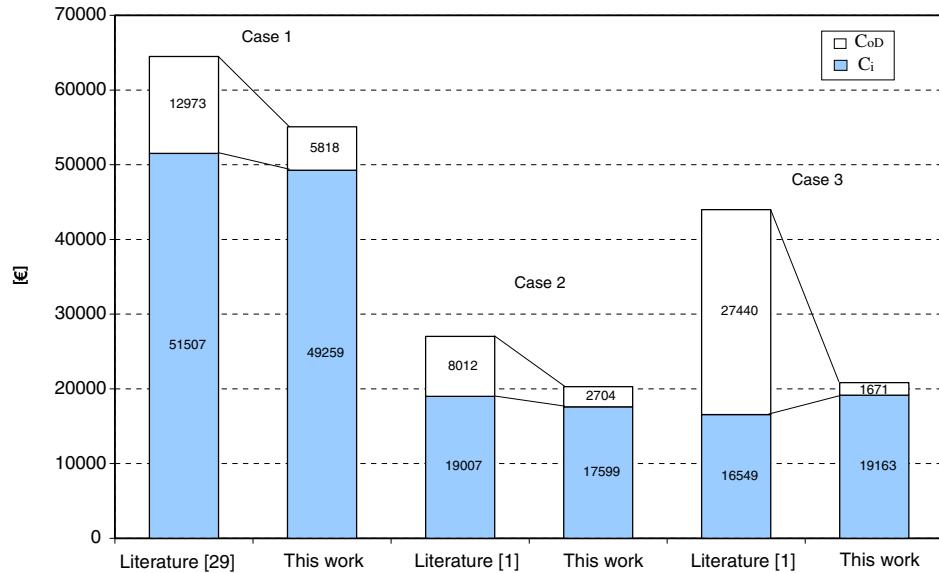


Fig. 2. Overall costs comparison.

**Table 3**  
Sensitivity to the electricity price (reference exchanger is case study #1)

	C <sub>E</sub> : 0.06 €/kW h	C <sub>E</sub> : 0.12 €/kW h	C <sub>E</sub> : 0.18 €/kW h
D <sub>s</sub> (m)	0.750	0.830	0.900
L (m)	3.980	3.379	3.021
B (m)	0.460	0.500	0.500
d <sub>o</sub> (m)	0.016	0.016	0.016
P <sub>t</sub> (m)	0.020	0.020	0.020
C <sub>l</sub> (m)	0.004	0.004	0.004
N <sub>t</sub>	1227	1567	1824
v <sub>t</sub> (m/s)	0.88	0.69	0.59
R <sub>e<sub>t</sub></sub>	13,963	10,936	9394
P <sub>r<sub>t</sub></sub>	5.7	5.7	5.7
h <sub>t</sub> (W/m <sup>2</sup> K)	4574	3762	3331
f <sub>t</sub>	0.028	0.031	0.032
ΔP <sub>t</sub> (Pa)	8741	4298	3483
a <sub>s</sub> (m <sup>2</sup> )	0.0696	0.0831	0.0901
D <sub>e</sub> (m)	0.011	0.011	0.011
v <sub>s</sub> (m/s)	0.53	0.44	0.41
R <sub>e<sub>s</sub></sub>	13,378	11,075	10,347
P <sub>r<sub>s</sub></sub>	5.1	5.1	5.1
h <sub>s</sub> (W/m <sup>2</sup> K)	1906	1740	1655
f <sub>s</sub>	0.347	0.357	0.361
ΔP <sub>s</sub> (Pa)	20,920	13,267	10,936
U (W/m <sup>2</sup> K)	707	660	626
S (m <sup>2</sup> )	245.50	262.80	277.00
C <sub>i</sub> (€)	46,776	49,259	51,280
C <sub>o</sub> (€/yr)	828	947	1163
C <sub>oD</sub> (€)	5088	5818	7148
C <sub>tot</sub> (€)	51,864	55,077	58,427

in order to avoid having a controlling film dictating the exchanger architecture.

The results of some trial runs of the GA with this modified objective function are shown in Table 4. The GA proved to be quite sensitive to the objective function as expected. However, taking  $\Delta h$  as an objective function proved to be counterproductive from the economic stand-

**Table 4**  
Results for a different objective function (OBJ) (reference exchanger is case study #1)

Literature values	OBJ = $\Delta h$		
	Run 1	Run 2	Run 3
D <sub>s</sub> (m)	0.894	1.710	1.400
L (m)	4.830	5.219	1.489
B (m)	0.356	0.250	0.390
d <sub>o</sub> (m)	0.020	0.051	0.016
P <sub>t</sub> (m)	0.025	0.063	0.02
C <sub>l</sub> (m)	0.005	0.013	0.004
N <sub>t</sub>	918	589	4935
v <sub>t</sub> (m/s)	0.75	0.18	0.22
R <sub>e<sub>t</sub></sub>	14925	9089	3458
P <sub>r<sub>t</sub></sub>	5.7	5.7	5.7
h <sub>t</sub> (W/m <sup>2</sup> K)	3812	1014	1491
f <sub>t</sub>	0.028	0.032	0.042
ΔP <sub>t</sub> (Pa)	6251	186	302
a <sub>s</sub> (m <sup>2</sup> )	0.0320	0.0856	0.1096
D <sub>e</sub> (m)	0.014	0.036	0.011
v <sub>s</sub> (m/s)	0.58	0.43	0.33
R <sub>e<sub>s</sub></sub>	18,381	34,381	8429
P <sub>r<sub>s</sub></sub>	5.1	5.1	5.1
h <sub>s</sub> (W/m <sup>2</sup> K)	1573	1014	1491
f <sub>s</sub>	0.330	0.301	0.372
ΔP <sub>s</sub> (Pa)	35,789	20,840	7529
U (W/m <sup>2</sup> K)	615	355	474
S (m <sup>2</sup> )	278.60	488.50	366.20
C <sub>i</sub> (€)	51,507	80,522	63,792
C <sub>o</sub> (€/yr)	2111	942	360
C <sub>oD</sub> (€)	12,973	5787	2210
C <sub>tot</sub> (€)	64,480	86,309	66,002
Δh (W/m <sup>2</sup> K)	0	0.19	0

point. The GA was able to make a negligible difference between the two heat transfer coefficients but at the expense of a much higher heat transfer surface area with respect to the original design. The additional capital

investment was not completely offset by the reduced pumping losses and a net increase of the total cost followed.

#### 4.2. Discussion

In all the examined cases, the operating costs were drastically cut and significant percent total cost reductions were obtained with respect to the original design. Even if the capital investment increased in one case, this was fully offset by the reduction in operating costs. The variation of capital investment ranged between  $-7.4\%$  and  $+15.8\%$  while a percent decrease of operating costs from  $-55.1\%$  to  $-93.9\%$  was obtained, leading to a total cost saving between  $-14.5\%$  and  $-52.6\%$ , thus confirming the effectiveness of the proposed approach. These kinds of results indicate that in general an optimisation of heat exchanger costs should be searched by acting on the pressure drops, through a proper choice of fluids velocity, rather than acting on the surface area, while a minimization of heat exchanger surface area in general does not represent a cost-effective strategy. In fact, the capital investment is dictated by the heat exchange area which is strictly linked to the overall heat transfer coefficient which, in turn, is scarcely sensible to variation in flow velocity (it depends approximately from  $Re^{0.55}$  shellside) and can be hardly improved by changing the design parameters. On the contrary, pumping losses are highly responsive to changes in flow velocity, being dependent on  $Re^{1.8}$ , thus leaving more space for a design optimisation. Moreover, besides cost minimization, the reduction of pumping losses supports efforts in resource conservation and energy saving which are a main concern of plant owners.

As a final remark it should be noted that the majority of the literature approaches to heat exchanger optimisation are based on the assumption of a simplified cost correlation as the one adopted in this study, owing to the difficulty of a detailed characterization of the manufacturing process. As a future work, instead, it is planned to analyse in greater detail the effect of the heat exchanger architecture and manufacturing process on the equipment cost in order to make the cost estimation more realistic. In fact, the length/diameter ratio of the GA-optimised design was always smaller than that of the original exchanger. While this might reflect the effects of a better balance between surface area and pressure loss, it is also the effect of choosing a capital cost correlation which is the function of the sole surface area of the exchanger instead of the exchanger weight. However, when the exchanger operates at high pressures an attempt is made to lower the shell diameter in order to reduce its thickness to avoid an excess weight of the vessel which translates in added cost. The preference of designers towards higher length/diameter ratios follows. Furthermore, it should be noted that the GA-optimised design generally leads to an exchanger architecture markedly different from that of the original design (for example, with a higher number of smaller length tubes). Thus, even by maintaining the same heat transfer area, different man-

ufacturing costs might result. This circumstance too is neglected by the adopted capital cost correlation. When the adopted cost function does not include such issues related to detailed mechanical design and manufacturing process, these aspects might be overlooked.

#### 5. Conclusions

In this paper, a solution method of the shell and tube heat exchanger design optimisation problem was proposed based on the utilization of a genetic algorithm. Referring to the literature test cases, reduction of capital investment up to  $7.4\%$  and savings in operating costs up to  $93\%$  were obtained, with an overall decrease of total cost up to  $52\%$ , showing the improvement potential of the proposed method. Furthermore, the genetic algorithm allows for rapid solution of the design problem and enables to examine a number of alternative solutions of good quality, giving the designer more degrees of freedom in the final choice with respect to traditional methods. As a future work, it is intended to deal in detail with issues of mechanical design and the equipment manufacturing process to significantly improve the estimation capability of the capital investment and obtain a more realistic heat exchanger optimisation procedure.

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