HEAT EXCHANGER DESIGN BASED ON ECONOMIC OPTIMIZATION.

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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 assume a configuration and gradually change design parameters until a satisfying solution is reached which meets the design specifications. However, such methods, besides being time consuming, do not guarantee the reach of an optimal solution. In this paper a procedure for optimal design for shell and tube heat exchangers is proposed which utilizes a genetic algorithm to minimize the total discounted cost of the equipment including the capital investment and pumping related annual energy expenditures. In order to verify the performances of the proposed method four case studies are also presented showing that total cost reductions greater than 15% are feasible respect traditionally designed exchangers.

Keywords. *Heat exchangers, economic optimization, genetic algorithm*

1.Introduction

When designing a heat exchanger usually a reference configuration of the equipment is chosen relying on experience or expert advice, and values of the design variables are defined based on the design specifications and the assumption of several mechanical and thermodynamic parameters. This implies a verification of the designer's choices and is typically based on iterative procedures involving many trials in order to finally meet design specifications with a reasonable design. At the end of this process a compromise solution between pressure drops and thermal exchange performances will result. In fact, the traditional heat exchanger design approach is to choose a geometric configuration, fix an allowable pressure drop value and choose the sizing parameters in order to have a satisfactory heat exchange coefficient leading to a suitable utilization of the heat exchange surface meeting the heat duty.

However, this kind of approach despite being well proven has some significant drawbacks. Firstly it is rather time consuming while manually performing the computation iterations is quite tedious, although several software tools helping in applying the above cited procedure are available on the market. Furthermore, it may be not effective as the obtained solution gives no guarantee of proper economic performances because no cost criteria are accounted for. Considering the functional importance and widespread utilization of heat exchangers in process plants, the minimum cost design of this kind of equipment is thus an important objective for designers. In this respect, being the investment cost dictated by the heat transfer area, which is constrained by the allowed overall heat transfer coefficient, it follows that marginal cost reduction opportunities exist as far as the investment cost is concerned. In fact, the convective heat transfer coefficient is scarcely sensible to variation in flow velocity (it depends approximately from $Re^{0.6}$) and can be hardly improved by changing the design parameters. On the contrary, cost minimization may be sought through a reduction of pressure drop which impacts on operating expenses. In fact, pumping losses are highly responsive to changes in flow velocity being dependent on $Re^{1.8}$. This is quite relevant as 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 from a long time, and the problem is the subject of ongoing research because better performing solution methods are still being searched. The proposed approaches vary in the choice of the objective function, in the adopted exchanger design procedure, in the number and kind of sizing parameters utilized and the numerical optimisation method employed.

In order to contribute to a solution to this problem, in this paper a methodological approach aimed at defining an optimal heat exchanger design through a minimization of total heat exchange-related costs is proposed.

In the paper after a critical review of the past approaches to this problem an optimal design method is discussed by choosing a proper cost function, a robust exchanger design procedure, a comprehensive set of design variables, and an effective optimisation algorithm resorting to evolutionary computation techniques based on genetic algorithms. Genetic algorithms, in fact, proved to be very effective in solving combinatorial optimisation problems.

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 able to meet the specification at the minimum total discounted cost. In the paper following a detailed description of the proposed method and the implemented solution algorithm, some application examples are also presented in order to assess the capabilities of the method.

2. Literature review

The problem of heat exchanger design optimization has attracted the attention of researcher from several decades. Therefore, a vast amount of literature has accumulated. The proposed approaches mainly differ in the following aspects:

- the kind of heat exchanger structure considered,
- the adopted exchanger design procedure,
- the number and kind of sizing and optimisation parameters utilized,
- the choice of the objective function,
- and the numerical optimisation method employed.

With reference to the configuration of heat exchange equipment most research works focus on shell-and-tube exchangers. However, alternative configurations have also been examined such as air cooled cross flow exchangers (Hedderich et al., 1982), and plate heat exchangers (Wang and Sunden, 2003). Bignardi et al. (1990) instead considered exchangers operating with organic fluids.

As far as the heat exchanger sizing procedures are concerned one both finds in the literature methods based on the logarithmic mean temperature difference approach as well as the NTU-efficiency method (Eryener, 2006). However, adopted sizing methods consist usually in simplified procedures based on reference textbooks approaches (Kern, 1950, Rosenhow and Hartnett, 1973, Shah and Bell, 2000, Hewitt, 1998) rather than detailed computer based methods currently utilized in industry. Passing to the optimisation variables considered, most works aim towards the simultaneous selection of several design parameters values at once, while some authors focus on the effects of choosing a single parameter. As an example of this latter case Saffar-Avval and Damangir (1995) determine general correlations for determining optimum baffle spacing. Their work is expanded also by Khalifeh Soltan et al. (2004) and Eryener (2005) through interesting parametric analyses. 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 pumping losses. However some authors consider only pumping costs (Mott et al., 1972), capital investment (Ramananda Rao et al., 1991), while others assume entropy generation as the objective function to be minimized (Bejan et al., 1995; Johannessen et al., 2002, Sun et al., 1993). Even the ratio of performances to cost has been analysed (Kovarik, 1989). With reference to the employed numerical optimisation method, a large number of techniques have been proposed starting from Lagrangian multipliers or gradient-based techniques and arriving to artificial-intelligence based methods.

However, many of the adopted methods are often affected by some limitations and none is completely general. They may require a high number of iterations or may get stuck in local minima. Often the adopted numerical method is deeply connected and affected by boundary conditions, and only allows a limited number of optimisation variables to be acted upon. Furthermore, a large number of fixed parameters values to be assumed limits the degree of freedom in pursuing optimal design solutions thus making rather stiff and constrained the overall design process.

In particular, Fax and Mills used Lagrange multipliers (Fax and Mills, 1957) as well as Unuvar and Kargici (2004) and Kovaric (1989). Palen et al. (1974) utilized the so called Complex method which asks for several feasible initial designs to be available before starting the optimisation and use six geometrical parameters. Fontein and Wassink (1978) adopted the simplex method, while Afimiwala (1976) utilized various non linear programming methods. Geometric programming was proposed by Paul (1982) and Radhakrishnan et al. (1980). Buzek and Podkanski (1996) examined the problem of optimizing series of heat exchangers. Parametric analysis methods were instead utilized by Jenssen (1969), Ramananda Rao et al. (1991), Eryener (2006), Poddar and Polley (2000). Passing to newer optimization techniques, Chauduri et al. (1997) adopted simulated annealing also including vibration constraints. Genetic algorithms have demonstrated to be an effective approach utilized by several researchers in recent times (Babu and Mohiddin, 1999; Babu and Munawar, 2000; Selbas et al., 2006; Tayal et al., 1999).

3. Proposed approach

In this work a computational procedure for shell-and-tube heat exchangers design optimisation is presented based on the minimization of the total discounted cost of owning and operating the heat exchanger including capital investment and pumping losses. The proposed solution procedure utilizes a genetic algorithm for minimizing the objective function. At present the procedure excludes heat transfer with phase change.

The procedure 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 capital investment, operating cost, and the objective function;
- 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 the mass flow rates of the two fluids, as well as the inlet and outlet temperatures of the fluids shell side T_{is}, T_{os}, and tubeside, T_{it}, T_{ot}.

Fixed parameters assigned by the user are the tubesheet patters (triangular or square), the number of tubeside passages (1, 2, 4…), and the thermophysical properties of both fluids.

The optimisation variables, with values assigned iteratively by the optimisation algorithm, are the following; shell diameter D_s, tube diameter d_0, baffles spacing B, tubes length L, tubes pitch P_t.

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 shell side and tubeside heat exchange coefficients h_{shell}, h_{tube}, the overall heat transfer coefficient U and the exchanger area S, the number of tubes N_t, 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 pressure losses and 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 to the design routine to define a new constructive solution of the heat exchanger. The process is iterated until a minimum of the objective function is found or a prescribed convergence criterion is met.

In the following, the heat exchanger design procedure is detailed as well the structure of the objective function. In the next section details are instead provided about the genetic algorithm optimisation routine. Finally some application examples are presented in order to assess the capabilities of the proposed method.

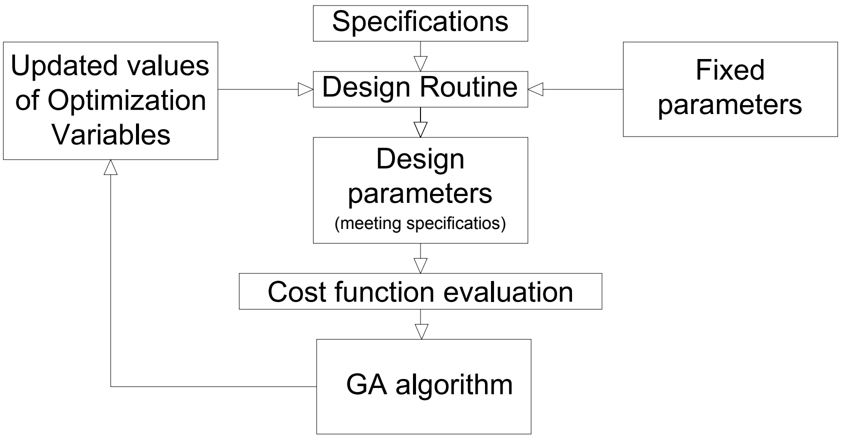

Figure 1: Proposed optimisation algorithm

3.1 Heat exchanger design procedure

The adopted design procedure is based on the mean logarithm temperature difference approach.

According to this method the heat exchange surface area is computed resorting to the following equation

$$S = \frac{Q}{U \cdot \Delta T \cdot F} \tag{1}$$

being Q the required heat duty, F the temperature difference corrective factor which is architecture-dependent and available from TEMA rules for the specified exchanger configuration, U the overall heat-transfer coefficient and ΔT the mean logarithm temperature difference resulting from specifications. The heat transfer coefficient is computed as a function of the shell side and tube side convective coefficients including the fouling resistances R_{foul} and neglecting the tube wall thermal resistance.

$$U = \frac{1}{\frac{1}{h_{shell}} + R_{foul,shell} + R_{foul,tube} + \frac{1}{h_{tube}}} \tag{2}$$

where the tubeside heat transfer coefficient h_{tube} is computed assuming the well known Sieder and Tate turbulent internal flow correlation (valid for Re>10000),

$$h_t = \frac{\lambda}{d_0} \cdot 0.027 \cdot \mathrm{Re}^{0.8} \cdot \mathrm{Pr}^{\frac{1}{3}} \cdot \left(\frac{\mu_t}{\mu_w}\right)^{0.14} \tag{3}$$

being λ the walls thermal conductivity, *Re* the Reynolds number, *Pr* the Prandtl number, d_0 is the tube diameter, μ_t and μ_w the fluid viscosity computed at the bulk and wall temperatures. The shell side heat transfer coefficient h_{shell} is instead computed resorting to Kern's formulation related to segmental baffled shell and tube exchangers.

$$h_s = \frac{\lambda}{D_e} \cdot 0.36 \cdot \mathrm{Re}^{0.55} \cdot \mathrm{Pr}^{\frac{1}{3}} \cdot \left(\frac{\mu_t}{\mu_w}\right)^{0.14} \tag{4}$$

being D_e the shell hydraulic diameter computed as

$$D_e = \frac{4 \cdot \left(Pt^2 - \pi d_0^{\,2} / 4\right)}{\pi d_0} \tag{5}$$

and utilized also for calculating Reynolds number.
The values of fouling coefficients are assigned from literature data based on fluid type and operating temperature.

The number of tubes is then computed from

$$N_t = C \cdot \left(\frac{D_s - 0.02}{d_0}\right)^z \tag{6}$$

where constants *C* and *z* are defined according to the number of passes and tubes arrangement as described in detail in the literature (Selbas et al., 2006).

Finally, the flow velocities are computed as follows:

$$v_t = \frac{m_t}{\frac{\pi d_o^{\,2}}{4} \cdot \rho_t} \cdot \frac{n}{N_t} \tag{7}$$

$$\mathrm{v_s} = \frac{\mathrm{m_s}}{\mathrm{a_s} \quad \rho_s}; \qquad \mathrm{a_s} = \frac{\mathrm{D_s} \quad \mathrm{B} \quad \mathrm{Cl}}{\mathrm{P_t}}; \quad \mathrm{Cl} = \mathrm{P_t} - \mathrm{d_0} \tag{8}$$

where m_t is the tubeside mass flow rate, *n* is the number of tubes passages, d_0 is the tube diameter, ρ_t the tube side fluid density, m_s the shell side mass flow rate, ρ_s the shell side fluid density, D_s the shell diameter, *B* the baffles spacing, *Cl* the clearance.

3.2 Objective function calculation

The objective function has been assumed as the total cost C_{TOT} (€) sum of the investment cost C_I (€) and the discounted annual operating costs deriving from pumping losses C_O (€/yr)

$$\mathrm{C_{TOT}} = \mathrm{C_I} + \mathrm{C_O}\,\mathrm{d_f} = \mathrm{C_I} + \mathrm{C_{OSD}} \tag{9}$$

where d_f is the discount factor (i.e. the uniform series present worth factor)

$$\mathrm{d_f} = \frac{(1+\mathrm{i})^{\mathrm{n}} - 1}{\mathrm{i}(1+\mathrm{i})^{\mathrm{n}}} \tag{10}$$

being *i* the discount rate (%/yr) and *n* the exchanger life (yr) and C_{OSD} (€) is the sum of discounted operating costs.
Therefore, the total cost is a function of thermo-physical G_t, economical G_e, flow related G_c and geometrical G_d parameters, i.e. $C_{TOT} = f(G_t, G_e, G_d, G_c)$.

The capital investment C_I (€) is computed as a function of the exchanger surface adopting Hall's correlation (Taal et al., 2003)

$$C_I = a_1 + a_2 S^{a3} \tag{11}$$

being $a_1=10000$, $a_2=324$ and $a_3=0.91$ for exchangers made with stainless steel for both shell and tubes.

The annual operating cost C_O is instead computed as

$$C_O = H C_E P \tag{12}$$

being H the annual operating hours (hr/yr), C_E (€/Wh) the energy cost, P (W) the pumping power.

The pumping power is assumed proportional to the sum of shellside and tubeside pressure drops (Kern, 1950)

$$P = \frac{1}{\eta}\left(\frac{m_t}{\rho_t}\Delta P_t + \frac{m_s}{\rho_s}\Delta P_s\right) \tag{13}$$

being η the mechanical efficiency, ρ_t and ρ_s the tubeside and shellside flow density, while m_t and m_s respectively the tubeside and shellside mass flow rates and ΔP_t and ΔP_s the tubeside and shellside pressure drop.

In turn, the tubeside pressure drop is computed as sum of pressure drop inside the straight tubes and the elbows

$$\Delta P_t = \Delta P_{tube\,length} + \Delta P_{tube\,elbow} = \frac{f_t L \cdot n \cdot \rho_t \cdot v_t^2}{2 \cdot \varphi_t \cdot d_0} + \frac{4 \cdot n \cdot \rho_t \cdot v_t^2}{2} \tag{14}$$

where $f_t = 0.316 \cdot \mathrm{Re}_t^{\ 0.25}$ valid for ($2300 < Re < 70000$) is the friction factor, L is the tube length, n is the number of tubes passages, ρ_t is the fluid density, v_t the flow velocity, and $\phi = (\mu_t/\mu_w)$.

The shellside pressure drop is instead

$$\Delta P_s = \frac{f_s L \cdot D_s \cdot \rho_s \cdot v_s^2}{B \cdot D_e \cdot \varphi_s \cdot 2} \tag{15}$$

where $f_s = e^{0.576 - 0.19\ \ln(\mathrm{Re}_s)}$ is the friction factor, D_s the shell diameter, D_e the shell hydraulic diameter, L the tubes length and B the baffle spacing.

4. Optimization algorithm

To carry out the above described optimisation procedure a genetic algorithm (GA) has been selected. GA is a stochastic global search method that mimics the process of natural biological evolution (Goldberg, 1989). It operates on a population of individuals representing candidate solutions to the problem and applies the principle of survival of the fittest to produce better performing individuals in subsequent evolutionary generations of the examined population. At each generation, individuals are selected according to their level of fitness and then are bred together. This process leads to the creation individuals better suited to their environment than their parents.

GAs show a number of advantages respect other optimisation techniques making this method suitable for combinatorial optimisation problems even of large size as in this case (Rardin, 1998). They examine a population of solution in parallel instead of evaluating a single solution at a time. GAs do not require any information on the derivative of the objective function (as happens in gradient methods) and the sole value of the objective function influences the direction of the search. Finally, GAs can give a certain number of potentially optimal solutions, and the final choice can be demanded to the user. This characteristic can be important when a problem has a group of optimal solutions, as happens in multi-objective optimization. The first step of GA is the creation of initial population: a certain number of individuals are randomly created so as to start the optimisation process. The individual of a population is a string of bits coding the characteristic of the individual itself. In this case the individual represents a candidate heat exchanger configuration satisfying the design specifications. Then, for each individual its cost is computed, and subsequently the calculation of the fitness function is performed. The fitness function indicates the quality of the single individual with respect to the entire population. In this case, an individual whose cost is less than the average cost has a fitness function higher than the average. Subsequent step is the selection process which consists in creating couples of individuals who will generate offsprings once the desired number of offsprings per generation has been fixed. This implies at first determining for each individual a probability of reproducing which is proportional to the value of its fitness function, and subsequently picking the couples of individuals for reproduction, also known as *sampling*, in a number able to produce the required number of offsprings. This is carried out resorting to proper selection algorithms. The successive phase is the creation of the new generation of individuals.

The new generation is composed by:

1. the best individuals copied from the previous generation
2. new individuals randomly generated
3. new individuals obtained by crossover recombination of the selected individuals of the previous generation
4. mutant individuals.

Copy of the previous generation best individual enables to maintain the best result reached so far. Random generation of new individuals enables to maintain the genetic variety, because such individuals are absolutely independent from the best individual of the present generation, and this enables to overcome local minima. Recombination of individuals to create offspring enables to reproduce the best features of existing individuals into new individuals. The last option is the mutation of some individuals. Mutation is the process which changes a part of an individual to create a new individual, different from the original. The purpose of the mutation process is to examine a wider range of solutions in order to maintain the genetic variety.

When the number of generations reaches the selected limiting value or a specified convergence criterion is met the iterative process is terminated and the resulting best individual represents the desired solution, which complies with the constraints at the lowest cost.

A flow chart of the adopted GA method is shown in figure 2.

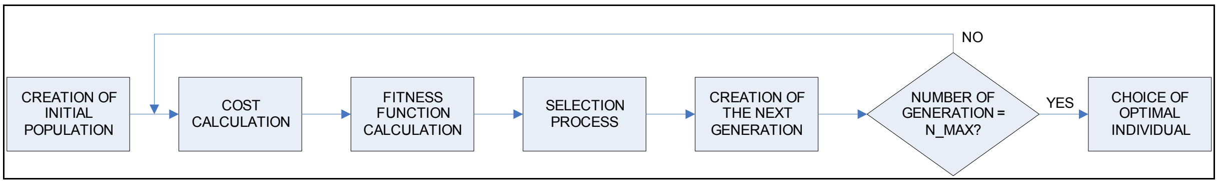

Figure 2

5. Numerical implementation of optimization method.

To perform computations, the described design optimisation procedure has been implemented in MATLAB GA toolbox. As known, GAs do not allow to know if the optimal solution or simply a sub-optimal, although very good, solution has been reached. Therefore, it is very important that the GA parameters are set at values allowing the system to perform satisfactorily while containing the computation time. In particular, main parameters to be fixed are the number of individuals ("PopulationSize") and the number of generations ("Generations").

To test stability and robustness of the implemented method several sample runs of the program were carried out changing the setting parameters of the GA, obtaining always comparable results. At the end of the experimentation phase the following values of the setting parameters were chosen. The adopted maximum number of generations was 500 although convergence was always obtained in maximum twenty generations. Figure 3 shows two examples of the convergence process. The convergence criterion was to stop iterations when the value of the objective function changed less than 10^{-5} €. The population was composed of 20 individuals. The two best individuals were copied from one generation to the successive. The crossover percentage was 70%, while the mutation process involved all individuals less those subject to crossover and the best individuals copied from the previous generation.

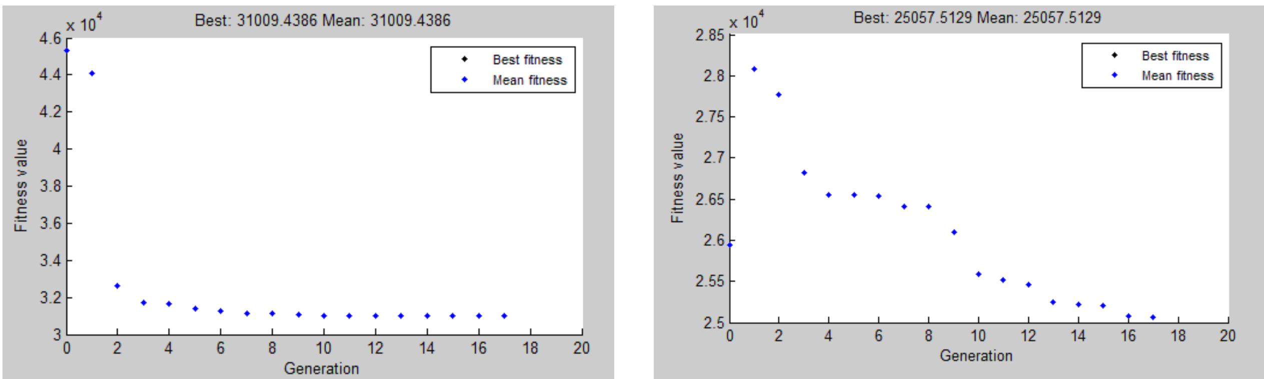

Figure 3

6. Analysis of case studies

In order to test the effectiveness of the described optimisation method it has been applied to a number of literature case studies in order to have reliable reference sizing data to be compared. The experimental campaign involved four different types of exchangers which were chosen as representative of a wide range of possible applications. In particular the following test cases were considered.

Two cases of large heat exchangers
- Case 1: kerosene – crude oil exchanger, duty 1.5 MW
- Case 4: distilled water-water exchanger, duty 2.1 MW

Two cases of medium sized heat exchangers
- Case 2: organic fluid-water exchanger, duty 645 kW
- Case 3: distilled water – raw water, duty 464 kW

For each case the design specifications were fed to the optimisation algorithm and the resulting design data were compared with the design solution given by the referenced author, also comparing the investment (C_I), the sum of discounted annual operating costs (C_{OSD})and the total discounted cost (C_{TOT}) of the original solution versus the one obtained by the GA optimisation method. The appendix supplies the detailed design data of the GA solution and the percentage variations respect the original design for sake of comparison.

Case 1: kerosene-crude oil exchanger. This case study was taken from Kern (1950, page 151). The design specifications are shown in Table 1. The original design assumed an exchanger with four passages tube side and one passage shell side. The same architecture has been retained in the GA approach. Table 2 shows original sizing data.

Table 1

Case 1	Mass flow	T input	T output	ρ	Cp	μ
	[kg/s]	[°C]	[°C]	[kg/m^3]	[J/kg K]	[Pa s]
Shell side	5,52	199	93	850	2700	$4*10^{-4}$
Tube side	18	38	77	995	2250	$36*10^{-4}$

Table 2

Case 1		Literature values
D_s	[m]	0,54
B	[m]	0,127
d_o	[m]	0,0254
L	[m]	4,88
P_t	[m]	0,0318
N_t		158
U	[W/m^2 K]	316
S	[m^2]	61,6
$\Delta P_{tubeside}$	[bar]	0,625
$\Delta P_{shellside}$	[bar]	0,238
P	[W]	1353

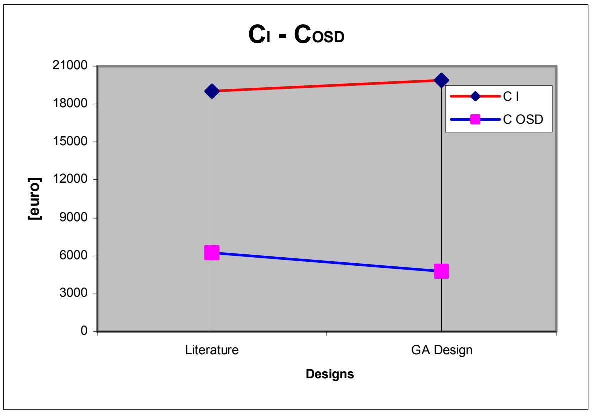

Figure 4

In this case while the capital cost increased by 4.4% owing to an increase of tubes number, which compensates their length reduction leading to a net increase of the heat exchanger surface, the total cost was reduced of about 2.52 % thanks to a 24.13% reduction of discounted operating costs caused by a reduction of tube side pressure drop.

Case 2: organic fluid-water exchanger. This is a one passage for both tube side and shell side exchanger taken from Ramananda Rao et alt. (1991)

Table 3

Case 2	Mass flow	T input	T output	ρ	Cp	μ
	[kg/s]	[°C]	[°C]	[kg/m^3]	[J/kg K]	[Pa s]
Shell side	8,05	99	40	1165	1340	$4,88*10^{-4}$
Tube side	22	33	40	1000	4187	$8*10^{-4}$

Table 4

Case 2		Literature values
D_s	[m]	0,4
B	[m]	0,08
d_o	[m]	0,019
L	[m]	6,096
P_t	[m]	0,02381
N_t		198
U	[W/m^2 K]	581
S	[m^2]	72
$\Delta P_{tubeside}$	[bar]	0,033
$\Delta P_{shellside}$	[bar]	4,58
P	[W]	3300

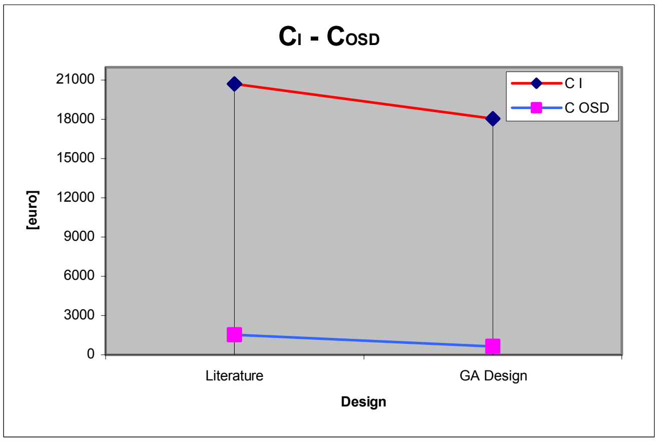

Figure 5

In this case a total cost reduction of about 15.76% is obtained from both reduction of capital cost (- 12.75%) and operating costs (-56.79%). Given the predominance of the capital investment, the total cost reduction was justified above all by the reduction of the exchanger surface, mainly through a reduction of the tubes number, which did not resulted into an increase of pumping power thanks to a significant reduction of shell side pressure drop respect the original design. However, in this case the pressure drop of the original design was not declared by the author and it was estimated resorting to the formula utilized by the GA and the exchanger geometry.

Case 3: distilled water – raw water exchanger. This case study was taken from Kern (1950, page 155). The design specifications are shown in Table 5. The exchanger structure is characterized by two passages tube side and one passage shell side.

Table 5

Case 3	Mass flow	T input	T output	ρ	Cp	μ
	[kg/s]	[°C]	[°C]	[kg/m^3]	[J/kg K]	[Pa s]
Shell side	22	34	24	995	4178	$9,65*10^{-4}$
Tube side	37	29	27	997	4178	$9,57*10^{-4}$

Table 6

Case 3		Literature values
D_s	[m]	0,38
B	[m]	0,305
d_o	[m]	0,019
L	[m]	4,88
P_t	[m]	0,0237
N_t		160
U	[W/m^2 K]	1470
S	[m^2]	46,7
$\Delta P_{tubeside}$	[bar]	0,51
$\Delta P_{shellside}$	[bar]	0,52
P	[W]	3000

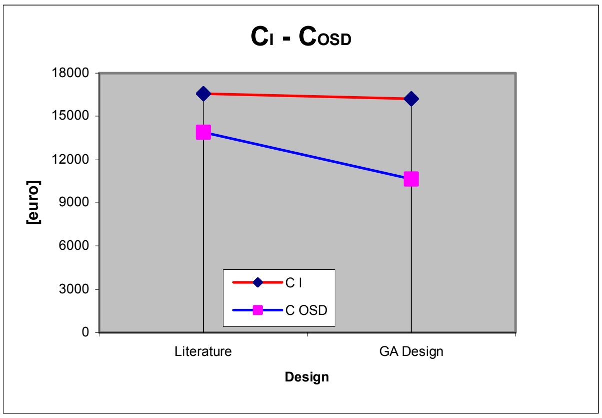

Figure 6

In this case too a reduction of both capital investment (i.e. surface area) and operating costs was obtained but, respect the case #2 the reduction of operating costs was much higher in absolute terms. Thanks to a significant decrease of shell side pressure drop the sum of discounted annual costs lowered by 23.36% which, paired to a of 2.12% reduction in capital cost led to a total costs decrease of 11.8%.

Case 4: distilled water-water exchanger. This is a two passages tube side and one passage shell side taken from Selbas and coworkers (Selbas et al., 2006). Table 7 shows design specifications and Table 8 compares the design data.

Table 7

Case 4	Mass flow	T input	T output	r	Cp	m
	[kg/s]	[°C]	[°C]	[kg/m^3]	[J/kg K]	[Pa s]
Shell side	33	20	35	995	4178	$3*10^{-4}$
Tube side	20	75	50	995	4178	$3*10^{-4}$

Table 8

Case 4		Literature values
S	[m^2]	57
$\Delta P_{tubeside}$	[bar]	0,037
$\Delta P_{shellside}$	[bar]	0,019
P	[W]	166

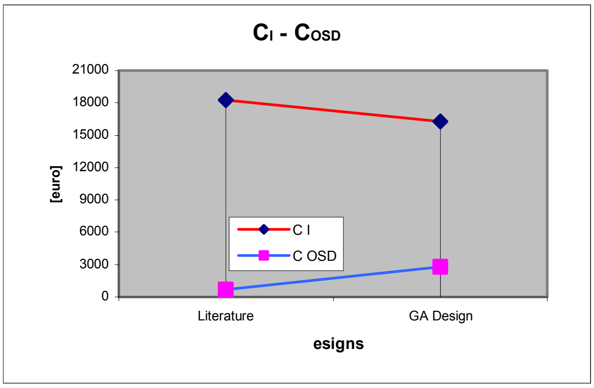

Figure 7

In this case an increase of the total cost, although very small (about 0.7 %), resulted from the GA application even if a significant reduction of the capital investment was instead obtained (-12 %). This was caused by a marked increase of the operating costs (+76 %). However, the pressure drops obtained by the GA algorithm appear more reasonable than those declared by the original design authors.

Obtained results confirm the effectiveness of the proposed optimisation method. In fact, significant percentage reductions in capital investment and/or the operating costs were obtained leading in most of the examined cases to a significant total cost decrease. The variation of capital investment ranged between – 12.75% and +4.4%. As far as operating costs were concerned, instead, the percentage decrease ranged from –23.36% to – 56.79% except the case #4. Finally, the saving in total cost ranged between –2.52% to –15.76%. This kind of results indicate that in many cases an optimisation of heat exchanger costs can be searched by acting on the pressure drops, through a proper choice of fluids velocity, rather than acting on the surface area. 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 $Re^{0.6}$) and can be only slightly improved by changing the design parameters. On the contrary, pumping losses are highly responsive to changes in flow velocity being dependent on $Re^{1.8}$. Even if the absolute saving from size reduction can sometime overweight the pumping loss reduction, the percent reduction of the latter term is usually more significant. This means that in relative terms the impact of the optimisation process on the pumping losses is higher. Therefore, the analysed case studies indicate that utilization of heat exchanger design optimisation procedures besides allowing significant reduction in total cost, enable drastic reduction of energy related operating costs, thus supporting efforts in resource conservation and energy saving which are a main concern of plant owners.

7. Conclusions

In this paper a formulation of the shell and tube heat exchanger design optimisation problem has been proposed based on the utilization of a genetic algorithm. Resorting to the analysis of a number of significant case studies, the paper demonstrates that this optimisation algorithm can effectively be utilized to improve the design process and significantly reduce the total discounted cost of heat exchange equipment respect traditional sizing procedures. While reduction in capital investment can surely be obtained, the best results are obtained through the reduction of operating costs related to pumping losses. Therefore, this method can be exploited to pursue also goals of energy saving in industrial plants. Referring to literature test cases reduction of capital investment up to 12% and savings in operating costs up to 57% have been obtained, with overall decrease of total discounted cost up to 16%. This shows the improvement potential coming from applying the proposed method. Nevertheless, while in some cases the cost improvement may seem small, it should be pointed out that in practice this can lead to significant overall savings when a large number of heat exchanging units exist in the same plant as is the case in most process plant applications. Furthermore, the genetic algorithm allows for rapid solution of the design problem and enables to examine a number of alternative solutions of good quality through the evolution of a population of individuals which improve their characteristics over a number of generations through a selection process. This gives the designer more degrees of freedom in the final choice respect traditional methods. As a final remark it should be pointed out that a distinct feature of the GA is the capability of rearranging the design parameters values in order to seek the optimal balance between capital investment and operating expenses.

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9. Appendix

Case 1		This work	Δ%
D_s	[m]	0,59	8,5
B	[m]	0,11	-15,5
d_o	[m]	0,0254	0,0
L	[m]	4,3	-13,5
P_t	[m]	0,0318	0,0
N_t		195	19,0
U	[W/m^2 °C]	357	11,5
S	[m^2]	67	8,1
$DP_{tubeside}$	[bar]	0,44	-42,0
$DP_{shellside}$	[bar]	0,28	15,0
P	[W]	1027	-31,7
$\mathbf{C_I}$	[€]	19895	4,6
$\mathbf{C_{OSD}}$	[€]	4750	-24,1
$\mathbf{C_{TOT}}$	[€]	24645	-2,5

Case 2		This work	Δ
D_s	[m]	0,39	-2,5
B	[m]	0,11	37,5
d_o	[m]	0,019	0
L	[m]	5,7	-6,50
P_t	[m]	0,024	0,80
N_t		165	-16,67
U	[W/m^2 °C]	518	-10,84
S	[m^2]	55,7	-22,64
$DP_{tubeside}$	[bar]	0,043	30,30
$DP_{shellside}$	[bar]	1,89	-58,73
P	[W]	1420	-56,97
$\mathbf{C_I}$	[€]	18062	-12,74
$\mathbf{C_{OSD}}$	[€]	656	-56,79
$\mathbf{C_{TOT}}$	[€]	18718	-15,76

Case 3		This work	Δ%
D_s	[m]	0,39	2,56
B	[m]	0,36	15,28
d_o	[m]	0,019	0
L	[m]	4,267	-14,37
P_t	[m]	0,0237	0
N_t		175	8,57
U	[W/m^2 °C]	1845	20,33
S	[m^2]	44,6	-4,71
$DP_{tubeside}$	[bar]	0,44	-15,91
$DP_{shellside}$	[bar]	0,32	-62,50
P	[W]	2300	-30,43
$\mathbf{C_I}$	[€]	16214	-2,16
$\mathbf{C_{OSD}}$	[€]	10645	-30,47
$\mathbf{C_{TOT}}$	[€]	26859	-13,38

Case 4		This work	Δ%
S	[m^2]	45,25	-25,97
$DP_{tubeside}$	[bar]	0,0663	44,19
$DP_{shellside}$	[bar]	0,1371	86,14
P	[W]	600	72,33
$\mathbf{C_I}$	[€]	16281	-12,20
$\mathbf{C_{OSD}}$	[€]	2775	76,36
$\mathbf{C_{TOT}}$	[€]	19056	0,69