ECONOMIC OPTIMIZATION OF HEAT EXCHANGER DESIGN AND MAINTENANCE POLICY

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Abstract. *In this paper a novel approach to shell and tube heat exchanger optimization is presented. The method allows a joint optimization of both the equipment design and the cleaning policy. Given the heat duty of the equipment a thermal design procedure defines the heat transfer area as well as flow velocities and the fouling rate so that capital investment, pressure losses and the cleaning interval required to reach a maximum allowed fouling resistance are calculated. A genetic algorithm is utilized to determine the optimal values of both geometric design parameters (i.e. shell diameter, tubes diameter, pitch pattern etc.) and the maximum allowable fouling resistance so that the minimization of a total cost function including capital investment, operational costs related to friction losses, and maintenance costs related to the cleaning schedule is obtained. The method is novel in that during the design phase the heat exchanger architecture is usually optimized neglecting maintenance expenses, while during the operational phase the heat exchanger architecture is already finalized and maintenance optimization approaches allow only to determine a minimum cost cleaning schedule based on the existing exchanger. In this work, instead, the problems of equipment sizing and cleaning schedule determination are solved simultaneously so that the entire life cycle cost is minimized. In the paper the optimal design approach is described and an application example is provided to show the capability of the method.*

Keywords*: heat exchangers optimization, fouling, genetic algorithm, heat exchanger cleaning.*

1. INTRODUCTION

In recent times a renewed interest in the optimal design of heat exchangers has been witnessed in the literature. This results from the availability of new optimization techniques, such as genetic algorithms, able to handle a large number of design parameters including both discrete and continuous variables. Considering the functional importance and widespread utilization of heat exchangers in process plants, their minimum cost design is also an important goal. In particular, the minimization of energy related expenses is critical in the optic of energy savings and resources conservation. However, given that most of the life cycle cost of an exchanger is made up of operational costs, including periodic cleaning to control fouling phenomena which deteriorate heat transfer, even the optimization of maintenance schedules is a relevant problem. In fact Kuppan (1989) reports, for instance, that the economic loss due to fouling in heat exchanger is about 10^9 \$/yr for American plants, whereas in Hewitt (1998) an estimate of economic loss up to 1.4 G\$/yr for U.S.A. and between 0.5 and 1 M\$ for English plants is given.

However, in the literature two basically different approaches have been pursued as far as heat exchanger optimization is concerned. The first approach aims to the optimal sizing of the heat exchanger usually based on a cost minimization goal, considering capital investment and energy related expenses, or on the maximization of some thermal performance. In this approach the impact of periodical cleaning of the exchanger is neglected and fouling phenomena are considered only when including an allowance for a fouling resistance in the thermal design procedure. The second approach, instead, assumes that the heat exchanger has been already built and that a maintenance schedule has to be optimized in order to minimize maintenance and energy related costs while satisfying the required heat duty.

Recent examples of design optimization methods are the works of Khalifeh Soltan et al. (2004), Unuvar and Kargici (2004), Selbas et al. (2006), Sena-Gonzalez et al. (2007), Ponce-Ortega et al. (2009). The optimization techniques utilized nowadays in heat exchangers optimization are frequently of genetic nature (Tayal et al., 1999; Selbas et al., 2006; Babu and Munawar, 2007; Caputo et al., 2008; Costa and Queiroz, 2008). When evaluating a cost based objective function the investment cost can be estimated by literature correlations (Taal et al., 2003) whereas the operating cost is always considered as an energy related cost, imputable to overcoming friction losses in the fluid flow.

Examples of maintenance schedule optimization approaches are provided instead by Putman (2001), Khan and Zubair (2004), O'Donnell et al (2001), Sheikh et al. (1996), Zubair et al. (1997), Zubair and Shah (2004). However, in this field a large contribution has been made to maintenance of heat exchanger networks (Georgiadis and Papageorgiou, 2000; Georgiadis et al., 2000; Lavaja and Bagajewicz, 2004 and 2005; Markowski and Urbaniec, 2005; Sanaye and Niroomand, 2007; Smaili et al., 2001, 2002a, 2002b). Only Wildi-Tremblay and Gosselin (2007) recently attempted to include maintenance concerns in the design optimization procedure by simply including a minimum surface area constraint to account for fouling.

In order to overcome the drawbacks of the current two-step approach (optimal design neglecting maintenance and subsequent optimization of the cleaning schedule for an existing exchanger), in this work a method for joint optimization of both the equipment design and the cleaning policy is described. A genetic algorithm is utilized to determine the optimal values of both geometric design parameters (i.e. shell diameter, tubes diameter, pitch pattern etc.) and the maximum allowable fouling resistance so that the minimization of a total cost function including capital investment, operational costs related to friction losses, and maintenance costs related to the cleaning schedule is obtained. In this manner a more precise approach respect that of previous literature works (Wildi-Tramblay and Gosselin, 2007) is obtained in that an overall cost minimization is pursued without imposing any predetermined constraint. Furthermore, the proposed approach enables to overcome the limitations of traditional optimization methods which neglect fouling impact (Bell, 2000) thus risking to undersize the heat exchanger. The paper is organized as follows. At first a discussion of heat transfer surfaces fouling is carried out with reference to its impact on the exchanger performances and the cleaning process. Then the proposed optimization method is described. Finally, an application example is provided to show the capability of the method. The method has been developed with specific reference to shell and tube heat exchangers owing to their widespread diffusion.

2. HEAT EXCHANGERS FOULING

During heat exchangers operations a progressive accumulation of fouling deposits occurs on both sides of the heat exchanging surfaces. This determines the gradual build up of an additional heat transfer resistance that reduces the overall heat transfer coefficient U thus lowering the heat flow between hot and cold fluids. The overall heat transfer coefficient can be, in fact, expressed as

$$1/U = \frac{1}{\alpha_s} + R_{f_o} + \frac{D_t \ln\left(D_t / D_{t_i}\right)}{2\,\lambda_w} + \frac{D_t}{D_{t_i}}\left(R_{f_i} + \frac{1}{\alpha_t}\right) \tag{1}$$

where R_{fo} and R_{fi} are the shell side and the tube side fouling resistance respectively (m^2 K/W), α_t and α_s (W/m^2 K) are the tube side and shell side convective coefficient respectively; D_t and D_{ti} the tube external and internal diameter (m), and λ_m is the thermal conductivity of tube walls (W/m K). The fouling phenomenon depends on several physical and operational parameters, namely the chemical nature of the fluids, the stream temperature, the surface roughness, the particulate load in fluid, and the stream velocity, to name the most influent ones. Only a few of such parameters are under control of the designer. Obviously, as both shell and tube side fouling resistance increase through the time, the fouling phenomenon is time dependent. Furthermore, a pressure drop increase is observed due to reduction of flow pass area and a growth of the surface roughness (Bott, 1995; Bryers, 1983; Garrett-Price at al., 1985; Muller-Steinhagen, 2000; Muller-Steinhagen et al., 2007).

An example of the progressive decay of the overall heat transfer coefficient (U) from its initial value U_{clean} is shown in figure 1 according to different fouling mechanisms.

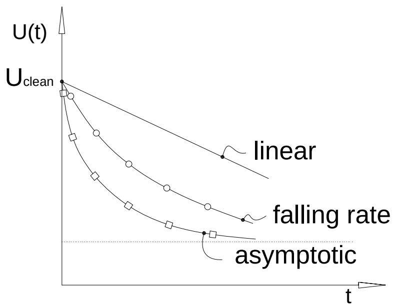

Figure 1. Different kind of fouling trends

While the discussion of different fouling mechanisms is beyond the scope of this paper the interested reader can consult a number of literature works on this subject (Jun and Puri, 2005; Sheikh et al., 2001; Jafari Nasr and Majidi Givi, 2006; LeClercq-Perlat and Lalande, 1991; McGurn and Thompson, 1995; Mwaba et al., 2006; Riverol and Napolitano, 2005; Zabiri et al., 2007). Whichever the fouling process, the growth of the fouling film through time can be expressed by

$$\frac{dR_f}{dt} = \varphi_d - \varphi_r \tag{2}$$

where ϕ_d is the fouling deposit growth rate and ϕ_r is the rate of deposit removal by the flow. The overall fouling layer growth law is very difficult to predict especially when it is time dependent. Generally, when deposits are brittle and the removal rate is proportional to the thickness, the fouling film growth law is linear with time. In other cases the deposit rate decreases with time and the growth law is asymptotic.

According to Ludwig (2001), in case of asymptotic fouling, the fouling resistance can be obtained integrating equation (2) thus obtaining the following law of fouling resistance growth (also known as Sern –Keaton correlation)

$$R_f = R_f^* \,(1 - e^{-\beta t}) \tag{3}$$

where R^*_f is an asymptotic resistance value related to the fluid velocity ν, the tube diameter D_t and to the kind of foulant, while β is the reciprocal of the fouling phenomenon time constant. Sample empirical correlations valid for typical fluids are the following (Hewitt, 1998),

$$R_f^* = \frac{0.101}{v^{1.33}\,D_t^{0.23}} \qquad \text{Calcium carbonate on surface at constant temperature} \tag{4}$$

$$R_f^* = \frac{0.55}{v^2} \qquad \text{Oil deposit at constant heat flux} \tag{5}$$

$$R_f^* = \frac{0.015}{v^{1.2}} \qquad \text{Sand deposit from water at constant heat flux} \tag{6}$$

At the beginning of heat exchanger's operating life the overall heat transfer coefficient U_{clean} is thus the sum of convective coefficients, tube and shell side, and the reciprocal of the thermal resistance of tube wall. Then the fouling resistances are gradually added and progressively increase until dirt accumulation eventually determines an excessive heat transfer resistance and a cleaning operation is required to bring back the equipment to its original condition. Figure 2 shows the trend of gradual decay of U and its periodical restoration for two different values of the allowable fouling resistance.

By fixing a threshold level for the maximum fouling resistance, and knowing the trend of fouling resistance growth over time, it is thus possible to determine the timing of the cleaning operations.

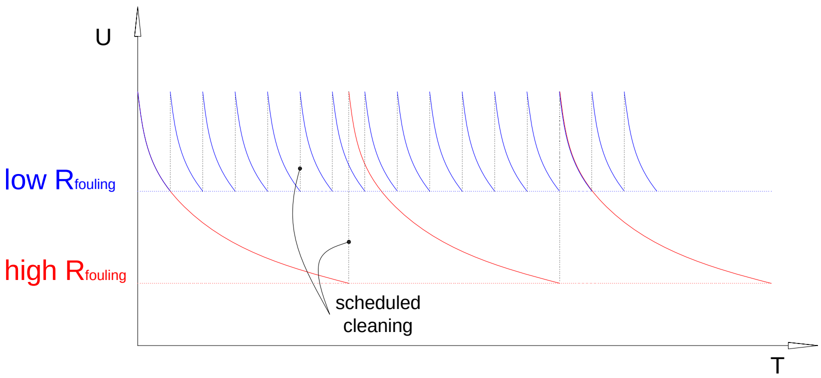

Figure 2. Influence of allowed fouling resistance on cleaning policy.

Referring to figure 2 it can be observed that, if the allowable fouling resistance is set to a small value, frequent maintenance stops are needed for cleaning. In this case, since the minimum value of U is not much lower than U_{clean}, only a small oversizing of the exchanger results, thus limiting the extra surface and the additional investment cost. On the contrary, if the value of maximum fouling resistance is high, the maintenance stops would be less frequent reducing the cleaning cost, but a higher extra surface area will be needed to allow for the lower minimum value of *U*. In practice, given the heat duty specification, the heat transfer area is determined by the assumption of a value for the minimum allowable heat transfer coefficient in dirty operating conditions U_{dirt}. The lower the U_{dirt} value is specified, the higher the oversizing of the heat exchanger results, but the lower the cleaning expenses. Conversely, if a high value of U_{dirt}.is assumed a small oversizing results but more frequent cleaning operations will be required, thus asking for a trade off analysis.

To ensure the effectiveness of the equipment for an acceptable working period, designers must necessarily overdesign the heat transfer surface, increasing the investment cost. This also asks for the need for regulating the mass flow during the initial period of operating life when exchange surfaces are still clean and the transfer coefficient is high. Generally, an oversizing from 25% up to 50% of the total surface area required in clean conditions is commonplace (Pak et al., 2005; Sanatgar and Somerscales, 1991). Often, probabilistic design methods are also employed to account for fouling effects (Zubair et al. 2000a, 2000b; Zubair and Qureshi, 2006).

However, an attempt to limit fouling by acting on the flow velocity can also be pursued. In fact, the deposition rate is influenced by the stream velocity, and an increase of stream velocity can be beneficial from this point of view as the higher the fluids velocity, the lower the tendency to foul, even if an increase of pressure losses occurs. This also determines a trade off situation.

The restoration of initial heat transfer conditions is made resorting to chemical or mechanical cleaning operations (Abdul-Latif et al., 1988; Frenier and Barber, 1998). This often requires to put the equipment off line and determines additional costs related to unavailability and operational problems. In particular, chemical cleaning requires the equipment off-line, while mechanical cleaning can be carried out both on-line and off-line.

Usually, mechanical cleaning is less expensive than the chemical one but when a strong mechanical action is required to remove deposits, even the mechanical cleaning methods needs that the exchanger is put off-line. Generally, tube 30° layout is more efficient from a thermal point of view but it doesn't allow a mechanical cleaning. On the contrary, tube 45° or 90° layout, with or without clean lines, are suitable to mechanical cleaning but they are thermally less efficient. Moreover, the feasible kind of cleaning is related to the ratio of shell internal diameter to tubes outside diameter. Table 1 (Hewitt, 1998) shows allowed shell-tube diameter combinations for mechanical cleaning.

Table 1. Allowed coupling Shell/Tube for mechanical cleaning
(dotted area: recommended, dashed area: possible)

D_t \ D_s	100	200	300	500	700	1000	1500
6							
10							
14							
20							
25							
38							
51							

Mechanical cleaning not allowed

Overall, setting a design value for U_{dirt}.determines a trade off between capital investment and cleaning costs, while setting a flow velocity impacts the trade off between capital investment, maintenance schedule and pumping costs. Moreover, the exchanger constructive arrangement determines the allowed cleaning method, thus further affecting the maintenance costs. Therefore, exchanger design and maintenance scheduling problems are strictly interrelated.

3. PROPOSED APPROACH

The proposed 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 including a maximum allowable fouling resistance $R_{f\ allowable}$;
- evaluation of the capital investment, operating costs, 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 3. 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.

A set of thermo physical properties, process data and fixed characteristics (heat exchanger TEMA type) of the equipment are assigned by designers. Starting from this input data a random starting value is given to a set of independent design variables (VIP). The VIP number and meaning depends on the equations used to size the equipment. In this work the Bell-Delaware method (Shah and Sekulic, 2003) has been used and the selected VIP 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 $L_{pt}Ratio$, the baffle cut B_c, the sealing strips number N_{ss}, the tube layout angle θ_{tp} and the tube pass number N_{tp}. A further independent variable is the maximum allowable fouling resistance $R_{f\ allowable}$. The other heat exchanger's characteristics (i.e. the dependent design variables, VDP) are then directly computed from the VIP. 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 and fluid dynamical equipment characteristic. Once the VDP are computed from the VIP the overall heat transfer coefficient (U) is estimated. This allows to determine the total heat exchanger's length L_{tt} (m), the surface area (S) and the baffles number N_b (as entire multiple of L_{bc} at least equal to calculated L_{tt}).

The computed values of flow velocities and the constructive details of the exchanger structure are then used, resorting to Eqs. 3-6, to predict the time (T_{clean}) when the limit fouling resistance $R_{f\ allowable}$ is reached and cleaning is to be performed. Then, computation of the objective function follows.

Here we assume that the objective function to be minimized is the total cost function (C_{tot}) which is a sum of the capital investment, and the present worth of the pumping cost for overcoming friction losses and cleaning costs occurring over the equipment life,

$$C_{tot} = C_I + \sum_{i=1}^{Nyr} \frac{C_{es}(1+f)^i}{(1+s)^i} + \sum_{i=1}^{Nyr} \frac{N_{clean,i}\, C_{clean,i}}{(1+s)^i} + \sum_{j=1}^{v} \alpha_j\, P_j\, C_{fitt} \tag{7}$$

where C_I is the capital investment (€), C_{es} is the annual operating cost due to pumping power caused by fluid friction losses (€/yr), f is the energy cost inflation rate (%/yr), s is the interest rate (%/yr), Nyr is the equipment life (yr), $N_{clean} = 1/\ T_{clean}$ is the annual cleanings number (1/yr), C_{clean} is the unit cost of each cleaning operation (€) dependent on the allowable kind of cleaning. The last term is a penalty function term which penalizes the cost of solutions violating one or more operating constraint imposed by user; a_i is a binary activation index for the *i-th* constraint (imposed by user: 0 is the constraint is omitted, 1 if it is considered), P_i is a binary violation index (0 if constraint is not violated, 1 otherwise), and C_{fitt} is a fictitious cost that lead to the rejection of solutions not satisfying one or more constraint. The total cost function is thus completely determined by specifying the constructive details of the heat exchanger and the allowable fouling resistance.

The genetic optimisation algorithm (GA), based on the value of the objective function, updates the trial values of the optimisation variables (VIP) which are then passed 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, as shown in the flow diagram of Figure 3. Further details of the adopted genetic algorithm for the optimization process are given elsewhere (Caputo et al., 2008).

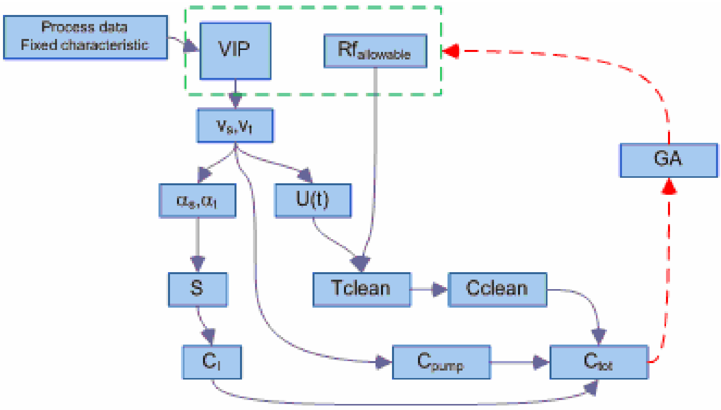

Figure 3. Flow chart of optimization procedure using GA and considering maintenance

It should be noted that the approach proposed in this work is centered on the trial assignment by the optimization algorithm of the allowable fouling resistance as an independent design variable, and on the computation of the cleaning frequency as dictated by the specified fouling resistance and the geometrical characteristic of the exchanger's through the resulting fluids velocities, as shown by equations 3 to 6. Knowing the fouling mechanism, it is, in fact, possible to determine the time dependence of fouling resistance as influenced by the constructive features of the equipment. Once the fouling resistance growth law for the actual heat exchanger and fluids is known, to fix a threshold value for the fouling resistance is equal to set a service interval.

The proposed approach then suggests to entrust the genetic algorithm with the autonomous choice of the optimal allowable fouling resistance once the overall heat transfer coefficient asymptotic decrease law is known (figure 1). In this manner the value to be assigned to this critical parameter is managed within the entire optimization procedure instead of being left to a trial and error approach or to the designer's arbitrary choice. The selected allowable fouling resistance determines, in fact, the extra heat transfer surface requirements and the cleaning frequency, thus setting the trade-off between capital investment and operating expenses. The resulting time schedule of maintenance cleaning is also the optimal schedule for the exchanger being designed.

4.CASE STUDY

In order to test the effectiveness of the proposed approach a case study is presented here. The case study is based on the process specifications and streams data taken from a textbook heat exchanger sizing example from Kern (1950). The considered equipment exchanges heat between distilled water and raw water streams. The process data are shown in Tables 2 and 3. In Kern's original formulation a minimal tube-side velocity higher than 1.8 m/s was imposed to prevent fouling. Here this constraint has been removed as the actual fouling resistance determined by fluid nature and velocity is considered. Moreover, no attempt will be made to compare the results of this model equipment sizing with the sizing approach of Kern in that in the original example no data were provided to determine the kind of fouling phenomenon which was considered.

Table 2. Heat exchanger thermal characteristics

Duty	[kW]	413.8	F_t	[-]	0.947
ΔTLM	[°C]	6.36	ΔTm	[°C]	6.0

Table 3. Process Data

Shell Side			Tube Side		
	Distilled water			Raw water	
Inlet temperature T_{si}	[°C]	34.0	Inlet temperature T_{ti}	[°C]	24.0
Outlet temperature T_{so}	[°C]	29.5	Outlet temperature T_{to}	[°C]	26.7
Mass flow rate M_s	[kg/s]	22.0	Mass flow rate M_t	[kg/s]	35.3
Fluid density ρ_s	[kg/m^3]	995	Fluid density ρ_t	[kg/m^3]	995
Fluid conductivity λ_s	[W/m K]	0.615	Fluid conductivity λ_t	[W/m K]	0.605
Fluid specific heat C_{ps}	[J/kg K]	4180	Fluid specific heat C_{pt}	[J/kg K]	4180
Fluid viscosity η_s	[mPa/s]	0.773	Fluid viscosity η_t	[mPa/s]	0.887

The design optimization procedure was implemented on a personal computer resorting to the Genetic Algorithm toolbox of the scientific computing environment MATLAB. The heat exchanger was sized using the Bell-Delaware method and investment cost was estimated adopting the Purohit approach (Taal et al., 2003). Total cost was evaluated over a working period of five years (40000 h). The main characteristics of the resulting design as obtained from the optimization procedure proposed in this work are shown in Table 4 (column labeled "Joint Optimization"), whereas TGA1, TGA2, TGA3 represent exchangers resulting from a traditional design optimization procedure carried out by a genetic algorithm similar to that discussed by Caputo et al. (2008). In the cases of columns labeled TGA1, TGA2, TGA3, the heat exchanger architecture has been thus optimized neglecting the maintenance cost, with the aim of minimizing an objective function represented solely by the capital investment and the present value of the life cycle energy cost for pumping. The resulting maintenance schedule determined the additional maintenance cost which were added to the former costs to estimate the overall cost. In cases of TGA1, TGA2, TGA3 an allowable fouling resistance of respectively $2.5x10^{-4}$, $5x10^{-4}$, $1x10^{-3}$ [m^2 K/W] was arbitrarily chosen by the designer as would happen in practice. In the "Joint Optimization" column, instead, the genetic algorithm minimized an overall cost function sum of capital investment and present value of life cycle pumping energy cost and maintenance costs as proposed in this work. In this case the algorithm autonomously determined an "optimal" allowable fouling resistance value of $1.5x10^{-4}$ [m^2 K/W]. Finally, for sake of comparison an additional column labeled "Hybrid TGA" has been included in Table 4. This column represent a design obtained through the traditional genetic algorithm optimization approach (i.e. neglecting the maintenance cost in the objective function) but assuming as the designer-imposed allowable fouling resistance the same optimal value of $1.5x10^{-4}$ [m^2 K/W] obtained through the joint optimization procedure. It interesting to note that in this case a different design, with a higher cost, results respect that obtained directly through the joint optimization approach.

Once the sizing algorithm in the optimization procedures has finalized the constructive details of the exchanger, the tubes diameter and flow velocity are determined and the asymptotic fouling resistance can be calculated (a time constant vale of 0.00008 e 0.00006 s^{-1} was assumed respectively shell-side and tube-side) thus allowing to determine the fouling growth curve and the time interval required to reach the imposed (TGA1, TGA2, TGA3, Hybrid TGA) or the self-determined (Joint Optimization) allowable fouling resistance. This allows to compute the number of cleaning operations, maintenance cost and the overall cost of the heat exchangers as shown in the Table. A minimum cost value of about 22200 € has been obtained by the proposed Joint Optimization approach. It is worth noting that even if all of the other exchangers (TGA1, TGA2, TGA3, Hybrid TGA) have been obtained by minimizing a total cost function including capital investment and energy cost, the fact that the allowable fouling resistance value was arbitrarily set by the designer, as happens in practice, determined higher overall costs than the ones incurred when even the allowable fouling resistance is treated, instead, as an independent variable to be optimized. This confirms that the arbitrary selection of a design value for the allowable fouling resistance may significantly impact the life cycle cost of an exchanger leading to poor economic performances.

Table 4. Proposed Heat exchanger optimized designs

Parameter			**TGA 1**	**TGA 2**	**TGA 3**	**Joint Optimization**	**Hybrid TGA**
Shell diameter	D_s	[mm]	795.0	891.0	670.0	641.0	609.8
Baffle cut	B_c	[%]	16.0	18.0	19.0	28.0	21.0
Number of baffles	N_b	[-]	13	17	14	27	12
Central baffle spacing	L_{bc}	[mm]	361.0	178.2	670.0	128.2	609.8
Extremal baffles spacing	L_{bi}, L_{bo}	[mm]	184.0	178.2	670.0	128.2	609.8
Pitch ratio	$L_{tpRatio}$	[%]	1.29	1.48	1.35	1.32	1.35
Tubes ext. diameter	D_t	[mm]	50.8	12.7	15.8	31.7	50.8
Tubes internal diameter	D_{ti}	[mm]	46.5	10.9	14.1	28.4	46.5
Tubes pitch	L_{tp}	[mm]	65.6	18.7	21.4	41.9	68.4
Tube layout angle	θ_{tp}	[deg]	45	90	45	90	30
Tube passes	N_{tp}	[-]	4	2	1	2	2
Tubes number	N_{tt}	[-]	85	1562	685	149	54
Tubes length	L_{tt}	[mm]	4732.4	3243.9	10077.3	3615.7	7952.4
Flow velocity (tube-side)	v_t	[m/s]	0.98	0.48	0.33	0.74	0.77
Flow velocity (shell-side)	v_s	[m/s]	0.21	0.23	0.09	0.87	0.23
Reynolds number (shell-side)	Re_s	[-]	15816.6	6743.0	2672.0	43603.5	15143.7
Prandtl number (shell-side)	Pr_s	[-]	5.3	5.3	5.3	5.3	5.3
Reynolds number (tube-side)	Re_t	[-]	56754.3	6586.3	5818.1	26508.6	44667.8
Prandtl number (tube-side)	Pr_t	[-]	5.5	5.5	5.5	5.5	5.5
Convective heat transfer coefficient (shell-side)	α_s	[W/m^2 K]	2656.4	3053.8	2550.1	3759.3	2379.9
Convective heat transfer coefficient (tube-side)	α_t	[W/m^2 K]	3918.4	647.7	351.4	3489.8	3235.2
Overall heat transfer coefficient	U_{dirt}	[W/m^2 K]	995.3	404.4	253.4	1359.0	903.1
Heat exchange area	S	[m^2]	63.32	197.63	342.40	52.96	68.10
Heat exchange area	S_{clean}	[m^2]	44.98	145.83	247.11	40.24	52.02
Heat exchange area	S_{extra}	[%]	28.96	26.21	27.83	24.01	23.61
Pressure drop (shell-side)	Δp_s	[kPa]	3.16	7.53	3.26	17.11	2.33
Pressure drop (tube-side)	Δp_t	[kPa]	6.22	2.65	1.45	2.30	2.76
Operating cost	C_{es}	[€/yr]	371.8	333.4	158.2	588.8	191.1
Allowed fouling resistance	$R_{f\ allowable}$	[m^2 K/W]	0.00025	0.0005	0.001	0.00015	0.00015
Cleaning interval	T_{clean}	[h]	2859	1886	684	4166	-
Overall cleanings number	N	[-]	14	22	59	9	-
Operating cost present value	$C_{es,tot}$	[€]	1409.5	1264.0	599.9	2990.1	724.5
Cleaning cost present value	$C_{clean,tot}$	[€]	1137.2	9477.0	4548.9	758.2	1137.2
Capital investment	C_I	[€]	20032.3	37499.8	53636.3	18467.0	20738.0
Total cost	$C_I + C_{es,tot}$	[€]	21441.8	38763.9	54236.2	21457.1	21462.5
Total cost	$C_I + C_{es,tot} + C_{clean,tot}$	[€]	**22579.1**	**48240.9**	**58785.1**	**22215.3**	**22599.7**

Table 4 also shows the resulting heat exchange area (S) in comparison with the theoretical heat transfer area (S_{clean}) required to satisfy the duty specifications had the fouling phenomenon been negligible. A fouling related percent increase of the heat transfer area (S_{extra}) thus follows. It can be observed that the optimal design value of the maximum fouling resistance implied an extra surface of 24% respect the value obtainable neglecting fouling phenomena.
In case of the optimal "Joint Optimization" exchanger a total of 9 cleaning operations are performed during the life cycle of 40000 h and the optimal maintenance interval is about 4160 hours, i.e about twice per year. The resulting trend of the overall heat transfer coefficients and the corresponding maintenance schedules are shown in Figures 4, 5, 6, 7 for the "Joint Optimization" exchanger as well as the TGA1, TGA2, TGA3 designs for sake of comparison.

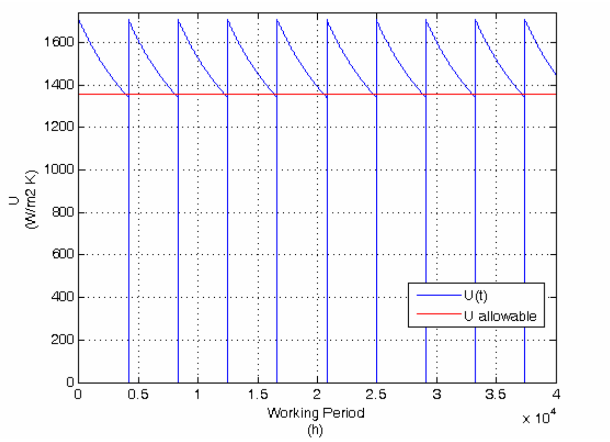

Figure 4. U trend through 5 years for the "Joint Optimization" exchanger.

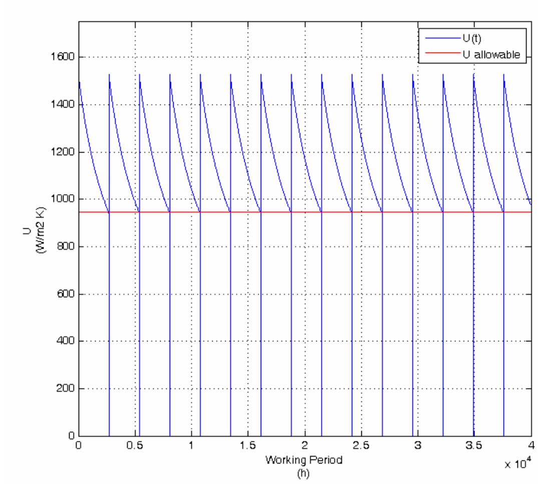

Figure 5. U trend through 5 years for the "TGA1" exchanger.

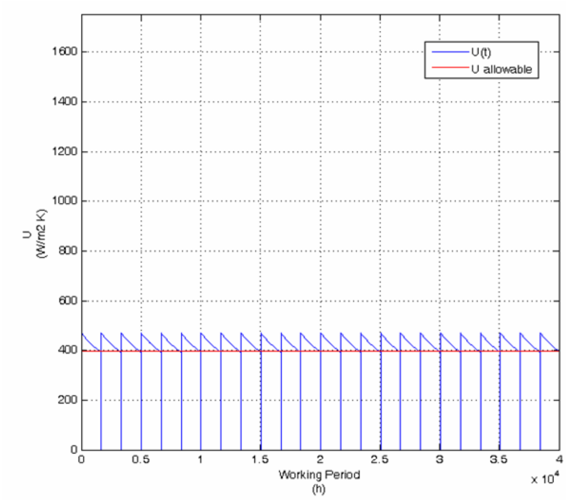

Figure 6. U trend through 5 years for the "TGA2" exchanger.

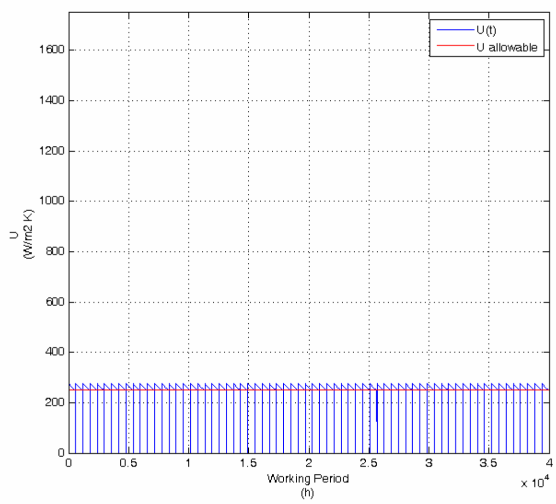

Figure 7. U trend through 5 years for the "TGA3" exchanger.

5. CONCLUSIONS

In the paper a new design approach has been developed for finding the optimal shell and tube heat exchanger configuration including the optimization of its cleaning schedule. The optimization is based on the minimization of a total cost function including capital investment and operational expenses connected to fluid moving and the equipment periodical cleaning. This model is based on considering the maximum fouling resistance as one of the design parameters to be optimized. This automatically resolves the trade off implied by the choice of surface area, cleaning schedule and flow velocities. The model is also able to evaluate the kind of cleaning methods allowable for the designed heat exchanger. The proposed model is a useful addition to the existing design optimization methods which neglect fouling phenomena and periodical cleaning issues, and is more effective that traditional cleaning schedule optimization methods which are applied to an exchanger of predefined configuration. In this method, in fact, the problem of finding a geometric configuration of the equipment is solved simultaneously considering the impact that the equipment architecture has on the fouling process and the resulting cleaning requirements.

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