HEAT EXCHANGER DESIGN OPTIMIZATION UNDER STOCHASTIC OPERATING CONDITIONS

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Abstract. *In this paper a procedure for optimal design of shell and tube heat exchangers working under variable, deterministic or stochastic, operating conditions is proposed. The methodology adopts a genetic algorithm to determine the best equipment architecture and design parameters in order to optimize a user-defined objective function including capital investment, operating costs, value of the transferred heat, and penalties for unmet specifications. The objective function is computed factoring in, for any specific design configuration, the actual equipment performances obtained during off design operations caused by deterministic time trends, or stochastic variations of process parameters according to predetermined probability density functions. The performance improvement obtained when passing from steady state design hypothesis to variable or stochastic operating conditions are then discussed resorting to some numerical examples.*

Keywords*: heat exchanger design, genetic algorithm, stochastic operating conditions.*

1. INTRODUCTION

Owing to the wide utilization of heat exchangers in industrial processes their cost minimization and the maximization of thermal performances 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 recent times a renewed interest in the optimal design of heat exchangers has been thus witnessed in the literature. This corresponds to 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 (Babu and Munawar 2007; Caputo, Pelagagge and Salini 2008a; Hilbert, Janiga, Baron and Thevenin 2006; Ponce-Ortega, Serna-Gonzalez and Jimenez-Gutierrez 2009; Tayal, Fu and Diwekar 1999).

Nevertheless, while a number of approaches have been proposed to automate and optimize the heat exchangers design, a number of issues still remain to be solved. For instance, most of computer aided optimal design procedures assume steady state operations thus neglecting either deterministic and stochastic variability in the operating conditions. The uncertainty in the heat transfer coefficients correlations is also neglect, and penalties for off-design performances are usually not included in the evaluation of economic objective functions.

Haseler, Owena and Sardesai (1983) as well as Clarke, Vasquez, Whiting and Greiner (2001) investigated the sensitivity of the uncertainties in the overall heat exchanger calculations to uncertainties in individual fluid properties, but they did not consider the effects of uncertainties in heat exchanger geometry or in process specifications, and did not addressed any optimization problem. Taylor, Hodge and James (1999) used uncertainty analysis to determine bounds on the predicted performance parameters in thermal systems while Bernardo, Pistikopoulos and Saraiva (2001) discussed the incorporation of robustness criteria in process equipment design problems under uncertainty. Cho (1986), instead, presented a statistical-based method for sizing a heat exchanger based on the probability or confidence level that it will meet its intended thermal-hydraulic duty but, again, without an optimization approach.

Therefore, the problem of optimal sizing of heat exchangers working under stochastic operating conditions still remains to be solved, even if optimization under uncertainty is a relevant research field (Sahinidis 2004) even in the area of process equipment design.

In order to contribute to a solution of this problem, in this paper a procedure for optimal design of shell and tube heat exchangers working under variable deterministic or stochastic operating conditions is proposed. The methodology adopts a genetic algorithm to determine the best equipment architecture and design parameters in order to optimize a user-defined objective function including capital investment, operating costs, value of the transferred heat, and penalties for unmet specifications. The objective function is computed factoring in, for any specific design configuration, the actual equipment performances obtained during off design operations caused by deterministic time trends, or stochastic variations of process parameters according to predetermined probability density functions.

The paper is organized as follows. At first a procedure for the optimal design of heat exchangers operating at constant nominal operating conditions is briefly resumed building on earlier work of the authors (Caputo, Pelagagge and Salini 2008a). Then, a sensitivity analysis is performed to assess the performance variability of a heat exchanger optimized for some nominal operating conditions when working under off design conditions. Afterwards, some new objective functions are proposed suitable for optimal sizing under variable operating conditions. Subsequently, the previous algorithm is modified to allow the heat exchanger optimization when variable but known operating conditions occur. Finally, an extension to the above algorithm is made in order to account for equipment optimization even under stochastic operating conditions. The effect of changing the objective function or the changes in the optimal equipment configuration when passing from steady state hypothesis to variable or stochastic operating conditions are then discussed. Finally, in order to verify the capability of the proposed method, some numerical examples are also presented showing that significant benefits are obtained when an exchanger optimized accounting for off-design conditions is compared to a similar exchanger optimized only for static average operating conditions.

2. HEAT EXCHANGER OPTIMIZATION FOR STATIONARY OPERATING CONDITIONS

In a previous work (Caputo, Pelagagge and Salini 2008a) a detailed computer model was developed for optimal design of heat exchangers operating in stationary conditions, resorting to an optimization procedure based on genetic algorithm.

The procedure includes the following steps.

- Computation 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 with a design routine based on the Kern (1950) procedure. 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 hot and cold fluids (M_h, M_c), as well as the inlet and outlet temperatures of the hot fluid T_{hi}, T_{ho}, and cold fluid, T_{ci}, T_{co}, the remaining parameter being determined by an energy balance. All of the above process parameters are assumed to remain constant during the exchanger operation. 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.

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. When the Kern procedure is adopted the VIP are the shell inside diameter D_s, tube outside diameter do, and baffles spacing B. In a subsequent work (Caputo, Pelagagge and Salini 2009a) the Bell-Delaware design method (Shah and Sekulic 2003) has been implemented and the selected VIP become the inside shell diameter, the tube outside diameter, the central baffle spacing L_{bc} (m), the extremal baffle spacings L_{bi} and L_{bo}, the pitch ratio $L_{ptRatio}$, the baffle cut B_c, the sealing strips number N_{ss}, the tube layout angle θ_{tp} and the tube pass number N_{tp}.

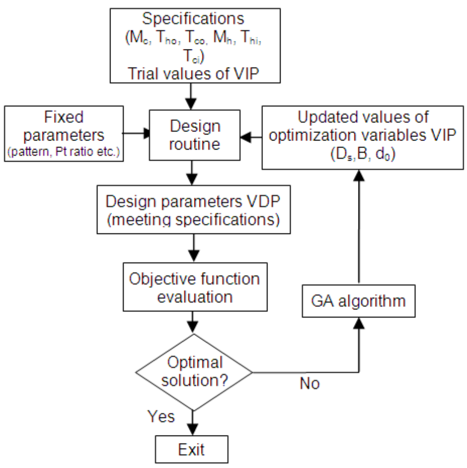

Figure 1. Original optimized design algorithm.

The remaining heat exchanger's design features (i.e. the dependent design variables (VDP)) are then directly computed from the VIP according to the selected design procedure. In particular the shell side and tube side heat exchange coefficients h_s, h_t, the overall heat transfer coefficient (U), the overall heat exchange area S, the number of tubes N_t, the shell and tubes length L and tube side and shell side flow velocities vs and vt, as well as the baffles number are determined, 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 (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 on the flow chart of Figure 1. More details of the design procedure and the optimisation algorithm are given elsewhere (Caputo, Pelagagge and Salini 2008a,b; 2009a).

The adopted objective function is the total present cost C_{tot} (€) sum of the capital investment C_I and the total discounted operating cost related to pumping power to overcome friction losses

$$C_{tot} = C_I + \sum_{k=1}^{ny} \frac{C_o}{(1+i)^k} \tag{1}$$

$$C_o = P \cdot C_E \cdot H \tag{2}$$

where C_E (€/kWh) is the electric energy cost, H (hr/yr) the annual operating hours, P (W) the pumping power, i the annual interest rate (%/yr), ny (yr) the equipment life. P depends from the equipment pressure drop and, in turn, from the selected exchanger geometry, see Caputo, Pelagagge and Salini (2008a) for details. The capital investment depends from the heat exchange area and equipment configuration. It can be estimated according to simplified correlations (Taal, Bulatov, Klemes, Stehlik 2003) such as Hall's equation

$$C_I(€) = 8000 + 259.2\, S^{0.91} \tag{3}$$

However, when a detailed cost optimization is sought, more precise cost estimation techniques are required (Caputo, Pelagagge and Salini 2008b; 2009b).

This approach was tested with reference to four literature case studies (Caputo, Pelagagge and Salini 2008a). In all of the examined cases operating costs were drastically cut and significant percent total cost reductions were obtained respect 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% to –52.6%, thus confirming the effectiveness of the proposed approach.

In a subsequent work (Caputo, Pelagagge and Salini 2009a) the model was extended to allow simultaneous optimization of both the heat exchanger architecture and the cleaning schedule in order to minimize the life cycle cost, sum of the capital investment, the present worth of the pumping cost for overcoming friction losses, and cleaning costs occurring over the equipment life. The following objective function was adopted

$$C_{tot} = C_I + \sum_{k=1}^{ny} \frac{C_o(1+f)^k}{(1+i)^k} + \sum_{k=1}^{ny} \frac{N_{clean,k}\, C_{clean,k}}{(1+i)^k} + \sum_{j=1}^{v} \alpha_j\, P_j\, C_{fitt} \tag{4}$$

In equation (4) f is the energy cost inflation rate (%/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 which penalizes the cost of solutions violating one or more operating constraint imposed by user; α_i is a binary activation index for the *j-th* constraint (imposed by user: 0 if 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. In particular, by specifying the fouling resistance growth law, the T_{clean} value is computed as the time required to reach a maximum allowable fouling resistance $R_{f\ allowable}$ which is a further VIP set by the genetic algorithm. The total cost function is thus completely determined by specifying the constructive details of the heat exchanger and the allowable fouling resistance and the fouling growth law which is application-specific. It is worthwile to point out that the main feature of this modified model is to consider 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, thus, improves existing design optimization methods, which neglect fouling phenomena or periodical cleaning issues, and is more effective that traditional methods for cleaning schedule optimization which are applied to an exchanger of predefined configuration

According to this approach, in fact, the problem of finding a geometric configuration of the equipment is solved simultaneously with the cleaning requirements, considering the impact that the equipment architecture has on the fouling process. The effectiveness of this approach has been shown referring to a case study (Caputo, Pelagagge and Salini 2009a).

3. SENSITIVITY ANALYSIS

Before dealing with a design approach able to take into account variable operating conditions it can be useful to analyze the relationship between operating conditions and equipment performances. This is made to assess whether the extent of the interactions is significant or not. This can be carried out by simply changing the input parameters respect the values used to design the equipment and computing the resulting changes in the outlet temperatures and the amount of exchanged heat.

For sake of simplicity we consider the kerosene – crude oil exchanger denoted as case 2 in (Caputo, Pelagagge and Salini 2008a) having a duty of 1.44 MW. Variations of ±40% in the hot and cold fluid inlet temperatures, and the hot fluid flow rate have been imposed respect the nominal operating conditions of 199 °C, 38.5 °C and 5.5 kg/s respectively. It was found that when the hot fluid inlet temperature changed ±40% the exchanged heat changed linearly in the range ±75% while the hot fluid outlet temperature changed linearly in the ±85.5% range, thus confirming the strong sensitivity of output parameters to input variables. However, the mean logarithmic temperature difference between hot and cold streams, which acts as the heat transfer driving force, changed non linearly in the −42.46% to +34% range as shown in Figure 2.

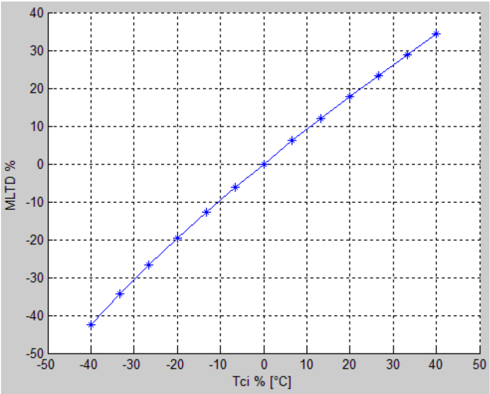

Figure 2: Percent variation of log mean temperature difference vs hot fluid inlet temperature.

When the cold fluid inlet temperature changed ±40% the same ±40% variation resulted in the exchanged heat (obviously the variations had opposite sign) while the cold fluid outlet temperature changed linearly in the ±20% range. Therefore, a lower sensitivity resulted in comparison to the previous case. The mean logarithmic temperature difference changed non linearly in the ±14% range instead.

Even the variation of the hot stream outlet temperature is non linear (+32.5%/−76%) with ±40% changes in the hot fluid flow rate as shown in Figure 3.

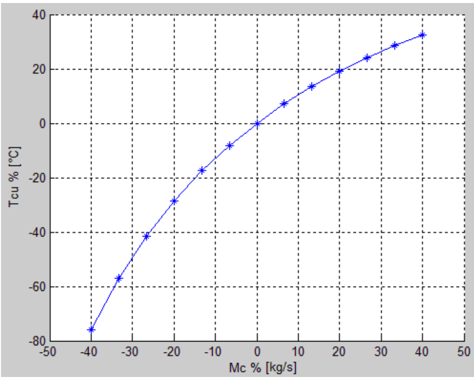

Figure 3: Percent variation of hot stream outlet temperature vs hot fluid flow rate percent change.

Changes in flow rate also determine variations in flow velocity and pressure drop (which is proportional to the square of fluid velocity) thus causing variations in operating costs in a strongly non linear fashion (+56%/−27% in operating cost variation respect ±40% changes in flow rate) as depicted in Figure 4.

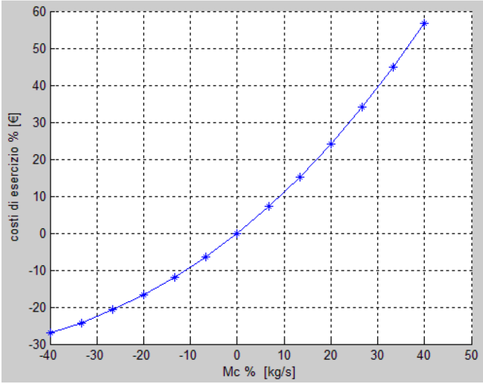

Figure 4: Percent variation of operating cost vs hot fluid flow rate percent change.

An increasingly non linear behavior occurs when two parameters are changed simultaneously. Figures 5 and 6 show the percent changes in exchanged heat and hot fluid outlet temperature respectively, when both hot fluid flow rate and inlet temperature are changed. In Figures 5 and 6 the points on the abscissa represent different combinations of the independent variables, which have been ordered as to have increasing values of the percent changes on the y axis.

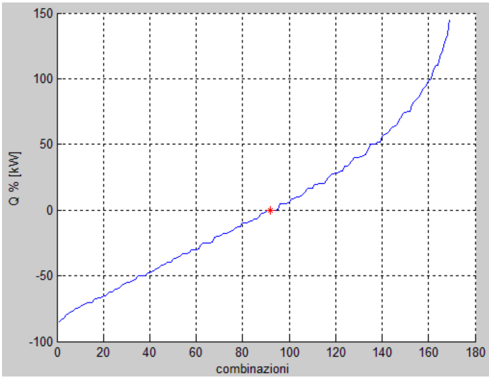

Figure 5: Percent variation of exchanged heat vs combined variation of hot fluid flow rate and inlet temperature.

Obviously, the percent variation of the output variable with combined variations of the inlet variables is much greater that the sum of the percent variations occurring when the inlet variables are changed separately, implying that effects cannot be summed.

Overall, this sample sensitivity analysis shows that even small changes in an input variable can have significant effects on the equipment output performances, often in non linear manner, and that simultaneous changes of some input variables lead to self amplifying effects. Therefore, given this non linearity and strong sensitivity, the actual variations of operating conditions cannot be neglected when designing a heat exchanger expected to operate in non stationary conditions. Moreover, a design based on average values will not be satisfactory, as same percent changes of opposite sign in input variables are likely to determine different effects. Finally, if changes above and below the average value have not the same probability of occurrence their impact can be even stronger.

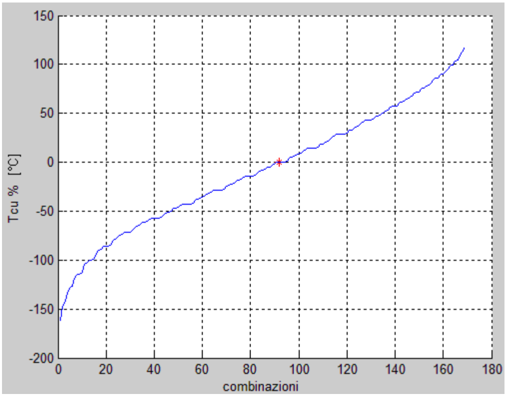

Figure 6: Percent variation of hot fluid outlet temperature vs combined variation of hot fluid flow rate and inlet temperature.

4. OPTIMAL DESIGN WITH DETERMINISTIC CHANGES IN OPERATING CONDITIONS

When an exchanger designed for stationary operation is used with changing operating conditions (i.e variable fluid temperatures and flow rates) its thermal performances are affected. This may cause a performance degradation that in some cases cannot be tolerated so that the entire process is upset. Additional costs may arise when the equipment fails to meet the specifications. In this respect, it is important to design a "robust" equipment, i.e. an exchanger sized in a manner that it is able to meet specifications even during the majority of foreseeable variations in operating conditions thanks to its scarce sensitivity to such changes.

From the perspective of this work it should be pointed out that available design routines in the literature always assume that streams properties are known, i.e. that values of mass flow rate of both streams and the inlet and outlet temperatures of the cold and hot streams are constant and known, being imposed by the designer. When one or more of such parameters are actually variable, instead, an average value is often assumed and it is treated as if it were the true value of that parameter, so that the design procedure can be carried out in the traditional manner. This approach is questionable because if the response to changes in operating conditions is nonlinear and when parameters variations are not symmetrical respect the nominal value, or excursions above or below the average value occur with different frequency or have a different economic impact, then to base the equipment design on an average value is not representative. In fact the effects of operating conditions above and below the nominal average conditions are not compensated. For instance, in heat recovery contexts when the hot stream increases its flow rate or inlet temperature above the nominal value, then a greater heat recovery follows which could represent an economic benefit. To exploit this benefit the designer could "oversize" the heat exchanger in order to fully benefit of this favourable off design condition. Nevertheless, if this condition only rarely occurs, then the added capital investment can not be fully offset and an optimal cost effective oversizing level should be sought. On the contrary, in different contexts, a heat recovery above the nominal requirement could be useless, while a heat recovery below the nominal level could cause shut down of the process if no auxiliary heat generator is installed.

Respect this traditional approach, in this work, instead, all of those streams properties which are actually constant are input to the optimization procedure as external specifications, while the parameters which are subject to operational variations are considered as VIP to be optimized. This is like using some "fictious" but constant input specification along with actual specifications in order to feed the design routine. The values of the "fictious" specifications are chosen by the genetic algorithm in order to optimize the objective function. Therefore, the heat exchanger is sized in a traditional manner but based on constant values of both real and fictious design specification, the latter being chosen in order to make the equipment as robust as possible to actual variations of the operating conditions. For instance, in case the cold fluid flow rate and the outlet temperatures are constant, but the hot fluid flow rate and the inlet streams temperatures change according to a deterministic trend, the values of M_c, T_{ho}, T_{co}, are input as external specifications, while the values of M_h, T_{hi}, and T_{ci}, are considered as optimization variables (VIP). At each iteration of the genetic algorithm some new trial constant values of M^*_h, T^*_{hi}, and T^*_{ci}, are generated and used as they were actual constant specifications by the sizing algorithm along with the known values of M_c, T_{ho}, T_{co}. After the candidate exchanger has been sized resorting to the computed values of the VDP, the actual time variation of M_h, T_{hi}, and T_{ci}, are used to simulate the operation of the equipment and to compute its operating performances and the objective function. The procedure is iterated until an optimum is found or a specified number of iterations is met. Actually, the design algorithm self-generates some of its (constant-valued) design specifications until the "optimal values" of such "fictious" specifications are found so that actual heat exchanger performance in the real time-varying operating conditions are maximized.

The flow chart of this modified algorithm is shown in Figure 7.

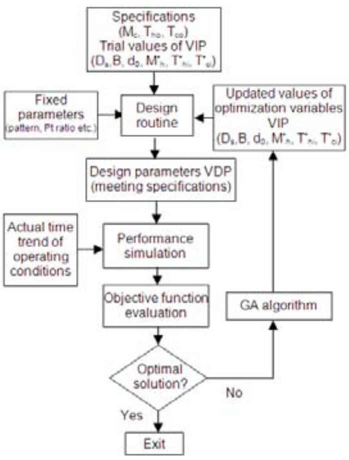

Figure 7. Optimized design algorithm for variable operating conditions.

In this revised optimization procedure the objective function is the total present cost, sum of the capital investment and the present worth of operating expenses including an Economic Performance Measure (EPM, €/yr) which accounts for economic value of the benefits or penalties coming from exceeding or failing to meet some of the specifications during the actual off design operation.

$$C_{tot} = C_i + \sum_{k=1}^{ny} \frac{\left(C_o + EPM\right)}{\left(1+i\right)^k} \quad (5)$$

While the definition of the EPM function depends from the specific problem at hand, in the following some sample formulations for a few representative scenarios have been hypothesized.

4.1 Scenario1

In this scenario the heat exchanger is a purely heat recovery unit and the higher the recovered heat the better. The EPM in this case is the economic value of the cumulative amount Q of recovered heat during the reference time interval,

$$EPM = -\,C_Q\,Q \quad (6)$$

where C_Q (€/kWh) is the thermal energy value. The minus sign indicates that this is a cost reduction contribution to the objective function representing the total cost to be minimized. A control of the hot stream flow rate can be enforced for temperature regulation and a upper limit to the hot stream outlet temperature can be included.

4.2 Scenario 2

In this case the heat exchanger has not the role of a heat recovery but has to maintain a process stream above a preset temperature limit. There is no use in going above this threshold temperature, but when the equipment fails to meet this specification and the controlled stream temperature falls below the threshold then a penalty (i.e. a fixed cost) C_F applies. In this case

$$EPM = C_F\,n \quad (7)$$

where n is the number of times that the temperature constraint is violated during the reference time interval.

4.3 Scenario 3

In this case a stream has to be heated above a given threshold so that when the controlled stream temperature falls below the preset threshold T_T a penalty applies, while the higher is the temperature above the threshold the better because fuel consumption for auxiliary heating is avoided. The EPM is a combination of the previous cases

$$EPM = C_F\,n - C_Q\,Q \quad (8)$$

where $Q = 0$ if $T < T_T$.

4.4 Scenario 4

This is the same scenario of the previous case except that when the stream temperature falls below the threshold value an auxiliary heat generator can be switched on, bearing a specific cost C_{AUX} (€/kWh), to supply the heat amount Q_{AUX} required to meet the specification. Therefore,

$$EPM = C_{AUX}\,Q_{AUX} - C_Q\,Q \quad (9)$$

As an application example, reference will be made to the previously mentioned heat exchanger, having nominal specifications M_c= 12.3 kg/s, M_h = 5.5 kg/s, T_{hi} = 199 °C, T_{ho} = 93 °C, T_{ci} = 38.5 °C, and T_{co} = 77 °C. Respect the nominal values, a deterministic variation pattern of inlet stream temperatures and hot stream flow rate is hypothesized as shown in Figure 8, while the other three variables are assumed to remain at their specified constant values. The proposed algorithm, implemented in the Matlab computational environment, was utilised to size an optimized exchanger in each of the above scenarios.

As expected, given the changes in the objective function according to each scenario, the optimization algorithm sized the exchangers in different manners, so that a specific configuration was obtained for each of the considered scenarios (see Table 1).

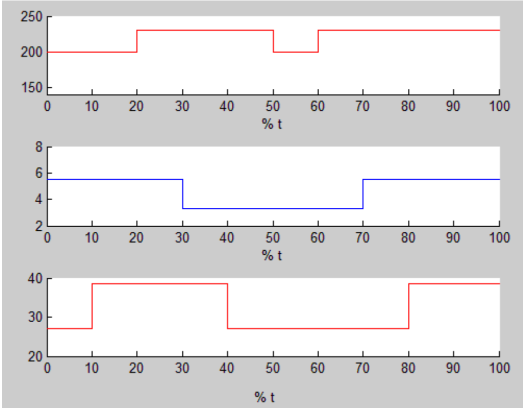

Figure 8. Deterministic variation of M_h, T_{hi} and T_{ci} operating conditions.

Table 1: Comparison of optimal exchangers designs

Design parameter	Scenarios			
	1	2	3	4
Shell diameter D_s (m)	0.62	0.63	0.62	0.62
Length L (m)	7.9	12.3	13.1	7.6
Baffles spacing B (m)	0.08	0.14	0.14	0.07
Tubes diameter d_o (m)	0.046	0.046	0.046	0.046
Tubes number N_t	37	37	37	37
Tube side velocity (m/s)	1.06	1.06	1.06	1.06
Heat transfer coefficient tube side h_t (W/m^2 K)	573	573	573	573
Pressure drop tube side ΔPt (Pa)	11910	17950	19100	11510
Shell side velocity (m/s)	0.41	0.43	0.45	0.45
Heat transfer coefficient shell side h_s (W/m^2 K)	543	557	572	574
Pressure drop shelside ΔPs (Pa)	28810	28050	32450	38170
Overall heat transfer coefficient (W/m^2 K)	250	253	256	256
Heat transfer area S (m^2)	46	72.7	77	45
Capital investment C_I (€)	17502	21413	22125	17231
Discounted sum of operating costs C_{oD} (€)	2203	3556	3987	2385
Total cost C_{tot} (€)	19705	24969	26112	19616

Results are computed assuming ny = 10 years, i = 10%/yr, C_E = 0.12 €/kWh, H = 7000 hr/yr. In order to highlight the difference in the equipment architecture, economic values in Table 1 refer only to capital investment and pumping energy expenses but not to the economic value of the performances (i.e EPM function).

The comparison of economic performance measures is made, instead, in Table 2. The Table compares the performances of exchangers sized for variable operating conditions (VOP), i.e. according to this work approach, and exchangers designed for nominal operating conditions (NOP), i.e. according to the method described in Section 2. In all cases the performances are computed referring to the actual variable operating conditions for a generic 8 hours daily shift. However, the performance measures are expressed in relative terms respect a benchmark represented as an optimal exchanger designed for the nominal operating conditions and working in those same stationary operating conditions. In greater detail, the Table shows the Economic Daily Performance EDP (€/day) which is the added cost (negative value) resulting from temperature threshold violation or expense for auxiliary energy, or the energy saving (positive value) respect a reference exchanger operating at the same constant nominal conditions. EDP is computed neglecting any capital investment or pumping cost as happens for EPM. Also shown is the overall daily exchanged heat Q (kWh/day) respect the nominal case of constant operating conditions. In this case a negative value represent less recovered heat respect the exchanger operating in stationary conditions while a positive values represents an increased heat recovery.

Table 2: Comparison of performance measures

Scenario	EDP (VOP)	EDP (NOP)	Q (VOP)	Q (NOP)
Scenario 1	393	163	614	255
Scenario 2	0	-7000	-59.2	255
Scenario 3	354	163	533	255
Scenario 4	0	-6836	57	-255

Table 2 shows that exchangers sized for variable operating conditions show better performances than exchangers sized for nominal operating conditions when the actual operating conditions change forcing the equipment to operate off design.

5. OPTIMAL DESIGN WITH STOCHASTIC CHANGES IN OPERATING CONDITIONS

In this case the variations of the operating parameters are assumed to be random instead of deterministic, and are generated by specifying a probability distribution for each of the variable operating parameters. As we are only interested in computing average or cumulative performance values over a long time interval, there is no need to generate random time histories of the operating conditions and the equipment life can be approximated as a sequence of stationary states. Each state is defined by a combination of one specific random value for each one of the variable parameters, and the state probability is computed as the product of the probability of occurrence of each parameter's value (i.e here the operating parameters are assumed to be independent).

If the number of variable parameters is N and each parameter can assume M distinct and independent values, then the number of possible states is M^N. Over an arbitrary time interval of length τ the *k-th* state, having probability of occurrence p_k, holds for an overall duration of $p_k\,\tau$. If in the *k-th* state a thermal power P_k is exchanged, then the overall exchanged heat is

$$Q_{TOT} = \sum_{k=1}^{M^N} P_k\, p_k\, \tau \tag{10}$$

This kind of approach can be extended to all other performance measures of interest and allows the computation of the objective function value with any specified probability distribution of operating parameters. The design optimization approach then follows the same flow chart of Figure 7, with the exception that the performance simulation with a specified deterministic time trend is substituted by a performance simulation over the entire set of possible system states, and the results for each state are weighted according to the states' probability of occurrence as specified by the user-defined frequency distributions of the operating parameters.

In order to show an application example of this further model, the same heat exchanger as before will be considered. However, the following three operating parameters are assumed to be randomly variable, namely the hot fluid flow rate and the inlet temperatures of both streams.

At first it is assumed that the above variables are uniformly distributed in the following ranges: M_c (3.3 to 7.2 kg/s), T_{hi} (150 to 230 °C), T_{ci} (27 to 50 °C). The sizing and economic data resulting from the optimization procedure are shown in Table 3.

Table 3: Comparison of optimal exchangers designs

Design parameter	Scenarios			
	1	2	3	4
Shell diameter D_s (m)	0.63	0.62	0.61	0.63
Length L (m)	8.16	12.75	9.52	9.09
Baffles spacing B (m)	0.08	0.15	0.08	0.09
Tubes diameter d_o (m)	0.047	0.047	0.047	0.047
Tubes number N_t	37	36	35	37
Tube side velocity (m/s)	1.07	1.10	1.13	1.07
Heat transfer coefficient tube side h_t (W/m^2 K)	574	587	600	574
Pressure drop tube side ΔPt (Pa)	12242	19499	15626	13518
Shell side velocity (m/s)	0.41	0.41	0.43	0.39
Heat transfer coefficient shell side h_s (W/m^2 K)	542	542	552	528
Pressure drop shell side ΔPs (Pa)	29541	24339	35673	26702
Overall heat transfer coefficient (W/m^2 K)	250	253	258	247
Heat transfer area S (m^2)	48	73	53	54
Capital investment C_I (€)	17725	21486	18500	18577
Discounted sum of operating costs C_{oD} (€)	2262	3549	2834	2383
Total cost C_{tot} (€)	19987	25035	21334	20960

Afterwards, it is assumed that the above variables have a Gaussian distribution with the following parameters: M_c (μ=5.5 kg/s, σ= 0.73 kg/s), T_{hi} (μ=199 °C, σ=26 °C), T_{ci} (μ=38.5 °C, σ=5.1 °C). The sizing and economic data resulting from the optimization procedure are shown in Table 4. Both Tables show that different stochastic behaviours of the operating conditions and different applications scenarios lead to different optimal configurations for heat exchangers having the same nominal operating conditions. As before, In Tables 3 and 4 only economic values related to capital investment and pumping costs are included.

Table 4: Comparison of optimal exchangers designs

Design parameter	Scenarios			
	1	2	3	4
Shell diameter D_s (m)	0.62	0.62	0.63	0.63
Length L (m)	12.92	11.88	11.62	12.92
Baffles spacing B (m)	0.19	0.12	0.14	0.17
Tubes diameter d_o (m)	0.047	0.047	0.047	0.047
Tubes number N_t	36	36	37	37
Tube side velocity (m/s)	1.10	1.10	1.07	1.07
Heat transfer coefficient tube side h_t (W/m^2 K)	587	587	574	574
Pressure drop tube side ΔPt (Pa)	19745	18244	16991	18775
Shell side velocity (m/s)	0.36	0.38	0.42	0.38
Heat transfer coefficient shell side h_s (W/m^2 K)	503	523	549	517
Pressure drop shellside ΔPs (Pa)	14968	24940	25119	18636
Overall heat transfer coefficient (W/m^2 K)	244	248	252	245
Heat transfer area S (m^2)	74	68	69	76
Capital investment C_I (€)	21627	20760	20813	21923
Discounted sum of operating costs C_{oD} (€)	3161	3100	3254	3212
Total cost C_{tot} (€)	24788	23861	24067	25135

Table 5 reports, instead, the corresponding values of performance measures, i.e. EDP (€/day) and overall daily exchanged heat Q (kWh/day). As previously made, the Table compares the performance measures for exchangers sized for variable operating conditions (VOP) and exchangers designed for nominal operating conditions (NOP), in this case both operating in stochastic conditions, and computed referring to the benchmark exchanger designed and operated at nominal conditions. Positive values of the performance measures indicate an improvement respect the benchmark exchanger. Results confirm that in this case too, heat exchangers designed taking into account of stochastic operating conditions outperform exchangers optimized for nominal operating conditions but operated in variable conditions.

Table 5: Comparison of performance measures

Uniform probability distribution				
Scenario	EDP (VOP)	EDP (NOP)	Q (VOP)	Q (NOP)
Scenario 1	291	60	454	95
Scenario 2	0	-285	-222	95
Scenario 3	290	60	454	95
Scenario 4	243	13	454	95
Gaussian probability distribution				
Scenario	EDP (VOP)	EDP (NOP)	Q (VOP)	Q (NOP)
Scenario 1	123	106	193	166
Scenario 2	0	-16	23	168
Scenario 3	114	106	179	166
Scenario 4	133	106	207	167

6. CONCLUSIONS

Heat exchangers usually operate in variable operating conditions. This makes the design process more complex because standard sizing procedures are based on assuming constant values of the reference operating parameters (stream temperatures and flow rates). To overcome this problems design specifications are often set referring to average value of the expected operating conditions. Nevertheless, this assumption often leads to unsatisfactory performances when excursions of the operating parameters above or below the nominal value have different impacts or different frequency.

Moreover, non linear behaviors occur. In this paper a method for optimal design of heat exchanger working in variable operating conditions has been described building on an earlier model for optimal design of heat exchangers in stationary operation. The model accounts for both deterministic and stochastic variations of the operating conditions, freely imposed by the user. In both cases it has been shown that exchangers sized for variable operating conditions have superior performances respect the corresponding exchangers sized referring to nominal average specifications but operated under variable conditions.

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8. RESPONSIBILITY NOTICE

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