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Design Optimization of Shell and Tube Heat Exchangers for Cost and Pressure Drop by Genetic Algorithm (GA)

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***Abstract*—Shell and tube type heat exchangers are one of the most commonly used heat exchangers for wide range of industrial applications. Being cost effective and energy efficient are the prime objective while designing the heat exchangers. The genetic algorithms are very effectualtool for the optimization of these parameters. Genetic algorithms can be used for optimization in various fields. Optimization of shell and tube heat exchangers is one such case. In this study three shell and tube heat exchangers encompassing varied uses are optimized for single objective functions of minimizing total cost and pressure drops by Genetic algorithm. These results in significant cost reductions compared to the original cases. Shell inside diameter, tube outside diameter and baffle spacing is used as the constraint variables. The results are also analysed for sensitivity against fluctuations in electricity prices.**

***Index Terms*— Shell and tube, Heat exchangers, Genetic algorithm, Optimization.**

I. Introduction

Among the variety of heat exchangers available, Shell and tube heat exchangers occupy a central role in the heat transfer process due to its wide spread use and ease of construction. It is the most widely used heat exchanger. For most practical applications the design of shell and tube heat exchangers can be limited to selection procedure from available catalogues. But in many cases it is required to design a heat exchanger that would meet specific needs. Various methods for designing a shell and tube heat exchangers are available like Kern method and Bell-Delaware method [1].Bell-Delaware method takes a larger number of parameters into consideration for the design. These designs are not necessarily the optimal ones for the required applications and would need further transformations in case optimum design is desired. Many studies have been carried out to optimize a shell and tube heat exchanger for various objectives using different methods and approaches to optimize. The most basic optimization of heat exchangers would be a single variable parametric study with prime focus on only the effects of varying a single parameter on the objective [2]. Other multi-parameter studies have also been carried out [3]. In the past literature authors have based their studies in optimizing for economic objective functions like minimizing investment cost, operating cost and total cost of heat exchange r [4]. Others have based their studies in Thermodynamic objective functions like

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maximizing heat transfer rate, minimizing entropy generation [5, 6] etc. Some of the studies have even attempted for a thermo-economic optimization like ratio of performance of heat exchanger to its cost [7]. Multi-objective optimizations with simultaneous optimization of two or more objectives mentioned above have been extensively carried out [8-10]. Genetic algorithm [11], particle swarm optimization (PSO) [12] and artificial bee colony algorithm (ABC) [13] are extensively used for designing and optimization of Shell and tube heat exchangers. From the literature study it is clear that the optimized design of shell and tube type heat exchanger exhibits better performance as compared to the existing industrial design. The parameters as shell inside diameter, tube outside diameter and the baffle spacing are the prime component of the design; and for getting the optimal performance, all three parameters has to be optimized. Further, among the available methods having different optimization algorithms, Genetic algorithm (GA)can be a suitable option for the heat exchanger applications, particularly for shell and tube type. In present research, the design of existing shell and tube type heat exchangers case studies have been optimized by means of genetic algorithmfor the objectives of minimum cost and pressure drop. Each objective is optimized as a single objective function. This study has been conducted considering fixed electricity cost, an additional analysis for sensitivity to changing electricity price to optimized parameters have also been performed. The thermodynamic modelling of shell and tube heat exchanger based on Kern methodology used to design the heat exchanger is presented along with brief explanation on genetic algorithm. The thermal mathematical model for the shell and tube type heat exchanger and the anatomy of genetic algorithms considered for the case studies has been elaborated in succeeding sections.

II. Mathematical model of Shell and tube Heat exchanger

The energy balance in Shell and tube heat exchanger is given by the equation

$$Q = \left(\dot{m}C_p\right)_t\left(T_{t,i} - T_{t,o}\right) = \left(\dot{m}C_p\right)_s\left(T_{s,o} - T_{s,i}\right) \tag{1}$$

where$Q$ is the heat duty of the heat exchanger, T is the temperature for the respective shell and tube inlets and outlets represented by subscripts s and t for shell side and tube side respectively and *i*and *o* for inlet and outlet. C_p is the specific heat capacity.

Number of tubes is calculated as given [14], where K_1 and n_1 are constants depending on the number passes and tube arrangement and D_s and d_o are the shell inner diameter and tube outside diameter.

$$n = K_1\left(\frac{D_s}{d_o}\right)^{n_1} \tag{2}$$

The tube pitch and inner tube diameter can be calculated as follows[15]

$$P_t = 1.25 d_o \text{ And } d_i = d_o - 2t \qquad (3),\ (4)$$

The heat transfer surface area A can be calculated by LMTD method in the following manner

$$A = \frac{Q}{U \Delta T_{lm} F} \tag{5}$$

where LMTDΔT_{lm} and correction factor F are calculated by Eq.(6),(7) [16]

$$\Delta T_{lm} = \frac{(T_{s,i} - T_{t,o}) - (T_{s,o} - T_{t,i})}{\ln((T_{s,i} - T_{t,o})/(T_{s,o} - T_{t,i}))} \tag{6}$$

$$F = \begin{cases} 1 & Tubepass = 1 \\ \frac{\sqrt{R^2+1}}{R-1} \cdot \frac{\ln\left(\frac{1-P}{1-PR}\right)}{\ln\left[\frac{2-P\left(R+1-\sqrt{R^2+1}\right)}{2-P\left(R+1+\sqrt{R^2+1}\right)}\right]} & Tubepass even number \end{cases} \tag{7}$$

Where, $R = \frac{(T_{s,i} - T_{s,o})}{(T_{t,o} - T_{t,i})}$ and $P = \frac{(T_{t,o} - T_{t,i})}{(T_{s,i} - T_{t,i})}$ (8), (9) U is the overall heat transfer coefficient and given by Eq.(10) [17]

$$U = \left[\frac{1}{h_t}\left(\frac{d_o}{d_i}\right) + R_t\left(\frac{d_o}{d_i}\right) + \frac{d_o \ln\left(\frac{d_o}{d_i}\right)}{2k_w} + R_s + \frac{1}{h_s}\right]^{-1} \tag{10}$$

Where ht and hs are the heat transfer coefficients and R_t and R_s are the fouling coefficients. K_w is the conductivity of the tube material.The heat transfer coefficient for a turbulent flow regime in the tube side is given by Eq.(11) [17]

$$h_t = 0.023\frac{k_w}{d_i} Re_t^{0.8} Pr_r^{\frac{1}{3}}\left(\frac{\mu_t}{\mu_{tw}}\right)^{0.14} \tag{11}$$

Where μ_t is the dynamic viscosity at bulk temperature of tube side and μ_{tw} is dynamic viscosity at wall temperature which is calculated by [15]

$$h_t(T_{tw} - T_{bt}) = U(T_{bs} - T_{bt}) \tag{12}$$

Where T_b is the bulk fluid temperature

Reynolds number is calculated by $Re_t = \frac{\rho_t v_t d_i}{\mu_t}$ (13)

The fluid velocity in tube side is calculated as $v_t = \frac{N_{pass}\dot{m}_t}{\rho_t n\pi(d_i^2/4)}$ (14)

Where, N_{pass} is the number of tube passes.

In a similar manner the shell side heat transfer is calculated by Eq.(15) [16],

$$h_s = 0.36\frac{k_w}{D_e} Re_s^{0.55} Pr_s^2\left(\frac{\mu_s}{\mu_{sw}}\right)^{0.14} \tag{15}$$

Where D_e is the shell hydraulic diameter, given by [16,17]

$$D_e = \frac{1.27}{d_o}(P_t^2 - 0.785d_v^2) \text{and} D_e = \frac{1.094}{do}(P_t^2 - 3.656d_o^2) \tag{16}$$

For square and triangular tube pitch respectively and μ_{sw} is evaluated at the outer tube wall temperature which is given by [15],

$$h_s(T_{bs} - T_{tw}) = U(T_{bs} - T_{bt}) \tag{17}$$

The shell side Reynolds number is given by $Re_s = \frac{\dot{m}_s D_e}{\mu_s A_s}$ (18)

As is the cross flow area on shell side given by [15, 16]

$$A_s = \frac{D_s B(P_t - d_o)}{P_t} \tag{19}$$

A. Pressure drop calculations are done by the following procedure [15],

The tube side pressure drop is given by

$$\Delta P_t = N_{pass}\left(\frac{4f_t L}{d_i} + 2.5\right)\frac{\rho_t v_t^2}{2} \tag{20}$$

Where f_t is the friction factor and is given by [15],

$$f_t = 0.046(Re_t)^{-0.2} \tag{21}$$

The pressure drop on shell side is given by [18],

$$\Delta P_s = f_s \left(\frac{\rho_s v_s^2}{2}\right)\left(\frac{L}{B}\right)\left(\frac{D_s}{D_e}\right) \tag{22}$$

f_s is the shell side friction factor and is calculate by following correlation [19],

$f_s = 2b_o Re_s^{-0.15}$, bo = 0.72 for Res< 40000. (23)

Total pumping power is calculated by [17],

$$P = \frac{1}{\eta}\left(\frac{\dot{m}_s \Delta P_s}{\rho_s} + \frac{\dot{m}_t \Delta \mathrm{P_t}}{\rho_t}\right) \tag{24}$$

$\eta$is the pump efficiency and considered as 70% in this study.The total cost is calculated as,

$$C_{tot} = C_i + C_{od} \tag{25}$$

Where C_i *and* C_{od} are capital investment cost and the discounted operating cost of the heat exchanger these costs are given by following equations [20]

$$C_i = a_1 + a_2 A^{a3} \tag{26}$$

$$C_{od} = \sum_{k=1}^{my} \frac{C_o}{(1+i)^k} \tag{27}$$

The heat exchangers in this case are considered to be made of stainless steel, the values for constants in cost calculations are a_1=8000, a_2=259.2 and a_3=0.91. The interest rate per annum is i=10% and the number of operational years are 10.The annual operating cost is calculated by [20],

$$C_o = PC_E H \tag{28}$$

Where C_E is the cost of unit energy, here taken as 0.12€/kWh and the operational hours in a year H are taken as 7000h/annum.

Table I. CASE STUDIES SPECIFICATIONS

	Mass flow rate(kg/s)	T_{in}(C)	T_{out}(C)	P(kg/m³)	C_p(kJ/kgK)	µ(Pas)	k_w(W/mK)	$R_{Fouling}$(m²K/W)
Case 1								
Shell Side: Methanol	27.8	95	40	750	2.84	0.00034	0.19	0.00033
Tube side: Sea water	68.90	25	40	995	4.20	0.00080	0.59	0.00020
Case 2								
Shell Side: Kerosene	5.52	199	93.3	850	2.47	0.00040	0.13	0.00061
Tube side: Crude Oil	18.80	37.8	76.7	995	2.05	0.00358	0.13	0.00061
Case 3								
Shell Side: distilled water	22.07	33.9	29.4	995	4.18	0.00080	0.62	0.00017
Tube side: raw water	35.31	23.9	26.7	999	4.18	0.00092	0.62	0.00017

III. GENETIC ALGORITHM

The Genetic Algorithm GA was used to optimize the cost and pressure drop in each case. The ga-Genetic Algorithm toolbox in MATLAB was used for this purpose. The population was set at 1000.

A. Case studies

The following three case studies have been chosen from the literature for analysis in this study.
Case 1: A methanol-brackish water exchanger is taken as study from [21]. The heat exchanger consists of two tube passes with triangular pitch and one shell pass. And heat duty of 4.34MW.
Case 2: A kerosene Crude oil heat exchanger from [16] is taken which has 4 tube pass with square pitch and one shell pass and Heat duty of 1.44MW.
Case 3: A distilled water- water heat exchanger is taken from [16] with two tube pass in triangular pitch arrangement and one shell pass. It has a heat duty of 0.46MW.

B. Design parameters, constraints and objective function

Two objective functions are considered separately for minimization in this study, namely the minimization of total cost along with investment cost and operational cost and minimization of pressure drop. The layout for the tube arrangements is either square or triangular based on the case study. The number of passes are also based on the case study values.

The constraints put in the optimization are as follows, The Shell diameter D_s rages from 0.1m to 1.5m. Tube outside diameter d_o value ranges from 0.015 to 0.051 and Baffle spacing is varying from 0.05m to 0.5m.

Objective functions

Objective 1: Minimizing Cost $\sum_{i=1}^{N_s} \sum_{j=1}^{N_p} C_{tot,ij}(D_{sij}, d_{oij}, B_{ij})$

Objective 2: Minimizing Pressure drop $\sum_{i=1}^{N_s} \sum_{j=1}^{N_p} P_{tot,ij}(D_{sij}, d_{oij}, B_{ij})$

IV. RESULTS AND DISCUSSION

Case 1: Methanol-Brackish water exchanger with two tubes side pass has 13.87% reduction in the total cost of the heat exchanger of which investment cost was only slightly reduced but the operating cost was halved compared to the original cost. The optimum dimensions for shell diameter remained same while the tube diameter and baffle spacing changed from 0.02m to 0.015m and 0.356m to 0.499m respectively. This was accompanied by a reduction in heat exchanger length too. The pressure drop minimization saw the pressure loss on shell side reduce to 3312Pa and on tube side to 264Pa. The optimum dimensions correspond to the extremes of the constraint values the tube diameter is 0.015m shell diameter 1.5m and baffle spacing 0.5m.

Table II. COMPARISON OF COST OPTIMIZATION BY GENETIC ALGORITHM WITH ORIGINAL STUDY

	Case1		Case 2		Case 3	
	Original	This study	Original	This study	Original	This study
D_s	0.894	0.847	0.539	0.835	0.597	0.595
d_o	0.02	0.015	0.025	0.051	0.015	0.015
B	0.356	0.499	0.127	0.05	0.5	0.5
L	4.748	3.0513	5.983	8.55	5.904	1.7282
N	918	1830	158	88	160	839
Re_t	14939	10250	8468	6375	40207	9963.1
h_t	3878	3924	1086	290	9799	4172
ΔP_t	5880	4972.5	53195	3782.8	65657	4186.2
A_s	0.05883	0.0845	0.01344	0.0084	0.0217	0.0595
D_e	0.0142	0.0107	0.02469	0.0505	0.01349	0.0107
Re_s	19739	10302	25344	83397	17155	4938.3
h_s	1903	1774	978.9	922	6186	3950
ΔP_s	37733	12626	25344	191420	88520	5367.8
U	634.3	660	268.1	165	1230	1014.4
A	273.7	263.13	74.21	120.553	56.35	68.327
C_{tot}	64127	55229	29974	37994	49751	22077

Case 2: The Kerosene-Crude oil heat exchanger with four tube passes had an increase of 26.75% in total cost. The corresponding dimensions for shell tube and baffle spacing were 0.835m, 0.051m and 0.05m respectively. The minimum pressure drop on shell side and tube side were around 96Pa and 237 Pa respectively, this condition saw an increase in dimensions which lead to even higher costs for both investment and operation.

TABLE III. OPTIMIZATION FOR MINIMUM PRESSURE BY GENETIC ALGORITHM

	Case1	Case 2	Case 3
D_s	1.5	1.5	1.5
d_o	0.015	0.021	0.015
B	0.5	0.5	0.5
L	1.3084	1.8167	0.5509
N	6460	2477	6460
h_t	1431	120	815
ΔP_t	264.61	237.09	47.3464
D_e	0.0107	0.0208	0.0107
h_s	1294	281	2376
ΔP_s	3312	96.8743	779.7
U	436	67	413.3
A	398.31	296.88	167.69
C_{tot}	69275	54137	35553
P_{tot}	3576	329	826

Case 3: The distilled water-raw water exchanger with two tube passes saw a large decrease of 55.62% in the total cost in optimizing for the cost. While the investment cost saw a rise, the operational costs saw a drastic reduction bringing the overall cost down. The minimum pressure drops were found to be 779Pa and 47Pa on shell side and tube side respectively. The total pressure drops on both sides being 826Pa.

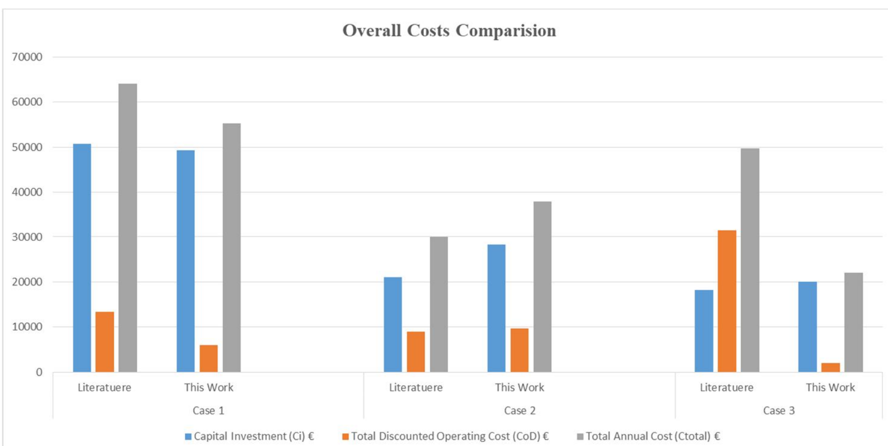

Figure 1. Cost distribution in each case

Further, the relation of baffles spacing, tube outer dia. And shell inside dia. With total system cost has been evaluated for three different rates of electricity costsas 0.06€/kWh, 0.12€/kWhand 0.18€/kWh. It is obvious that the system total cost is proportional to the cost of electricity, although the rate of proportions differs for all three parameters. As Fig. 2 depicts, the total cost is much higher for lower baffle spacing of 0.2 m and it reduces drastically on the increment of spacing. After a certain decrement, total cost gets saturated and will no longer reduce marginally for further increment in baffle spacing. From Fig. 3, it can clearly be observed that the tube outer dia. does not have much significant impact on the total cost of the system. Fig. 4 presents that the total cost reduces marginally with increment in shell diameter up to 0.9 m. Further, increasing in the shell dia. lead to minute increment in total cost and also the impact of electricity cost gets diluted at higher shell diameters.

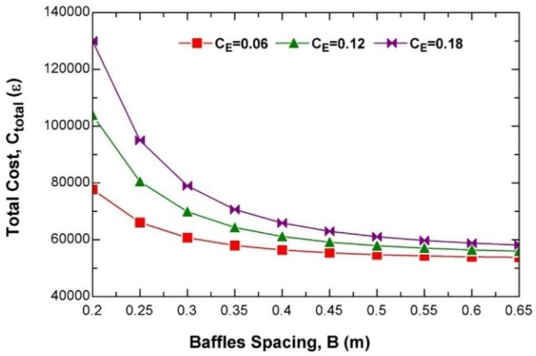

Figure 2. Cost distribution in each case

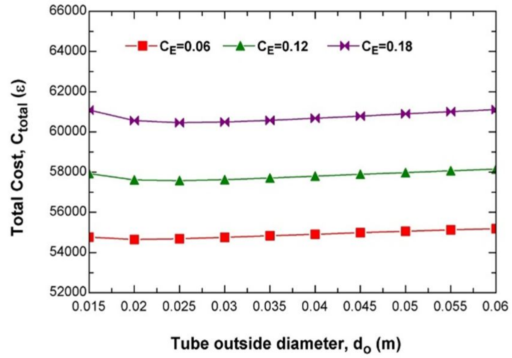

Figure 3. Cost distribution in each case

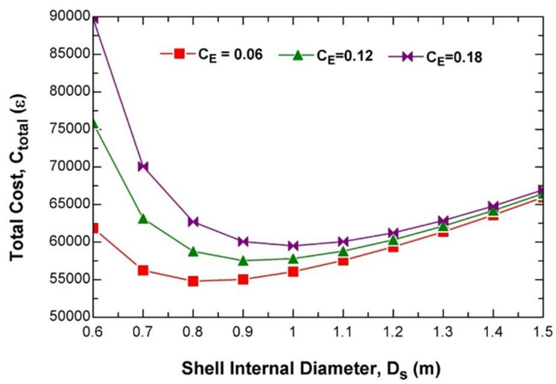

Figure 4. Cost distribution in each case

V. Sensitivity to changing electricity prices

The optimized data for case 1 was analysed to see the effect of fluctuating electricity prices on the cost and optimum parameters. Three different prices of €0.12, €0.06 and €0.18 per kWh of electricity were considered. €0.12 was taken as standard for all optimization while the €0.06 €0.18 are taken as 50% decrement and increment in electricity costs for sensitivity analysis. When the electricity costs are higher the operating costs are higher hence the optimization algorithm tries to reduce the pressure losses which contribute to operating costs. This is evident in the results with rise in costs for higher electricity price and lower costs for lower electricity price.

Table IV. Sensitivity to Electricity Cost variation

	Case1		
C_E	0.06	0.12	0.18
D_s	0.763	0.847	0.917
d_o	0.015	0.015	0.015
B	0.461	0.499	0.5
L	3.5623	3.0513	2.6876
N	1456	1830	2180
Re_t	12910	10250	4604.4
h_t	4720	3924	3412
ΔP_t	8568.5	4972.5	3272.7

A_s	0.0703	0.0845	0.0917
D_e	0.0107	0.0107	0.0107
Re_s	12379	10302	9496.8
h_s	1963	1774	1697
ΔP_s	20189	12626	10336
U	712	660	629
A	243.913	263.13	276.09
C_{tot}	51482	55229	57835

VI. CONCLUSION

The optimization by means of genetic algorithm presents better results than the original case and returns higher reduction in costs and pressure drops for respective objective functions. The algorithm also responds correctly to the changing electricity prices. The costs have maximum reduction of around 55% in the Case 3, the pressure drops in all three cases have been drastically reduced but it has been accompanied by rise in large number of tubes, which can be a constraint in actual manufacturing. The influence of parameters like shell diameter, tube outer diameter and baffles spacing has been evaluated for variable electricity costs. Future,the sensitivity analysis on the changing electricity prices has also been executed. As a future scope, the study can be extended with optimizing pressure drop with more constraints like number of tubes, length etc. that can give a more realizable result.

APPENDIX A. NOMENCLATURE

A	heat transfer area (m^2)
a_1, a_2,a_3	constants
A_s	Cross flow area(m^2)
B	Baffle spacing (m)
C_i	Capital cost(€)
C_E	Energy cost(€/kWh)
C_o	Annual operating cost(€)
C_{od}	discounted operating cost(€)
C_{tot}	total cost (€)
D_e	Hydraulic diameter of shell(m)
f	Tube side friction factor (-)
H	annual operating time(h/year)
U	Overall heat transfer coefficient(W/m^2K)
K_1	constant
k_w	Thermal conductivity of tube material(W/mK)
L	Tube length(m)
$\dot{m}$	mass flow rate(kg/s)
n	number of tubes
n_1	Constant
n_y	life of the equipment(year)
P	Pumping power (W)
R	fouling factor(m^2K/W)
T	Temperature(K)
v_t	fluid velocity in tube(m/s)

Greek symbols

ΔT_{lm}	LMTD (K)
ΔP	Pressure drop(Pa)
μ	Dynamic viscosity(Pas)
ρ	Density(kg/m^3)
η	Pump efficiency

Subscripts

c	cold side
h	hot side
in	inlet
out	outlet
s	shell side
t	tube side
w	tube wall

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