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Thermal and Economic Optimization of Shell and Tube Type Heat Exchanger Using Genetic Algorithm

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Abstract:
In the present work the design of shell and tube type heat exchanger is optimized in terms of cost and heat transfer rate for triangular and square orientation of heat exchanger tubes using genetic algorithm. The results of genetic algorithms are compared with other optimization algorithms. As per the findings the genetic algorithm performs best optimization on both cost and heat transfer rate as compared to the other optimization techniques observed in the literature. Moreover, the triangular configuration of the tubes gives utmost heat transfer rate at lowest cost than the square configuration.

Keywords: *Shell and Tube; Heat Exchanger; Optimization; Genetic Algorithm*

I. Introduction

Heat exchangers (HE) are the devices used for transferring the heat energy from one substance to another. Among the various types of HEs, the shell and tube type heat exchangersare the most commonly used HEfor wide range of industrial applications. For shell and tube type heat exchangers, the cost of manufacturing and rate of heat transfer are the most important design parameters[1]. Enhanced heat transfer rate boosts the performance of thermal devices [2–4]. Further, different configurations like triangular and square orientation can be employed for the performance enhancement.

In the present work, the design of shell and tube type heat exchanger is optimized by minimizing the cost and maximizing the heat transfer rate. The triangular and square configuration of tube is also investigated. The Genetic Algorithm (GA) optimization technique is used to serve the purpose. The outcomes of GA are compared with other optimization techniques found in literature. The considered optimization techniques along with GA are Big Bang-Big Crunch (BB-BC), (FA), (BAT), Back-tracking Search Algorithm (BSA) and hybrid Back-tracking Search Algorithm - Sine-Cosine Algorithm (BSA-SCA). The mathematical model of the shell and tube type heat exchanger is presented in Section 2.The outcomes of the work are shown in Section 3 and finally, the conclusion is given in Section 4.

II. Mathematical Model

The schematic diagram of the basic shell and tube type heat exchanger is presented in Figure 1. The energy balance in Shell and tube heat exchanger is given by following expression,

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

where, Q is the heat duty of the heat exchanger, T is the temperature of respective shell and tube inlets and outlets represented by subscripts s and t for shell side and tube side respectively and I and

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o for inlet and outlet. $C_p$is the specific heat capacity.

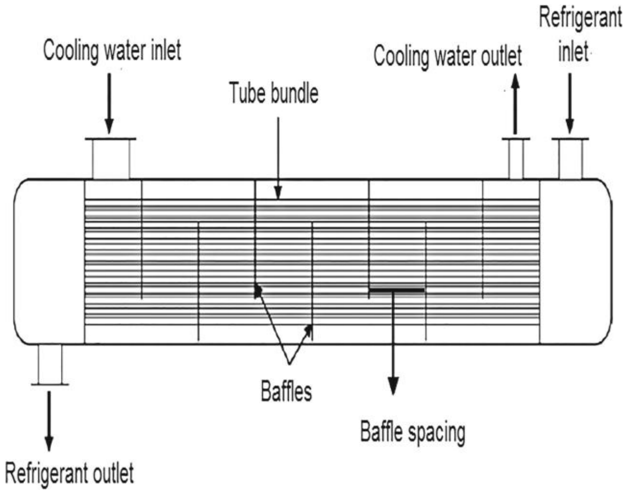

Figure 1. Schematic depiction of basic shell and tube type heat exchanger[5]

Number of tubes is calculated as given[6], 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.

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

The tube pitch and inner tube diameter can be calculated as follows [7],

$$P_t = 1.25 d_o \quad (3)$$

$$d_i = d_o - 2t \quad (4)$$

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

$$A = \frac{Q}{U \Delta T_{lm} F} \quad (5)$$

Where, Logarithmic Mean Temperature Difference ΔT_{lm} and correction factor F are calculated by Eq.(6),(7) [8],

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

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

Where,

$$R = \frac{(T_{s,i}-T_{s,o})}{(T_{t,o}-T_{t,i})} \quad (8)$$

$$P = \frac{(T_{t,o}-T_{t,i})}{(T_{s,i}-T_{t,i})} \quad (9)$$

U is the overall heat transfer coefficient and given by Eq.(10) [9],

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

Where h_t and h_s 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) [9],

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

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 [7],

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

Where T_b is the bulk fluid temperature

Reynolds number is calculated by

$$Re_t = \frac{\rho_t v_t d_i}{\mu_t} \quad (13)$$

fluid velocity in tube side is calculated as

$$v_t = \frac{N_{pass}\, \dot{m}_t}{\rho_t n \pi (d_i^2/4)} \quad (14)$$

where$N_{pass}$is the number of tube passes.

In a similar manner the shell side heat transfer is calculated by Eq.(15)[8],

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

Where D_e is the shell hydraulic diameter, given by [8,9],

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

for square and triangular tube pitch respectively and μ_{sw} is evaluated at the outer tube wall temperature which is given by[7],

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

The shell side Reynolds number is given by

$$Re_s = \frac{\dot{m}_s D_e}{\mu_s A_s} \quad (18)$$

A_s is the cross flow area on shell side given by [7,8],

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

Pressure drop calculations are done by the following procedure [7],

The tube side pressure drop is given by

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

Where $f_t$is the friction factor and is given by [7],

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

The pressure drop on shell side is given by [10],

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

f_s is the shell side friction factor and is calculate by following correlation [11],

$$f_s = 2 b_o Re_s^{-0.15}, \; b_o = 0.72 \text{ for } Re_s < 40000. \quad (23)$$

Total pumping power is calculated by [9],

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

$\eta$is the pump efficiency and considered as 70% in this study.

The total cost for heat exchanger is calculated as

$$C_{tot} = C_i + C_{od} \quad (25)$$

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 [12],

$$C_i = a_1 + a_2 A^{a3} \quad (26)$$

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

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 [12],

$$C_o = P C_E H \quad (28)$$

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 7000hr/annum.

The triangular and square configurations of the tubes inside the heat exchanger shell used for the design optimization are shown in Figure 2.

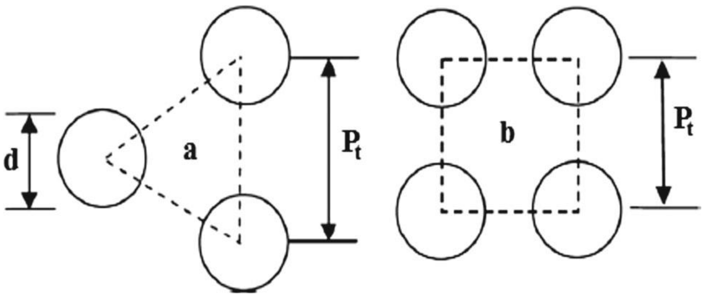

Figure 2. Triangle and square configuration of the tubes inside the heat exchanger[5]

III. Results

As elaborated in the introduction, the shell and tube type heat exchanger is optimized for lowest possible system cost and maximum possible heat transfer rate based on the mathematical model presented in Section 2. The optimization obtained through the GA with triangular and square configuration of tubes is compared with five different optimization algorithms as shown in

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Table 1 and Table 2. While minimizing the cost, the GA algorithm gives the lowest value of the cost in case of triangular orientation; whereas the cost for square orientation is also comparative, but a little higher. However, while maximizing the heat transfer rate, the GA algorithm gives superior results for both triangular andsquare tube configuration.

Table 1. Cost minimization using various optimization techniques

	BB-BC	FA	BAT	BSA	BSA-SCA	GA	GA
D_s (m)	0.1835	0.1786	0.1768	0.1771	0.1726	0.1670	0.1660
L (m)	3.7320	3.9192	4.0206	4.0261	4.1800	4.3964	4.4724
B (m)	0.3549	0.3692	0.3682	0.4102	0.3807	0.3870	0.3210
d_o (m)	0.01	0.01	0.0101	0.01	0.01	0.01	0.01
N (−)	1	1	1	1	1	1	1
Pitch configuration	Triangular	Triangular	Triangular	Triangular	Triangular	Triangular	Square
P_t (m)	0.0125	0.0125	0.0126	0.0125	0.0125	0.0125	0.0125
C_l (m)	0.0025	0.0025	0.0025	0.0025	0.0025	0.0025	0.0025
d_i (m)	0.0080	0.0080	0.0080	0.0080	0.0080	0.0080	0.0080
N_t (m)	131	124	120	122	115	107	106
A_s (m^2)	0.0130	0.0132	0.0130	0.0145	0.0131	0.0129	0.0107
N_b (m)	9	9	9	8	9	10	12
ΔP_t(Pa)	14943.270	17507.089	18879.873	18684.318	21755.399	30055	31081
D_e (m)	0.0071	0.0071	0.0071	0.0071	0.0071	0.0071	0.0099
R_{es} (−)	3055.9250	3013.0012	3062.4823	2731.6713	3017.6259	2855.4	4820
h_s (W/m^2K)	4736.5930	4705.1683	4732.6541	4466.0236	4719.2978	4645	4452
ΔP_s (Pa)	8867.2290	8443.8621	8527.2199	6340.7574	8234.9745	8696.2000	9775.3000
U (W/m^2 K)	968.7082	975.7089	980.2854	967.2231	988.5327	1011.3000	1003.5000
S (m^2)	30.8264	30.6052	30.4623	30.8737	30.2082	29.5572	29.7871
C_i (€)	13868.870	13830.538	13805.763	13877.071	13761.668	13649.000	13689.000
C_o (€/year)	84.1047	88.2321	92.0846	80.0157	97.6812	114.0567	121.8623
C_oD (€)	516.7868	542.1481	565.8203	491.6617	600.2088	700.8290	148.7911
C_{tot} (€)	**14385.650**	**14372.686**	**14371.583**	**14368.731**	**14361.873**	**14349**	**14437**

Table 2. Overall heat transfer coefficient maximization using various optimization techniques

	BB-BC	FA	BAT	BSA	BSA-SCA	GA	GA
D_s (m)	0.1937	0.1763	0.1817	0.1601	0.1539	0.1001	0.1040
L (m)	3.5409	4.6113	3.8651	4.6919	5.1358	9.7517	9.3752
B (m)	0.2403	0.2306	0.2219	0.2167	0.2290	0.2050	0.2085
d_o (m)	0.0102	0.0114	0.0100	0.0103	0.0104	0.0100	0.0100
N (−)	2	1	2	1	1	1	1
Pitch	Triangular	Triangular	Triangular	Triangular	Triangular	Triangular	square

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configuration							
P_t (m)	0.0126	0.0143	0.0126	0.0129	0.0130	0.0125	0.0125
C_l (m)	0.0025	0.0029	0.0025	0.0026	0.0026	0.0025	0.0025
d_i (m)	0.0080	0.0091	0.0080	0.0082	0.0083	0.0080	0.0080
N_t (m)	120	90	118	91	82	35	38
A_s (m^2)	0.0130	0.0081	0.0081	0.0069	0.0070	0.0041	0.0043
N_b (m)	9	18	16	20	21	46	43
ΔP_t (Pa)	18879.873	22525.689	37615.7	33859.65	44312.01	471190	392280
D_e (m)	0.0071	0.0081	0.0071	0.0073	0.0074	0.0071	0.0099
R_{es} (−)	3062.4823	5566.0388	4941.8090	5891.4110	5832.1680	8990.2	11845
h_s (W/m^2K)	4732.6541	5793.1969	6162.7510	6616.2201	6544.4040	8729	7299
ΔP_s (Pa)	8527.2199	32593.713	35507.180	49190.050	47853.340	177610	105520
S (m2)	30.4623	29.7583	28.7838	27.6266	27.2942	21.4452	22.3844
C_i (€)	13805.763	13683.541	13513.9	13311.83	13253.63	12218	12386
C_o (€/year)	92.0846	228.6388	281.5672	344.7214	363.6109	1994.6	1448.7
C_oD (€)	565.8203	1404.8864	1730.1080	2118.1640	2234.2310	12256	8901.8
C_{tot} (€)	14371.583	15088.427	15244.01	15429.99	15487.86	24474	21288
U (W/m^2K)	**980.2855**	**1003.4754**	**1037.4510**	**1080.9028**	**1094.0683**	**1393.9**	**1335.4**

IV. Conclusion

The optimization by genetic algorithm gives better results compared to other considered techniques in terms of both, cost minimization and heat transfer rate maximization. Here, the triangular and square orientation of the tubes is also investigated.The optimized total costs of the system obtained from GA technique are €14349 and €14437 for triangular and square configurations respectively. Further, the optimized values of heat transfer rate were observed to be 1393.9W/m^{2}K and 1335.4W/m^{2}K for triangular and square configurations respectively, which is significantly higher among other optimization methods. Hence, the Genetic Algorithm is the most suitable method for the design optimization of shell and tube type heat exchangers. Moreover, future studies in the field can be conducted on the optimization of pressure drop with all the constraints like number of tubes, length etc. that can give a more realizable result.

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