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Optimization by Using Bat Algorithm on Shell and Tube Heat Exchangers

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ABSTRACT
In this study, the key objective optimization on shell and tube heat exchanger is based on the total cost which includes operating cost and investment cost. The optimization is carried out using bat algorithm for the three case studies given in the open literature. Bats are the only mammals with wings and their behavior is based on the echolocation, which is helpful for them to detect prey, avoid obstacles, and locate their roosting crevices in the dark. The various design parameters are tube length, tube diameter, pitch size, baffle spacing, and number of tubes. The results are reported and compared with previous literature. It is found that the cost obtained from bat algorithm is quite matching with other algorithms. The objective is to optimize various dimensions of the shell and tube heat exchanger so as to get maximum heat transfer coefficient and minimum design cost. The method used is Kern for finding pressure drops and heat transfer coefficient. The number of iterations taken for finding the optimal solution for a single objective function is <20 iterations and this ensures less computational time.

Key words: *Heat exchanger, Bat algorithm, Pumping power.*

1. INTRODUCTION

A heat exchanger is a device that is used to transfer thermal energy between two or more fluid, between a solid surface and a fluid or between solid particulates and a fluid at a different temperature and in thermal contact shell and tube. An expansion joint is an important component in the case of fixed tubesheet exchanger for certain design conditions. The selection criteria for a proper combination of these components depend on the operation pressures, temperatures, thermal stress, corrosion characteristics of fluids, fouling, cleanability, and cost. Because of the desired heat transfer in the heat exchanger take place across the tube surface, the selection of tube geometrical variables is important from a thermal performance point of view. Tube size is specified by outside diameter and wall thickness. Smaller diameter tubes yield higher heat transfer coefficients and so result in a compact heat exchanger. However, larger diameter tubes are easier to clean, more rugged, and they are necessary when the allowable tube side pressure is small. Fora given surface area, the most economical exchange is possible with a small shell diameter and long tubes, consistent with the space and the availability of handling facilities [1,2].

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2. BAT ALGORITHM

The echolocation characteristics of microbats can develop various bat inspired algorithms. In the basic bat algorithm developed by Yang (2008) [3], the following approximate or idealized rules were used.

1. All bats use echolocation to sense distance, and they also know the difference between food prey and background barriers in some magical way.
2. Bats fly randomly with velocity v_i at position x_i with a frequency f_{min}, varying wavelength λ and loudness A_0 to search for prey. They can automatically adjust the wavelength of their emitted pulses and adjust the rate of pulse emission $r \in [0, 1]$ depending on the proximity of their target.
3. Although the loudness can vary in many ways, it can be assumed that the loudness varies from a large (positive) A_0 to a minimum constant value A_{min}.

3. MATHEMATICAL MODELS

The heat exchanger surface area is given by Kern [4], Sinnot *et al*. [5]:

$$A=\frac{Q}{U_o \Delta T_{lm} F} \tag{1}$$

Where Q is the heat load, U is the overall heat transfer coefficient, ΔT_{LM} is the logarithmic mean temperature difference, F is the correction factor.

The heat transfer rate is given by:

$$Q=m_s c_{ps}(T_{is}-T_{os})=m_t c_{pt}(T_{ot}-T_{it}) \quad (2)$$

$$U_o = \frac{1}{\frac{1}{h_s} + R_{fs} + \frac{d_0}{d_i}(R_{ft} + \frac{1}{h_t})} \quad (3)$$

3.1. Tube Side

$$h_t = 0.027 \frac{k_t}{d_i} Re_t^{0.8} Pr_t^{\frac{1}{3}} \left(\frac{\mu_t}{\mu_w}\right)^{0.14} \quad (4)$$

Where f_t is the Darcy friction factor given as follows [6]:

$$Pr_t = \frac{\mu_t C_{pt}}{k_t} \quad (5)$$

$$Re_t = \frac{\rho_t v_t d_i}{\mu_t} \quad (6)$$

Where Pr_t and Re_t re tube side Prandtl and Reynolds number.

$$v_t = \frac{m_t}{\frac{\pi d_i^2}{4}} \cdot \frac{N_p}{N_t} \quad (7)$$

Where Np is the number of passes and N_t is the number of tubes [5,7]:

$$N_t = k_1 \left(\frac{D_s}{d_0}\right)^{n_1} \quad (8)$$

Where k_1 and n_1 are coefficients that are taken values according to flow arrangement and number of passes.

3.2. Shell Side

$$h_s = 0.36 \frac{k_s}{D_e} Re_s^{0.55} Pr_s^{\frac{1}{3}} (\frac{\mu_t}{\mu_w})^{0.14} \quad (9)$$

$$D_e = \frac{4(p_i^2 - (\pi d_0^2 / 4))}{\pi d_0} \quad (10)$$

(for square pitch)

Where D_e is the shell hydraulic diameter and computed as given by Kern [4], Sinnot *et al*. [5]:

$$pr_s = \frac{\mu_s C_{ps}}{k_s} \quad (11)$$

$$Re_s = \frac{\rho_s v_s D_e}{\mu_s} \quad (12)$$

$$v_s = \frac{m_s}{A_s \rho_s} \quad (13)$$

Where v_s is the flow velocity for the shell side and can be obtained [4,5]:

$$a_s = D_s . B\left(1 - \frac{d_o}{P_t}\right) \quad (14)$$

Where A_s is the cross section area normal to flow.

3.3. Logarithmic Mean Temperature Difference

$$\Delta T_{lm} = \frac{(T_{si} - T_{to}) - (T_{so} - T_{ti})}{\ln((T_{si} - T_{to}) / (T_{so} - T_{ti}))} \quad (15)$$

3.4. Pressure Drop and Objective Function

In all the heat exchanger, there is close physical and economical affinity between heat transfer and pressure drop. For a constant heat capacity heat exchanger, increasing the flow velocity will cause a rise of heat transfer coefficient. However, an increase of flow velocity will cause more pressure drop which results in additional running cost.

$$\Delta P_t = \Delta P_{tubelength} + \Delta P_{tubeelbow}$$

$$\Delta P_t = \frac{\rho_t v_t^2}{2} \cdot \left(\frac{L}{d_i} f_t + p\right) . N_p \quad (16)$$

Different values of constant p are considered by different authors. Kern [4] assumed p=4 and Sinnot *et al*. [5] assumed p=2.5.

$$\Delta P_s = f_s \left(\frac{\rho_s v_s^2}{2}\right) \cdot \left(\frac{L}{B}\right) \cdot \left(\frac{D_s}{D_e}\right) \quad (17)$$

$$P = \frac{1}{n}\left(\frac{m_t}{\rho_t} \Delta P_t + \frac{m_s}{\rho_s} \Delta P_s\right) \quad (18)$$

$$C_{ii} = a_{11} + a_{22} A^{a33} \quad (19)$$

Where C_{ii} is the capital investment a_{11} = 8000, a_{22} = 259.2 and a_{33} = 0.91 for exchangers made with stainless steel for both shell and tubes [8].

$$C_{oo}=P.C_{ee}.H \tag{20}$$

$$C_{oDd} = \sum_{k=1}^{n_y} \frac{C_{00}}{(1+i)^k} \tag{21}$$

$$C_{tt} = C_{ii}+C_{oDd} \tag{22}$$

Where C_{tt} is total cost taken as the objective function, which includes capital investment (C_{ii}), energy cost (C_{ee}), annual operating cost (C_{oo}) and total discounted operating cost (C_{oDd}) [4].

4. RESULTS AND DISCUSSION

4.1. Case 1

A heat exchanger for distilled water and raw water with heat load is 0.415 MW. This heat exchanger has two tube side passages with triangle pitch pattern and one shell side passage.

The following upper and lower bounds for the optimization variables were imposed, shell internal diameter D_s ranging between 0.2 m and 1.2 m, tubes outside diameter d_o ranging from 0.015 to 0.05 m, baffles spacing B ranging from 0.05 to 0.5 m and tube length L ranging from 1 to 5 m. All values of discounted operating costs were computed with n_y=10 years, annual discount rate i=10%, energy cost C_e= 0.12 €/ kW h and an annual amount of work hours H =7000 h/ year similar to other researches. The operating and physical parameters of the heat exchanger are shown in Table 1.

Design parameters are assumed to solve this problem with BA with triangular pitch pattern. As can be seen in Table 2, the results obtained from BA are better than results of other algorithms. Heat exchanger area in BA method has been reduced comparison with other methods of heat exchanger design. Reduction of heat exchange area resulted in less number of tubes and reduction of the tube length, the number of tubes decreased as compared to reference GA, ABC, BBO and increased as a reference. The capital investment decreased correspondingly 15.84, 6.69, 2.72 and 2.67% in comparison with the original design, GA, ABC and BBO, respectively. The combined reduction of capital investment and operating costs led to a reduction of the total cost of about 21.68% in comparison with original design and 8.31, 5.13, 0.57 and 0.16% in comparison with GA, ABC, and BBO, respectively. Cost comparison of present approach and other methods is shown in Figure 1 for the case study 1. The number of iterations taken for convergence is shown in Figure 2 is less than five iterations.

Table 1: The operating and physical parameters of the shell and tube heat exchanger.

Variables	Case study	
	Shell side (distilled water)	Tube side (raw water)
m (kg/s)	27.80	68.9
T_i (°C)	95	25
T_o (°C)	40	40
P (kg/m^3)	750	995
C_p (kJ/kgK)	2.84	4.2
M (Pa s)	0.00034	0.0008
k (W/mK)	0.19	0.59
R_f (m^{2}k/W)	0.00033	0.0008

5. CONCLUSION

In this study, a solution method of the shell and tube heat exchanger design optimization problem was

Table 2: Case 1.

Variables	Original design	GA	ABC	BBO	Present work BA
D_s (m)	0.894	0.830	1.3905	0.801	0.7621
L (m)	4.830	3.379	3.963	2.040	1.531
B (m)	0.356	0.500	0.4669	0.500	0.382
d_o (m)	0.020	0.016	0.0104	0.010	0.0156
P_t (m)	0.025	0.020	-	0.0125	0.0196
C_l (m)	0.005	0.004	-	0.0025	0.0042
N_t	918	1567	1528	3587	1036
v_t (m/s)	0.75	0.69	0.36	0.77	1.081
Re_t	14,925	10,936	-	7642.497	16869.4
Pr_t	5.7	5.7	-	5.7	5.69
h_t (W/m^2 K)	3812	3762	3818	4314	5459.55
f_t	0.028	0.031	-	0.034	0.02729
ΔP_t (Pa)	6251	4298	3043	6156	8523.57
D_e (m)	0.014	0.011	-	0.007	0.0111
v_s (m/s)	0.58	0.44	0.118	0.46	0.635
Re_s	18,381	11,075	-	7254.007	15639.71
Pr_s	5.1	5.1	-	5.1	5.08
h_s (W/m^2 K)	1573	1740	3396	2197	2137
Fs	0.33	0.357	-	0.379	0.338
ΔP_s (Pa)	35789	13267	8390	13799	14041
U_o (W/m^2 K)	615	660	832	755	783
A (m^2)	278.6	262.8	-	229.95	221.73
C_{ii} (€)	51,507	49,259	44,559	44,536	43334
C_{oo} (€/year)	2111	947	1014.5	984	1166
C_{oDd} (€)	12973	5818	6233.8	6046	7165
C_{tt} (€)	64480	55077	50793	50582	50499

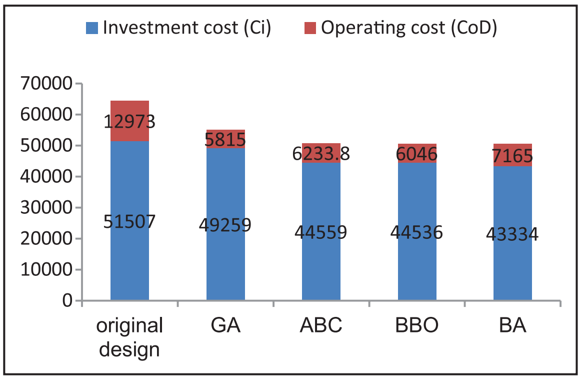

Figure 1: Overall costs comparison for case study 1.

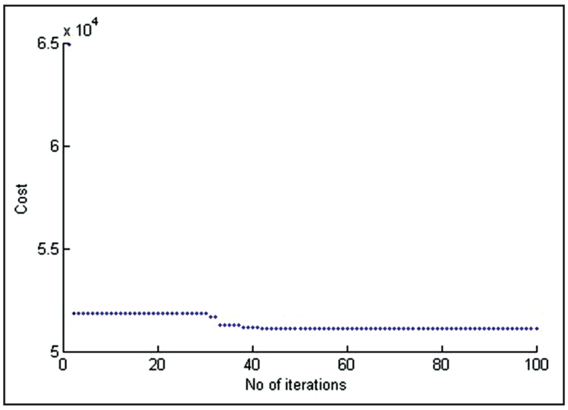

Figure 2: Convergence of BA for case study 1.

Nomenclature

a_{11}	Numerical constant	K_1	Numerical constant
		Q	Heat duty (W)
a_{22}	Numerical constant	Pr_t	Tube side Prandtl number
a_{33}	Numerical constant	Pr_s	Shell side Prandtl number
A	Heat exchanger surface area (m^2)	Re_s	Shell side Reynolds number
B	Baffle spacing (m)	Re_t	Tube side Reynolds number
C_l	Clearance	R_{fs}	Shell side fouling resistance (m^2K/W)
C_{ee}	Energy cost (€/kWh)	R_{ft}	Tube side fouling resistance (m^2K/W)
C_{ii}	Capital investment (€)	P_t	Tube pitch (m)
C_{oo}	Annual operating cost (€/year)	T_{is}	Shell side inlet fluid temperature (K)
C_{oDd}	Total discounted operating cost (€)	T_{os}	Shell side outlet fluid temperature (K)
C_p	Specific heat (kJ/kg K)	T_{it}	Tube side inlet fluid temperature (K)
C_{tt}	Total annual cost (€)	T_{ot}	Tubeside outlet fluid temperature (K)
D_e	Hydraulic shell diameter (m)	U_o	Overall heat transfer coefficient (W/m^2K)
d_i	Tube inside diameter (m)	v_s	Shell side fluid velocity (m/s)
d_o	Tube outside diameter (m)	v_t	Tube side fluid velocity (m/s)
D_s	Shell inside diameter (m)		
F	Temperature difference correction factor		
f_s	Shell side friction coefficient	Greek letters	
f_t	Tube side friction coefficient	ΔP	Pressure drop (Pa)
H	Annual operating time (h/year)	ΔT_{lm}	Logarithmic mean temperature difference (K)
h_s	Shell side convective coefficient (W/m^2K)	Π	Numerical constant
h_t	Tube side convective coefficient (W/m^2K)	P	Density (kg/m^3)
I	Annual discount rate (%)	μ	Dynamic viscosity (Pa s)
K	Thermal conductivity (W/mK)	η	Overall pumping efficiency
L	Tube length (m)	Subscripts	
m_s	Shell side mass flow rate (kg/s)	c	Cold stream
m_t	Tube side mass flow rate (kg/s)	e	Hydraulic
N_p	Number of tube passes	h	Hot stream
n_1	Numerical constant	i	Inlet
		o	Outlet
n_y	Equipment life (year)	s	Shell side
N_t	Number of tubes	T	Tube side
P	Pumping power (W)	W	Wall

proposed based on the utilization of a bat algorithm. Based on proposed method, a computer code was developed and three test cases were solved by it. Referring to the literature, the total cost is minimum compared to original design and it is quite matching with other algorithms. Furthermore, the BA algorithm allows for a rapid solution for the design problem. Because the number of iterations took for convergence is less than 10 and it saves the computational time. It enables to examine a number of alternative solutions of good quality, giving the designer more degrees of freedom in the final choice with respect to traditional methods. Hence, the present algorithm can be used for any optimization applications more confidently.

6. REFERENCES

1. Y. A. Kara, O. Guraras, (2004) A computer program for designing of shell and tube heat exchangers, ***Applied Thermal Engineering,*** **24:** 1797-1805.
2. A. L. H. Costa, E. M. Queiroz, (2008) Design optimization of shell-and-tube heat exchanger, ***Applied Thermal Engineering,*** **28:** 1798-1805.
3. X. S. Yang, (2008) ***Nature Inspired Metaheuristic Algorithm,*** 2nd ed. UK: Luniver Press.
4. D. Q. Kern, (1950) ***Process Heat Transfer,*** New York: McGraw-Hill.
5. R. K. Sinnot, J. M. Coulson, J. F. Richardson, (2005) Chemical engineering. In: ***Chemical Engineering Design,*** Vol. 6. Oxford: Butterworth-Heinemann.
6. G. F. Hewitt, (1998) ***Heat Exchanger Design Handbook,*** New York: Begell House.
7. R. K. Shah, K. J. Bell, (2000) ***Handbook of Thermal Engineering,*** Florida: CRC Press.
8. M. Taal, I. Bulatov, J. Klemes, P. Stehlik, (2003) Cost estimation and energy price forecast for economic evaluation of retrofit projects, ***Applied Thermal Engineering,*** **23:** 1819.

***Bibliographical Sketch**

I am Tharakeshwar T.K. Pursuing Ph.D. in Siddaganga Institute of Technology, Tumkur. I have 9 years of Teaching work experience in Mechanical Engineering Department, S.I.T, Tumkur. Currently the position occupied is Assistant Professor. The work carried out by me is a research work. Because it uses a novel computational technique called Bat algorithm. This algorithm is substantiated by the conclusion and its results. Finally, the Bat algorithm is compared with other algorithms to find out better optimal results. This article is nowhere published anywhere.