Economic Optimization of Shell and Tube Heat Exchanger Using Nanofluid

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Abstract—Economic optimization of shell and tube heat exchanger (STHE) is presented in this paper. To increase the rate of heat transfer, copper oxide (CuO) nanoparticle is added into the tube side fluid and their optimum results are compared with the case of without additive nanoparticle. Total annual cost (TAC) is selected as fitness function and nine decision variables related to the heat exchanger parameters as well as concentration of nanoparticle are considered. Optimization results reveal the noticeable improvement in the TAC and in the case of heat exchanger working with nanofluid compared with the case of base fluid (8.9%). Comparison of the results between two studied cases also reveal that the lower tube diameter, tube number, and baffle spacing are needed in the case of heat exchanger working with nanofluid compared with the case of base fluid.

Keywords—Shell and tube heat exchanger, nanoparticles additive, total annual cost, particle volumetric concentration.

I. INTRODUCTION

RECENTLY, many researchers have proposed methods of using a nanofluid as the working fluid in the heat exchangers [1]-[3]. The effect of Cu–H₂O nanofluids on the efficiency of a flat-plate solar collector was investigated experimentally by He et al. [4]. Many efforts have been made in the recent years of heat management about increasing the convective coefficient of the heat transfer fluid [5]. Heat transfer processes are widely used in numerous areas including heat exchanger [6]. For example, Ghozatloo et al. focused on developing higher convective heat transfer behavior of nanofluids through the STHE under laminar flow [7]. The heat-transfer characteristics of TiO₂–water nanofluids as a coolant in concentric tube heat exchanger were also presented by Khedkar et al. [8].

The main aim of this study is exploring the influence of nanoparticle CuO on economic optimization of STHE (Fig. 1). For this purpose, TAC is considered as objective function and nine decision variables including the nanoparticle concentration are selected in this regard.

II. THERMAL MODELING

Effectiveness for E type of shell is obtained as follows [9]:

$$\varepsilon = \frac{2}{(1+C^*)+\sqrt{(1+C^{*2})}\ \coth(\frac{NTU}{2}\sqrt{(1+C^{*2})})} \quad (1)$$

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where C^* and NTU are defined as:

$$NTU = \frac{U_o A_o}{C_{min}} \quad (2)$$

$$C^* = C_{min}/C_{max} \quad (3)$$

Here A_o and U_o are the total heat transfer surface area and overall heat transfer coefficient expressed as:

$$A_o = A_t = \pi L d_o N_t \quad (4)$$

$$U_o = \frac{1}{\frac{1}{h_o}+R_{o,f}+\frac{d_o\ln(d_o/d_i)}{2k_w}+\frac{d_o}{d_i}R_{i,f}+\frac{d_o}{d_i}\frac{1}{h_i}} \quad (5)$$

where L is the tube length, N_t is the tube number, d_o is the tube outside diameter is, $R_{o,f}$ is the shell side fouling factor, d_i is the tube inside diameter, k_w is the wall conductivity, and $R_{i,f}$ is tube side fouling factor. In addition, h_i and h_o are the convection heat transfer coefficients in tube and shell side defined in the following sub-sections:

A. Heat Transfer Coefficient and Friction Factor in Tube Side

In this study, additive nanoparticle in water and in tube side is used. Moreover, nanoparticles CuO have the following properties [10]:

$$\rho_{nf} = (1-\phi)\rho_{bf} + \phi\rho_{np} \quad (6)$$

$$c_{p,nf} = \frac{(1-\phi)(\rho c_p)_{bf} + \phi(\rho c_p)_{np}}{(1-\phi)\rho_{bf} + \phi\rho_{np}} \quad (7)$$

$$\frac{k_{eff}-k_f}{k_f} = 3.761\phi + 0.0179(T-273.15)-0.307 \quad (8)$$

$$\frac{\mu_{eff}}{\mu_f} = 1.475 - 0.319\phi + 0.51\phi^2 + 0.009\phi^3 \quad (9)$$

where ϕ is the particle volumetric concentration.

Nusselt number and friction factor for CuO nanofluid are

estimated as [11]:

$$Nu_{nf} = 0.065\left(Re_{nf}^{0.65} - 60.22\right)\left(1 + 0.0169\phi^{0.15}\right)Pr_{nf}^{0.542} \quad (10)$$

$$f_{nf} = 0.3164\,Re_{nf}^{-0.25}\left(\frac{\rho_{nf}}{\rho_{bf}}\right)^{0.797}\left(\frac{\mu_{nf}}{\mu_{bf}}\right)^{0.108} \quad (11)$$

Above correlations are valid for Reynolds number and particle volumetric concentration (PVC) in the range of 4000-16000 and 0-6% [11]. Furthermore, Re is the Reynolds number expressed as:

$$Re = \frac{4\dot{m}}{\pi d_i \mu}\left(n_p / N_t\right) \quad (12)$$

where $\dot{m}, n_p$, and μ are the mass flow rate, number of tube pass, and viscosity in tube side, respectively.

Using the Nusselt number, convection heat transfer coefficient is obtained as follows:

$$h_{nf} = \left(Nu_{nf} \times k_{eff}\right)/d_i \quad (13)$$

In addition, pressure drop in tube side is estimated as [9]:

$$\Delta P_t = \frac{\dot{m} \times n_p}{2\rho A_o^2}\left[\frac{4fL}{d_i} + \left(1 - \sigma^2 + K_c\right) - \left(1 - \sigma^2 - K_e\right)\right] \quad (14)$$

where ρ, K_i, K_o, and σ are the tube side density, tube inlet and outlet pressure drop coefficients as well as ratio of minimum free flow area to the frontal area, respectively. Moreover, A_o is the tube side flow cross section area in each pass estimated as:

$$A_o = \frac{\pi d_i^2}{4}\left(N_t / n_p\right) \quad (15)$$

B. Heat Transfer Coefficient and Pressure Drop in Shell Side

In this study, Bell-Delaware process is applied to obtain the heat transfer coefficient and pressure drop in the shell side [12]. In this method, an ideal heat transfer coefficient for cross flow stream in tube bundle is estimated and then corrected with some factors. Ideal heat transfer coefficient is estimated as [12]:

$$h_{id} = \left(j_s c_p \frac{\dot{m}}{A_s}\right)\left(\frac{k}{c_p \mu_s}\right)^{2/3}\left(\frac{\mu_s}{\mu_w}\right)^{0.14} \quad (16)$$

where c_p, $\dot{m}$, A_s, k and μ_s are the heat capacity, rate of mass flow, cross flow area, fluid thermal conductivity and

viscosity, respectively. In addition, $\mu_{s,w}$ is the viscosity of fluid in the shell side at wall temperature, and j_s is the Colburn factor obtained by [12]:

$$j_s = a_1\left(\frac{1.33}{p_t / d_o}\right)^a \left(Re_s\right)^{a_2} \quad (17)$$

$$a = \frac{a_3}{1 + 0.14\left(Re_s\right)^{a_4}} \quad (18)$$

$$Re_s = \frac{d_o\dot{m}}{\mu_s A_s} \quad (19)$$

$$A_s = \frac{D_s}{p_t}\left(p_t - d_o\right)BS \quad (20)$$

where p_t, Re_s, and BS are the tube pitch, the Reynolds number in the shell side, and the baffle spacing, respectively. Moreover, a_1-a_4 are constants which could be found in [12]. In addition, D_s is the shell diameter which is estimated as [12]:

$$D_s = p_t\sqrt{4CL \times N_t /\left(\pi \times CTP\right)} \quad (21)$$

where CL and CTP are the tube layout and the tube count constants respectively which are depend on tube arrangement and pass.

As it is mentioned, the ideal shell side heat transfer coefficient is corrected using some factors as follow [12]:

$$h_o = h_s = h_{id} J_c J_l J_b J_s J_r \quad (22)$$

where J_c, J_l, J_b, J_s, and J_r are respectively correction factors associated to baffle configuration, baffle leakage, bundle/pass bypass streams, inlet/outlet baffle spacing and adverse temperature gradient.

Friction factor in the Bell-Delaware method is estimated by [12]:

$$f_s = b_1\left(\frac{1.33}{p_t / d_o}\right)^b \left(Re_s\right)^{b_2} \quad (23)$$

$$b = \frac{b_3}{1 + 0.14\left(Re_s\right)^{b_4}} \quad (24)$$

where b_1-b_4 are constants and could be found in [12].

Finally, overall pressure drop in the shell side is obtained as:

$$\Delta P_s = \Delta P_{cr} + \Delta P_{i-o} + \Delta P_w \quad (25)$$

where ΔP_{cr}, ΔP_{i-o} and ΔP_w are the pressure drop in cross-flow, the inlet/outlet, and the window sections, respectively.

Details of calculating pressure drop, Colburn and friction factors are mentioned in [9], [12].

III. FITNESS FUNCTIONS, DECISION VARIABLES AND CONSTRAINTS

In this study, TAC is considered as objective or fitness function which is estimated as follow:

$$TAC\ (\$ / year) = a_f C_{in} + C_{op} \qquad (26)$$

$$C_{in} = d_1 + d_2\left(A_t^{d_3}\right) + C_{np} \qquad (27)$$

$$a_f = \frac{i}{1-(1+i)^{-n}} \qquad (28)$$

where a_f, C_{in}, C_{np}, i, and n are the annual factor, the capital cost, the nanoparticle cost, and the rate of interest, respectively. In addition, d_1-d_3 are constants, and C_{op} is operational cost related to the pumping and is approximated as:

$$C_{op} = \left(\left(\frac{\dot{m}}{\rho\eta_P}\Delta P\right)_t + \left(\frac{\dot{m}}{\rho\eta_P}\Delta P\right)_s\right)\tau\,\varphi_e \qquad (29)$$

where subscripts t and s show the tube and shell side streams, respectively. Furthermore, η_P, φ_e, and τ are the pump efficiency, the electricity tariff, and the system operational hours in a year, respectively.

In this study tube arrangement, tube diameter, tube pitch ratio (ratio between tube pitch and tube outside diameter), tube length, tube number, baffle spacing ratio (ratio between baffle spacing and shell inside diameter), baffle cut ratio (ratio between baffle cut and shell inside diameter), flow allocation as well as PVC are considered as nine decision variables.

The constraint is also considered as follows:

$$3 < L/D_s < 12 \qquad (30)$$

$$\varepsilon > 0.5 \qquad (31)$$

IV. CASE STUDY

In this paper, oil with mass flow rate of 8 kg/s and inlet temperature of 78.3 °C are considered as hot stream which is cooled by water with inlet temperature of 30 °C. In addition, CuO is selected as nanoparticle in the tube side of STHE. Furthermore, input parameters listed in Table I are selected for input date. In addition, density of CuO considered 3950 kg/m³, and the following correlation is used to estimate the CuO heat capacity as a function of temperature [13]:

$$c_{p,np} = -40.92 + (4.024\mathrm{T}) - (5.0048\mathrm{T}^2)10^{-3} \\ + (2.8852\mathrm{T}^3)10^{-6} - (6.2488\mathrm{T}^4)10^{-10} \qquad (32)$$

TABLE I
CONSTANTS CONSIDERED AS INPUT PARAMETERS IN THE OPTIMIZATION

Parameter	Value
Water fouling factor (m²W/K)	0.000074
Oil fouling factor (m²W/K)	0.00015
Rate of interest (-)	0.10
System life time (year)	10
d_1	8500
d_2	409
d_3	0.85
Operational hours in year (hour)	5000
Electrical tariff ($/kWh)	0.02
Pump efficiency (-)	0.6

V. RESULTS AND DISCUSSION

A. Optimization

Decision variables or design parameters as well as their range of variation are listed in Table II.

TABLE II
DECISION VARIABLES AND THEIR RANGE OF VARIATION

Parameter	Lower range	Upper range
Tube pattern (°)	30-45-90	
Tube inside diameter (mm)	10.3	17.3
Tube pitch ratio (p_t/d_o)	1.25	2
Tube length in each pass (m)	3	8
Number of tube (-)	100	600
Baffle spacing ratio (BS/D_s)	0.2	1.4
Baffle cut ratio (BC/D_s)	0.19	0.32
Allocation of cold stream flow	Shell or Tube side	
PVC (%)	0	6

The optimization is performed using Genetic Algorithm (GA) for two cases including without additive nanoparticle (base fluid) and with additive nanoparticle (nanofluid), and their results are compared. Variation of TAC versus generation for two cases of with/without additive nanoparticle using GA are shown in Fig. 1 for the water mass flow rate of 4 kg/s. Due to the semi stochastic performance of GA, each optimization procedure is performed for three times, and the best results are chosen. As it is shown in this figure, significant cost reduction is found in the case of additive nanoparticle compared with the case of without additive additive nanoparticle. As it is listed in Table III, 8.9% reduction is observed by using additive nanoparticle compared with the base fluid case.

The optimum values of decision variables for the results presented in Fig. 1 are listed in Table III. As it is illustrated in Table III, the same tube arrangement, tube pitch ratio, tube length and baffle cut ratio are selected in the both cases of with/without additive nanoparticle. On the other hand, the lower tube inside diameter, lower tube number, and lower baffle spacing ratio are chosen in the case of additive nanoparticle compared with the case of without additive nanoparticle. Finally, PVC=5.5% is selected in the case of

additive nanoparticle.

To generalize the optimization problem, the optimum annual costs are also obtained for two other cases including the water mass flow rate of 2 kg/s and 6 kg/s, and their results are presented in Figs. 2 (a), (b). 3.7% reduction in annual cost is observed by using additive nanoparticle in the case of water mass flow rate of 2 kg/s, while no significant improvement is observed in the case of water mass flow rate of 6 kg/s.
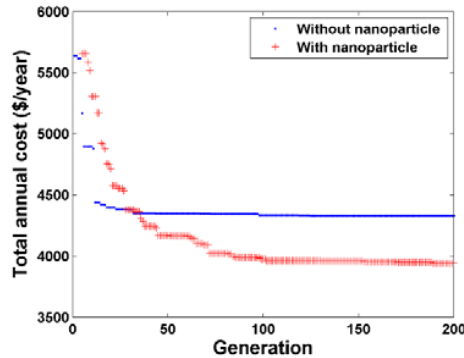

Fig. 1 Progress of objective function versus generation for two cases of without/with additive nanoparticle (m_w=4 kg/s)
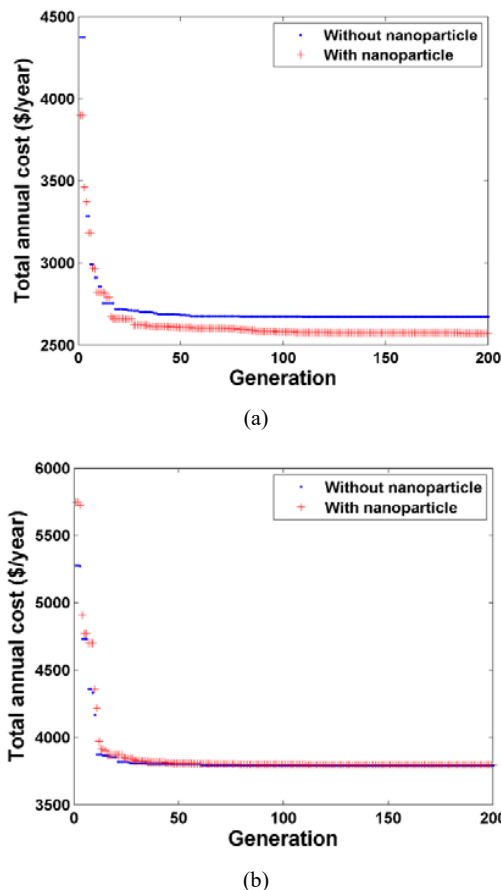

(a)

(b)

Fig. 2 Progress of objective function versus generation for two cases of without/with additive nanoparticle (a) m_w=2 kg/s, (b) m_w=6 kg/s

TABLE III
OPTIMUM VALUES OF DESIGN PARAMETERS IN THE CASES OF WITH/WITHOUT ADDITIVE NANOPARTICLE AND IN CASE OF M_w=4KG/S

Parameter	Without nano	With nano
Tube arrangement (°)	30	30
Tube inside diameter (mm)	10.8	10.3
Tube pitch ratio (p_t/d_o)	1.61	1.62
Tube length in each pass (m)	5.52	5.54
Tube number (-)	354	225
Baffle spacing ratio (BS/D_s)	0.55	0.48
Baffle cut ratio (BC/D_s)	0.25	0.26
Cold stream flow allocation (-)	Tube side	Tube side
PVC (%)	-	5.5
TAC ($/year)	4327.4	3941.3

VI. CONCLUSIONS

Economic optimization of STHE was presented in this paper. To increase the rate of heat transfer, CuO nanoparticle was added into the tube side fluid, and their optimum results were compared with the case of without additive nanoparticle. For this purpose, TAC was considered as objective function, and nine design parameters including tube arrangement, tube diameter, tube pitch ratio, tube length, tube number, baffle spacing ratio, baffle cut ratio, flow allocation as well as PVC were selected. The optimum result shows significant cost reduction (8.9%) in the case of additive nanoparticle compared to the case of without additive nanoparticle. In addition, the same tube arrangement, tube pitch ratio, tube length and baffle cut ratio were selected in the both cases of with/without additive nanoparticle. On the other hand, the lower tube inside diameter, lower tube number and lower baffle spacing ratio were chosen in the case of additive nanoparticle compared to the case of without additive nanoparticle. Finally, 5.5% nanoPVC was selected in the case of additive nanoparticle.

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