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Multi-objective optimization of shell and tube heat exchangers

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ABSTRACT

The effectiveness and cost are two important parameters in heat exchanger design. The total cost includes the capital investment for equipment (heat exchanger surface area) and operating cost (for energy expenditures related to pumping). Tube arrangement, tube diameter, tube pitch ratio, tube length, tube number, baffle spacing ratio as well as baffle cut ratio were considered as seven design parameters. For optimal design of a shell and tube heat exchanger, it was first thermally modeled using ϵ–*NTU* method while Bell–Delaware procedure was applied to estimate its shell side heat transfer coefficient and pressure drop. Fast and elitist non-dominated sorting genetic algorithm (NSGA-II) with continuous and discrete variables were applied to obtain the maximum effectiveness (heat recovery) and the minimum total cost as two objective functions. The results of optimal designs were a set of multiple optimum solutions, called 'Pareto optimal solutions'. The sensitivity analysis of change in optimum effectiveness and total cost with change in design parameters of the shell and tube heat exchanger was also performed and the results are reported.

1. Introduction

Shell and tube heat exchanger is widely used in many industrial power generation plants as well as chemical, petrochemical, and petroleum industries. There are effective parameters in shell and tube heat exchanger design such as tube diameter, tube arrangement, baffle spacing and baffle cut ratio. Some authors considered the cost of heat transfer surface area or capital investment as an objective function to be minimized [1,2]. While others considered the sum of investment (related to the heat transfer surface area) and operational (fluid head losses) costs as an objective function for optimizing a shell and tube heat exchanger [3–8]. The sum of entropy generation of streams as an objective function was also reported in [9–11]. Multi-objective optimization of total annualized cost and the amount of cooling water required for shell and tube heat exchanger was studied in reference [12]. Hilbert et al. [13] also, used a multi-objective optimization technique to maximize the heat transfer rate and to minimize the pressure drop in a tube bank heat exchanger. Liu and Cheng [14], optimized a recuperator for the maximum heat transfer effectiveness as well as minimum exchanger weight and pressure loss.

In this paper after thermal modeling of an industrial shell and tube heat exchanger (using ϵ–*NTU* method and Bell–Delaware approach for estimating the shell side heat transfer coefficient and pressure drop), the exchanger was optimized by maximizing the effectiveness as well as minimizing the total cost. Genetic algorithm optimization technique was applied to provide a set of Pareto multiple optimum solutions. The sensitivity analysis of change in optimum values of effectiveness and total cost with change in design parameters was performed and the results are reported.

As a summary, the followings are the contribution of this paper into the subject:

- Multi-objective optimization of shell and tube heat recovery heat exchanger was performed with effectiveness and total cost as two objectives (not selected in other available literature) using genetic algorithm.
- The tube arrangement, tube diameter, tube pitch ratio, tube length, tube number, baffle spacing ratio as well as baffle cut ratio were selected as design parameters (not selected as a group of variables in other available literature).
- A closed form equation for the total cost in term of effectiveness at the optimal design point was proposed. This equation can be modified without change in its procedure of deriving for any new input values.
- Sensitivity analysis of change in objective functions when the optimum design parameters vary was performed.

2. Thermal modeling

The heat exchanger effectiveness for our selected E type TEMA shell and tube heat exchanger was estimated from [15]:

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Nomenclature

$A_{o,t}$	tube side flow area per pass (m^2)
A_t	total tube outside heat transfer area (m^2)
A_s	cross flow area at or near the shell centerline
BC	baffle cut (m)
c_p	specific heat in constant pressure (j/kg K)
C_{min}	minimum of C_h and C_c (W/K)
C_{max}	maximum of C_h and C_c (W/K)
C^*	heat capacity rate ratio (C_h/C_{max})
C_{in}	Total investment cost ($)
C_{op}	Total operating cost ($)
C_o	annual operating cost ($/yr)
C_{total}	total cost ($)
CL	tube layout constant (–)
CTP	tube count calculation constant (–)
d_i	tube side inside diameter (m)
d_o	tube side outside diameter (m)
D_s	shell diameter (m)
f	friction factor (–)
h_i	tube side heat transfer coefficient (W/m^2 K)
h_o	shell side heat transfer coefficient (W/m^2 K)
i	annual discount rate (%)
j	Culburn number (–)
K_c	entrance pressure loss coefficient (–)
K_e	exit pressure loss coefficient (–)
k_{el}	price of electrical energy ($/kWh)
k	thermal conductivity (W/m k)
L	tube length (m)
L_{bc}	baffle spacing (m)
m	mass flow rate (kg/s)
ny	equipment life (yr)
n_p	number of tube pass (–)
N_t	number of tube (–)
NTU	number of transfer units (–)
p_t	tube pitch (m)
P	pumping power (W)
Pr	Prandtl number (–)
$R_{o,f}$	fouling resistance shell side (m^2 K/W)
$R_{i,f}$	fouling resistance tube side (m^2 K/W)
Re	Reynolds number (–)
T	temperature (°C)
U	overall heat transfer coefficient (W/m^2 K)

Greek abbreviation

ϵ	thermal effectiveness (–)
Δp	pressure drop (pa)
μ	viscosity (pa s)
η	pump efficiency (–)
τ	hours of operation per year (h/yr)
σ	ratio of minimum free flow area to frontal area (–)

Subscripts

s	shell-side
t	tube side
w	tube wall
i	inner or inlet
o	outer or outlet

$$\epsilon = \frac{2}{\left(1+C^*\right)+\left(1+C^{*2}\right)^{0.5}\coth\left(\frac{NTU}{2}\left(1+C^{*2}\right)^{0.5}\right)} \tag{1}$$

where the heat capacity ratio (C^*), and the number of transfer units (NTU), are defined as:

$$NTU_{max} = \frac{U_o A_t}{C_{min}} \tag{2}$$

$$C^* = \frac{C_{min}}{C_{max}} = \frac{\min(C_s, C_t)}{\max(C_s, C_t)} = \frac{\min\left((\dot{m}c_p)_s, (\dot{m}c_p)_t\right)}{\max\left((\dot{m}c_p)_s, (\dot{m}c_p)_t\right)} \tag{3}$$

where A_t is the total tube outside heat transfer surface area and U_o is the overall heat transfer coefficient which are computed from:

$$A_t = \pi L d_o N_t \tag{4}$$

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

where L, N_t, d_i, d_o, $R_{i,f}$, $R_{o,f}$, k_w are tube length, tube number, tube inside and outside diameter, tube and shell side fouling resistances and thermal conductivity of tube wall respectively.

2.1. Tube side

The tube side heat transfer coefficient (h_i) was estimated from [15]:

$$h_i = h_t = (k_t/d_i)0.024\mathrm{Re}_t^{0.8}\mathrm{Pr}_t^{0.4} \quad \text{for } 2500 < \mathrm{Re}_t < 1.24\times 10^5 \tag{6}$$

where k_t and Pr_t are tube side fluid thermal conductivity and Prandtl number, also Re_t is tube flow Reynolds number which is defined as:

$$\mathrm{Re}_t = \frac{m_t d_i}{\mu_t A_{o,t}} \tag{7}$$

where m_t is the mass flow rate and $A_{o,t}$ is the tube side flow cross section area per pass estimated as:

$$A_{o,t} = 0.25\,\pi\, d_i^2 N_t/n_p \tag{8}$$

and n_p is the number of tube passes.

Furthermore, the tube side pressure drop was also estimated from [15]:

$$\Delta p_t = \frac{G^2}{2\rho_i}\left[\left(1-\sigma^2+K_c\right)+2(\rho_i/\rho_o-1) + \frac{4f_t L}{d_i}\rho_i(1/\rho)_m - \left(1-\sigma^2-K_e\right)\rho_i/\rho_o\right] \tag{9}$$

where Δp_t included the pressure drop due to flow contraction, acceleration, friction, and expansion, four terms in Eq. (9). K_c and K_e are tube entrance and exit pressure loss coefficients. Furthermore f_t is the tube side friction factor estimated as: