Modelling of Economical Design of Shell and Tube Type Heat Exchanger Using Specified Pressure Drop

M. M. El-Fawal[*1], A. A. Fahmy[2] and B. M. Taher[3]

[1]National Center for Nuclear Safety and Radiation Control, Atomic Energy Authority, Cairo, Egypt
[2]Reactor Department, Nuclear Research Center, AEA, Cairo, Egypt
3Dept. of Engineering Physics and Mathematics, Faculty of Engineering, Cairo University, Cairo, Egypt
[*]mohamed_elfawal@hotmail.com

Abstract: Shell and Tube-type heat exchanger have wide application in nuclear industry where they play an important role in the transfer of heat from core to the heat sink, their cost minimization is an important target for both designers and users. In this paper a computer program for economical design of shell and tube heat exchanger using specified pressure drop is established to minimize the cost of the equipment including the sum of discounted annual energy expenditures related to pumping. The design procedure depends on using the acceptable pressure drops in order to minimize the thermal surface area for a certain service, involving discrete decision variables. Also the proposed method takes into account several geometric and operational constraints typically recommended by design codes, and may provide global optimum solutions as opposed to local optimum solutions that are typically obtained with many other optimization methods. While fulfilling heat transfer requirements, it has anticipated to estimate the minimum heat transfer area and resultant minimum cost for a heat exchanger for given pressure drops. The capability of the proposed model was verified through two design examples. The obtained results illustrate the capacity of the proposed approach through using of a given pressure drops to direct the optimization towards more effective designs, considering important limitations usually ignored in the literatures.
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1. Introduction

Shell-and-tube heat exchangers are probably the most common type of heat exchangers applicable for a wide range of operating temperatures and pressures. They have larger ratios of heat transfer surface to volume than double-pipe heat exchangers, and they are easy to manufacture in a large variety of sizes and flow configurations. Their construction facilitates disassembly for periodic maintenance and cleaning. Shell-and-tube heat exchangers find widespread in process industries, in conventional and nuclear power stations, steam generators, etc. Fig. 1 presents a schematic diagram for shell and tube heat exchanger.

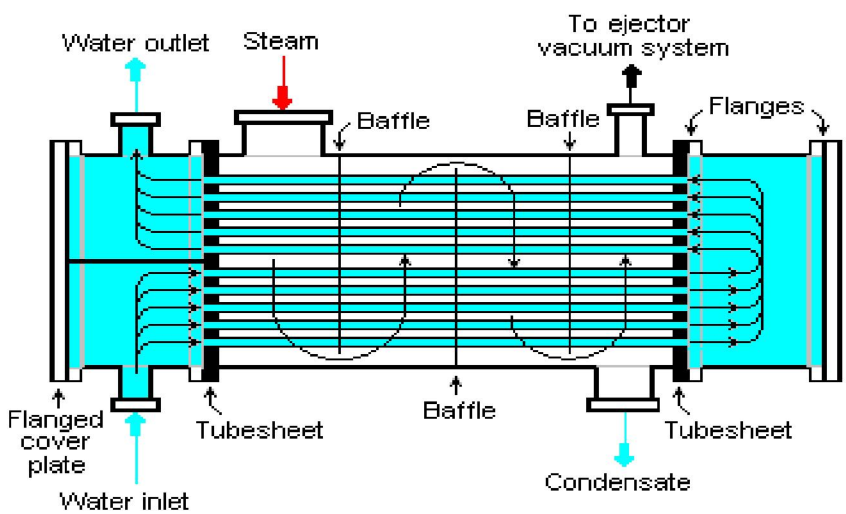

Fig. 1: Schematic diagram of the shell and tube heat exchange

In nuclear reactors, the amount of reactor power generation is limited by thermal rather than by nuclear considerations. The reactor core must be operated at such a power level, that with the best available heat removal system, the temperature of the fuel and cladding any where in the core must not exceed safe limits. Otherwise, accidents leading to fuel element meltdown could happen causing radiological releases. Thus, the optimum design of reactor cooling system would result on extracting heat from the reactor core without exceeding the design SAFE LIMITS [1]. Heat generated in reactor core is transferred to the Ultimate Heat Sink (UHS) through Reactor Cooling System (RCS).

The essential components in RCS are: heat source (reactor core), heat sink (heat exchanger), pumps, piping, valves, control and safety instrumentation interlocks and other related subsystems. Thus, heat exchanger represents one of the essential components in RCS. Due to the important role of shell-and-tube heat exchangers, a considerable number of papers has been devoted to the design optimization problem, employing different techniques, such as, numerical resolution of the stationary point equations of a nonlinear objective function, graphical analysis of the search space, simulated annealing, genetic algorithms, mixed integer nonlinear programming, systematic screening of tube count tables…etc [2-19].

In the context of the development of new design technique, this paper presents an optimization procedure integrated with practical design guidelines, aiming to provide a feasible alternative in an engineering point of view. These design rules are usually ignored in the literature, which restrains the effective application. An optimum design is based on the best or most favorable conditions. In almost every case, these optimum conditions, can ultimately be reduced to a consideration of costs or profits. Thus an optimum economic design could be based on conditions giving the least cost per unit of time or the maximum profit per unit of production. When one design variable is changed, it is often found that some costs increase and others decrease. Under these conditions, the total cost may go through a minimum at one value of the particular design variable, and this value would be considered as an optimum.

Two types of quantitative problems are commonly encountered by the design engineer when he deals with heat-transfer calculations. In the first type, all of the design variables are set, and the calculations involve only the determination of the indicated non variant quantities. By choosing various conditions, the engineer could ultimately arrive at a final design that would give the least total cost for fixed charges and operation. Thus, the second type of quantitative problem involves conditions in which at least one variable is not fixed, and the goal is to obtain an optimum economic design.

In general, increased fluid velocities result in larger heat-transfer coefficients and, consequently, less heat-transfer area and exchanger cost for a given rate of heat transfer. On the other hand, the increased fluid velocities cause an increase in pressure drop and greater pumping costs. The optimum economic design occurs at the conditions where the total cost is a minimum. The basic problem, therefore, is to minimize the sum of the variable annual costs for the exchanger and its operation. The objective function is the total annual cost for heat exchanger. In this paper the optimum design of a heat exchanger is developed by use of a new technique.

The technique was employed according to distinct problem formulations in relation to: (i) objective function: heat transfer area or total annualized costs (i.e. capital costs of the heat exchanger and pumps associated to fluid flow operating costs); (ii) constraints: heat transfer and fluid flow equations, pressure drop and velocity bounds, etc.; and (iii) decision variables: selection of different search variables and its characterization as integer or continuous (e.g., tube diameter can be considered a fixed parameter, a continuous variable or a discrete variable).

2. Procedure of Heat Exchangers Design

The classical approach to shell-and-tube heat exchanger design involves a significant amount of trial-and-error because an acceptable design needs to satisfy a number of constraints. Typically, a designer chooses various geometrical parameters such as tube length, shell diameter and baffle spacing based on experience to arrive at a possible design. If design does not satisfy the constraints, a new set of geometrical parameters must be chosen. Even if constraints are satisfied, the design may not be optimal, so it is necessary to optimize the design either in terms of capital cost or running cost. Capital cost involves minimization of heat transfer surface area to meet heat transfer service while running cost involves with minimum pressure drops. In this work, a methodology is proposed that calculates the approximate free flow areas on tube and shell side where a minimum shell-side pressure drop was considered as constraining criteria for optimum design. Once these are obtained, geometrical dimensions can be tried to satisfy heat transfer requirements.

Inlet data for both fluids are: inlet temperature (T_{in}), outlet temperature (T_{out}), mass flow rate (m), density (ρ), heat capacity (C_p), viscosity (μ), thermal conductivity (k), allowable pressure drop (ΔP_{design}), fouling factor ($r_{ddesign}$) and area as well as, pumping cost data. The mechanical variables to be optimised are tube inside diameter (d_{in}), tube outside diameter (d_{ex}), tube arrangement (a_{rr}), tube pitch (p_t), tube length (L), number of tube passes (N_{tp}) and number of tubes (N_t), for the tube-side. To the shell-side, the desired variables are the external diameter (D_s), the tube bundle diameter (D_{otl}), baffles number (N_b), number of shells (N_S) baffles cut (l_c) and baffle spacing (B). Finally, thermal-hydraulic variables to be calculated are heat duty (Q), heat exchange area (A), tube - side and shell side film coefficients (h_t and hs), dirty and clean overall heat transfer coefficient (U_o), pressure drop (ΔP_t and ΔP_s), log mean temperature difference (LMTD), the correction factor of LMTD (F_t) and the fluids location inside the heat exchanger. The main equations of the model are given below.

2.1 Tube Side

The tube side single phase heat transfer coefficient may be estimated by a variety of the available of correlations, empirical and semi-empirical, suggested by several workers. Petukhov [20] postulated an equation of the heat transfer coefficient for the turbulent flow region, $Re_t > 2100$, in the form:

$$h_t = Nu_t\, k_t / d_t \quad (1)$$

where

$$Nu_t = \{(f/2) Re_t Pr_t\} / \{1.07 + 12.7\,(f/2)^{0.5}[(Pr_t)^{0.66} - 1]\} \quad (2)$$

$$Re_t = (\rho\, V\, d/\mu)_t \quad (3)$$

$$Pr_t = (\mu\, C_p / k)_t \quad (4)$$

$$f = (1.58\, Re_t - 3.8)^{-2} \quad (5)$$

The number of tubes is given by the following equation [21]:

$$N_t = a\,\pi\,(D_s)^2 / [A_t] \quad (6)$$

Where

$$A_t = b\,(P_T)^2 \quad (7)$$

The value of tube count constant (a) for different tube passes and values of the layout constant (b) are given by:

a = 0.93 for one pass;
a = 0.9 for two tube passes;
a = 0.85 for more than two tube passes;
b = 1.0 for 90° and 45° tube pattern and
b = 0.87 for 30° and tube pattern

From equations 6 and 7 the number of tubes is given by:

$$N_t = a\,\pi\,(D_s)^2 / [4b(P_R)^2\,(d_f)^2] \quad (8)$$

Where

$$P_R = P_T / d_f \quad (9)$$

The shell inside diameter has the form:

$$D_s = (b/a)^{0.637}\,\{A_{of}\,[(P_R)^2\, d_f/L]^{0.5}\} \quad (10)$$

Where (A_{of}) is the outside heat transfer surface area based on the outside diameter of the low finned tube (d_f), and can be calculated from:

$$A_{of} = \pi\, d_f\, L\, N_t \quad (11)$$

The tube side pressure drop is calculated using the following equation [22]

$$\Delta P_t = 4\, N_p\,[f(L/d_t) + 1][\rho_t (u_t)^2/2] \quad (12)$$

Where

$$F = 1/[1.58 \ln (Re_t)^{-3.28}]^2 \quad (13)$$

2.2 Shell Side

The single phase heat transfer coefficient is given by Sinnott as [23]:

$$h_f = 0.155\,(R'e)^{0.6}\, P_r^{0.33}\,(\mu/\mu_w)^{-0.14}\,(k/d') \quad (14)$$

where

$$R'e = G_s\, d' / \mu$$

$$G_s = m_s / \{NB[(P_T - d_r) - (d_f - d_r) t_f N_f]\}$$

$$N = (\pi D_s)/(4\, P_T)$$

$$d' = [(d_f)^2 - (d_t)^2]^{0.5}$$

$$P_T = 1.25\, d_f$$

Sach [24] recommended a correlation for the single phase heat transfer coefficient presented by Rabas et al [25] which has the form:

$$St\,(Pr)^{0.66} = 0.29\,(Re)^n\,(P_f/d_r)^{1.115}\,(P_f/L_f)^{0.257}\,(t_f/P_f)^{0.666} \times (d_f/d_r)^{0.473}\,(d_r/t_f)^{0.772} \quad (15)$$

$$n = -0.415 + 0.0346 \ln (d_f/P_f) \quad (16)$$

The Stanton number is given by the equation:

$$St = Nu / Re\,Pr \quad (17)$$

Then the heat transfer coefficient (h_f) is given by:

$$h_f = 0.29 (Re)^{n+1} (Pr)^{0.333} (k/d_h)(P_f/d_r)^{1.11}\,(P_f/L_f)^{0.257} \times (t_f/P_f)^{0.666}\,(d_f/d_r)^{0.473} (d_r/t_f)^{0.772} \quad (18)$$

where Reynolds number is based on the hydraulic diameter and given by:

$$Re = G_s\, d_h / \mu \quad (19)$$

The hydraulic diameter (d_h) can be calculated from the equation:

$$d_h = 4B\{[(P_T - d_r) - (d_f - d_r) t_f N_f] / (\pi/2)[((d_f)^2 - (d_r)^2 + d_t t_f) N_f + (1 - t_f N_f) d_r]\} \quad (20)$$

The pressure drop can be calculated using the correlation developed by Rabas et al [25]

$$\Delta P_f = 2\, f\,(G_s)^2 / \rho\, N_r \quad (21)$$

Where

$$N_r = D_s / P_T \quad (22)$$

The friction factor is given by:

i. for $1000 \le Re \ge 25000$, $n_t \ge 6$ *and* $P_L \le P_T$

$$f = [3.805\,(Re_t)^{-2.336}\,(S_f/d_f)^{0.2512}\,(L_f/S_f)^{0.7292} \times (d_r/P_T)^{0.709}\,(P_T/P_L)^{0.3791} \quad (23\text{-}a)$$

ii. for $1000 \le Re \ge 713000$, $20^o \le T_{LA} \ge 40^o$, $P_T/d_r \le 4$ *and* $n_t \ge 4$

$$f = [1.748\,(Re_t)^{-2.33}\,(L_f/S_f)^{0.522}\,(d_r/P_T)^{0.599} \times (d_r/P_L)^{0.1738} \quad (23\text{-}b)$$

iii. for $1000 \le Re \ge 800000$, $n_t \ge 10$ *and* $n_t \ge 10$

$$f = [4.71 (Re_t)^{-2.86}[(P_T/d_r) - 1]^{-0.36} (L_f/S_f)^{0.51} \times [(P_T - d_r)/(P_L - d_r)]^{0.536}$$

(23-c)

$Re = G_s\, d_r / \mu$ (24)

The above equations were used in the present model to allow the designer to have a variety options to select the suit heat exchanger design. The calculation of pressure drops on both shell and tube side will be reflected on the power consumption by the system circulation pumps. The pumping and area cost as ($/year) was estimated using the following formulas [26]:

Area cost = $123A^{0.59}$ (34)

The heat transfer surface area is given by:

$A = Q / U\, \Delta tm$ (35)

Also the flow rate of utility fluid (m_u) can be calculated from the equation [20]:

$m_u = q / \{(C_p)_o [(\Delta t_1 - (\Delta t_2) + T_1 - T_2]\}$ (36)

2.4 Number of Tubes and Inside Tube Flow Area

The inside tube flow area per pass and the number of tubes are given by the following equations [20]:

$(A_i)_t = m_i / G_i$ (37)

$(N_t) = (4\, n_p\, S_i) / \pi\, (D_i)$ (38)

Where

D_i is the inside tube diameter

G_i is the inside mass velocity, kg/hr.m^2

$N_{i\ is}$ is the number of tubes

n_p is the number of tube passes

A_i is the cross sectional inside tube flow area, m^2

m_i is the fluid mass flow rate; kg/hr

2.5 Tube Length

The value of tube length is obtained from the optimum heat transfer area and the total number of tubes. Thus for a given tube optimum diameter the tube length L is calculated from the equation [20]:

$L = (A) / \pi\, (D_t)\, N_t$ (39)

2.6 Number of Clearances and Baffles

The number of clearances N_c,and baffles were computed directly from the following equations[20]:

i. with square pitch and $N_t > 25$

$N_c = 1.37\, (N_t)^{0.475}$ (40)

ii. With equilateral triangular pitch and $N_t > 25$

$N_c = 0.94 + [(N_t - 3.7) / 0.907]^{0.5}$ (41)

The number of baffles n_p was estimated from the following equation [19]

$n_p = (N_c D_c\, L) / S_o$ (42)

Where

D_c is the clearance between tubes, (m) and S_o is the shell side flow area, (m^2).

2.7 Constraints for the Model

To get a practical design, the shell and tube heat exchanger must satisfy the given heat duty and the following operational and geometric constraints [27]:

$\Delta P_t \le \Delta P_{t,max}$

$\Delta P_s \le \Delta P_{s,max}$

$V_{t,min} \le V_t \le V_{t,max}$

$V_{s,min} \le V_s \le V_{s,max}$

$D_s \le D_{s,max}$

$L \le L_{max}$

$R_{bs,min} \le R_{bs} \le R_{bs,max}$

$R_{smsw,min} \le S_m / S_w \le R_{smsw,max}$

where ΔP is the pressure drop, V is the velocity, D_s is the shell diameter, L is the total length, R_{bs} is the ratio baffle spacing to shell diameter, R_{smsw} is the ratio cross flow area to window flow area, S_m the cross flow area and S_w the window area. The first four equations are thermo-hydraulic constraints and the last four equations represent geometric constraints. Typical design limits to be used in this set of constraints are given by Muralikrishna and Shenoy [27].

A computer program is developed based on the model described above. Baffle spaces at inlet and outlet of the exchanger are assumed to be equal for simplicity. The program allows the user to choose the shell-side fluid and also to select optimization constraints, i.e., one is minimum shell side pressure drop and the other is allowable shell-side pressure drop. The flow diagram of the computer program is illustrated in the Fig. 2.

3. Results

The performance of the proposed model is illustrated through the analysis of the results obtained in two examples of design tasks and comparing the solution reached with a previous recent literature approach.

3.1 Case Study One

The first example presented here was the one reported by Polley, Panjeh Shahi and Nunez [8] to demonstrate the inverse design methodology. The original example involved water on the tube side of the exchanger with an assumed film heat transfer coefficient of 6000 W/m^2 K. The fluid on shell side is viscous oil. The tube side and shell side pressure drops for this situation are 11.66 kPa and 13.7 kPa. These are the allowable ΔP subsequently used in our design. The sample operating conditions and the design data are shown in Tables 1 and 2. The flow rates, temperatures, allowable pressure drops, and physical properties of streams are fixed. It is required to determine the optimum area and optimum cost of shell-and-tube heat exchanger.

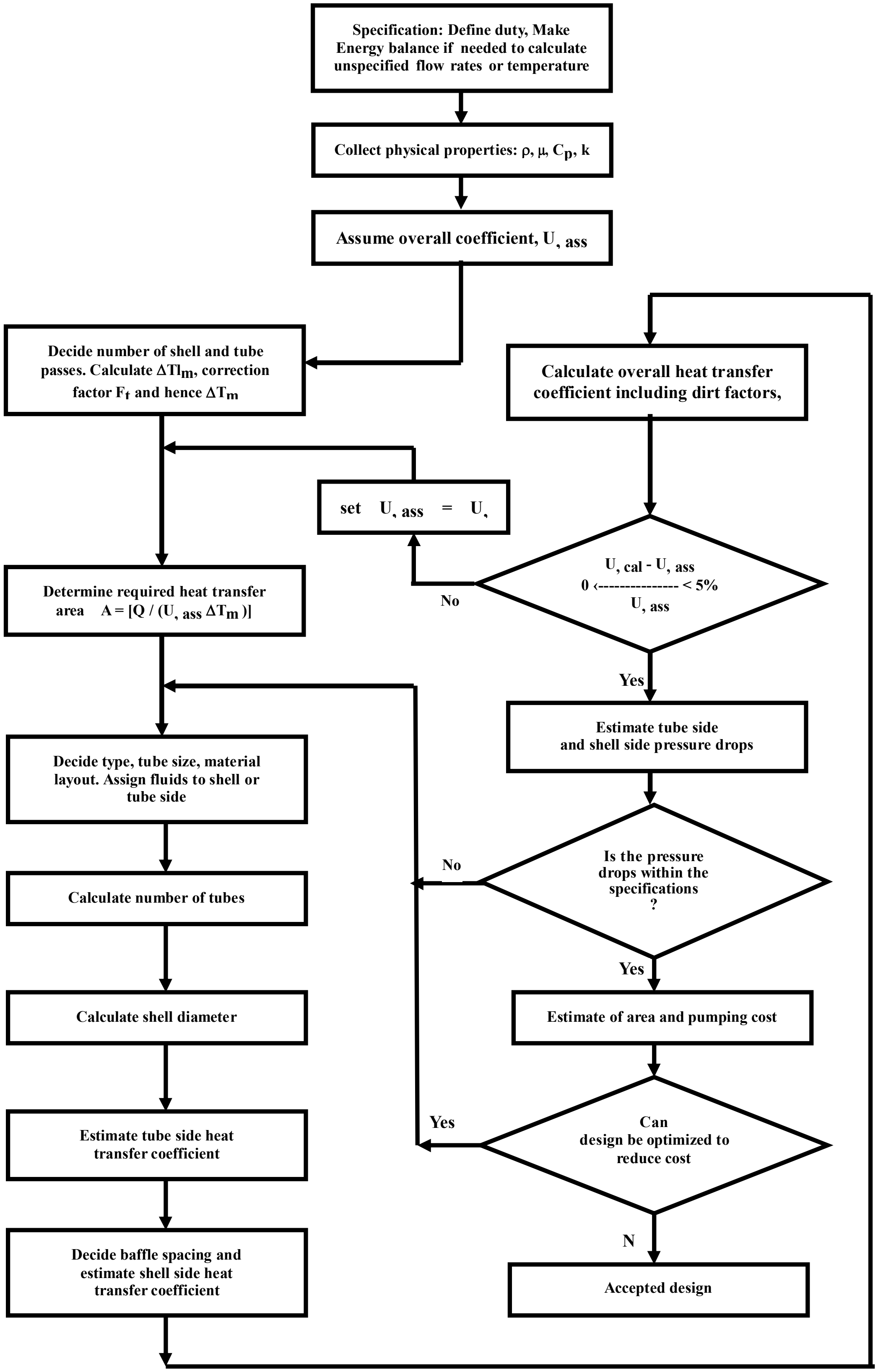

Fig. 2: Flow diagram of the computer design program

Table 1: Shell and tube exchanger design data[28] – physical properties

Physical Properties	Shell side	Tube side
Fluid	Oil	Water
Flow rate (kg/s)	22.4	77.96
Fluid density (kg/m^3)	740	1000
Heat capacity (J/kg. K)	2407	4187
Viscosity (Pa. s)	0.494	1.000
Thermal conductivity (W/m. K)	0.105	0.61
Inlet temp. (deg K)	373	280
Outlet temp (deg K)	315	290
Allowable ΔP (kPa)	14	12
Material	C32100	SS 316
Heat Duty (kW)	3100	

Table 2: Shell and tube exchanger design problem [28] – geometry

Geometry	Values
Tube OD (mm)	16
Tube ID (mm)	14
Tube layout (deg)	30
Tube pitch (mm)	21
Baffle-to-shell clearance (mm)	6
Tube-to-baffle clearance (mm)	0.6
Bundle-to-shell clearance (mm)	10

Table 3: Comparison of shell and tube heat exchanger designs

Geometry	Polley, Shahi, and Nunez [28]	Present work
Shell diameter (mm)	563	520
Tube length (mm)	1815	1728
Baffle cut (%)	29.3	26.8
Baffle spacing (mm)	253	228
No. of baffles	6	5
No. of tubes	574	548
No. of tube passes	2	2
Required area (m^2)	52.3	49.39
Installed area (m^2)	52.3	49.39
Shell side Re	21398	27926
Shell side ΔP (kPa)	13.7	13.493
Tube side ΔP (kPa)	11.69	11.61
Shell side heat coefficient (W/	1406	1471
Tube side heat coefficient (W/	6641	6750
Pumping cost ($/year)	950	2424
Area cost ($/year)	3150	2826

A summary of the calculated results using the proposed model compared with the corresponding values reported by Polley, Panjeh Shahi and Nunez [28] is given in Table 3. From the Table it could be seen that the obtained results using the present model are in good agreement with the corresponding values reported by Polley, Shahi, and Nunez [28].

3.2 Case Study Two

The second example presented here was reported by Mizutani et al. [9]. The sample operating conditions and the design data are shown in Tables 4 and 5. For the solution of this example, a constraint in the tube length of 4.88 m was imposed. A summary of the results obtained with the proposed model compared with the corresponding values reported by Mizutani et al is given in Table 6. They used a disjunctive programming optimization method to solve this problem. From Table 6 it can be observed that the proposed model provided a better solution than the one obtained by Mizutani et al where the constraints, were not taken into consideration in their technique.

Table 4: Sample operating conditions

Sample operating conditions	Shell side	Tube side
Fluid	Water	Water
Fouling resistance (m^2k/W	0.000176	0.00076
Mass flow rate (kg/s)	13.88	8.33
Inlet temperature (K)	340	290
Outlet temperature (K)	-	310
Limitations	Maximum allowable pressure drop =12 000 (Pa)	
Tube material	Carbon steel, thermal conductivity = 60 (W/mK)	

Table 5: The design data of the example

Design data	Shell side	Tube side
T_{in} (K)	370	300
T_{out} (K)	590	590
Mass flow rate (kg/s)	27.78	68.88
k [W/(m K)]	0.190	0.59
Fluid density (kg/m^3)	1000	1000
specific heat [J/(kg K)]	2840	4200
viscosity (Pa. s)	0.00034	0.0008

Table 6: A summary of the calculated results obtained from this work compared with previous work obtained by Mizutani et al [9]

Calculated results	Results obtained by Mizutani et al [9]	Present work
Area (m^2)	202	230
U (W/m^2 K)	860	720
Number of tubes	832	700
Number of tube passes	2	6
Inside tube Diameter (mm)	12.6	22
Outside tube Diameter (mm)	15.9	24
Number of baffles	8	8
Shell diameter (m)	0.687	1.11
Tube length (m)	4.88	5
Baffle spacing (m)	0.54	0.52
ΔP_{tube} (Pa)	22676	11000
ΔP_{shell} (Pa)	7494	4740
Pumping cost ($/year)	2424	964
Area cost ($/year)	2826	3043

4. Conclusions

In this work, an optimization model for the design of a shell and tube heat exchanger has been proposed. The optimization strategy based upon the presented analytical optimization analysis is developed as a computer aided design package. Important additional constraints, usually ignored in previous optimization schemes, are included in order to approximate the solution to the design practice. Two cases for optimal design of shell and tubes heat exchanger based upon the devised computer program were presented. In case study one the obtained results in the present work are consistent with the corresponding values reported by Polley, Shahi, and Nunez [28]. In case two the comparison showed that the proposed model is more efficient in terms of providing excellent optimum solutions than standard optimization method reported by Mizutani et al [9]. Also the result of the study cases ends up with the final conclusion that the use of the model provides the best solutions with higher quality together with short duration of real time.

Nomenclature:

a is the tube count calculation constant (dimensionless)
a_{rr} tube arrangement
A: Surface area (m^2)
B is the passes and layout constant (dimensionless)
B: Baffle spacing (m)
C_p: Specific heat (kJ/kg K)
d: Tube diameter (m)
d_f: Finned tube outside diameter, (m)
d_h: Hydraulic diameter defined by eq.(20)
d_r: Tube diameter measured to the fins root, (m)
D: Diameter (m)
f: Friction factor (dimensionless)
F: Fouling resistance (m^2 ºC/W)
G: Mass velocity (kg/m^2 s)
h: Heat transfer coefficient (W/m^2 K)
H: Fluid enthalpy (kJ/kg)
k: Thermal conductivity (W/m ºC)
L: Heat exchanger length (m)
L_f: Fin height (m)
LMTD: Logarithmic Mean Temperature Difference (ºC)
m: Mass flow rate (kg/s)
n: Number of tubes per pass
N_f: Number of fins per unit length
N_p: Number of passes
N_r: Number of tubes in central row of bundle
n_p Number of baffles
n_t: Number of vertical rows in heat exchanger
N_t: Total number of tubes
Nu: Nusselt number (dimensionless)
P_f: Fin pitch (m)
P_p: Pumping power (kW)
P_r: Prandtle number (dimensionless)
P_R: Tube pitch ratio (dimensionless)
P_L: Longitudinal tube pitch (m)
P_T: Transverse tube pitch (m)
Q: Heat exchanger load (kW)
Q_s: Flow rate of utility (m^3/h)
Re: Reynolds number (dimensionless)
S_f: Fin gap (m)
S_t: Stanton number (dimensionless)
t_f: Fin thickness (m)
t: Tube side fluid temperature ºC)
T: Temperature or shell side fluid temperature (ºC)
u_t: Tube side fluid velocity (m/s)
U_O: Heat exchanger overall heat transfer coefficient (W/m^2. ºC)

ΔP: Pressure drop (Pa)
ΔT: temperature difference (ºC)

Greek Symbols:

ρ: Fluid density (kg/m^3)
µ: Fluid dynamic viscosity (Pa.s)

Subscripts:

ass: assumed
b: Bundle
cal: calculated
e: Exit or edge of baffle Spacing
f: Finned or fin
i: Inside
Inc: Increment
lm: Logarithmic mean
m: Mean
o: Outside
out: Outlet value
in: Inlet value
r: Fin root value
S: Shell side value
t: Tube side value
tot: Total value
w: Wall value

Corresponding author
M. M. El-Fawal
National Center for Nuclear Safety and Radiation Control, Atomic Energy Authority, Cairo, Egypt
*mohamed_elfawal@hotmail.com

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