World Academy of Science, Engineering and Technology
International Journal of Industrial and Manufacturing Engineering
Vol:6, No:5, 2012

Optimum Design of an Absorption Heat Pump Integrated with a Kraft Industry using Genetic Algorithm

B. Jabbari, N. Tahouni and M. H. Panjeshahi

***Abstract*—**In this study the integration of an absorption heat pump (AHP) with the concentration section of an industrial pulp and paper process is investigated using pinch technology. The optimum design of the proposed water-lithium bromide AHP is then achieved by minimizing the total annual cost. A comprehensive optimization is carried out by relaxation of all stream pressure drops as well as heat exchanger areas involving in AHP structure. It is shown that by applying genetic algorithm optimizer, the total annual cost of the proposed AHP is decreased by 18% compared to one resulted from simulation.

***Keywords*—**Absorption Heat Pump, Genetic Algorithm, Kraft Industry, Pinch Technology

I. INTRODUCTION

THE pulp and paper industry is a very large energy consumer industry, in the form of heating and cooling energy to dry liquor in evaporation section and to maintain critical streams blow temperature limits, respectively [1]. More recently, the advanced energy conversion technologies such as absorption heat pump (AHP) and tri-generation are used to improve energy efficiency [2], [3]. The utilization of AHPs for heat upgrading in pulp and paper industry has been investigated [4]. Also, modeling, design and construction of a lithium bromide water absorption cycle has been investigated [5], [6] with shell and tube heat exchangers which are widely used in industry [7]. Optimum design of a shell and tube heat exchanger has been studied in many works [8], [9]. The geometry of shell and tube heat exchangers for minimizing their cost has been optimized by several methods like genetic algorithm (GA) [10]–[13]. However, the optimization methods are mainly carried out for single phase flow heat exchangers and optimum design of them with presence of phase change, like their application in refrigeration systems and heat pumps has received less attention. The design and optimization of a shell and tube heat exchanger with phase change, like shell and tube condenser has been investigated in some works [11], [14]. [15].

B. Jabbari is MSc Student with the School of Chemical Engineering, College of Engineering, University of Tehran, Iran (e-mail: b.jabbari@ut.ac.ir).

N. Tahouni is Assistant Professor with the School of Chemical Engineering, College of Engineering, University of Tehran, Iran (phone: 98-21-66957788; fax: 98-21-66957784; e-mail: ntahuni@ut.ac.ir).

M. H. Panjeshahi is Professor with the School of Chemical Engineering, College of Engineering, University of Tehran, Iran (e-mail: mhpanj@ut.ac.ir).

II. AHP INTEGRATION WITH KRAFT INDUSTRY

A. Kraft Process – Case Study

The Kraft process is a chemical process in which the paper pulp is produced by wood chips in a digester using delignification liquor. Paper pulp is an important source for producing many kinds of paper products [16]. The delignification liquor decomposes lignin and separates the cellulosic fibers. The spent delignification liquor (Black liquor) contains valuable chemical materials that could be recovered. Moreover, the residual wood materials could be burnt to utilize of its energy content [17]. To make the liquor combustible, its solid content must be increased. As the black liquor concentration section is sometimes the highest energy consumer in Pulp and paper mill, this section of an Iranian pulp and paper mill is considered for efficiency improvement. The black liquor (BL) coming from the washing section has about 6.8% solids content. To prepare the liquor to be burnt, its solid content is increased to 53% in a five-effect falling film evaporator. Table I shows the stream data of the BL concentration section. Using pinch technology tools, the minimum heating and cooling requirements for the concentration section is obtained as 15.2 and 17.2 MW, respectively.

TABLE I
STREAM DATA FOR CONCENTRATION SECTION

			***T*in (°C)**	***T*out (°C)**	**Q (MW)**
Heating demand	effect1	BL	93.5	99.7	0.24
		BLev	99.7	99.7	15
	effect2	BL	86	93.5	0.5
		BLev	93.5	93.5	14.5
	effect3	BL	69.1	86	1.4
		BLev	86	86	13
	effect4	BL	45.8	69.1	2.4
		BLev	69.1	69.1	10.6
Cooling demand	effect5	BLev	45.8	45.8	10.6
	vap1 to effect2		99.6	99.6	15
	vap2 to effect3		93.5	93.5	14.5
	vap3 to effect4		86	86	13
	vap4 to effect5		69.1	69.1	10.6
	vap5 to condenser		45.8	45.8	17.3
Cold utility	fresh water		31	44.8	17.2
Hot utility	fresh steam		188.0	188.0	15.2

BL= Black Liquor, ev: evaporation

B. AHP – Scheme Proposal

In our previous work a single effect water-lithium bromide AHP was designed to upgrade low temperature heat in the concentration section [18]. A schematic diagram of the AHP with its five heat exchangers is given in Fig. 1. It consists of a generator, a condenser, an evaporator, an absorber and a solution heat exchanger (SHX).

Moreover, there is a steam turbine in the mill which produces power and delivers steam at a lower pressure that it can be used to drive the generator of an AHP. The outlet steam of the steam turbine supplies the heat of generator, QG. A low-temperature process stream supplies the heat duty of evaporator and the useful heat is released via condenser, QC, and the absorber, QA, in which QC+QA is more than QG.

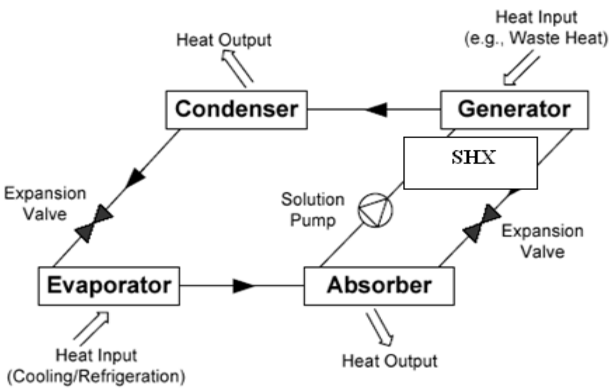

Fig. 1 Absorption heat pump

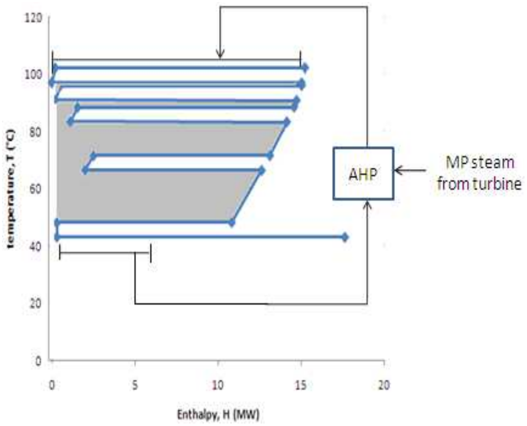

Fig. 2 Integration of a proposed AHP with the BL concentration section

The Grand Composite Curve can be used in this work to select a hot stream below the pinch point to supply the energy demand of the evaporator and a cold stream above the pinch point to receive the heat duties released from the condenser and absorber, as it can be seen in Fig. 2. After identification the source and sink, the phase equilibrium diagram is used to determine the temperature of evaporator, condenser, absorber and generator of AHP [1]. Therefore, by applying the AHP, the net heating and cooling demand in the concentration section is reduced.

From the simulation of the AHP by Aspen plus software it was found that 17.6 ton/hr of Medium Pressure steam (MP) discharging from steam turbine is used to supply the generator load, QG (9.2 MW). The total amount of useful heat, QC+QA is 15.2 MW and evaporator duty, QE is 5.9MW. The simulation of the AHP with Aspen Plus software is shown in Fig. 3.

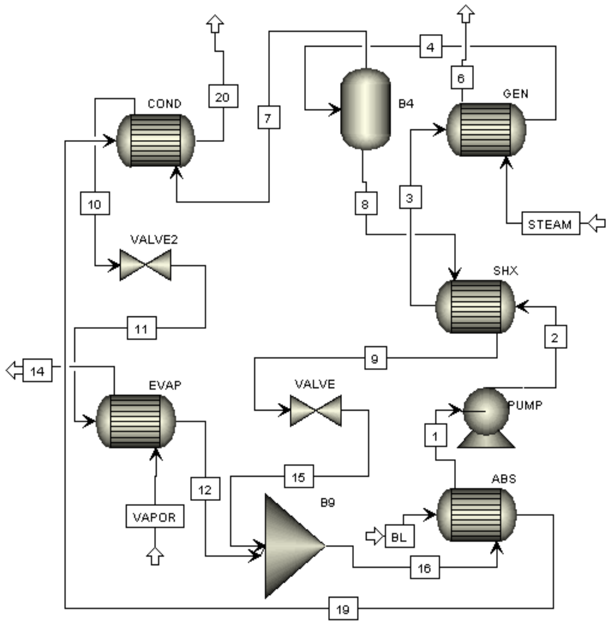

Fig. 3 Simulation of the proposed AHP

As it described the AHP is considered to supply 15.2 MW of heating demand of concentration section, from its condenser and absorber and also reducing cooling demand of this section by its evaporator. Our purpose is to optimize the AHP heat exchangers design, with regard to its configuration as well as producing 15.2 MW heating energy and 5.9MW cooling energy at desired temperatures. Some unknown temperatures and duties are obtained again from optimization. All heat exchangers are considered to be shell and tube heat exchangers.

III. Objective Function

There is a trade-off between required heat transfer surface area and pressure drop of streams in design of heat exchangers involved in AHP. Therefore, the following cost components should be considered in heat exchanger optimization: First, the annualized capital cost of the heat exchanger, Second, capital cost and operating cost of pumps (for liquid streams) and the capital cost and operating cost of compressors (for gas streams).

The total cost as the objective function includes investment cost (IC) and operating cost (OC) [19]:

$$TAC = IC * AF + OC\ , \quad AF = \frac{i(i+1)^n}{(1+i)^n - 1} \tag{1}$$

Where AF, i and n are annual factor, annual interest rate and heat exchanger lifetime, respectively.

The investment cost includes the capital cost of heat exchangers (C_{HX}) and required pumps (C_{Pump}) and compressors (C_{Comp}), which are calculated as follows [20]:

$$C_{HX} = a_1 + a_2 A^{a_3} \tag{2}$$

$$C_{Pump} = b_1 + b_2 (\frac{m^\circ}{\rho} \Delta P)^{b_3} \tag{3}$$

$$C_{Comp} = c_1 + c_2 (\frac{m^\circ}{\rho} \Delta P)^{c_3} \tag{4}$$

Where a_i, b_i, c_i are relative constants, A, ΔP, m°, ρ are heat exchange surface area, pressure drop of streams, mass flow rate and density of streams, respectively.

The operating cost is related to power consumption by pump and compressor to drive fluids for shell side (E_s) and tube side (E_t) [21]:

$$OC = \frac{(E_s + E_t) \times op \times ec}{1000} \tag{5}$$

Where op is the annual operating time and ec is electricity cost. The power is computed from below equation, where η is the pump or compressor efficiency:

$$E = \frac{\Delta P m^\circ}{\rho \eta} \tag{6}$$

Also the power required for driving two-phase flow (E_{tp}) is related to two-phase pressure drop (ΔP_{tp}) and can be computed by Equation 7 [11]:

$$E_{tp} = \frac{\Delta P_{tp} m^\circ (\rho_g + \rho_h)}{2 \eta \rho_g \rho_h} \tag{7}$$

ρ_g and ρ_h are the density of gas and liquid streams, respectively.

Here, we aim to minimize the total cost of the proposed AHP by varying the heat exchangers geometry. The geometry of heat exchangers has a strong effect on the overall heat transfer coefficient and pressure drops and consecutively on the total cost. Therefore, first we have to define the required equations for heat transfer coefficients and pressure drops as a function of design variables.

As there are phase changes in AHP heat exchangers, we have to consider two-phase heat exchanger equations as well as single phase equations in the optimization procedure.

IV. Heat Transfer and Pressure Drop Calculations

A. Heat Transfer Rate

The heat transfer rate (Q) between the shell and tube fluids can be determined from the following basic equation:

$$Q = UAF \Delta T_{lm} \tag{8}$$

Where U is the overall heat transfer coefficient and ΔT_{lm} is log mean temperature difference. The correction factor F is used when the number of tube passes is more than 1.

The overall heat transfer coefficient can be expressed by the general equation:

$$U = (\frac{1}{h_s} + R_s + \frac{d_o \ln\left(\frac{d_o}{d_i}\right)}{2k_w} + \frac{d_o}{d_i}\left(\frac{1}{h_t} + R_t\right))^{-1} \tag{9}$$

Where h_s and h_t are heat transfer coefficients of shell and tube sides, and R_s and R_t are fouling resistances for these sides. d_o, d_i and k_w are tube outside diameter, tube inside diameter and wall thermal conductivity.

The heat transfer rate can also be computed from the hot side or cold side in following two ways:

$$Q = m^\circ c_p (T_i - T_o) \tag{10}$$

$$Q = m^\circ h_{fg} \tag{11}$$

Where the equation 10 is related to sensible heat transfer and the equation 11 is related to latent heat transfer in heat exchanger sides [22]. c_p is heat capacity, h_{fg} is latent heat and T_i, T_o are inlet and outlet temperatures of streams.

B. Tube Side Heat Transfer Coefficient and Pressure Drop in Single Phase Flow

The film heat transfer coefficient for tube side (ht) can be calculated as follows [23]:

$$Nu = 0.023 Re^{0.8} Pr^{0.33} (\frac{\mu}{\mu_w})^{0.14} \tag{12}$$

Where Nu, Re, Pr are Nusselt number, Reynolds number and Prandtl number. Equation12 can be re-arranged to give:

$$h_t = 0.023 (\frac{k}{d_i})(\frac{\rho di}{\mu})^{0.8} Pr^{0.33}\, u^{0.8} \tag{13}$$

Where k, μ, u, μ_w are conductivity, dynamic viscosity, velocity of tube fluid and wall dynamic viscosity, respectively.

The pressure drop through a single tube is given by the fanning equation:

$$\Delta P = 2f (\frac{l}{d_i}) \rho u^2 \tag{14}$$

l is tube length and f is friction factor given by following equation:

$$f = 0.046\, Re^{-0.2} \tag{15}$$

The velocity of a fluid through a single tube is a function of volumetric flowrate (V) and the number of tubes (N):

$$u = \frac{4V}{N \pi d_i^2} \tag{16}$$

And the surface of the heat exchanger is calculated by:

$$A = N \pi d_o l \tag{17}$$

C. Shell Side Heat Transfer Coefficient and Pressure Drop in Single Phase Flow

Kern's formulation is used for computing shell side heat transfer coefficient and pressure drop. According to kern's correlation we consider the assumption that the baffle cut is 25% [24]:

$$h_s = 0.36(\frac{k}{d_e})(Re)^{0.55}Pr^{0.33}(\frac{\mu}{\mu_w})^{0.14} \quad (18)$$

Where d_e is tube bundle equivalent diameter. The pressure drop is given by:

$$\Delta P = 0.5f(\frac{D_s(N_b+1)}{d_e})\rho u^2 \quad (19)$$

With:

$$f = 1.79\ Re^{-0.19} \quad (20)$$

Where D_s and N_b are shell diameter and number of baffles. The heat transfer surface area is given by:

$$A = N\pi d_o l = N\pi d_o(N_b+1)L_b \quad (21)$$

Where:

$$D_s(N_b+1) = \frac{4P_t^2}{\pi^2 d_o}.\frac{A}{D_s L_b} \quad (22)$$

$$N = (\frac{\pi}{4})(\frac{D_s}{P_t})^2 \quad (23)$$

L_b is baffle spacing, and P_t is tube pitch. Flow velocity for shell side can be calculated by:

$$u = \frac{V}{\left(\frac{D_s}{P_t}\right)(P_t-d_o)L_b} \quad (24)$$

D. Evaporation Case

In this work it is assumed that the evaporation occurs in tube side of all heat exchangers. In the case of vaporization the following relationship can be expected to hold [25]:

$$\frac{h_{tp}}{h_l} = (\frac{\Delta P_{tp}}{\Delta P_l})^n \quad (25)$$

h_{tp} and ΔP_{tp} are heat transfer coefficient and pressure drop for two phase flow. The exponent, n is related to the Reynolds number exponent in heat transfer correlation, b, and the Reynolds number exponent in friction factor equation, y:

$$n = \frac{b}{2-y} \quad (26)$$

Two-phase pressure drop is obtained from the equation developed by Chisholm [26]:

$$\frac{\Delta P_{tp}}{\Delta P_l} = 1 + \frac{C}{x_{tt}} + \frac{1}{x_{tt}^2} \quad (27)$$

Where:

$$C = (\frac{\rho_l}{\rho_g})^{1/2} + (\frac{\rho_g}{\rho_l})^{1/2} \quad (28)$$

$$x_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}(\frac{\mu_l}{\mu_g})^{0.1} \quad (29)$$

Where x_{tt} is Lockart-Martinelli parameter, C is a constant and x is mass quality. By an integral over the mass quality change, the mean value for heat transfer coefficient can be obtained:

$$h_{tp} = \frac{h_{lo}}{\Delta x}\int_{x_1}^{x_2}(1-x)^{1-b}(\frac{\Delta P_{tp}}{\Delta P_l})^n.dx \quad (30)$$

The subscript lo refers to total flow with liquid phase properties.

For calculating overall pressure drop, Chisholm's equation can also be integrated over the mass quality change:

$$\Delta P_{tp} = \Delta P_{lo}\int_{x_1}^{x_2}(1-x)^{2-y}\left(1+\frac{C}{x_{tt}}+\frac{1}{x_{tt}^2}\right).dx \quad (31)$$

Which h_{lo} and ΔP_{lo} can be determined directly by using single phase equations for tube side which were described in part B.

E. Condensation Case

In this study, we assume that condensation occurs in shell side of heat exchanger. Following the procedure recommended by Smith we can calculate the heat transfer coefficient for condensing fluid in the shell ($h_{s,C}$) [7]:

$$h_{s,C} = 1.35\ k_l(\frac{\rho_l^2 d_o g N}{\mu_l m^{\circ}})^{1/3} \quad (32)$$

For pressure drop calculation Chisholm's correlation can be used [27]:

$$\emptyset_{lo}^2 = 1 + (Y^2-1)(x-x^2)^{0.815} + x^{1.37} \quad (33)$$

Where $\emptyset_{lo}^2$ and Y^2 are, respectively, the two-phase multiplier and Chisholm's parameter, which are defined as below:

$$\emptyset_{lo}^2 = \frac{(\partial P/\partial z)}{(\partial P/\partial z)_{lo}} \quad , Y^2 = \frac{(\partial P/\partial z)_{go}}{(\partial P/\partial z)_{lo}} \quad (34)$$

Again the subscripts lo and go refer to total flow with liquid phase properties and total flow with gas phase properties, respectively. $(\partial P/\partial z)_{go}$ and $(\partial P/\partial z)_{lo}$ can be determined by using single phase equations for shell side in part C. By an integral over the mass quality change, the following expression for pressure drop with shell side condensation can be obtained:

$$\Delta P = (\frac{\partial P}{\partial z})_{lo}\ L\int_{x_1}^{x_2}[1+(Y^2-1)(x-x^2)^{0.815}+x^{1.37}].dx \quad (35)$$

V. Optimization Procedure Using GA

Finding a geometry leading to the lowest cost plays an important role in optimization of AHP heat exchangers. By considering tube inside diameter (d_i), tube outer diameter (d_o), tube length (l), shell diameter (D_s) and baffle spacing (L_b) and tube pitch (P_t) as GA variables for each heat exchanger, we perform the optimization to find the minimum

Total annual cost (TAC). Following criteria must be satisfied during the optimization:

$$
\begin{aligned}
&0.2\ m \le D_s \le 2\ m \\
&0.014\ m \le d_o \le 0.05\ m \\
&3 \le \frac{l}{D_s} \le 15 \\
&0.2 \le \frac{L_b}{D_s} \le 1 \\
&\Delta P_t \le \Delta P_{t,max} \\
&\Delta P_s \le \Delta P_{s,max}
\end{aligned}
\tag{36}
$$

The required data are presented in table II to VI. We suppose that during phase changes (condensation or evaporation), the temperature is constant. Also, the unknown temperatures are reported in data tables as T_1 to T_7.

TABLE II
REQUIRED STREAM DATA OF SHX

SHX	Shell	Tube
Mass(kg/s)	149.6	152.4
P(bar)	1.2	1.2
R(k/m^{2}w)	0.00018	0.00018
Inlet		
Phase	L	L
T(°c)	T_1	101.8
ρ(kg/m^3)	1565	1682
Cp(J/kg k)	1956	1621
μ (CP)	1	1
K(W/m k)	0.38	0.37
Outlet		
Phase	L	L
T(°c)	T_2	T_3
ρ (kg/m^3)	1696	1557
Cp(J/kg k)	1600	1900
μ (CP)	1	1
K(W/m k)	0.36	0.39

SHX = solution heat exchanger, P = inlet stream pressure, R = fouling resistance, T = temperature, ρ = density, Cp = heat capacity, μ = viscosity, K = thermal conductivity

TABLE III
REQUIRED STREAM DATA OF GENERATOR

Generator	Shell	Tube	
Mass(kg/s)	4.3	152.45	
P(bar)	12	1.2	
R(k/m^{2}w)	0.00018	0.00018	
Inlet			
Phase	V	L	
T(°c)	254.2	T_4	
ρ (kg/m^3)	5.1	1557	
Cp(J/kg k)	2044	1900	
μ (CP)	0.02	1	
K(W/m k)	0.04	0.39	
Outlet		Two phase	
Phase	L	L	V
T(°c)	188	T_5	T_5
ρ (kg/m^3)	878	1565	0.57
Cp(J/kg k)	4453	1956	1944
μ (CP)	0.1	1	0.02
K(W/m k)	0.67	0.38	0.032

TABLE IV
REQUIRED STREAM DATA OF CONDENSER

Condenser	Shell	Tube	
Mass(kg/s)	2.78	10.92	
P(bar)	1.2	1	
R(k/m^{2}w)	0.00018	0.00018	
Inlet		Two phase	
Phase	V	L	V
T(°c)	104.8	99.6	99.6
ρ (kg/m^3)	0.57	958.4	0.59
Cp(J/kg k)	1944	4217	1908
μ (CP)	0.02	0.3	0.01
K(W/m k)	0.032	0.68	0.024
Outlet		Two phase	
Phase	L	L	V
T(°c)	104.8	99.6	99.6
ρ (kg/m^3)	954.6	958.4	0.59
Cp(J/kg k)	4224	4217	1908
μ (CP)	0.2	0.3	0.01
K(W/m k)	0.68	0.68	0.024

TABLE V
REQUIRED STREAM DATA OF EVAPORATOR

Evaporator	Shell		Tube
Mass(kg/s)	7.25		2.78
P(bar)	0.1		0.07
R(k/m^{2}w)	0.00018		0.00018
Inlet			
Phase	V		L
T(°c)	45.8		40.2
ρ (kg/m^3)	0.07		992.2
Cp(J/kg k)	1877		4178
μ (CP)	0.01		0.6
K(W/m k)	0.02		0.63
Outlet	Two phase		
Phase	L	V	V
T(°c)	45.8	45.8	40.2
ρ (kg/m^3)	990	0.07	0.05
Cp(J/kg k)	4179	1877	1875
μ (CP)	0.5	0.01	0.01
K(W/m k)	0.63	0.02	0.02

TABLE VI
REQUIRED STREAM DATA OF ABSORBER

Absorber	Shell		Tube	
Mass(kg/s)	152.45		10.92	
P(bar)	0.07		0.07	
R(k/m^{2}w)	0.00018		0.00018	
Inlet	Two phase			
Phase	L	V	L	
T(°c)	T_6	T_6	93.5	
ρ (kg/m^3)	1699	0.04	963	
Cp(J/kg k)	1601	1897	4207	
μ (CP)	1	0.01	0.3	
K(W/m k)	0.36	0.025	0.67	
Outlet			Two phase	
Phase	L		L	V
T(°c)	T_7		99.6	99.6
ρ (kg/m^3)	1682		958	0.59
Cp(J/kg k)	1619		4217	1908
μ (CP)	1		0.3	0.01
K(W/m k)	0.37		0.68	0.024

The number of adjustable variables is 30. An initial population of Chromosomes is randomly generated. The population in each generation is taken as 100 and crossover probability and mutation probabilities are chosen to be 0.7 and 0.1 respectively.

The results of optimal design found by the GA are presented in table VII.

TABLE VII
OPTIMAL HEAT EXCHANGER GEOMETRIES FOUND BY GA

	SHX	Gen	Cond	Evap	Abs
di(mm)	11.5	10.8	19.8	17.2	22.3
do(mm)	14.4	14.1	23.6	20.1	27.4
Pt(mm)	21.6	21.15	35.4	30.15	41.1
L_b(m)	0.432	0.77	0.497	0.967	0.995
Ds(m)	0.864	1.54	0.994	1.935	1.991
Nt	1256	4142	618	3222	1842
L(m)	5.897	4.62	4.33	5.806	6.265
Area (m^2)	236.97	445.19	400.71	1.73×10^3	2.96×10^3
ΔP_s (pa)	2.07×10^4	1.43×10^3	1.72×10^4	1.38×10^3	1.01×10^3
ΔP_t (pa)	1.49×10^4	6.19×10^3	9×10^3	591.15	1.1×10^4
hs (W/ m^2k)	2.45×10^3	764.42	635.86	909.1	470.6
ht (W/ m^2k)	3.45×10^3	2.48×10^3	6.3×10^4	1.62×10^4	8.09×10^3
U (W/ m^2k)	843.9	449.25	499.11	637.1	371.61
TAC ($/year)			383998		

SHX = solution heat exchanger, Gen = Generator, Cond = Condenser, Evap = Evaporator, Abs = Absorber, mm = millimeter, m = meter, di = Tube inside diameter, do = Tube outside diameter, Pt = Tube pitch, Lb = Baffle spacing, Ds = shell diameter, Nt = Number of tubes, L = tube length, ΔPs = shell side pressure drop, ΔPt = tube side pressure drop, hs = shell side heat transfer coefficient, ht = tube side heat transfer coefficient, U = overall heat transfer coefficient.

The unknown temperatures were found after optimum design, which can be seen in table VIII:

TABLE VIII
STREAM TEMPERATURES FOUND BY GA

Temperature	°C
$T1 = T5$	184.6
$T2$	107.8
$T3 = T4$	178
$T6$	105.51
$T7$	101.9

VI. CONCLUSION

In this work, the optimum design of an absorption heat pump integrated with a pulp and paper industry is carried out, by considering shell and tube heat exchangers for the AHP components. Six variables related to geometry of each heat exchanger are considered to get the best design with lowest total annual cost by GA, regarding to the considered AHP configuration and desired heat duties. By comparison between the cost resulted by GA (383998 $/year) and the one resulted from general simulation by Aspen Plus software before optimization (471163 $/year), the TAC is decreased by 18%.

ACKNOWLEDGMENT

The authors would like to thank Ms. Shokoufe Tayyebi for her constructive help when warping up the work. Without her assistance this manuscript could not meet the paper submission deadline.

REFERENCES

[1] L. Savulescu, B. Poulin, A. Hammache, S. Bedard, S. Gennaoui, “Water and Energy Savings at a Kraft Paperboard mill using Process Integration”, P&PC 106, 29-31 (2005).
[2] B. Bakhtiari, E. Mateos, R. Legres, J. Paris, “Integration of an Absorption Heat Pump in the Kraft Pulping Process”: Feasibility study, Repr. PAPTAC Ann. Meet., A, 235-239, Montreal (2007).
[3] A. Costa, V. Neuhann, J. Vaillancourt, “Applications of Absorption Heat Pumps in the Pulp and Paper industry for Increased Efficiency and Reduction of GHG Emissions”, 191-195, Montreal (2004).
[4] A. Costa, B. Bakhtiari, S. Schuster, J. paris, “Integration of Absorption Heat Pumps in a Kraft Pulp Process for Enhanced Energy Efficiency”, Energy 34, 254-260 (2009).
[5] H. Jarlos, “Thermodynamic Modeling of Absorption Heat Pumps”, Master’s Thesis, Department of Energy and Environment, Chalmers University, 2010.
[6] S. Kalogirou, G. Florides, S. Tassou, L. Wrobel, “Design and Construction of A Lithium Bromide Water Absorption Refrigerator”, CLIMA 2000/Napoli 2001 World Congress – Napoli (I), 15-18 September 2001
[7] R. Smith, “Chemical Process Design and Integration”, Wiley: New York, 2005.
[8] A. Costa, E. Queiroz, “Design Optimization of Shell and Tube Heat Exchangers”, Applied Thermal Engineering 28 (2008) 1798–1805.
[9] M. Serna, A Jime´nez, “An efficient method for the design of shell and tube heat exchangers”. Heat Transfer Engineering 2004; 25:5–16.
[10] R. Selbas,O¨ . Kizilkan, M. Reppich, “A new design approach for shell and tube heat exchanger using genetic algorithms from economic point of view”, Chemical Engineering and Processing 45 (2006) 268275.
[11] B. Allen, L. Gosselin, “Optimal Geometry and Flow Arrangement for Minimizing the Cost of Shell and Tube Condenser”, International Journal of Energy Research 32, 958–969 (2008).
[12] J. M. Ponce, M. Serna, V. Rico, A. Jime´nez, “Optimal design of shell and tube heat exchangers using genetic algorithm”, 16th European Symposium on Computer Aided Process Engineering and 9th /International Symposium on Process Systems Engineering, 2006.
[13] Ph. Wildi-Tremblay, L. Gosselin, “Minimizing shell and tube heat exchanger cost with genetic algorithm and considering maintenance”, International Journal of Energy Research 31, 867–885 (2007).
[14] TW. Botsch, K. Stephan, “Modelling and simulation of the dynamic behavior of shell-and-tube condenser”, International Journal of Heat and Mass Transfer 1997; 40:4137–4149
[15] JL. Alcock, DR. Webb, “An experimental investigation of the dynamic behavior of a shell and tube condenser”, International Journal of Heat and Mass Transfer 1997; 40: 4129–4135.
[16] G.A. Smook, “Handbook for Pulp Paper Technologists”, Angus Wild Publication Inc., Vancouver (2002).
[17] M. Marinova, E. Mateos-Espejel, B. Bakhtiari, J. Paris, “A New Methodology for the Implementation of Trigeneration in Industry”: Application to the Kraft Process, 333-351, (2007).
[18] B. Jabbari, N. Tahouni, M. H. Panjeshahi, “Improving Energy Efficiency in Pulp and Paper industry, Using a CCHP System”, Proceeding of the 7th International Chemical Engineering Congress & Exhibition, (IChEC), Kish Island, Iran, 21-24 Nov, 2011.
[19] M. R. Jafari Nasr, G. T. Polley, “An Algorithm for Cost Comparison of Optimized Shell-and-Tube Heat Exchangers with Tube Inserts and Plain Tubes”, Chem. Eng. Technol. 23 (2000) 3.
[20] R. S. Hall, J. Matley, and K. J. McNaughton, “Current Costs of Process Equipment”, *Chem. Eng.* 89(7), 80-116 (Apr. 5, 1982).
[21] RK. Sinnot, Coulson and Richardson’s Chemical Engineering. Butterworth-Heinemann: Stoneham, 1996.
[22] RK Shah, DP. Sekulic, “Fundamentals of Heat Exchanger Design”, Wiley: New Jersey, 2003
[23] V.K. Patel, R.V. Rao, “Design optimization of shell-and-tube heat exchanger using particle swarm optimization technique”, Applied Thermal Engineering, Vol. 30, pp. 1417-1425, (2010).

[24] D.Q. Kern, Process Heat transfer, McGraw-Hill, New York (1950), p. 161-167
[25] J. C. Chen, “Correlation for boiling heat transfer to saturated fluids in convective flow”, Ind Eng Chem Proc Des Dev, 5: 322-329, 1996.
[26] D. Chisholm, “A theoretical basis for the Lockhart-Martinelli correlation for two phase flow”, Int J Heat Mass Trans, 10: 1767-1778, 1967.
[27] IDR. Grant, D. Chisholm, “Two-phase flow on the shell side of a segmentally baffled shell and tube heat exchanger”, International Journal of Heat Transfer 1979; 101:38–42.

Banafshe Jabbari was born in Amol, Iran in 1987. She received her Bachelor degree in chemical engineering from Amirkabir University of Technology, Tehran, Iran, in 2009. She is currently an M.S. student in chemical engineering at the University of Tehran, Tehran, Iran and working on design and optimization of CCHP systems using pinch and exergy analysis with pressure drop consideration.

Nassim Tahouni was born in Tehran, Iran in 1975. She studied her Bachelor degree in chemical engineering at Sharif University of Technology, Tehran, Iran and the Master and Doctoral degrees in chemical engineering at the University of Tehran, Tehran Iran. Nassim Tahouni earned her Ph.D. in 2008. She is now assistant professor of Chemical Engineering at the University of Tehran, School of Chemical Engineering. She has carried out many research projects as well as consultancy services in the fields of Process Integration, Energy and Environment.Dr. Tahouni is a member of Iranian Association of Chemical Engineering and also Iran Energy Association. She has been working in the field of Process Integration over the past 12 years, mainly dealing with Process Design & Optimization, Retrofit of Process Industries, Sub-ambient Systems and Low-temperature Separation. She has also published more than 25 papers in scientific journals and conference proceedings.

M. Hassan Panjeshahi was born in Tehran, Iran in 1957. He studied his Bachelor degree in chemical engineering at Sharif University of Technology, Tehran, Iran and the Master and Doctoral degrees in chemical engineering at the University of Manchester, Manchester, UK. Hassan Panjeshahi earned his Ph.D. in 1991.He is now a professor of Chemical Engineering at the University of Tehran, School of Chemical Engineering, and also adjunct professor in the Department of Chemical and Petroleum Engineering, University of Calgary, Canada and visiting professor in the Department of Chemical Engineering, Universiti Teknologi Petronas. Between 2003 and 2008, he was chancellor of the Petroleum University of Technology in Iran.Professor Panjeshahi is currently a member of editorial board of eight scientific journals and also member of several scientific societies, associations and institutions, including Iran Academy of Sciences. He is the founder and the first president of the Iran Energy Association and also the founder and the first Editor-in-chief of the Iranian Journal of Energy affiliated to the World Energy Council. Prof. Panjeshahi has been working in the field of Process Integration over the past 23 years, mainly dealing with Process Optimization, Energy Conservation, Emission Reduction, Water and Waste Water Minimization in Process Industries. He has done many consultancy services for different process industries and also carried out numerous applied research projects in different aspects of Process Integration. He has also published more than 110 papers in scientific journals and conference proceedings.