THERMO FLUIDIC ANALYSIS OF SHELL & TUBE TYPE HEAT EXCHANGER BY GENETIC ALGORITHM

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ABSTRACT: Heat exchangers are one of the mostly used equipment in the industries. Heat exchangers are the device which are used to transfer heat between two streams. Among various heat exchanger shell and tube type heat exchangers have been widely used equipment in power plants, petroleum refining, steam generation and many more providing relatively large ratios of heat transfer area to volume and weight and way of easy cleanliness. Due to their wide utilization and acceptance, their cost minimization is an important objective. Pressure drop and area required for a certain amount of heat transfer, provides an indication about power requirements and capital cost (Running cost) of a heat exchanger. This paper deals with optimization of shell and tube heat exchanger for its length and total cost (including running cost and capital cost) by optimizing tube outside diameter, baffle spacing and shell inside diameter by implementing software based (MATLAB) multi-objective genetic algorithm. The other purposes of this paper are to deliver information on construction, design and operation of heat exchangers with theoretical demonstration of how to apply theories of heat transfer and fluid mechanics and how to find solutions for practical problems which are due to design, material selection, accommodating of heat exchangers.
KEYWORDS: Shell and tube heat exchanger, genetic algorithm, optimizations, thermal design

I. INTRODUCTION

Heat Exchanger is a device which provides a flow of thermal energy between two or more fluids at different temperatures. Heat exchanger finds application in space heating, refrigeration, air conditioning, power plants, chemical plants, petrochemical plants, natural gas processing, and sewage treatment whereas in industries, heat exchangers are used in industrial process to recover heat between two process fluids. Heat exchanger may also be used to sterilize, pasteurize, fractionate, distill, or control process fluid. Generally there are no moving parts in a heat exchanger with some exceptions such as rotary regenerators. At present heat exchangers are available in different configurations. Depending upon their application, mode of heat transfer process fluids and flow, heat exchangers can be classified.

Though there are various types of heat exchangers available in the industry, however the Shell and Tube Type of heat exchanger is probably the most widely used heat exchanger. Shell-and-tube heat exchanger (STHE), in which one flow goes along a bundle of tubes and the other within an outer shell, parallel to the tubes, or in cross-flow. The heat transfer between the two fluids takes through the wall of the tubes. The principal components of a shell and tube heat exchanger are:

- Shell
- Tubes
- Shell cover
- Tube sheet
- Channel
- Channel cover
- Baffles and
- Nozzles

Other components of shell and tube heat exchanger include tie-rods and spacers, pass partition plates, sealing strips, impingement plate, longitudinal baffle supports, and foundation. The Standards of the Tubular Exchanger Manufacturers Association (TEMA)describe these various components in detail.

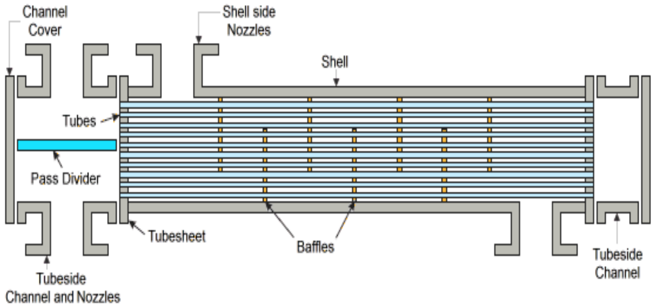

Fig 1.1: Schematic diagram of component of shell and tube type heat exchanger

II. THERMAL DESIGN OF SHELL AND TUBE HEAT EXCHANGER

The design of shell and tube heat exchanger involves a large number of geo- metric and operating variables as a part of the search for heat exchanger geometry that meets the heat duty requirement and a given set of design constrains. Before discussing actual thermal design, let us look at the data that must be required before design can begin:

- Flow rates of both the fluids
- Inlet and outlet temperatures of both the fluids
- Operating pressure of both streams: This is not really necessary for liquids, as their properties do not vary with pressure. required for gases, especially in those cases in which the gas density is not furnished
- Allowable pressure drop for both streams: This is a very important parameter for shell and tube heat exchanger design. Generally, for liquids, a value of 0.5–0.7 kg/cm2 is permissible. For gases, the value is generally 0.05–0.2 kg/cm2, with 0.1 kg/cm2 being generally acceptable.

- Fouling resistance for both streams: If the value of fouling resistance is not furnished, the designer should adopt the values of fouling resistance specified in the TEMA standards or based on past experience of designer.
- Physical properties of both streams: It includes properties such as viscosity, density, thermal conductivity and specific heat, preferably at both inlet and outlet temperatures.
- Heat duty:The heat duty specified should be consistent throughout for both the tube side and the shell side.
- Preferred tube size:Tube size is represented generally as O.D. * thickness * length. Taking inventory considerations into account some plant owners have a preferred O.D. * thickness and the available plot area will be used to calculate the maximum tube length.
- Maximum shell diameter: The maximum value of shell diameter depends upon tube-bundle removal requirements and is limited by crane capacities.
- Materials of construction:If the tubes and shell are made of same materials, all components should be of this material. However, if the shell and tubes are made of different metallurgy, the materials of all principal components should be specified to avoid any ambiguity.
- Special considerations:Special considerations include cycling, alternative operating scenarios, upset conditions and whether operation is continuous or intermittent.

Thermal design of shell and tube heat exchanger is divided into two categories:

- Tube side design
- Shell side design

TUBE SIDE DESIGN

Tube side calculations are quite straight forward, since tube side flow represents a simple case of flow through a circular conduit (pipe). Pressure drop and heat-transfer coefficient both vary with tube side velocity, the latter more strongly so.

Heat-transfer coefficient

The tube side heat-transfer coefficient is a function of the Reynolds number, the Prandtl number, and the tube diameter. These can be splinted down into the following fundamental parameters: physical properties such as viscosity, thermal conductivity, and specific heat; tube diameter; and, most important heat transfer coefficient depends upon mass velocity. The variation in liquid viscosity has the most dramatic effect on heat-transfer coefficient.

This can be understood by this fundamental equation for turbulent heat-transfer inside tubes is:

$$h_t = 0.023\frac{k_t}{d_0}Re_t^{0.8}Pr_t^{\frac{1}{3}} \quad (1)$$

From above equation it is quite clear that viscosity influences the heat-transfer coefficient in two opposing ways — as a parameter of the Reynolds number, and as a parameter of Prandtl's number. Thus, from Eq. 1

$$h \propto \mu^{0.33-0.8}$$
$$h \propto \mu^{-0.47}$$

These two factors lead to some interesting generalities about heat transfer.

Pressure drop

Tube side pressure drop varies to the square of mass velocity (can be understood easily by considering pressure drop relation that exist in flow through conduit case). Thus, with increase in mass velocity results in rapid increase in pressure drop than does the heat-transfer coefficient. Consequently, there will be an optimum mass velocity above which there is no benefit to increase mass velocity further. Furthermore, very high value of velocities lead to erosion.

SHELL SIDE DESIGN

The shell side design calculations are far more complex than those for the tube side. The main reason behind this complexity is that on the shell side there is one principal cross-flow stream and four leakage or bypass streams in place of one flow stream. There are various shell side flow arrangements, baffling designs as well as various tube layout patterns, which together influence the shell side stream analysis.

Shell configuration

TEMA (tubular exchange manufacturing association) defines various shell patterns based on the flow of the shell side fluid through the shell named as E, F, G, H, J, K, and X. it is quite important to furnish the shell configuration. Every configuration has its own benefits and limitation clearly defined by TEMA.

Tube layout patterns

There are four tube layout patterns, as shown in Figure 2 triangular (30°), rotated square (45°), rotated triangular (60°), square (90°).

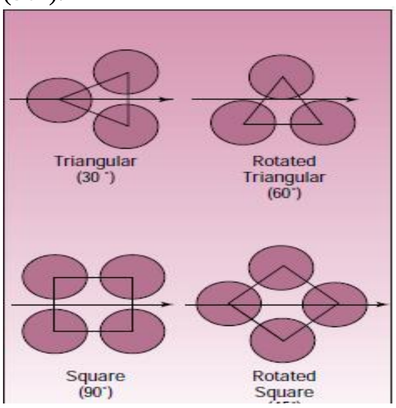

Figure 2: Types of layout

Tube pitch

Tube pitch is defined as the shortest distance between two adjacent tubes in a shell and tube heat exchanger. For each pattern, TEMA specifies a minimum tube pitch in terms of tube O.D.

Baffling

They force the fluid of shell side to flow across the tubes to ensure high heat transfer rates and also provide support for tube bundle.

Type of baffles: Baffles are of two types: plate and rod. Plate baffles may be single-segmental, double-segmental, or triplesegmental, as shown in Figure 3.

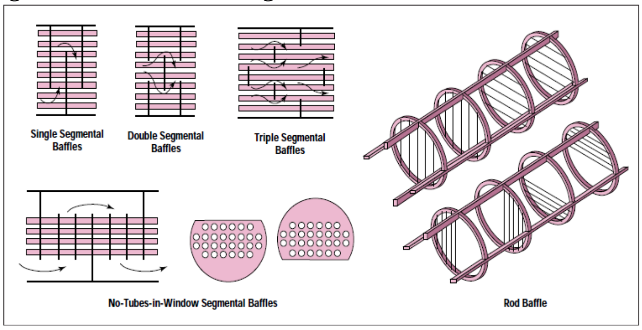

Figure3: Types of baffles

Baffle spacing. Baffle spacing is defined as the centerline-to-centerline distance between adjacent baffles. It is the most important parameter in STHE design. The TEMA standards defines the minimum value of baffle spacing as 0.2 of the shell inside diameter or 2 inch, whichever is maximum. If closer spacing is used it will result in poor bundle penetration and stream distribution by the shell side fluid and also produces difficulty in mechanically cleaning the outsides of the tubes. The maximum value of baffle spacing is equal to the shell inside diameter.

Baffle cut. As shown in Figure 4, baffle cut is defined as the height of the segment that is cut in each baffle to permit the shell side fluid to flow across the baffle. This is generally expressed as a percentage of the shell inside diameter.

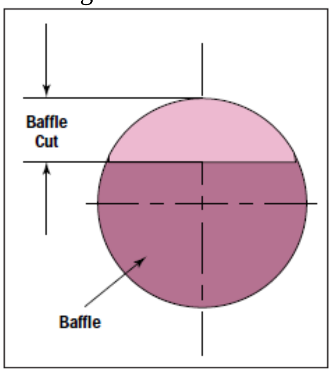

Figure 5: Baffle cut

Both very small and very large baffle cuts are detrimental to efficient heat transfer on the shell side due to large deviation from an ideal situation, as illustrated in Figure 6.

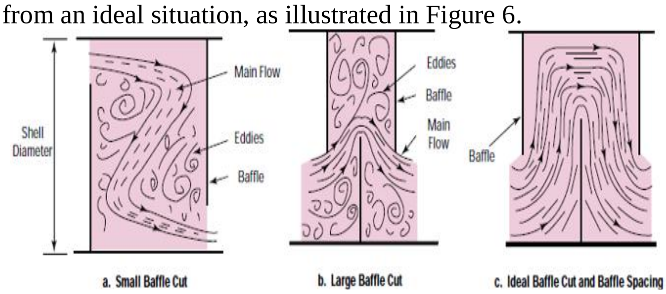

Figure 6: Effect of small and large baffle cuts

III. PROBLEMS DEFINITION

Due to the wide acceptance of shell and tube heat exchangers in industrial processes their cost minimization is an important target for both designers and users.

In order to obtain maximum heat exchanger performance at the lowest possible operating and capital costs without comprising the reliability, the following features are required of a shell and tube heat exchanger:

(1) Higher heat transfer coefficient and larger heat transfer area

(2) Lower pressure drop

Pressure drop and heat transfer rates are the entities that are inter-related and both pressure drop and heat transfer rates influence the capital and operating costs of a heat exchanging system in a very decisive way. The sum total of the pressure drops that occurs across the shell side and tube side is used to determine the pumping power requirement and furthermore the annual operating cost of shell and tube heat exchanger.

IV. GOVERNING EQUATION

The design procedure used is based on the Mean logarithm temperature difference

According to adopted method mean logarithm temperature difference is calculated by:

$$A = \frac{Q}{U\Delta T_{ML}F}$$

The overall heat transfer coefficient is determined through the following equations:

$$U = \frac{1}{\left[\frac{1}{h_s} + R_{fs} + \frac{d_0}{d_i}\left(R_{ft} + \frac{1}{h_t}\right)\right]}$$

According to the flow regime,the tube-side heat transfer coefficienth_t is calculated by the following correlations:

$$h_t = 0.027\frac{k_t}{d_0}Re_t^{0.8}Pr_t^{\frac{1}{3}}\left(\frac{\mu_t}{\mu_{wt}}\right)^{0.14}$$

The above given relation is valid for

$$Re_t > 10{,}000$$

The tube side Reynolds number $Re_t$is given by,

$$Re_t = \frac{\rho_t v_t d_i}{\mu_t}$$

Flow velocity for tube side is found by,

$$v_t = \frac{m_t}{\left(\frac{\pi}{4}\right)d_t^2\rho_t}\left(\frac{n}{N_t}\right)$$

$N_t$is the number of tubes and n is the number of tube passes which can be found approximately from the following equation,

$$N_t = C\left(\frac{D_s}{d_o}\right)^{n_1}$$

C and n_1 are coefficients that are taking values according to flow arrangement and number of passes. These coefficients are shown in Table 1 for different flow arrangements.

The tube side Prandtl number Pr_tgiven by,

$$Pr_t = \frac{\mu_t C_{pt}}{k_t}$$

Darcy friction factor f_t is given as,

$$f_t = (1.82log10^{Re_t} - 1.64)^{-2}$$

Kern's method for shell side design

For segmental baffle shell-and-tube exchanger Kern's formulation is used for calculating shell side heat transfer coefficienth_s,

$$h_s = 0.36\frac{k_t}{d_e}Re_s^{0.55}Pr_s^{\frac{1}{3}}\left(\frac{\mu_t}{\mu_{wts}}\right)^{0.14}$$

Reynolds number for shell side flow is calculated by following correlation

$$Re_s = \frac{m_s d_e}{A_s \mu_s}$$

Where d_e= shell hydraulic diameter and computed as,
Shell side hydraulic diameter is calculated by following correlation
For square pitch

$$d_e = \frac{4\left(S_t^2 - \left(\frac{\pi d_0^2}{4}\right)\right)}{\pi d_0}$$

For triangular pitch

$$d_e = \frac{4\left(0.43S_t^2 - \left(\frac{\pi d_0^2}{4}\right)\right)}{0.5\pi d_0}$$

Cross Section area normal to flow direction is computed by,

$$A_s = D_s B\left(1 - \frac{d_0}{S_t}\right)$$

$$A_s = \frac{m_s}{\rho_s A_s}$$

Prandtl number for shell side is given by,

$$Pr_s = \frac{\mu_s C_{ps}}{k_s}$$

The overall heat transfer coefficient (U) depends on both the tube side and shell side heat transfer coefficients and fouling resistances are given by,

$$U = \frac{1}{\left[\frac{1}{h_s} + R_{fs} + \frac{d_0}{d_i}\left(R_{ft} + \frac{1}{h_t}\right)\right]}$$

Taking into consideration of the cross flow between adjacent baffle, the logarithmic mean temperature difference (LMTD) is given by following correlation,

$$LMTD = \frac{(T_{hi} - T_{CO}) - (T_{ho} - T_{ci})}{\ln\left[\frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}}\right]}$$

The correction factor F for the flow configuration is given by,

$$F = \frac{\sqrt{R^2+1}}{R-1}\ln[\frac{\left(\frac{1-P}{1-PR}\right)}{\left(2-PR+1-\sqrt{R^2+1}\right)/(2-PR+1+\sqrt{R^2+1})}]$$

Which is a function of dimensionless temperature ratio.
Where R = correction coefficient
P = efficiency
Correction coefficient (R) is given by,

$$R = \frac{(T_{hi} - T_{ho})}{T_{co} - T_{ci}}$$

P is the efficiency given by,

$$P = \frac{(T_{co} - T_{ci})}{T_{hi} - T_{ci}}$$

Considering overall heat transfer coefficient, the heat exchanger surface area (A) is computed by,

$$A = \frac{Q}{UFLMTD}$$

The heat transfer rate, for sensible heat transfer is given by,

$$Q = m_h C_{ph}(T_{hi} - T_{ho}) = m_C C_{ch}(T_{co} - T_{ci})$$

The necessary tube length (L), based on total heat exchanger surface area (A) is given by,

$$L = A/\pi d_0 N_t$$

Total pressure drop (total sum of pressure drop on tube side and shell side) is given by,

$$\Delta P_t = \Delta P_{tube\ lengt\,h} + \Delta P_{tube\ elbow}$$

Where

$$\Delta P_{tube\ lengt\,h} = \text{Pressure drop on tube side}$$

$$\Delta P_{tube\ elbow} = \text{Pressure drop on shell side}$$

Pressure drop on tube side is given by,

$$\Delta P_t = \frac{\rho_t v_t^2}{2}\left(\frac{L}{d_i} f_t + p\right) n$$

Different authors considered different values of constant p. Kern assumed p = 4.
The shell side pressure drop is given by,

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

Where,

$$f_s = \text{Friction factor on shell side}$$

Friction factor f_s is given by,

$$f_s = 2b_0 Re_s^{-0.15}$$

And $b_0 = 0.72$, valid for Res< 40,000
Pumping power is given by,

$$P = \frac{1}{\eta}\left(\frac{m_t}{\sigma_t}\Delta P_t + \frac{m_s}{\sigma_s}\Delta P_s\right)$$

Where, η is pumping efficiency.
The objective function has been assumed as the total present cost

$$C_{total} = C_i + C_{oD}$$

The capital investment C_i, a function of the exchanger surface is determined by considering Hall's correlation

$$C_i = a_1 + a_2 S^{a_3}$$

Where the value of constants $a_1, a_2 \& a_3$ is as follows:
a_1= 8000,
a_2 = 259.2 and
a_3= 0.91
The value of above constants are valid if the material for exchangers is stainless steel for both shell side and tube side.
Considering pumping power (used to overcome losses due to friction) the total discounted operating cost is given by following correlation,

$$C_{oD} = \sum_{k=1}^{ny} \frac{C_o}{(i+i)^{k'}}$$

Where,

$$C_o = \text{Annual operating cost}$$

Annual operating cost is computed by,

$$C_o = PC_E H$$

P = Pumping power;
C_E= the energy cost in \$/Wh
H = The annual operating hours

V. OPTIMIZATION ALGORITHM

Optimization procedure is carried out by selecting a genetic algorithm (GA) in MATLB. It is a global search method i.e., this method does not allow to know if a sun-optimal or optimal solution, although good solution has been reached in other words it is similar to the process of natural biological evolution given by Goldberg in 1989). In general, main parameters that are to be fixed in this method are population size (number of individuals) and generation (number of generations). It works on a population of individuals representing candidate solutions to the given problem and the principle used to find out the survival of the fitness is applied and several runs of the program were carried out to produce better performing individuals. Individuals are selected at each generation according to their level of fitness and then are bred together. The adopted procedure leads to the creation individuals better suited to their environment than their parents. A number of advantages is showed by GA as compare to other optimization techniques. This method is suitable for combinatorial optimization problems even of large size. This method examines a population of solution in parallel in place of evaluating a single solution at a time. It does not require any information about the derivative of the objective function as generally happens in gradient methods and the sole value of the given objective function affects the direction of the search. Finally, GAs can give a certain number of possible optimal solutions, and the final choice can be demanded to the user. This characteristic possessed by GAs can be important when the given problem has a group of optimal solutions, as it generally happens in multi-objective optimization.

GA method starts with the creation of an initial population in which a certain number of individuals are randomly created to start the process. The individual of a population represents a string of bits coding the characteristic of the individual itself. In this particular case the individual represents the configuration of heat exchanger which satisfies the design specifications. After that for each individual its cost is calculated, and then the calculation of the fitness function is performed. The fitness function represents the quality of the single individual with respect to the entire population. In this case, an individual whose cost is less than the average cost has a fitness function higher than the average. Next step is the selection process which includes the creation of couples of individuals who will generate off springs when the required number of off springs per generation is definite. It means that at first determining a probability of reproducing for each individual which is proportional to the value of its fitness function, and subsequently picking the couples of individuals for reproduction, also known as sampling, in a number able to produce the required number of off springs. This is carried out resorting to proper selection algorithms. The successive phase is the creation of the new generation of individuals.

The new generation is furnished by:

- Best individuals which is copied from the previous generation
- Randomly generated new individuals
- New individuals which are obtained by crossover recombination of the selected individuals obtained from previous generation
- Mutant individuals.

Best individual which is copied from previous generation enables to maintain the best result reached so far. Genetic variety is maintained by random generation of new individuals, because of absolutely independency of such individuals from the best individual of the present generation, and by doing so it enables to overcome local minima. By recombining of individuals to create offspring enables to regenerate the best features of existing individuals into new individuals. The last step in GAs is the mutation of some individuals; a process used to change a part of an individual to create a new individual which is different from the original. The mutation process enables to examine a wider range of solutions, another way to maintain the genetic variety.

The iterative process is terminated when a specified convergence criterion is met or the number of generations reaches limiting value which is selected earlier. By doing so, the resulting best individual obtained represents the desired solution. Thus result is complied with the constraints at the lowest cost.

A flow chart of the adopted GA method is shown in figure 7.

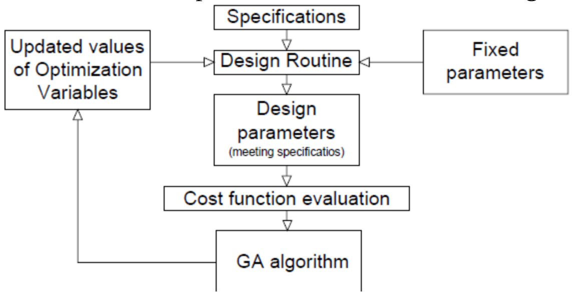

Figure 7: Adopted GA method

A total of 4 design variations are considered for the optimization.

Case 1: n = 2, Triangular tube layout
Case 2: n = 4, Triangular tube layout
Case 3: n= 2, Square tube layout
Case 4: n = 4, Square tube layout

The three design variables are

- Tube outside diameter (d0)
- Baffle spacing (B)
- Shell inside diameter (D_s)

There are two objective functions which are simultaneously optimized to obtain a set of solutions for the best values for both functions.

- Length of shell and tube heat exchanger
- The cost associated with the initial investment and operation of heat exchanger

Empirical and Statistical data

The shell side fluid and tube side fluids are distilled water and raw water respectively. Empirical and Statistical datais shown in table 1.

Table1: Empirical and Statistical data

Fluid/ Properties	Specific heat in $\frac{KJ}{KgK}$	Fouling factor in m^2k/W	Density in kg/m^3	Dynamic viscosity in $pa\ s$	Temp. In/Out in °C	Thermal conductivity of material in W/mk	Mass flow rate in kg/s
Tube side	4.18	0.00017	999	0.008	23.9 /26.7	0.62	22.07
Shell side	4.18	0.00017	995	0.00092	33.9 /29.4	0.62	35.31

Table 2: Values of constants for different layout

NO. OF PASSES	TRIANGULAR TUBE PITCH	SQUARE TUBE PITCH

	K_1	N_1	K_1	N_1
2	0.249	2.207	0.156	2.291
4	0.175	2.285	0.158	2.263

Constants:
$d_i = 0.8d_0$
$p_t = 1.25d_0$
$b_0 = 0.72$
$n_y = 10\ years$
$i = 10\%$
$H = 7000\ yr/hr$
$C_e = 0.12\ units/KWh$
Assumptions:

- $Re_t > 10000$
- $Re_s < 40000$
- p=4 (Kern assumption)

Variable Bounds:

- $0.015 \le d_0 \le 0.051$
- $0.5 \le Ds \le 1.5$
- $0.05 \le B \le 0.5$

VI. RESULT & DISCUSSIONS

Case 1: n = 2, Triangular tube layout
Minimum value of objective 1 = 2.512074
When d0=0.016132, B=0.056833, Ds=1.498633 with maximum cost value = 423595.5
Minimum value of objective 2 = 50203.7
When d0=0.01661, B=0.487396, Ds=0.530987 with maximum length = 7.750892
Moderate value of both objective 1 & 2 = 4.88362 & 58633.08
When d0=0.016274, B=0.411127, Ds=0.756528
Case 2: n = 4, Triangular tube layout
Minimum value of objective 1 = 1.449332
When d0=0.015005, B=0.052773, Ds=1.499951 with maximum value of the cost function = (=1.28E+07)
Minimum value of objective 2 = 66155.38
When d0=0.049461, B=0.49999, Ds=1.499961 with maximum length of heat exchanger = 7.775957
Acceptable value of both function respectively = 7.775957 & 69090.95
When d0=0.031607, B=0.466695and Ds=1.499853

Case 3: n= 2, Square tube layout
Minimum value of objective 1 = 1.2538
When d0=0.15005, B=0.050005, Ds=1.5 with maximum value of objective 2 = 1992995
Minimum value of objective 2 = 53242.4
When d0=0.015005, B=0.49999, Ds=0.7557 with maximum value for length =3.7674
Moderate value of both objective 1 & 2 = 1.7878 & 66804.41
When d0=0.015005, B=0.4386 and Ds=1.28

Case 4: n = 4, Square tube layout
Minimum value of objective 1 = 241.9545
When d0=0.015001, B=0.052165, Ds=1.499999 with maximum value of objective 2 = 2.08E+08
Minimum value of objective 2 = 1382230
When d0=0.051, B=0. 5, Ds=0.948281 with maximum value of objective 1 = 416.4354
Moderate value of both objective & 2 = 300.0257 & 3975894
At d0=0.020535, B=0.363654 and Ds=1.122275

VII. CONCLUSION

Resorting to the analysis of a number of runs with different cases, the paper describes that the given optimization algorithm can effectively be used to improve the design process of shell and tube heat exchanger. It is evident that both length and annual cost of shell and tube heat exchanger are opposing entities. With increase in value of one entity, invariably reduce the value of other function. By adopting this method the total discounted cost of heat exchange equipment can be reduced instead of using traditional sizing procedures through the reduction of operating costs related to pumping losses. This is due to the fact that daily operating cost is affected by average duration of runs per day. The net surface area for shell and tube heat exchanger increases, if the length of heat exchanger increase. Due to this overall efficiency of heat transfer for a given duty cycle and given temperature difference increases. Hence pumpneeds to work for fewer hours. It can be seen that we get highest values of both length andcost function for square tube layout and 4 tube passes among all the four cases.

VIII. FUTURE SCOPE

Therefore, this present method can also be implemented to pursue goals of energy saving in industries. Results obtained from the present approach shows the improvement potential coming. Nevertheless, while there is very small improvement in cost in some cases, it should be noted that in practice this can lead to significant overall savings when a large number of heat exchanging units exist in the same plant as generally happens in most process plant applications. Further, the GA allows the designer for rapid solution of the design problem and enables to analyze a number of alternative solutions having good quality. This gives the designer more degrees of freedom in the final choice respect traditional methods.

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